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## Match 1:

Linke/B rner/He Cylindrical Gears Heinz Linke J rg B rner Ralf He Cylindrical Gears Calculation – Materials – Manufacturing with contributions by Dr.-Ing. J. B rner, Dr.-Ing. R. He , Dr.-Ing. E. R hle, Prof. Dr.-Ing. I. R mhild, Dr.-Ing. M. Senf, Dipl. Phys. Dipl. Ing. (FH) W. Sonntag, Dr.-Ing. A. Spengler, Dr.-Ing. F. Spirling, Dipl.-Ing. G. Tripp, Dr.-Ing. St. Wengler HANSER The Editors: Prof. Dr.-Ing. habil. Heinz Linke, Dresden, Germany Dr.-Ing. J rg B rner, Friedrichshafen, Germany Dr.-Ing. Ralf He , Bocholt, Germany Distributed in North and South America by Hanser Publications 6915 Valley Avenue, Cincinnati, Ohio 45244-3029, USA Fax: (513) 527-8801 Phone: (513) 527-8977 www.hanserpublications.com Distributed in all other countries by Carl Hanser Verlag Postfach 86 04 20, 81631 Munich, Germany Fax: +49 (89) 98 48 09 www.hanser-fachbuch.de The use of general descriptive names, trademarks, etc., in this publication, even if the former are not especially identified, is not to be taken as a sign that such names, as understood by the Trade Marks and Merchandise Marks Act, may accordingly be used freely by anyone. While the advice and information in this book are believed to be true and accurate at the date of going to press, neither the authors nor the editors nor the publisher can accept any legal responsibility for any errors or omissions that may be made. The publisher makes no warranty, express or implied, with respect to the material contained herein. The final determination of the suitability of any information for the use contemplated for a given application remains the sole responsibility of the user. Cataloging-in-Publication Data is on file with the Library of Congress. Bibliografische Information der deutschen Bibliothek: Die Deutsche Bibliothek verzeichnet diese Publikation in der Deutschen Nationalbibliografie; detaillierte bibliografische Daten sind im Internet ber <http://dnb.d-nb.de> abrufbar. All rights reserved. No part of this book may be reproduced or transmitted in any form or by any means, electronic or mechanical, including photocopying or by any information storage and retrieval system, without permission in writing from the publisher. Carl Hanser Verlag, Munich 2016 Editorial Management: Dipl.-Ing. Volker Herzberg, Julia Stepp Translation Management: Pawel Kolobkow, Lingua-World Dresden, Germany Production Management: Arthur Lenner, Der Buch macher, M nchen, Germany Cover Concept: Marc M ller-Bremer, www.rebranding.de, M nchen, Germany Cover Design: Stephan R nigk, M nchen, Germany Cover Illustration: Atelier Frank Wohlgemuth, Hamburg (based on an illustration from Dipl.-Ing. Uwe Trempler, Klingelnberg GmbH) Typesetted by Berit Herzberg, Freigericht, Germany Printed and bound by Books on Demand GmbH, Norderstedt, Germany Printed in Germany ISBN 978-1-56990-489-3 E-Book ISBN 978-1-56990-490-9 V Dedicated in reverence to Richard Stribeck, Karl Kutzbach, Enno Heidebroek and Stefan Fronius. To students, young engineers and scientists as a basis for successful studies, accompanying thoughts and thinking ahead. VII This book was edited with the commendable support of the gear industry, particularly from Siemens – Business Unit Mechanical Drives, Bocholt and the companies listed in the preface. IX Preface Gear units with their positive characteristics will continue to hold their position in power transmission technology in the future. The extremely high degree of efficiency, the relatively small installation space and the favourable mass-power ratio are the key factors in this regard. Their development, production and cooperation are becoming increasingly more international in scale. In this process, the documentation is often written in English. In international panels, the technical discussions are also usually conducted in English. At conferences of an international nature, the English language is used almost exclusively for the written contributions, the lectures and the discussions. Even the internationally agreed-upon standards (ISO) are written in English. Therefore, a number of colleagues expressed the wish that the book "Stirnradverzahnung" should also be published in the English language. We took up this challenge. The English edition was revised by extending the responsibility to the co-authors Dr.-Ing. J. B rner and Dr.-Ing. R. He . This English edition is based on the second edition of the German book "Stirnradverzahnung", published by Carl Hanser Verlag in 2010. In addition to the DIN standards, the internationally applicable standards (ISO, EN) were added during the editing of the manuscripts for this edition. Supplements perceived as useful, as well as clarifications and corrections, were also undertaken. The English edition was revised with the commendable support of the industry, particularly of Siemens AG – Business Unit Mechanical Drives (Bocholt, Germany) and Harmonic Drive AG (Limburg/Lahn, Germany); also worthy of mention are Eickhoff Antriebstechnik GmbH (Bochum, Germany); FRENCO GmbH (Altdorf, Germany); GKN Walterscheid Getriebe GmbH (Sohland/Spree and Lohmar, Germany); HQM H rtetechnik GmbH (Leipzig, Germany); KAPP Werkzeug-maschinen GmbH (Coburg, Germany); KWD Kupplungswerk Dresden GmbH (Dresden, Germany); Reishauer AG (Wallisellen, Switzerland); Voith Turbo GmbH & Co. KG (Crailsheim, Germany); WELTER zahnrad GmbH (Lahr, Germany); and ZF Friedrichshafen AG (Friedrichshafen, Germany). The following colleagues were of particular assistance in reviewing individual sections: Dipl.-Ing. Ines B rner (ZF Friedrichshafen, Materials Technology); Prof. Heinz-Joachim Spies (Bergakademie Freiberg, Materials Technology); Dr.-Ing. Gerhard Gajewski (Siemens Bocholt, Lubrication Technology); Dipl.-Ing. Maik Liesegang (Siemens Bocholt, Noise Behaviour); Robert Schr der (TU Dresden, Computer Science); Dipl.-Ing. Uwe Trempler (Klingelnberg GmbH, H ckeswagen, Load Capacity, Computer Science); Dr.-Ing. Wolfgang Uhlig (TU Chemnitz, Materials Technology); Dip.-Ing. Gerhard Tripp (formerly Flender Bocholt, Mechanical Manufacturing), Dr.-Ing. Steffen Wengler (Otto von Guericke University Magdeburg, Quality and Standards). We owe a special thanks to these colleagues. Our thanks also particularly go to Pawel Kolobkow and his team for their good cooperation and careful and diligent translation. We would also like to express our thanks to Carl Hanser Verlag both for their readiness to publish the English edition of our book and the good cooperation – especially to Volker Herzberg and Julia Stepp for their organisational support, Julia Phelps for her work as proof-reader, Arthur Lenner for the production management and Berit Herzberg for her work on the layout of the book. X Preface Finally, we would like to thank our families for all the understanding and patience they showed during our long commitment to the editing of this book. May the English version of this book serve research, development and mutual understanding in both the national and international framework as well as foster cooperation in collegiality, understanding and friendship. H. Linke, J. B rner, R. He XI The Editors Heinz Linke was born in 1935 in Chem- nitz, Germany. On completing primary school, he learned to become a technician and subsequently worked in the Zschopau motorcycle plant as a toolmaker. From 1952 to 1955, he studied machine tool con-struction at the Mechanical Engineering School in Schmalkalden, after which he worked as a calculation engineer in the VEB Entwicklungsbau Pirna (a publicly owned corporation). His fields of activity primarily consisted in performing strength calculations of gears, elements of gas tur-bines and hydraulic drive system assem-blies. During this industrial work, he took a correspondence course in fluid mechan-ics at the Dresden TU and increasingly dedicated himself to development and re-search tasks. In 1969, he received his doctorate on the subject "drive dynamics". In 1971, Dr. Linke started work at the Chair of Machine Elements at the Dresden TU. In 1973, he became a lecturer for design technology, and – following his habilitation in 1979 – was appointed full Professor for Design Technology (Design Theory / Machine Elements). He continued to exercise this function at the Dresden TU until he reached the age limit for professors in 2000. Alongside teaching and research, his tasks also included the development of a unified standard for the calculation of load capacity of gears in the member countries of Comecon (Council for Mutual Economic Assistance), the economic area comprising the Soviet Union and its satellite states. Subsequently, he was responsible for developing the fundamentals of DIN 743 (Load capacity of shafts and axles) and the VDI 2737 Guideline (Calculation of the tooth root load capacity of internal gears). Prof. Linke is an overseas member of the American Gear Manufacturers Association (AGMA). He possesses 60 years of experience in the area of the calculation of machine elements – particularly gear drives. Prof. Linke is still actively engaged in development and research for the industry up to the present day. Many of his research findings and experiences have been published in journal articles and at both national and international conferences. XII The Editors J rg B rner was born in 1960 in Elsterwerda, Germany. From 1980 to 1985, he studied Fundamentals of Mechanical Engineering at the Technische Universit t, Dresden, spe-cialising in Design Technology. From 1985 to 1989, he was a research student at the Chair for Machine Elements at the Dresden TU, receiving his doctorate in 1989 with his doctoral thesis on "Developing the minimum calculation model for internal dynamic tooth forces". Up to 1997, he was employed at the Institute for Machine Elements and Machine Design at the Dresden TU, engaged in the development of calculation programs for the load and stress distribution on gears as well as in basic investigations on stress concentration. From 1997 to 2000, Dr. B rner worked as a calculation engineer in gear development at the Tech Center of Caterpillar Inc, in Peoria/Illinois, USA. Since 2000, he has been engaged as a specialist for gear design and recalculation in the Gear Development Department at the ZF Friedrichshafen AG Company in Friedrichshafen. His work focuses on the further development of calculation software for gears based on theoretical analyses as well as his experiences gained from testing and using ZF products. He is also actively involved in the Forschungsvereinigung Antriebstechnik (Drive Technology Research Association), where he works in the Cylindrical Gearing working group. Ralf He was born in 1957 in Neustrelitz, Germany. From 1978 to 1983, he studied Fundamentals of Mechanical Engineering at the Technische Universit t, Dresden, spe- cialising in Design Technology. To 1986, he worked on his doctoral thesis as part of a research study at the Chair for Machine Elements at the Dresden TU, and received his doctorate on the subject "Influence of shafts and bearings on the load distribution in cylindrical gears". In 1986, he began work as a research assistant in the VEB Getriebe und Kupp-lungen Magdeburg (Gears and Clutches, a publicly owned corporation). In 1990, Dr. He joined the A. Friedr. Flender AG Company, which as a result of a merger is now the Mechanical Drives Division of Siemens AG. There he worked for four years on the technological development of bevel gears, before moving to the Technical Calculation Department. He headed this department from 2003 to 2013. Since 2014, he has been employed at Mechanical Drives as Senior Key Expert. Dr. He is also a member of the DIN working groups for Tooth Load Capacity Calculation, Bevel Gears, and Cylindrical Gear Tolerances. He is a member of AGMA and was actively engaged in drawing up the AGMA 6006 standard. As part of the ISO TC60 Gears committee, Dr. He is an active member of the Accuracy of Gears and Bevel Gears working groups. XIII Index of Authors Dr.-Ing. J. B rner ZF Friedrichshafen AG, Friedrichshafen Section 5.3; 6.2; 6.3; Appendix 5; 9 Review of the translation Dipl.-Ing. I. B rner ZF Friedrichshafen AG, Friedrichshafen Dr.-Ing. R. He Siemens AG, Bocholt Prof. Dr.-Ing. habil. H. Linke TU Dresden, Institute of Machine Elements and Machine Design Appendix 17.3 Support in the review of the translation; Selection of standards Section 1; 2; 3.1; 3.2; 4; 5.1; 5.2; 6.1; 6.5.1; 6.5.2; 6.5.3; 6.5.6; 6.5.7; 7.2; 9.1; 9.2; Appendices Dr.-Ing. E. R hle (Retiree, TU Dresden, Institute of Machine Elements and Machine Design) Section 3.3 Prof. Dr.-Ing. I. R mhild HTW Dresden, Professorship for Engineering and Power Transmission Technology Section 7.1; 7.3 Dr.-Ing. M. Senf TU Dresden, Institute of Machine Elements and Machine Design Section 6.4 Dipl.-Phys., Dipl.-Ing. (FH) W. Sonntag (Retiree, TU Dresden, Institute of Machine Elements and Machine Design) Section 6.5.4; 6.5.5; 6.6; Appendix 13, 14 Dr.-Ing. A. Spengler (Retiree, Bergakademie Freiberg) Section 7.4; 9.3; 10.3; Appendix 11, 12 Dr.-Ing. F. Spirling (Retiree, Ingenieurb ro Antriebstechnik Dr.-Ing. Spirling, Dresden) Section 6.7 Dipl.-Ing. G. Tripp (Retiree, A. Friedr. Flender AG, Bocholt) Section 10.1; 10.2; Appendix 15 (The first edition was written by Prof. W. Thyssen; the second edition – and the English edition – was extensively revised by Dipl.-Ing. Tripp) Dr.-Ing. St. Wengler Otto-von-Guericke-Univ ersit t Magdeburg, Institute for Manufacturing Technology and Quality Assurance Section 8; Appendix 3.2 (The first edition of Section 8 was written by Dr. P. Sch cke; the English edition was written by Dr. Wengler) XV Remarks on the Special Features of the Book The second German edition, published in 2010, has been supplemented, clarified and corrected in several details for this English edition. The internationally valid ISO standards, DIN EN ISO standards and DIN EN standards, in particular, have been supplemented along with the DIN standards. The information on the DIN standards has been retained because they are still used internationally. Appendices 17.1 and 17.2 contain a compilation of the important standards used in this book, supplemented by selected AGMA standards. Appendix 17.3 provides information on material standards and terms that used to be customary in some countries. New is also Appendix 18, which contains a special compilation of the basic gear geometry commonly used in the USA in parallel with the DIN and ISO standards, based on "diametral pitch" and "circular pitch". Following the tendency particularly prevalent in the English-speaking world, the decimal point is used to separate decimal places from integer values. When using the ISO, EN and DIN standards (particularly in the previous German publications of this book) the comma is used as the decimal separator in numbers. This also applies to a few exceptions remaining in this book that use the previous notation. The mathematically depicted contexts are based in some cases on the authors' own data and deri- vations – but primarily on the ISO standards. Nevertheless, the descriptions of the load capacity computation and the calculation equations are not identical reproductions of the ISO standards. In several cases, the authors have deviated from them if – from experience – clarifications have been judged as favourable or additions as useful. The statements on load capacity, in particular, also represent explanations and supplements to applicable standards. The ISO developments have resulted in the introduction of new symbols in relation to the DIN standards. In some cases, they were introduced for newly published standards during the editing of this book. However, it is not possible for any misunderstandings to arise from the retention of some symbols from previous and still valid DIN standards in this book. The understanding is en-sured by the list of symbols included at the end of each section. It also appeared reasonable to use the symbol M for the torque (torsion moment). It represents only one component of the gen- eral 3-axis torque system. The three components required are usually denoted by the symbol M. During the editing of the English edition, a different basis for gear geometry was formulated based on DIN EN ISO 21771. A particular feature here is that – unlike DIN 3960 – the diameter and the centre distance are also calculated as positive quantities for internal gears. This edition of the book is based on the previous consideration of DIN 3960 that the derivation of the geometry for externally toothed systems is uniformly applicable for external and internal gearing. Accord-ing to DIN 3960, both the diameter and the centre distance in internally toothed gears have nega-tive signs, because the number of teeth of internal gears is negatively defined. Irrespective of this, most guidelines specifically applicable for this lay down that for drawing data, the diameter of the internal toothing and the centre distance (positively measurable quantities as absolute amounts) have to be specified with a positive sign in the production documents. XVII Contents Preface .............................................................................................................................. .................................... 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........ 835 Index .............................................................................................................................. ....................................... 837 1 1 Overview and General Principles 1.1 Development of Toothing 1.1.1 Development of the Application of Gears and Gear Drives This introductory review is designed to assist the reader in placing present-day developments into their correct perspective and gain the necessary detachment from the “hustle and bustle” of today. Even in this area, the brilliant achievements of the great scientists, designers and inventors of previ-ous decades arouse not only wonder and awe but also inspiration for this fascinating field of technology, which still today contains a sufficient number of problems awaiting solution both for us and future generations. Their approach, courage and depth of thought represent both a motivation and an example of technical achievements for us. As this section is concerned with depicting the historical development, the more recent standards (ISO, EN, and so on) will not be dealt with here (but only in Section 2 and the subsequent sections). Special literature exists on the history of gear manufacturing and gears, such as by G. Matschoss and Count von Seherr-Thoss [1/1], [1/2]. Only a short summary based on these and other works can and should follow at this place. These include the representation by K. Kutzbach , “Grundlagen und neuere Fortschritte der Zahnraderzeugung” (Foundations and recent progress in producing gear wheels), which provides an excellent insight into the manufacturing technology then existing (1925) [1/3]. This tome offers many interesting aspects, including suggestions for current problems, albeit already having been assigned to history. It is, of course, impossible to specify the precise date of birth of the gear wheel. Structures similar to gear wheels, dating back over 4000 years, bear witness to examples constructed at the dawn of civilisation. They were probably intended to depict suns. Even today, the gear wheel possesses symbolic significance for the mechanical engineering industry as a company logo and an emblem of associations. The first traditional applications included antique instrument making . In 1901, a device dating back to the 1st century BC was discovered near a Greek island [1/2]. From the end of the 13th century, the use of gear wheels became more frequent – initially through the construction of wheel clocks . Such a mechanism is shown in Figure 1/1. This device is the clockwork of a celestial globe made by H. Diepel ; refer to [1/4]. In 230 BC, Philon described a water pumping station, thereby providing evidence for the existence of gear wheels. From a later period, the magnificent descriptions of mining machinery built by G. Bauer , known as Agricola , provided by the doctor, Mayor of Chemnitz, and mining expert have been handed down to us [1/5]. Figure 1/2 gives an impression of the way such gear wheels were used for pumping water (draining) out of mines using a treadle, while Figure 1/3 depicts the pumping of drinking water from a well by means of a drive using horses (horse-capstan drive). The material for these gear wheels was provided by wood. Lantern gear toothing and replaceable cogs were typical construction features. Figure 1/4 shows one such replaceable wooden tooth in a worn condition. 2 1 Overview, General Principles Figure 1/1 Planetary clockwork and calendar mechanism of the celestial globe by H. Diepel (ca. 1565), Royal Cabinet of Mathematical and Physical Instruments, Zwinger Museum, Dresden 1.1 Development of Toothing 3 Descriptions of water mills are known to us from the Roman architect and builder Marcus Vitruvius Pollio thanks to his 12- volume work (ca. 24 BC). Mill construction provided a strong spur to the further development of (gear wheel) gearing tech-niques at that time. The rules derived were, of course, of an empirical nature, and the representation was, in part, extremely origi-nal to our present-day perception, as shown by the sample of text (Figure 1/5) and the excerpt (Figure 1/6) taken from the book “Vollst ndige M hlenbaukunst” (Complete architecture of Mills), written by the Pro-fessor for Mathematics at the University of Frankfurt/Oder and Chief Building Director in Braunschweig, L. Chr. Sturm (1718, 1 st edition) [1/8]. Following the water mills, which were the oldest form of engine, the windmills arrived on the scene in the 9 th century. Following the demise of wind power, which came to be regarded as out of date, wind power stations have enjoyed a renaissance in the 20 th century. Today they present high demands on gear manufactu-ring. Figure 1/2 Water pump with threadle (Heinzekunst), G. Agricola , 1556 [1/5], [1/6] Figure 1/3 Water extraction system with horse-drawn Figure 1/4 Worn wooden tooth of a capstan drive; well house; Augustusburg, Saxony horse-drawn capstan (ca. 1575) 4 1 Overview, General Principles Figure 1/5 Sample of text from “Vollst ndige M hlenbaukunst” (Complete architecture of mills), L. Chr. Sturm [1/8] (To clarify: The book was published in 1718 in accordance with the usual rules at the time of spelling and grammar and written and printed with the letters usual at that time. The author, L. Chr. Storm , describes a design and implementation of the teeth. It is, according to his remarks, very beneficial. The production could be made easily by every practiced miller or carpenter. The preparation of teeth of the gear pair on a device by carving he called a “mess”.) Figure 1/6 Gear wheel pairing, L. Chr. Sturm [1/8] The achievements of Leonardo da Vinci in this field are admirably wide-ranging. Following up on his suggestions, the Dutch constructed windmills with rotatable roofs and became leaders in this field. Although Leonardo da Vinci had previously proposed an apparatus for rolling steel using a transmission, it was only much later that this proved possible to construct. Also known from him is the design for a file cutting machine driven by gears. His designs of gear pairings have been preserved, refer to Figure 1/7 ([1/9]). Gears played a key role in hoists at an extremely early stage. As early as 250 BC during the First Punic War, a large war galley is supposed to have been pulled across land to water by means of a winch, designed by Archimedes and driven by a worm drive. The representation of a pulley with gears formed as a cylindrical gear and angle drive, designed by Ramelli , has been handed down to us [1/10]. The further development of machine tool construction, particularly the invention of the leadscrew lathe by H. Maudsley (1800) also resulted in an increasing need both for gears and their greater precision. 1.1 Development of Toothing 5 Figure 1/7 Cylindrical gear pair, Leonardo da Vinci (ca. 1500, Madrid Codex) The development of engines led to a marked increase in the transmission of power and torques. In 1782, J. Watt chose the planetary gear train in order to convert the back-and-forth motion of his steam engine into rotary motion. For ship propulsion, the steam engine possessed a far-too-low rotational speed for the propeller that had been invented in the meantime. For the “Great Western” propeller-driven ocean steamer built by I.K. Brunel in 1839, it proved necessary to incorporate a gear to increase the speed gear ratio. The reverse situation arose towards the end of the 19th century when steam turbines came into use. Their far-too-high rotational speed had to be reduced by means of a reduction gear. A turbine locomotive by Krupp (developed in 1924 by R. Lorenz ), with a 2800 hp performance at 6800 min -1, is also known. A reduction gear was used to reduce its rotational speed to n = 300 to 400 min-1. The increasing use of electric motors and piston fuel engines (Otto, Diesel) ultimately required an ever wider use and development of gear drives, since the most efficient solution up to that date had only been direct operation in exceptional cases. While in 1909, a ship's turbo transmission with a power P = 6000 hp, rotational speed n = 1500 min -1 and a contact ratio i = 5 represented an absolute peak performance (built by the Pittsburgh Westinghouse Machine Co.), the peak power achieved today is in the order of P = 100000 kW and a rotational speed of n = 150000 min-1 (Table 1/1). Nevertheless, even from today's perspecti ve, many achievements remain astonishing, such as the single-stage cylindrical gear by E. Sykes (1921) with i = 63, z1 = 1, z2 = 63, which with a rotational speed of n1 = 1000 min-1 transmitted a power of P = 10 hp [1/2]. 6 1 Overview, General Principles Table 1/1 Threshold values achieved of listed gears Weight/ Power kg/kW 2.0 …0.1 1.0 … 0.2 2.5 … 0.2 3.0 … 0.7 2.0 … 0.5 4.5 … 2.0 80 … 8 10 … 4 6 … 1.5 5 … 1 4 … 1 Volume/ Power dm3/kW 0.5 … 0.2 0.4 … 0.15 0.8 … 0.4 1.0 … 0.5 0.7 … 0.3 0.6 … 0.2 20 … 2 2 … 0.5 4 … 0.5 3 … 0.4 1 … 0.25 Efficiency (stage) in % to 99.5 99.5 99 < 90 97 2) 98 98 98 94 98 from 97 98 97 50 20 2) 90 97 96 93 96 Gear ratio i extreme 1000 (3…35)1) 8 1 … 50 1 … 300 10 10 20 15 12 normal 1…800 3 … 13 1 … 5 4 … 8 5 … 50 1 … 6 1 … 6 1 … 5 1 … 10 1 … 10 Velocity / in m/s extreme 200 100 120 50 70 50 40 120 40 70 normal 80 80 40 30 25 3) 25 10 60 30 40 Max. rotational speed min-1 150000 (20000) 50000 20000 30000 3) 10000 10000 200000 6000 30000 Power P in kW extreme 150000 35000 4000 1000 3000 1000 3) 200 3000 3000 4000 (400) normal 2000 5000 500 300 1000 90 3) 20 200 150 100 100 Gear Cylindrical gear Planetary gear Bevel gear Hypoid gear Bevel-cylindrical gear Worm gear Friction gear Chain transmission Flat-belt transmission V-belt transmission Toothed-belt transmission Transmission with toothed gear Traction drive 1) High contact ratio up to 106 2) Ș falling with increasing gear ratio 3) The mass- produced gears generall y have smaller values. 1.1 Development of Toothing 7 The advent of power electronics has superceded several types of gears. In the field of machine tools, the continuously variable drive achieved by the motor, with the tendency to use the AC asynchronous motor controlled by the frequency inverter or the regulated DC motor, represents the modern solution, having replaced the hitherto-used switching gears. In most cases, the change gears have also been dispensed with in favour of the “electronic gearbox” (electronically controlled). On the other hand, however, the development of the technology has resulted in strong growth in the demand for gear drives. The motor-transmission combination is the most economical solution – particularly where it is a question of modifying high torques to a different (mostly lower) rotational speed level. The gear drive assembly will thus continue to play a key role in the future for exca-vators, haulage trucks, rolling stock, tracked vehicles, ships and turbo systems. Their correct design and further development of calculation and design methods will remain challenging tasks for engineers to pit their wits against in the future. A new level of quality of the available aids in this regard is provided by the advent of information technology. 1.1.2 Development of Gearing Geometry The science of gears emerged following or alongside the development of the principles of mechanics – including by Chr. Huygens (1633 to 1673; the terms evolute and involute), G. W. v. Leibniz (1646 to 1716; emergence of the involute), J. Bernoulli (1667 to 1748; instantaneous axis), and L. Euler and I. B. D'Alembert (1707 to 1783 or 1717 to 1783; propositions on kinematics). Ph. De La Hire (1640 to 1718; 1695: “Traite de Mecanique”) [1/2] is regarded as the founder of the scientific theory of gears . He was the first person to develop a gear with a constant gear ratio according to a set of rules. He specified the way to determine the mating flank for a given tooth flank. He had also previously written about producibility using roller cams. The roller cam was used by Abbe Ch.-E.-L. Camus (1699 to 1768) to generate a paired tooth system. He also discovered sliding in tooth flanks as the cause of friction and wear, The involute as a form of tooth was first considered by L. Euler (from 1752). He proved that the rolling motion of tooth flanks is a superposition of sliding and rolling motions, derived rules for applying the toothing in practice, developed drawing techniques and provided an equation for determining the distance of the centre of curvature of the curve generated in rolling from the pitch point. This equation is known in kinematics as the Euler-Savary Equation (refer to Section 2.3.2.2) according L. Euler and F. Savary , who also derived it in 1845. In spite of these remarkable achievements, the practical application of gears still remained long submerged in the depths of empirics . It was first necessary to raise this existing theoretical treasure. Along with other great designers, R. Willis (1800 to 1875) and K. Kutzbach (1875 to 1942) made outstanding contributions in this field. The necessary use of “metal” as a material in the newly developed engines, such as steam engines, steam turbines, piston fuel engines, and electric motors, made improvement in toothing geometry inevitable. The feeding in of the stronger workpiece in the milling process to the extent hitherto known was no longer possible. It was towards the end of the 19th century that the involute teeth prevailed. The decisive factor in this regard was the technical benefits to be gained, such as insensitivity to centre distance alteration and the profile displacement possibilities. 8 1 Overview, General Principles The scientists who realised these advantages at an early stage included R. Willis 1), J. L. Wei bach 2), F. Redtenbacher 3), C. v. Bach 4), G. B. Grant 5), and J. R. Brown 6) (refer to Table 1/2). As early as 1851 [1/27], for example, J. Wei bach held the view that “the involute tooth is in any case the most consummate of all gear constructions”. M. F lmer (1873 to 1941) provided decisive impulses to gear calculation theory . He laid the foundation for the widespread theory, which particularly owed its development to K. Kutzbach , Professor at Dresden University of Technology. In the USA, it was Professor E. Buckingham (born in 1887) who was responsible for the dis- semination of modern calculation methods. He introduced the notation inv Į (inv Į = tan Į - Į). Kutzbach summed up the situation of gear manufacturing existing in 1925 in his “Grundlagen und neuere Fortschritte der Zahnraderzeugung” (Principles and recent progress in gear generation) [1/3]. The historical development of the rolling process is also presented in another work by Kutzbach . Old problems of gear manufacture were shown in publications, including those by H. Fischer “Das Erzeugen der Zahnform f r R der” (Generating the tooth form for gears, see VDI Magazine 1898, pages 11 to 16) and T. Ritterhaus (VDI Magazine 1898, pages 165 to 166). Kutzbach provided clarification in the notations and designed a calculation rule for profile displaced tooth systems (DIN 870, 1931). In doing so, the difference between the sum of profile shift ( x 1 + x 2) m and the centre distance alteration ( a - a d) was calculated by means of an approximation relationship. Additional information on the development of toothing geometry is depicted in Table 1/2. The German Standards Committee, founded in 1917, played a decisive role in its subsequent work by promoting the required standardisation, dissemination and application of new methods. In this regard, mention should be made of DIN 780 (module series; 1923), DIN 867 (Tooth form for cylindrical gears and bevel gears, basic rack, Į = 20 , 1927; refer to Figure 1/8), DIN 868 (Gears, terms, notations, abbreviations; 1928) and DIN 870 (Profile displacement with involute gearing; 1931). This, however, by no means meant that the development was completed. M. Bergstr sser revised the involute geometry and in 1952 urged against continuing the use of the approximation equations in accordance with the DIN 870 standard. He specified exact equations based on the involute functions. Subsequently, the direct relationships between the service pressure angle and the sum of the profile displacement factors triumphed completely (refer to Sections 2.1.2 and 2.2.3). The use of profile displacement became increasingly common in positively influencing the stresses. In order to avoid leaving this investigation to the designers, who were occupied with “a large number” of things, guidelines both for the selection of the sum of profile displacement factors and their distribution were created – finding its expression in DIN 3992 (Profile displacement of externally toothed cylindrical gears). Due to the then newly created principles of a more precise calculation of load-carrying capacity of gears, the time had become ripe for reviewing and possibly revising these guidelines. A major problem was also presented by the determination of the permissible geometric deviations (the tolerance system). After internal regulations had already existed in a few companies, systematic investigations began in 1936 headed by Professor O. Kienzle (at the Technical University, Berlin). Professor G. Berndt (University of Technology, Dresden) and Dr. B rger (Physical Technical Institute of the Reich, Berlin - Charlottenburg) were also in volved with the problem. 1) Willis; 1800 to 1875; Professor at Cambridge 2) Wei bach; 1806 to 1871; Professor for Mathematics and Mechanics at the Mining Academy in Freiberg 3) Redtenbacher; 1809 to 1863; Professor in Karlsruhe 4) v. Bach; 1847 to 1931; Professor in Stuttgart 5) Grant; 1849 to 1917; USA, founder of the American gear machine industry 6) Brown; 1810 to 1876; USA, tool manufacturer 1.1 Development of Toothing 9 Table 1/2 Developments in Gear Geometry from the Beginning of the 19th Century through to the Start of Work on Standards in Germany (A Selection) Buchanan, Robert (English engineer; 1770 to 1816) • Further developed the teachings of Camus for the mechanical engineering industry; formulated a precursor to the Law of Gears (1808) White, James (English designer) • Invented the double helical and herringbone gear • Patent application for the helical gear (1808) Remark: White obtained the idea for the helical gear from step-like (half-pitch division only) mutually offset teeth halves Hawkins, John Isaac (English doctor and engineer; 1772 to 1865) • Recognised (1837) the insensitivity of involute teeth to centre distance alteration • Multiple meshing and thus distribution of load • Benefits of involute teeth in the frictional behaviour compared to epicycloid teeth Willis, Robert (English scientist; Professor in Cambridge; 1800 to 1875) • Practicable gear science (1841); basic definitions (e.g., diametral pitch, circular pitch, **backlash**, pressure angle) • Set of wheels • Constant pressure angle of the set of wheels (proposed 15 ) • Predicted the advantages of the involute teeth Saxton, Joseph (USA); born ca. 1840 • Invented first gear hobber based on the rolling process Reuleaux, Franz (German scientist/kinematician; Pro- fessor in Zurich and Berlin; 1829 to 1905) • Method of constructing the mating flank for a given flank based on the laws of motion (1865); • More exact definition of “line of action” • Favours involute teeth (1862) Sang, Edward (Scottish scientist; 1805 to 1891) • “A tooth rack is a gear with an infinite radius” (1852) Wiebe, Carl Hermann (Professor in Berlin; 1818 to 1881) • Precise formulation of the fundamental law of gears (involute teeth): normals, contact points, tangents, direction of force (1861) Pl cker, Julius (German Professor of mathematics and physics in Bonn; 1801 to 1868) • Founded the theory of screws (theory of helical gears, worm wheel gears and spatial gears) (1868) Saalsch tz, Luis (German university lecturer on mathe- matics and mechanics in K nigsberg) • Published the first German-language specialist work “On the Theory of Involute Teeth” K nigsberg (1870); law of gears and characteristics of involute teeth Pfauter, Hermann (German engineer; 1854 to 1914) • Invention of the general screw rolling process (Chemnitz; 1897) Schiebel, Adalbert (Professor for machine elements in Prague; 1872 to 1931) • Definition of axial overlap in helical gears (overlap ratio) (1913) • Detailed information on tooth friction loss (1913) • Value table for involute function (1922) Lasche, Oscar (German engineer; 1868 to 1923 • Corrected gear toothing by changing the tooth depth in association with constant distribution of the reference circle pitch on the tooth thicknesses (1899) (subsequent toothing by AEG Company) Maag, Max (Swiss engineer; 1883 to 1960) • Combined application of profile shift and pressure angle change (1908) • First (or one of the first) application of the involute function (1908) F lmer, Max (German engineer; 1873 to 1941) • Further implementation of the involute function • Wide application of profile displacement in association with constant tool (Hobs. Terms: V-gear), V-transmission ... (1919)) • Clarification of the issues to prevent undercut (1919) 10 1 Overview, General Principles As early as 1942, a tolerance system had already existed on an independent basis for gears used in precision engineering, which was also extended to the mechanical engineering industry. There then followed in 1952 the publication of the DIN 3962 standard (Tolerances for cylindrical gearing based on DIN 867, permissible individual errors) and in 1953 DIN 3963 (Tolerances for cylindrical gearing based on DIN 867, permissible tooth alignment and errors, permissible group errors, tooth thickness quantity). This work on standards is particularly associated with the name of A. Budnick [1/13]. In the further standards, such as DIN 3961 (1978) and the Council for Mutual Assistance (CMEA) (ST RGW 641; 1977), a fundamental distinction was made between quality and pass systems [1/20]. The quality characteristics are assigned to functional groups (for ST RGW 641: “kinematic precision”, “running uniformity”, “flank contact”). A leading role in these works in the CMEA was played by B. A. Taiz (Moscow). Valuable input was also provided by H. Weinhold (Magdeburg) and W. Krause (Dresden). The continually ongoing work for ISO (and subsequently for DIN) on gear tooth tolerances, load-carrying capacities and other recommendations include more precise and additional information, the clarification of the scope, a correction of various numerical values and a finer gradation. Even today, involute teeth are used with non-standard basic racks . This was initially justified by the tools still available. The 15 toothing and 14.5 toothing , in particular, thus continued to exist for the time being. At the present time, basic racks with Į 20 are used in favour of greater load-carrying capacity and reduced losses (e.g., 28 ) or lower noise, particularly for extra depth gearing – including an enlarged teeth depth and Į = 15 or 17.5 . The aim of using profiles with Į < 20 is to achieve transverse contact ratios İ Į 2 with less gear noise, such as in vehicles. Basic racks with Į > 20 are also still to be found in use today in situations where the tooth root and tooth flank stress are the key factors. Substantially fewer losses are achievable by means of smaller tooth depths for Į >> 20 – particularly with helical gears. Further increases in load- carrying capacity are possible using gears with unsymmetrical profiles. Finally, protuberance tools, used to avoid grinding steps, create final contours deviating from the normal basic rack (DIN 867) in the tooth root area (residual undercut, tooth root depth enlargement) and provide the greatest tooth root load-carrying capacity with the recently applied elliptical tooth root transition curves and shot peening. The efforts undertaken to determine the gearing data as optimally as possible have led to the development of the various special involute gearings and recommendations on profile displacement. Attempts are being made to achieve clarity in the large range of possibilities. Mention should be made here of the historically important works of H. Winter , who investigated key gearing systems in the 1950s [1/11]. Although the efforts made to use non-involute gears in attempts to outperform the load-carrying capacities hitherto achieved did not prove successful, nevertheless the profile modifications do to some extent result in a significant influence on the load-carrying capacity and actually already represent non-involute gears – albeit with relatively minor deviations (lying within the tooth deformation scaling) to the involute. The internal toothing is not usually included in the considerations. With the introduction of a nega-tive number of teeth, however, practically all the equations for the external toothing automatically merge into the equations hitherto used for the internal toothing (DIN 3993, Geometrical design of cylindrical internal gear pairs). The special characteristics of internal gears, however, include the various interference phenomena, which particularly occur in association with small differences in the number of teeth. In summary, the investigations of G. Schreier [4] and (from a later period) of DIN 3993 are available. The more recent mathematically based investigations into the spatial toothing geometry include those by F. L. Litvin [13]. 1.1 Development of Toothing 11 For the purpose of quickly developing the technology in a targeted manner on a legal basis, the former East Germany published (starting about 1949) its own standards under the TGL symbol (until 1956 the German abbreviation for “technical standards, quality regulations and delivery”, and then used as a general symbol for standards). Figure 1/8 Shape of tooth for cylindrical gears and bevel gears in accordance with the original version of the 1927 DIN 867 standard (reproduced by courtesy of DIN) In 1962, the XVIth RGW Congress (the former Council for Mutual Economic Assistance (CMEA) of the countries belonging to the sphere of influence of the Soviet Union) decided to form a Standing Committee of the Council for Standardisation (SKS) based in East Berlin as well as an Institute of the Council for Standardisation in Moscow. These bodies also revised and unified practically all the gearing standards for the CMEA (German: RGW) area, which were effective until about 1990. The symbol for these standards was ST RGW and for the East German standards based on these TGL RGW . 12 1 Overview, General Principles In 1988, East Germany became a member of ISO, an organisation founded in 1946 and emerged from the ISA (itself established in 1926), and also began work in the TC 60 department of the ISO (Technical Committee; gears/gear drives). With the turning back to DIN from 1989, both historical development and economic necessity were complied with. For an overview of standards, refer to the compilation included as Appendix 17.1 at the end of this book. Non-involute gears are only still used today in exceptional circumstances. These include cycloid gears and their special forms (cyclo gears, VBB gears [2]), the circular arc gear teeth (clocks, watches), the cylindrical lantern tooth system (large slewing gears, hoists) and also, formed from circular cones, the Wildhaber-Novikov gear (used in gear manufacturing on a large scale in the USSR) and the Sym-Marc gear (used in Japan) [2]. 1.1.3 Development of the Calculation of Load-Carrying Capacity 1.1.3.1 Tooth Root Stress In former times, “recipes” existed. They were the resu lt of experience and generally led to viable constructions, provided that the materials and both the rotational speed and torque ranges were adhered to. One example is R. Buchanan (refer to [1/2], page 262), who as early as 1808 specified the rules for watermill gears: “You make the teeth as wide in inches as the number of horse-power required to offer resistance” . One of the first mathematical rules for determining the tooth proportions based on flexural stress comes from the Leiden naturalist P. van Muschenbroek (1692 to 1761). In 1729, he established an equation for the relative “strength” of the material: 2BHQ = m L (1/1) Q load H thickness L length m material constant B width Very early on, the English engineer T. Tredgold assumed that a key effect was made by the load distribution across the face width. In 1822, he published a relationship for the chordal thickness, which he derived from the assumption that the entire tooth force acts on the outermost tooth edge as a result of deviations or an external body (Figure 1/9). He derived a relationship that proved particularly durable for cast-iron gears: 3 4Hd = / (1/2) d chordal thickness (in inches) / tangential velocity at reference circle (feet/second) H power (horse-power) 1.1 Development of Toothing 13 Similar assumptions were also later made by C. v. Bach (Figure 1/10, [1/30]) and F. Grashof (text sample in Figure 1/11 [1/29]). Tredgold set a third of the chordal thickness as the wear limit. Figure 1/9 Tooth loading made by Tredgold (1822), in accordance with [1/2] Figure 1/10 Tooth loading in analogy to Tredgold made by C. v. Bach [1/30] F. Grashof Uebrigens kann sich bei ungenauer Lagerung der Wellen, mangelhafter Ausf hrung der R der oder beim Dazwischenkommen eines kleinen K rpers der noch ung nstigere Fall ereignen, dass der Druck P sich an einer Zahnecke concentriert und dieselbe abzubrechen droht, am wahrscheinlichsten in einer Bruchfl che, welche unter 45 gegen die Stirnfl che des Zahnes geneigt ist. Diesem Fall entspricht die Maximalspannung Figure 1/11 Grashof on the assumption of the force application (text sample from [1/29]) (Incidentally, imprecise shaft bearings, faulty construction of the gears or the intervention of a small body may result in the still less favourable case that the pressure P will concentrate at an edge of a tooth and threaten to break it off, most probably on a fracture face inclined to the abutting face of the tooth at an angle less than 45 . The maximum tension corresponds to this case.) In 1869, F. Releaux formulated for the endurance limit (in contemporary notation) 3 zulKσυ= (1/3) )zul permitted tension, K constant, / tangential velocity (at reference circle) He was also the first to recognise the usefulness of calculating with a width ratio of b/t (b face width, t tooth pitch). The practical calculation was fundamentally influenced by C. v. Bach , who was ennobled for his services to mechanical engineering. Although initially taking the same approach as Tredgold , he finally provided the relationship in the form known today: t b k = P (1/4) P tooth load t tooth pitch k load factor b face width In order to take into account the wear of the teeth, Bach recommended the same Equation (1/4) for the wear, established before him by W. A. H. v. Kankelwitz [1/16]. Bach actually only specified the Equation (1/4) for cast gears and recommended values (wear) independent of rotational speed for the load factor k. 14 1 Overview, General Principles The practical-minded Bach found the simplicity of Equation (1/4) and the inclusion of two different types of damage (tooth breakage, wear) pleasing, but the concealment of the physical content involved proved to be an obstacle to further development. Once O. Lasche – stimulated by the increasing demands of the electric motor industry – had added a case of load application to the tooth tip for uniform load distribution to the case of the unfavourable face bearing (according to Tredgold ) to enable precise statements (1899), Karl Kutzbach continued the development of the calculation. In addition to force application on the tooth tip and uniform face bearing, he also assumed a sufficient tooth root curvature (Figure 1/12). Figure 1/12 Tooth loading ( Kutzbach , H tte, 24th edition, 1923) Figure 1/13 Tooth root stress calculation after Lewis (1892); parabola of equal strength [1/2] He also took inertia forces into account using endurance limit coefficients (1923). He also generally recommended b = 2t (b face width, t tooth pitch). M. ten Bosch of the Swiss Technical University (ETH) in Zurich drew attention to the fact that due to the transverse contact ratio, the full tooth forces were not engaged at the tooth tip. Nevertheless, for a long time he continued to recommend the calculation with the tip engagement for safety reasons due to the occurrence of component and profile errors. He did, however, take into account the bending and compressive stress [1/28]. The American C. Lewis had a lasting influence on the tooth root load-bearing capacity calculation. In 1892, he determined the tooth root cross section for the calcu-lation of the tooth root stress by placing a parabola of equal strength (known since 1638 thanks to Galileo Galilei ) in the tooth (Figure 1/13). Since he assumed physical considerations, his method was accepted as convincing and enjoyed widespread use for a long time. The English engineer H. E. Merrit also pointed out that due to the transverse contact ratio the full tooth forces were not engaged at the tooth tip and determined the external single engagement point as the contact point in accordance with the current calculation. H. Hofer [1/17] substituted the determination of the tooth root cross section using the Lewis parabola by a 30 tangent to the tooth root curve (Figure 1/14). Although this approach deviated from purely physical considerations, this assumption was demonstrated to still be reliable within the framework of the other conjectures and enabled the evaluation of the fundamen-tal influences. From this, the conclusion followed that a significant increase in load-bearing capacity is possible in regard to tooth root stress by means of profile displacement, and therefore only profile-displaced gears should be used for z = 10 to 30. This, however, was subsequently demonstrated to be inaccurate to the extent hoped for in relation to fatigue stress. 1.1 Development of Toothing 15 As early as 1925, the Swiss engineer V. Baud was the first to determine the notch effect photo-elastically. His fin- dings were evaluated in 1926 by St. Timoschenko . He introduced a rela- tive stress concentration factor. This would be written in the notation of today as Srel S1.6 = YY (1/5) Fn S Fn0.151 = + sYN (1/6) Although results on the notch effect had existed since 1925 and pointed to a significant influence, and the 1946 American standard on the calculation of load-carrying capacity had included the notch effect (Dolan and Brog-hamer), the notch effect change remained ignored in Germany and many other countries for a long time. As the notch effect is dependent on the shear component, it also proved possi-ble to observe a lever arm dependency of the nominal stress tolerated – particularly in pulsator tests. Figure 1/14 Tooth root stress determination Hofer (1942), 30 - tangent on the tooth root curve [1/17] G. Niemann and H. Glaubitz tried to take that into account by considering an equivalent stress as a result of bending, compressive and (mean) shear stress and introduced the coefficient for shear stress with a = 2.5; see Figure 1/15 and Equation (1/7) [1/18]. This was intended to make the allowable stress independent of the lever arm. They established the relationship () ( ) 2 2 vb d = + Ĳ ıı ı a − with a = 2.5 (1/7) Long after the notch stress theory had been solidly established for the calculation of shafts and rod-like structures, attention was also refocused on this problem for gears. As a result of taking measurements using a strain gauge, M. Hirt determined the stress concentration factors independently of the root fillet and the lever arm [1/25], which formed the basis of the ISO and DIN design equations (ISO 6336; DIN 3990). The RGW standard ST RGW 5744-86 and the updated version of TGL 10545 (1989) were based on calculations performed at the Dresden University of Technology – these, in turn, being based on the singularity method (example in Figure 1/16). The CMEA created its own calculation basis (ST RGW 5744-86), building on ISO standards. 16 1 Overview, General Principles Figure 1/15 Tooth root stress (Niemann and Glaubitz ) In this regard, the names Reschetow , Kudrjawzew , Sablonski , Filipowitsch (former USSR), Svoboda (former CSSR), Erney (Hungary) and Arnaudow (Bulgaria) particularly stand out. East Germany (Dresden University of Technology) was respon-sible for the processing. The ST RGW 5744-86 also formed the basis of the new load-carrying capacity standard TGL 10545 (1989) of the former East Germany, also worked out by the TU Dresden. These values are somewhat lower than those laid down in DIN and ISO [1/20, 21] (by an order of 10%). There then ensued investigations into gears with internally toothed grinding steps [1/22, 1/23]. Figure 1/16 Stress distribution in the tooth root curve with and without grinding, calculated in accordance with the singularity method [1/20, 1/21]; YS = YSO YSrel 1.1.3.2 Tooth Flank Stress – Pitting The derivation made by H. Hertz (1881) forms the basis for the calculation of the pitting resistance. A. F ppl recognised the significance of this theory, developed it further, presented it to designers and republished it in 1897. The theory was further developed by his son, L. F ppl (1936), who calculated the stresses in the interior of the bodies and thus played a key role in expanding the knowledge gained concerning the stress situation of the contact of cylindrical bodies. 1.1 Development of Toothing 17 Occasioned by missing values for the permissible load capacity in relation to roller bearings, R. Stribeck performed extensive tests involving point and line contacts (1898-1899). He confirmed and expanded the application of the Hertzian Theory, obtaining limiting loads that were usable in practice. For line contact in particular, he obtained the relationship 1 P = k DL (1/8) P load L roller width (length) D roller diameter k1 permissible specific load (deformation limit; stress value) Thanks to R. Stribeck , the fundamental investigations on the Hertzian contact are regarded as completed. Credit is due to C. v. Bach , who significantly advanced the dissemination of the Hertzian Equations by including them in his “H tte” engineering handbook in 1908. Application of the insights into gears gained by H. Hertz , A. F ppl , L. F ppl and R. Stribeck ensued relatively late. E. Videcky (1908, Budapest) was the first person to link the Hertzian Equations with gears. For the wear problem, he investigated tooth deformation and the Hertzian contact stress along the line of action. For the earlier requirements, the equation derived by Bach (1/4) proved to be well suited for cast iron. In order to apply a tried and tested relationship for steel in view of the increasing demands, K. Wissmann fell back on the Hertzian Equation by taking the evaluation of the wear as correct. He considered the single pair mesh factor as applicable for the evaluation of the flank stress, and in 1908 specified the equation 2 N 12 2 11 22 1 10.175 + + EE F = pbE E ⋅ NN (1/9) p Hertzian contact stress E modulus of elasticity FN normal tooth force N radius of curvature (profile) b face width In the initial applications, however, the contact stress was normally used in the form characterised by Stribeck (Stribeck contact stress). These include the relations obtained by Buckingham (1920), Kutzbach (1926), and Niemann (1938) (refer to [1/26] as well as A. K. Thomas [12], H. Trier [11], G. Schreier [4]). The Stribeck contact stress k for gears is N m Fk = bN (1/10) with 12 122 m=+NNNNN (1/11) N radius of curvature (profile) FN normal tooth load b face width The relationship for the Hertzian contact stress p is given by 22.86 pk = E (1/12) E modulus of elasticity Based on the idea of the wear as a function of the service life (in hours), Niemann provided the permissible value ( kperm. = k5000 Q; Q depending on the number of operating hours) [1/26]. 18 1 Overview, General Principles From today's perspective, we make a clear distinction between wear, which – as a consequence of the improvement in lubrication and increase in hardness of the materials used – is only critical in exceptional cases, and the pitting load capacity. Research efforts in this field are concentrated on determining other influences. These include waviness and roughness, the shear load and tempe-rature stress in association with the hydroelastic lubrication condition (shear), as well as the bending overlapping the flank stress above the area of the tooth root. In general, the lubricant film only causes low deviations from the value obtained by the Hertzian contact stress. The overlapping shear and the stress incurred from the change in temperature due to local heating cause other effects, which have not been satisfactorily determined at the present time. The waviness and roughness are also key factors, which must be taken into account on a statistical basis following future developments. Relationships to the Hertzian contact stress, supplemented by shear force, are provided in Section 6.5.1.1. Statements concerning the elastohydrodynamic effects are to be found in Section 6.5.4.1. 1.1.3.3 Scuffing Load The increase in the power transmitted per spatial unit was also associated with a growth in the size of losses and the heating of the gears . The aviation industry, in particular, has provided new requirements and cases of damage. As scuffing has now been identified more frequently, efforts were undertaken to take into account this phenomenon in the calculation. H. Hofer (ZF Friedrichs- hafen) places the band resulting from (d b) in relation to the power transmitted or lost. This was considered as a heat accumulation value (or as a variable for scuffing safety) (1926): 1 a = 20 zm bSN (1/13) z1 number of teeth (pinion) b face width in mm m module in mm N power in hp This did not yet provide the link to the work to determine wear that had been carried out considerably earlier, including by Poncelet (1826), Wei bach (1852), Bach (1881), Stribeck (1894) and Schiebel (1911). These persons had derived the mechanical work due to friction more precisely, including as a function of (1/ z 1 + 1/ z2) or the pitch circle arcs. The equation subsequently specified (1941) for the heat accumulation value Sa then had the form 21n a = 1 10 ≥zmb m mSVN hp (1/14) with the degree of loss 21 1214.1Vzz zz = , (-) for internal gears; here 21zz≥ (1/15) mn normal module b face width z1 number of teeth (pinion) N transmitted power The American engineer J. O. Almen (1935) developed a new calculation formula derived from numerous cases of scuffing damage and preservation of operating arcuate-cut bevel gear rear-axis drives. He discovered that the product of the Hertzian contact stress and the sliding velocity is crucial. Later, in 1943, he added to this product the greatest distance of the contact point on the line of action from the pitch point (termed the PVT formula; refer to [6]). Aa g a a=Fp gυ (1/16) 1.2 Functions and Classification of Gears and Teeth 19 AA G r e n z FF≤ (1/17) pa Hertzian contact stress on the tooth tip /ga sliding velocity on tooth tip ga distance of point of tip contact from the pitch point; partial contact length The parameter FA was calculated at the head of the pinion and the gear. The formulation derived by Almen particularly proved its worth with small aircraft transmissions [6]. In 1937, the Dutch engineer H. Blok established an equation that is still in use today [1/24]. He assumed a mass temperature and a locally occurring increase in temperature at the contact point, the temperature flash, from the flash temperature hypothesis. Working on a theoretical basis, he obtained an equation which expresses the size of the temperature flash t B on the assumption of the lubrication layer failing (refer to [6] and Section 6.5.4.2): () ()tT T 2 1 BB max T1 T2 H 12ȝ | | = 0.83 + Ftt bff b−≤// //1) (1/18) ȝ friction coefficient b face width Ft tangential force f ȜȖc /T tangential velocity Ȝ thermal conductivity of tooth flank Ȗ specific weight vertical to line of c specific heat action ( /T1 - /T2 = /g) bH half Hertzian contact deformation width /g sliding velocity The limiting value tBmax is dependent on the combination of material and lubricant used. Since the relationship derived by Blok possesses the incontestable advantage of being based on theoretical derivations, it is still firmly embedded in the process of gear calculations. Although the effect caused by the hydroelastic condition on t BGrenz has since been determined, the Blok hypothesis has remained as the basis of the calculation of scuffing load capacity. In Section 6.5.4.2, the calculation method used, as well as other developments (integral temperature method), is explained in detail. 1.2 Functions and Classification of Gears and Teeth Following the historical overview provided in the previous section, the focus (in overview form) in this section is on the functions and classifications of gears and teeth. Gears are a component of the drive . They have been – and will in the future remain – necessary, because the motor alone cannot meet all of the requirements and characteristics. As far as present-day requirements are concerned, a gear can no longer be considered in isolation, since the deformation and vibrational behaviour of the surrounding components exert a significant influence on the load-bearing capaci ty. In view of the additional loads caused by forced and parameter-regulated vibrations as well as the non-linearity usually present, it is necessary to consider the entire system – at least initially. For this reason, we start with the functions of the gears in the drive and only subsequently the classification of the gears and teeth. The functions of the gears consist of 1) The factor 0.83 in Equation (1/18) was later changed to 1.11. This therefore better corresponded to the elliptical distribution of the pressure than the parabolic distribution, while taking into account the fact that – instead of only half of the Hertzian contact deformation width bH – the entire Hertzian contact deformation width 2 bH was used under the root in the denominator. 20 1 Overview, General Principles • changing kinematic-dynamic variables, e.g., rotati onal speed, sense of rotation, torque, • changing the position of the output, e.g., centre distance, angular position between drive and output (geometrical adjustment), • changing the characteristics or generating speci fic characteristics, e.g., stepped adjustment of the motor characteristics to the hyperbola. From an abstract perspective , a gear consists of – at a minimum • a driving element (driving shaft), • an output element (output shaft), and • a frame (housing). With these three basic elements, it is possible to externally conceive the introduction of the three momenta to the drive (refer to Figure 1/37). Given 3 ja na bG j=1 = + + = 0MM M M (1/19) an = in(put) ab = out(put) then due to the box torque M G, it is necessary to achieve a change in torque |M an| |Mab|. From a mechanical perspective, this distinguishes a drive from a clutch, where no frame moment MG is present. Thus for a clutch the following always applies: Man = –Mab (refer to Section 1.4.3). The classification of gears can be undertaken from various perspectives. One very generalised and well-characterising distinction is between • non-uniform transmission gears (i constant), and • uniform transmission gears (i = constant). Figure 1/17 depicts an example of a non-uniform transmission gear drive. Vibrations occur as a result of i constant rotation angle accelerations existing at the output in association with the constant motion of the drive. This phenomenon is taken into account with some drives. For the drive in Figure 1/17a, the mean gear ratio is given by | iM| = 1. The instantaneous gear ratio i is (refer to Section 1.4.1) 12 2 1ȦȦi = = r r − or () () () 1 21 1 c o s + 1 ĳ 2 2i = a / b + - b a ′′ ′ ′ − (1/20) In addition to the aforementioned classification of gears from kinematic perspectives (non-uniform and uniform transmission gears), other criteria are also used for the differentiation, such as • design (pedestal gears, planetary gears), • intended use (vehicle gears, turbo gears, crane gears), • possibilities of changing the gear ratio (switchable and non-switchable gears), • physical action mechanism (mechanical gears, hydraulic gears), • gear body form (cylindrical gears, bevel gears, worm gears). 1.2 Functions and Classification of Gears and Teeth 21 Figure 1/17 Non-uniform transmission gear pairs a) Elliptical gear pair [1/14] b) Eccentrically mounted spur gear paired with an elliptical spur gear (Dresden TU Model Collection of Transmission Technology), approximation of the elliptical pairing As far as the classification of the gears or wheel pairings is concerned, a classification can ensue based on the motion conditions at the gears and between rolling gears and cross-helical gears (termed cross-helical gear units according to DIN 868). In contrast to rolling gears, cross-helical gears feature a sliding along the line of contact (pitch line, instantaneous axis of the paired rolling bodies). Cross-helical gears operate on non-intersecting, non-parallel axes. If they meet in a line, then their basic bodies (rolling bodies) are in the form of a hyperboloid. The hyperboloid is created by a “generator” (straight line) rotating around the axis of the rotating central body in question. The cylindrical gear and the bevel gear represent special cases. The generator is the pitch line and the instantaneous axis of the paired hyperboloid, which make contact along this line (Figure 1/19). This results in various types of gears depending on the size of the axis offset (Figure 1/20). For cylindrical gears, the distinction made on the basis of the tooth trace configuration for the (still in use) types lead only to straight, helical and double-helical gears (Figure 1/21a, b and c). Older forms that are no longer used include the “genuine” herringbone gear (Figure 1/21d) and the double herringbone gear (Figure 1/21f, [1/3], [1/14]). In contrast to bevel gears, arcuate teeth are rarely used with cylindrical gears (Figure 1/21e, [1/3]). For differentiation by profile form , the following fundamental types may be named: • cycloid gears • cylindrical lantern gears • Wildhaber-Novikov gears • involute gears In general, involute teeth are used. They are described in detail in Section 2. The tooth profile of cycloid gears is formed by hypocycloids and epicycloids [2], [5], [14]. Their advantages include the high degree of load-carrying capacity (pitting) and the fact that smaller numbers of teeth are more readily realisable than in involute straight gears. Against this, however, are the significant disadvantages that tools with cycloid gears are not straight-flanked – which makes them expensive – and cycloid gears are extremely sen sitive to centre di stance deviations. For this reason, the generating circle must match the working pitch circle. Cycloid gears are currently only used in exceptional cases. 22 1 Overview, General Principles Figure 1/18 Development of the basic blanks as hyperboloids by an imaginary straight line (“generator”) rotating around the gear axis a) special case: cylinder; b) special case: cone; c) hyperboloid (general case) Figure 1/19 Pairing of wheels with crossing axes (hyperboloid gears) . The gears known that are classified by the blank form are developed for specific limiting cases of the angle of intersection Ȉ and centre distance a (limiting case Ȉ = 0: cylindrical gear pairing; limiting case a = 0: bevel gear pair; limiting case Ȉ > 0 , a 0: axis-offset bevel gears or hypoid gears, approximation of the mapped pairings B 1, B2) The cylindrical lantern gear represents a degenerate cycloid gear ( point gear extended by pins). This type of gear is predominately found in large slewing gears used in conveying technology. Due to the wear and the pinion tooth form usually being approximated as an involute, only small peripheral speeds are considered with cylindrical lantern gears [2]. The efforts to develop gears with a high degree of flank load-carrying capacity (pitting) has led to profile pairings with the concave-convex curvature (in analogy to the internally-externally toothed gearing). 1.2 Functions and Classification of Gears and Teeth 23 Figure 1/20 Gear body forms or transmission types for intersecting axes; transmission type dependent on axis offset Figure 1/21 Differentiation of cylindrical gears or cylindrical gear transmissions by tooth trace configuration I: Currently commonly used gears II: Older, currently little or no longer used gears a) spur gearing; d) herringbone gearing; b) helical gearing; e) arcuate gearing; c) double-helical gearing; f) double herringbone gearing [1/3] In recent times, the circular arc gearing after Wildhaber and Novikov has become important (refer to Figure 1/22). In this gear, the circular arc profile can lie in the normal section ( Wildhaber) and the transverse section ( Novikov ). The former version is more favourable from a manufacturing per- spective. The tool, (say, a hob), can then produce gears with any helix angle desired. The 24 1 Overview, General Principles Wildhaber-Novikov gear possesses a line of action running parallel to the axis of the gear at a distance l. Only a point contact exists between the tooth flanks when there is no load. When load is applied, a length of contact is established, whose position on the path of the thread changes with the rotation of the wheel. An overlap ratio İ ȕ > 1 must be present. Figure 1/21g) Double herringbone gear (suspension railway in Dresden Loschwitz, former drive, photo: K. Ketschau) Figure 1/22 Non-involute gears; a) cycloid gear 1.2 Functions and Classification of Gears and Teeth 25 Figure 1/22 b) Cylindrical lantern gear Figure 1/22 c) Wildhaber-Novikov gear The advantage provided by the Wildhaber-Novikov gear in comparison with the involute gear is the 1.5- to 3-fold greater tooth flank load-carrying capacity (pitting). The major disadvantages include the mostly lower tooth root load-carrying capacity in comparison to the involute gear, the sensitivity in association with centre distance alteration (noise, vibrations), the greater costs involved in its manufacture (tool costs) as well as the limitation usually required for heat-treated gears (inflow; hitherto no grinding device usually available). Novikov gear drives were used in the former Soviet Union on an industrial scale. As compared to the case-hardened, ground involute gears, however, they do not possess any better characteristics, so further dissemination of these special gears is not anticipated. Terms for gear drives and gear pairings are standardised in the DIN 868 and DIN 3998 standards. 26 1 Overview, General Principles 1.3 Law of Gears for Cylindrical Gearing For gears, the demand practically always consists of ensuring a constant output rotational speed in association with a constant input rotational speed; that is, the gear ratio i should not periodically change. If the centre distance a and gear ratio i of the gear pairing are provided, the pitch circles are fixed with their radii r w1,2: w11 = ( - )ari (1/21) w2 w1ra r=− (1/22) w2 w1 = rir− (1/23) Note: i < 0 for external gear pairings (external gear – external gear) i > 0 for internal gear pairings (internal gear – external gear) A reversal of the direction of rotation is expressed by i < 0. For internal gears, rw and z are negative, as is the corresponding centre distance a. However, for the pitch circles to be present in the ratio desired, the gearing must satisfy the Law of Gears . If rolling circles are selected to generate the tooth profile (cycloid gears), then tooth profiles are automatically created, which ensure the transmission of motion with i = constant, when J 1. The following text is primarily designed to provide a general ans-wer to the question under which conditions the gear ratio is constant for given tooth profile forms (i.e., no periodic fluctuation in the rota-tion). From this condition – known as the Law of Gears – it should be possible to derive how to determine the counter-profile for a given tooth profile to satisfy the condition i = constant for a given r w1,2. We will take Figure 1/23 as the starting point to derive the law for cylindrical gears. Two tooth flanks meet at the point P y. The flank 1 rotates with 71 around the centre point 01 and flank 2 with 7 2 around 0 2. At the point of contact, the tooth flanks possess the velocity ȣy1 or ȣy2: Figure 1/23 Velocities at paired tooth flanks 1.3 Law of gears for cylindrical gearing 27 y1 y1 y2 y2 12 , ȦȦrr== −// (1/24) In order to prevent any flank separation, both flanks must possess an equal velocity ȣn in the direc- tion of the normals NN. With ȣy1 or ȣ y2, this includes the angle Įy1 or Į y2: nn y1 y 2 yy12cosĮ = , cos Į = // // (1/25) with Į y1, Įy2 expressed by the radii rb and ry. This then gives b1b 2 1y 2 y1 y2cosĮ = , cos Į = yrr rr (1/26) From Equations (1/25) and (1/26), we then obtain b1b 2 y1 y2 y1 y2 = rr rr// with y1 1 y1Ȧr=/ , 2 2 y1Ȧr=−/ , b1w 2 b2w 1rr rr=, 1,2 1,2wr0 C= (1/27) which results in (1/28) The pitch circle radii rw1,2 are given by the position of the point C . Point C, called the pitch point, is determined by the point of intersection of the normals NN of the touching tooth flanks with the line joining the gear centre points 0102. The gear ratio i from Equation (1/28) is therefore constant, provided that the common normal NNpasses through a point of constant bearing, the pitch point C. The Law of Gears1) can therefore be formulated as follows: “If an angular velocity with uniform gear ratio is to be transmitted from one shaft to another shaft by tooth flanks, then the common normals of both curves used as tooth profiles must pass the pitch point C at each point of contact”. This law enables the unambiguous determination of the counter-profi le for a given tooth profile and given pitch circles. This approach is illustrated in the following example. We shall assume the tooth profile of the gear 1 depicted in Figure 1/24. The counter-profile B is to be determined for a constant gear ratio. According to the Law of Gears, the construction of this flank is possible on a point-by-point basis. We first describe this process for one point. The pitch circles r w1, rw2 are derived from the gear ratio i = -rw2/rw1 and the centre distance according to Equations (1/21) and (1/22). If the gear is paired with a rack, then the pitch circle can be freely selected. Not every quantity, however, is practical. From Equation (1/21) or (1/22), the pitch point C is unambiguously the point of contact of the pitch circles. A point ( A 1) is now selected on the given profile ( A). The task now is to determine the point ( B1) of the counter-profile ( B) that comes into contact with this point on the given profile. The normal to the profile ( A) at the pointA 1 intersects the pitch circle of the gear 1 (Point 1 ). Point A1 touches the counter-profile when this normal passes through the pitch point C. For this purpose, gear 1 must have rotated far enough for point 1 to match the position of pitch point C. 1) For spatial gears (non-intersecting, non-parallel axes), the Law of Gears is formulated differently. 1 2b2w 2 b1w 1Ȧ Ȧ = = rri = rr−− 28 1 Overview, General Principles Following this rotation, E1 cor- responds to the new position of point A1 as well as the sought-after point of the mating flank ( B) (the triangles 01A11 and 01E1C are congruent). Based on the above statement, the flanks meet at E 1, since the normal passes through the pitch point C. It now only remains for the position of the point sought in the initial position to be determined. As the pitch circles mesh without slipping (no sliding), gear 2 is also rotated about the arc length p1C= (p1C′). The initial po- sition of the point of the profile ( B) of gear 2 touching at E1 is obtained by backward rotation about this arc. It then assumes the position B 1; B1 is the point sought (the point associated with A 1) of the counter-profile ( B) for the drawn position of the profile A . Figure 1/24 Constructing a point on the counter-profile The individual rotations are made clear in Figure 1/24 by means of hatched triangles. To transmit the arc lengths rolled off on the pitch circle, e.g., p1Cfrom gear 1 to gear 2 (p1C′) when using normal technical drawing aids, the circumference of both pitch circles in the vicinity of the pitch point is divided into small arcs of equal length (provided by the points 1, 2, 3, 4, 5 ...; 1 , 2 , 3 , 4 , 5 ... ). The length of these sections is to be chosen so that the arc and chord lengths do not considerably differ from each other. By now applying this approach to the points A1, A2, ..., An distributed over the entire profile A (Figure 1/25), then by connecting the points obtained, E1, E2, ..., En, we obtain both the so-called line of action as the geometric location of the point of contact of both profiles and (from the points B1, B2, ..., Bn) the sought-for counter-profile, e.g., the determining tool. For this purpose, it is expedient to make use of the previously mentioned division of the pitch circles into equal arc segments. The normals to the given profile are drawn from the points on the pitch circle to which the given profile belongs (points A 1, A2, ...). For each of these points ( A1, A2, ...), the counter- profile point is constructed (B 1, B2, ...), as described for A1, B1. The construction of the generating tool for a given straight-flanked profile of a shaft (“spline shaft”) is shown in Figure 1/26, taking the example of point A 5. 1.3 Law of gears for cylindrical gearing 29 If the given profile (tooth) is consi- dered as a cutting tool (e.g., rack cutter or hob), which of necessity is guided and rolls on the pitch circle of the profile to be generated, the counter-profile corresponding to the Law of Gears is automatically created (Figure 2.1/8). The construction principle is often used to determine the tooth root tran-sition curve. Figure 1/27 depicts this curve construction, described by the tooth edge ( K) of the counter-profile B (tool) when rolling relative to the profile to be generated (or given) A. It corresponds to the root curve created on the counter-profile generated, if the given profile represents a tool profile. The path of the tip point (denoted here with K) is considered relative to the counter-profile when rolling. For this purpose, both pitch circles (rack pitch line and pitch circle) are initially divided into small sections of equal arc length. When rolling the pitch cir-cle of the gear-generating gear (pro-file) on the pitch circle of the mating gear (profile) , then point 1 is congru- ent with 1 , 2 with 2 , 3 with 3 , and so on at specific positions. The tooth edge K of the gear-generating profile firmly attached to the pitch circle ( 2) possesses in each case a constant distance to the selectable points on pitch circle (1) of the gear-generating profile, independently of the relative position of pitch circle 1 to pitch circle 2. This means that when point 10 comes to rest on 10 , the tip point K possesses a distance N 10 = 10 K′ from point 10. Initially, it is sufficient to know that in this position, where K lies on a circular arc made on 10 with a radius N 10, draws through this cir- cular arc, and performs this construc-tion for a sufficient number of points, the root curve sought is automatically obtained as an envelope. Figure 1/25 Constructing the counter-profile Figure 1/26 Constructing the rolling tool for a given, straight- flanked (spline) shaft profile 30 1 Overview, General Principles Figure 1/27 Constructing the relative path of the tip point; generated root curve for a tool without tip rounding (cutting wheel) A further example of the construc- tion of the root curve can be seen in Figure 1/26, where the tooth tip of the tool (B 5) generates the undercut ( F5A5) determined by the envelope. In order to obtain the lowest possible stress concen-tration (notch effect) in the tooth root, a root curve with the largest possible radii of curvature is striven for. Due to the action of the mating gear (relative tip edge path of the mating gear), however, this cannot be arbitrarily selected. Of necessity, it has to lie below the envelope described by the tooth tip of the mating gear. If the counter-profile possesses a tip rounding (e.g. gear-generating rack-shaped tool), the relative path of the centre point of the tip rounding is first constructed according to the described procedure depicted in Figure 1/27. Then the root curve is dete rmined as the equidistant to the radial line (Figure 1/28). Figure1/28 Constructing the relative path of the rounded tip of the counter-profile (tooth rack profile) as an envelope to the equidistant of the centre point M of the tip rounding; root curve when manufacturing gears using a hob or rack-type cutter. 1.3 Law of gears for cylindrical gearing 31 This ensues by drawing a larger number of circles with the tip radius Na, whose centre point lies on the path of M. The resulting envelope is the sought-for root transition curve. It represents a cycloid . It should be noted, however, that not every profile constructed according to the procedure described is useful. A profile is not useful if arbitrary circles around the centre point of the gear in question can intersect the profile more than once (Figure 1/29), and a concave profile comes to rest on a concave profile (or a convex profile on a concave profile) with a quanti-tatively smaller radius of curvature (Figure 1/30). The case depicted in Figure 1/30 can also be recognised by the course of the line of action formed from the points E 1, ..., E n (for points Ei refer to Figure 1/25). An unsuitable profile is one in which more than one point of the line of action possesses the same distance to the centre point. The practical part of the profile used is externally limited by the tip circle (usable tip circle) and internally limited by the usable root circle. Figure 1/29 Unsuitable gear profile Figure 1/30 Usable (a and b) and unusable gear pairings (c and d) Once it has been demonstrated how the counter-profile could be constructed for a given profile, the question now arises as to under what conditions gears not directly manufactured by mutual rolling can be paired with each other, in order to enable smooth running, i.e., i = constant. This characteristic is known as the interchangeable gear property . Figure 1/31 is intended to show the particular properties [1/3]. The profile of the gear denoted by A can be generated by rolling the tooth rack profile B 0. Profile B would otherwise be rolled out by A0. Since the gear- generating profiles distinctive as tooth racks represent complementary profiles (see bottom right in Figure 1/31), then A can roll with B and A 1 with B1: If therefore a suitable rolling gear pair A + B is available, then all B 0, B1, ..., B n gears, which would work together with A, fit all A 0, A1, ..., A n gears, which would work together with B [1/3]. 32 1 Overview, General Principles In order to manufacture the A1, ..., An gears (of the subset A) and the B1, ..., Bn gears (of the subset B), two tools are thus required ( A0, B0). This toothing system is known as a paired tooth system. It would appear to be a reasonable step to develop the gear-generating profiles (tooth racks) in a centrally symmetrical manner so that A 0 = B0 is obtained (Figure 1/32). Under this condition, the differences between subsets A 1, ..., An and B1, ..., Bn cease to apply, while all the gears with different numbers of teeth that can be produced with a single tool are mutually pairable in any combination. This characteristic is known as the interchangeable gear property: According to Releaux, interchangeable gears are equally divided gears, which form a set and can function properly with each other irrespective of the pairing [1/31] (refer to [2]). Figure 1/32 depicts a centrally symmetrical straight generating profile (plane tooth system) which ensures the interchangeable gear properties (profile A0 identical to B0). In its simplest form, it possesses straight tooth flanks (involute toothing!). The straight generating profile or plane tooth system – or, as a borderline case, the toothing mapped on a rack – is known as the basic rack (Section 2.1.1.2). In each case, it generates usable tooth profiles, provided that the tooth depth h, pressure angle Į and helix angle ȕ are selected so that during the rotation of the gears at least one flank pair is always engaged, that is, the total contact ratio is İ Ȗ 1. Figure 1/32 Interchangeable gear properties via a centrally symmetrical reference profile (centrally symmetrical rack profile) Figure 1/31 Paired tooth systems (Kutzbach [1/3]) 1.4 Fundamental Relationships 33 1.4 Fundamental Relationships 1.4.1 Gear Ratio The adjustment mentioned in Section 1.2 ensues by changing the input parameter to the output parameter. Quantitatively, this is expressed by the gear ratio i. The gear ratio i is the relation of the drive angle velocity to the output angle velocity . For rotational input and output motion, taking into account the relationship (1/29) between the angular velocity Ȧ and the speed of rotation n Ȧ2ʌ = n (1/29) The gear ratio i is therefore an ab an ab / = / ȦȦi = nn (1/30) As a rule of sign , the following applies: when the input and output shafts are rotating in the same direction, the gear ratio i is positive; when rotating in the opposite direction, i is negative. It is often the case – including for non-parallel shafts – that only the absolute value for i is specified, with no additional comment or distinction as to the direction of rotation. The magnitude of the gear ratio for a gear unit depends on the particular dimensional proportions. We now derive the gear ratio for gear units from the dimensions. Similarly to the case of friction gears, tooth gears can be regarded as two gear bodies in rolling contact with each other. As in this case the transmission in fact takes place in an interlocking manner (by the teeth meshing), the circumferential speed is the same at the point of contact of the rolling gear bodies (rolling cylinders); refer to Figure 1/33. 12 ==/// (1/31) The speed / for rotary motion is Ȧ = r/ (1/32) Figure 1/33 Velocity at a cylindrical gear pair a) external gear pair; b) internal gear pair 34 1 Overview, General Principles Consequently, for the speeds from Figure 1/33 for the external gear / external gear pair 1212 = ȦȦrr− (1/33) and thus for the gear ratio i with d = 2r for gear units , we obtain 21 12ȦȦ = / i = dd− (1/34) Since the diameter and number of teeth of the internal gear are defined in Section 2.1.1.3 as negative quantities, Equation (1/34) also applies to internal gears. The gear ratio i of a gear pairing is equal to the negative relationship of the (pitch circle) diameter of the output gear to that of the input gear . Often, only the magnitude of the gear ratio is determined, and the negative sign in Equations (1/33) and (1/34), expressing the reversal of rotation, is omitted. For toothed gears , the diameter is proportional to the number of teeth, so that with d 1 = z 1m, d2 = z 2 m (m = proportionality factor; module) we obtain for external and internal gears (for internal gears, z2 < 0) (1/35) The gear ratio of multi-stage gears can be derived from the gear ratios of the individual gear stages by – starting with the input or output angular velocity – expressing the angular velocity of the idler gears from the transmission ratios of the individual gear stages. For any given gear stage ( j), say, the second gear stage ( j = 2) in accordance with Figure 1/34, we have ; / = abj anj j n n i anj ab(j-1) = ; nn abj an ( j+1) = nn Figure 1/34 Gear stages in series arrangement (schematically represented) to derive the gear ratio and the efficiency of multi-stage gear transmissions If nab is assumed and this factor is substituted by n ank/ik and this is continued as far as the first stage, then generally speaking an ab 123 k = ... nniii i (1/36) Making use of Equation (1/30), we thus obtain 123 k... ii i i i (1/37) The overall transmission ratio i of a multi-stage gear is equal to the product of the gear ratios i1 ... ik of the individual gear stages arranged in series. 12 2 1Ȧ/Ȧ / iz z== − 1.4 Fundamental Relationships 35 An example of a multi-stage gear is depicted in Figure 1/35. The application of the rules to determine the total transmission ratio for gear chains (Figure 1/36) leads to a special result. In this case, the intermediate gears lying between the input and output gears directly engage with the driving and driven gears. For ( k + 1) given gears present, a total of k meshes result, to which the step transmission ratio i j = − zj+1/zj is assigned. If the product of the individual transmission ratios is calculated in accordance with Equation (1/37) k j j=1 i = i∏ (1/38) then this gives ab an ( 1)zk zi = (1/39) where k is the number of meshes, and thus k + 1 the number of gears. Equation (1/39) shows that for gear chains, the absolute amount of the gear ratio is only dependent on the number of teeth of the output and input gears. The intermediate gears (from the second to the k th gear) only exert an effect on the sense of rotation. The rotational speed of an epicyclic gear (planetary gear) is obtained from the superimposition (sum) of the number of revolutions of the ideal fixed-axle gear transmission (motion relative to the arm) and the number of revolutions, which can be considered as the rotation of the entire housing. There exist special tried and tested methods for the practical calculation [17], [18], [19]. Figure 1/35 Multi-stage cylindrical gear 21 I II II I I ; = / ; zz ii i i i =− III I I 43 65 / ; = / ; iz z z z i =− − 21 43 65 = ( ) ( ) ( )iz z z z z z− Figure 1/36 Gear chain 36 1 Overview, General Principles 1.4.2 Efficiency The efficiency Ș is defined as the absolute magnitude of the ratio between the output power P ab and the input power Pan. ab anȘ / = PP (1/40) The power balance of a gear is given by ȈP = 0, provided that the input power is defined as positive and the output power (including power loss) as negative. () an ab V 0 = PP P++ (1/41) Pan input power Pab output power PV power loss With PV = (1 Ș)Pan, then we obtain ȘPan + Pab = 0 or (1/42) It is often customary to only perform calculations with the absolute magnitudes of the powers concerned. With regard to planetary gear trains, for which the sum is to be formed from powers, rotational speeds, and torques, the definition of these quantities as signed quantities has proven to be extremely beneficial. If the powers are expressed by the torques and angular velocities, then the following equation is obtained for the efficiency: ab ab an anȦȘ ȦM = M− (1/43) If the torque ratio is denoted as Mab/Man = i M, then by using the previously defined angular velocity ratio Ȧan / Ȧab = i, we obtain (1/44) As gear drives are constructed from positive transmission elements, losses only occur with the torque ratio i M (in contrast to friction gears and other non-positive gears and clutches). The total efficiency Ș of an input or gear results from the efficiency of the individual steps of the gears or gear steps arranged in series. Based on Figure 1/34, we obtain for the jth element abj j anj = ;ȘP P− anj ab(j 1)= ;PP − − P P 1) an(j+ abj=− If Pabj is replaced by (– PanjIj) in each case, and this is continued step by step beginning from the last step ( k) through to the first step, then we obtain ab an 12 3 k = ... Ș ȘȘ Ș PP ⋅ (1/45) Taking account of Equation (1/42), the efficiency is then given by (1/46) 12 3 kȘ = ... Ș ȘȘ ȘMȘi = i−ab anȘ / = PP− 1.4 Fundamental relationships 37 The following sentence thus applies: The total efficiency is the product of the efficiency of the individual steps i 1 ... i k of the (gear) elements arranged in series. A comparison of typical efficiency values between cylindrical gears (one single step) and other gears is shown in Table 1/1. It can be seen that a cylindrical gear pairing possesses the minimum losses of all gears. This statement also remains valid in comparison to other basic systems of the input, such as motors, turbines, and clutches subject to slip. The determination of the efficiency factor of a gear pairing depending on the gear data and hydro- dynamic conditions appears in Section 6.6.2. 1.4.3 Torques The fundamental relationship between the power, angular velocity and torque is as follows: Ȧ M = P / (1/47) Thus with ηP an + P ab = 0 [from Equation (1/42)], or with η = −M ab ω ab / (M an ω an) [from Equation (1/43)] and i = Ȧ an /Ȧ ab [from Equation (1/30)], the output torque M ab is derived as (1/48) For a positive gear ratio i, i.e., for the same direction of input and output rotation, the output torque (as the external torque acting on the gear) is opposed to the sense of rotation of the shaft. Here Mab acts in the sense of inhibiting the rotary motion. Because, according to Equation (1/48), this also results in a change in the torque in the gear, i.e., generally speaking M ab Man, then because M = 0Σ (1/49) there must be a third torque at work. This is in fact the reaction torque acting from the baseplate, with which the fixing elements of the gear box are subject to load (Figure 1/37). Equation (1/49) in its full form then reads an G ab 0 MM M++ = (1/50) an abȦȦi= ()an ab V 0 PP P++= an an anȦ > 0 MP = ; ab ab abȦ 0 MP =< ab anȘ PP=− , ab anȘ MM=− Figure 1/37 External torques acting on the gear ab an = Ș− M i M 38 1 Overview, General Principles Substituting Mab from Equation (1/48) into Equation (1/50), we now obtain for the reaction torque MG acting on the housing from the baseplate (1/51) The sign of the gear ratio i is to be included in the equation. It is only as a result of a torque MG 0 that it is at all possible to bring about a change in the amount of the output torque in relation to the input torque in a gear [refer to Equation (1/50)]. If MG = 0, then it is possible to comprehend the corresponding entity as a coupling, for which the relationship Man = −Mab is always true. 1.5 Symbols and Symbol Explanations A mm centre distance d mm diameter, reference diameter i - gear ratio iM - moment ratio k - number of meshes m mm module n min-1 number of revolutions r mm radius P kW power PV kW power loss M Nm torque ȣ m/s velocity z - number of teeth Į pressure angle; profile angle İȖ - total contact ratio Ș - efficiency N, ȡ mm radius of curvature 7 s-1 angular velocity Indices ab out(put) an in(put) b base cylinder G frame; box n normal plane t transversal plane w pitch circle 1 pinion 2 wheel G an = (Ș1i - )M M 39 2 Gear Teeth Geometry 2.1 Geometry of Spur Gear Teeth 2.1.1 Gear Teeth Geometry of a Spur Gear 2.1.1.1 Involute Generally, a tooth design is produced by a length of the trajectory of a circle (rolling circle) rolling on or in a circle (pitch circle). The resulting curves are cycloids. In the limiting case of an infinitely large radius of the rolling circle, a line rolls off a circle. In this process every arbitrary point on this line defines a curve, which is called the involute. The circle on which the line rolls off is the base circle. It is the geometric location of all centre points of the circles of curvature of the profile (also called the evolute). The advantages of involute teeth include • straight-flanked (simple) tools are possible (Section 2.1.1.2), • gears with the same base pitch can be paired, • they are kinematically insensitive to centre distance variation (parallel shift of axes with İ Ȗ ≥ 1), • a profile shift is possible (Section 2.1.2), • the tooth force direction is constant. The disadvantages include • an undercut when there is a small number of teeth (Section 2.1.1.3), • for external gear teeth, there is a contact of flanks which both have convex profiles (unfavourable for contact stress). The formation of the involute is usually portrayed by a taut string coming off a fixed base circle (Figure 2.1/1) . For the procedure demonstrated in Figure 2.1/1, if one imagines the entire system rotating in such a way that the string maintains a constant direction, then it is also possible to imagine this being generated by a string coming off the rotating base circle (Figure 2.1/2). The point P – imagine here it is the tip of a pencil – draws a curve – the involute (identical to the involute as per Figure 2.1/1) – on a piece of paper assumed to be attached to the rotating base cylinder, Figure 2.1/2. The principle shown in Figure 2.1/2 is used to determine the profile deviation. On a disc (the base circle disc), which is affixed to the gear to be examined, a rule is rolled off in a way corresponding to the sequence of motion shown in Figure 2.1/2 (no sliding between the rule and the base circle). 40 2 Gear Teeth Geometry Between the measurement de- tector attached to the rule and the involute, every deviation from the theoretically correct profile will be displayed as a deviation from the profile (refer to Section 8). In the production processes which are actually utilised, for precision and longevity reasons tooth flanks are usually generated not by point-shaped cutting tips, but by a cutting edge; Figure 2.1/3a. Principally it is possible to pro-duce the same involute with tools that have different pressure angles (Figures 2.1/3b to 3d). Figure 2.1/1 Formation of the involute (of a circle) by a point P on a raised, taut string They must generate the same base circle. The normal for the tool profile ( Į0 = Į w) must be tangent to a base circle with a constant diameter. Here it follows that the working pitch diameters dw are different, dependent on Įw: b w w = cosĮdd (2.1/1) The roll angle has a strong influence on the tooth root fillet being produced because of the changed relative tip edge path (refer to Section 2.3.2). To realise this in practice with generating tools (e.g., hobs), the changed tool addendum and dedendum as well as the changed tooth thickness would have to be observed. To generate the involute, the motion of the point P y on a line tangent to the base cylinder with /b = 7rb therefore corresponds to the velocity component of the tool in the direction of the base circle. The tool rolls off the pitch circle with the velocity /w. It has the radius rw. Because the cutting edge is slanted (angle Į0), this does not match the base circle anymore. The velocity /b on the circumference of the base circle, expressed as a component of pitch speed, is then /b = /w cos Įw (2.1/2) The velocity on the circumference of the pitch circle is /w = 7rw (2.1/3) 2.1 Geometry of Spur Gear Teeth 41 Figure 2.1/2 Generation of involutes attached to the rotating base circle by points PI,II;… on the string being pulled off the base circle From the speed conditions and the moving (generating) point Py on the cutting edge, one can already see that, for a tool rolling off with a line, the result is straight-flanked cutting edges for the generation of the involute, which is one of its excellent benefits; (Figure 2.1/3). Graphically, the involute is best determined with the envelope construction; (Figure 2.1/4). This is done by first drawing the tangents (line rolling off or raised string) on the base circle, for example, in point 3 ƍ. Now a point is selected on this line, for example, the point 0, which should be on the involute. Starting from the contact point of the tangent with the base circle (in Figure 2.1/4, point 3ƍ) one divides the rolling line into equal lengths ;3,2 2,1 and ,;34 4 , 5 and transfers these as lengths to the base circle (e.g., oo; 3,2 = 3 2 2,1 = 2 1 ′′ ′′ ) so that when rolling, point 2 lands on 2ƍ, 1 on 1ƍ, and so on. For the sake of expediency, one chooses these distances in such a way that the difference between arc and chord length is small, which makes it possible to transfer the distances from the line to the circle with dividers. With the distance 0,1 one then describes a circle by point 1ƍ, with 0,2 by 2ƍ, with 0,3 by 3ƍ and so on. The involute results as an envelope of the circular arc group (envelope construction). At the point of intersection of the rolling line with the involute, that is, the points 1, 2, and so on, the circles with the radii N1, N2, and so on, correspond exactly to their radii of curvature . The centre points of all circles of curvature whose radii of curvature are N1, N2, N3, and so on are on the base circle. 42 2 Gear Teeth Geometry The involute function invĮy is defined based on Figure 2.1/1. It plays a decisive role in gear calcu- lations. The basis for slip-free rolling is the path PT equal to the arc length p0T and therefore () by b y ytanĮĮ inv Į rr =+ After solving for invĮ y the result is y yy invĮ = tanĮ Į− (2.1/4) Values for the involute function are listed in Table 2.1/1 (inv20 = 0.01490438). With invĮy given, Įy can only be determined by iteration in several steps: α y (n + 1) = yn yn y 2 yntanĮĮ invĮ tanĮ−+ + + ynĮ (2.1/4a) As a usable, zeroed approximation, the result is (from a series expansion ) Equation (2.1/4b): y y3 0 3 inv () Į Į = (2.1/4b) The involute function is applied among others in the calculation of tooth thickness and profile shift. With a given radius r y, according to Figure 2.1/1 the associated angle Įy results from b y ycosĮ = r r (2.1/5) Figure 2.1/3 a) Generation of an involute by a rack as generating tool 2.1 Geometry of Spur Gear Teeth 43 Figure 2.1/3 b), c), d) Generation of the same involute by tools (e.g., grinding discs) with different profile angles Figure 2.1/4 Envelope construction of the involute 44 2 Gear Teeth Geometry Table 2.1/1 Involute Function inv Įy Įy 15 16 17 18 19 20 0.00 0.00614980 0.00749271 0.00902471 0.01076043 0.01271506 0.01490438 0.01 0.02 0.03 0.04 0.05 0.06 0.07 0.08 0.09 0.00616234 0.00617490 0.00618748 0.00620007 0.00621268 0.00622531 0.00623795 0.00625061 0.00626329 0.00750707 0.00752144 0.00753584 0.00755026 0.00756470 0.00757915 0.00759362 0.00760812 0.00762263 0.00904103 0.00905738 0.00907374 0.00909013 0.00910653 0.00912296 0.00913941 0.00915587 0.00917236 0.01077887 0.01079733 0.01081581 0.01083431 0.01085283 0.01087138 0.01088995 0.01090854 0.01092715 0.01273576 0.01275649 0.01277724 0.01279801 0.01281881 0.01283964 0.01286048 0.01288135 0.01290224 0.01492752 0.01495068 0.01497386 0.01499707 0.01502030 0.01504356 0.01506685 0.01509016 0.01511349 0.10 0.00627599 0.00763716 0.00918887 0.01094579 0.01292316 0.01513685 0.11 0.12 0.13 0.14 0.15 0.16 0.17 0.18 0.19 0.00628871 0.00630144 0.00631419 0.00632696 0.00633974 0.00635255 0.00636537 0.00637821 0.00639107 0.00765171 0.00766628 0.00768087 0.00769547 0.00771010 0.00772474 0.00773941 0.00775409 0.00776880 0.00920540 0.00922195 0.00923851 0.00925510 0.00927172 0.00928835 0.00930500 0.00932167 0.00933836 0.01096444 0.01098312 0.01100182 0.01102055 0.01103929 0.01105806 0.01107685 0.01109566 0.01111449 0.01294410 0.01296506 0.01298605 0.01300706 0.01302810 0.01304916 0.01307024 0.01309134 0.01311247 0.01516024 0.01518365 0.01520709 0.01523055 0.01525404 0.01527755 0.01530109 0.01532465 0.01534824 0.20 0.00640394 0.00778352 0.00935508 0.01113335 0.01313363 0.01537185 0.21 0.22 0.23 0.24 0.25 0.26 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0.00965948 0.00967658 0.01134223 0.01136135 0.01138050 0.01139966 0.01141885 0.01143807 0.01145730 0.01147656 0.01149584 0.01336789 0.01338933 0.01341079 0.01343228 0.01345379 0.01347533 0.01349689 0.01351847 0.01354008 0.01563329 0.01565721 0.01568116 0.01570513 0.01572913 0.01575315 0.01577720 0.01580127 0.01582537 0.40 0.00666519 0.00808201 0.00969371 0.01151514 0.01356172 0.01584950 0.41 0.42 0.43 0.44 0.45 0.46 0.47 0.48 0.49 0.00667844 0.00669171 0.00670499 0.00671830 0.00673162 0.00674496 0.00675832 0.00677170 0.00678510 0.00809714 0.00811228 0.00812745 0.00814264 0.00815784 0.00817307 0.00818831 0.00820358 0.00821886 0.00971087 0.00972804 0.00974523 0.00976244 0.00977968 0.00979693 0.00981421 0.00983151 0.00984883 0.01153447 0.01155381 0.01157318 0.01159258 0.01161199 0.01163143 0.01165089 0.01167037 0.01168988 0.01358337 0.01360505 0.01362676 0.01364849 0.01367024 0.01369202 0.01371382 0.01373564 0.01375749 0.01587365 0.01589783 0.01592203 0.01594626 0.01597052 0.01599480 0.01601911 0.01604344 0.01606780 2.1 Geometry of Spur Gear Teeth 45 Table 2.1/1 Involute Function inv Įy (continued) Įy 15 16 17 18 19 20 0.50 0.00679851 0.00823417 0.00986617 0.01170941 0.01377937 0.01609218 0.51 0.52 0.53 0.54 0.55 0.56 0.57 0.58 0.59 0.00681194 0.00682539 0.00683886 0.00685235 0.00686585 0.00687938 0.00689292 0.00690648 0.00692006 0.00824949 0.00826484 0.00828020 0.00829558 0.00831098 0.00832641 0.00834185 0.00835731 0.00837279 0.00988353 0.00990091 0.00991832 0.00993574 0.00995319 0.00997066 0.00998814 0.01000565 0.01002319 0.01172896 0.01174853 0.01176813 0.01178775 0.01180739 0.01182705 0.01184674 0.01186645 0.01188618 0.01380127 0.01382319 0.01384514 0.01386711 0.01388910 0.01391112 0.01393317 0.01395524 0.01397733 0.01611659 0.01614103 0.01616549 0.01618998 0.01621450 0.01623904 0.01626361 0.01628820 0.01631282 0.60 0.00693365 0.00838829 0.01004074 0.01190594 0.01399945 0.01633746 0.61 0.62 0.63 0.64 0.65 0.66 0.67 0.68 0.69 0.00694727 0.00696090 0.00697455 0.00698822 0.00700191 0.00701562 0.00702934 0.00704309 0.00705685 0.00840381 0.00841935 0.00843491 0.00845049 0.00846609 0.00848171 0.00849735 0.00851301 0.00852869 0.01005831 0.01007591 0.01009352 0.01011116 0.01012882 0.01014650 0.01016420 0.01018192 0.01019967 0.01192572 0.01194552 0.01196534 0.01198519 0.01200506 0.01202495 0.01204487 0.01206481 0.01208477 0.01402159 0.01404376 0.01406595 0.01408817 0.01411041 0.01413267 0.01415496 0.01417727 0.01419961 0.01636213 0.01638683 0.01641156 0.01643631 0.01646108 0.01648588 0.01651071 0.01653557 0.01656045 0.70 0.00707063 0.00854439 0.01021743 0.01210476 0.01422197 0.01658536 0.71 0.72 0.73 0.74 0.75 0.76 0.77 0.78 0.79 0.00708443 0.00709825 0.00711208 0.00712594 0.00713981 0.00715370 0.00716761 0.00718154 0.00719549 0.00856011 0.00857585 0.00859161 0.00860739 0.00862319 0.00863901 0.00865485 0.00867071 0.00868659 0.01023522 0.01025303 0.01027086 0.01028871 0.01030658 0.01032448 0.01034239 0.01036033 0.01037829 0.01212476 0.01214479 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0.03203586 0.03207569 0.03211556 0.03215546 0.03219540 0.03223537 0.03227539 0.03611276 0.03615620 0.03619969 0.03624320 0.03628676 0.03633036 0.03637399 0.03641766 0.03646137 0.60 0.01893689 0.02181541 0.02499155 0.02848475 0.03231543 0.03650512 0.61 0.62 0.63 0.64 0.65 0.66 0.67 0.68 0.69 0.01896426 0.01899167 0.01901909 0.01904655 0.01907404 0.01910155 0.01912909 0.01915666 0.01918426 0.02184567 0.02187596 0.02190627 0.02193662 0.02196699 0.02199740 0.02202784 0.02205830 0.02208880 0.02502488 0.02505824 0.02509164 0.02512506 0.02515852 0.02519201 0.02522553 0.02525908 0.02529266 0.02852135 0.02855799 0.02859466 0.02863136 0.02866809 0.02870486 0.02874167 0.02877851 0.02881538 0.03235552 0.03239563 0.03243579 0.03247598 0.03251621 0.03255647 0.03259677 0.03263710 0.03267747 0.03654890 0.03659273 0.03663659 0.03668049 0.03672443 0.03676840 0.03681242 0.03685647 0.03690056 0.70 0.01921188 0.02211932 0.02532628 0.02885229 0.03271788 0.03694469 0.71 0.72 0.73 0.74 0.75 0.76 0.77 0.78 0.79 0.01923954 0.01926722 0.01929493 0.01932267 0.01935043 0.01937823 0.01940605 0.01943390 0.01946178 0.02214988 0.02218046 0.02221108 0.02224172 0.02227240 0.02230310 0.02233384 0.02236460 0.02239540 0.02535992 0.02539360 0.02542732 0.02546106 0.02549483 0.02552864 0.02556248 0.02559635 0.02563025 0.02888923 0.02892620 0.02896321 0.02900025 0.02903732 0.02907443 0.02911158 0.02914876 0.02918597 0.03275832 0.03279880 0.03283932 0.03287987 0.03292046 0.03296108 0.03300174 0.03304244 0.03308317 0.03698886 0.03703307 0.03707731 0.03712160 0.03716592 0.03721028 0.03725468 0.03729912 0.03734359 0.80 0.01948969 0.02242622 0.02566419 0.02922322 0.03312394 0.03738811 0.81 0.82 0.83 0.84 0.85 0.86 0.87 0.88 0.89 0.01951762 0.01954559 0.01957358 0.01960160 0.01962965 0.01965772 0.01968583 0.01971396 0.01974213 0.02245708 0.02248797 0.02251888 0.02254983 0.02258080 0.02261181 0.02264285 0.02267391 0.02270501 0.02569816 0.02573216 0.02576619 0.02580025 0.02583435 0.02586848 0.02590264 0.02593683 0.02597106 0.02926050 0.02929781 0.02933516 0.02937254 0.02940996 0.02944741 0.02948490 0.02952242 0.02955997 0.03316475 0.03320559 0.03324647 0.03328738 0.03332833 0.03336932 0.03341034 0.03345140 0.03349250 0.03743266 0.03747725 0.03752189 0.03756656 0.03761126 0.03765601 0.03770080 0.03774562 0.03779048 0.90 0.01977032 0.02273614 0.02600531 0.02959756 0.03353363 0.03783539 0.91 0.92 0.93 0.94 0.95 0.96 0.97 0.98 0.99 0.01979854 0.01982678 0.01985506 0.01988336 0.01991170 0.01994006 0.01996845 0.01999687 0.02002531 0.02276730 0.02279849 0.02282970 0.02286095 0.02289223 0.02292354 0.02295488 0.02298625 0.02301766 0.02603960 0.02607392 0.02610828 0.02614266 0.02617708 0.02621153 0.02624602 0.02628053 0.02631508 0.02963518 0.02967284 0.02971053 0.02974826 0.02978602 0.02982382 0.02986165 0.02989951 0.02993741 0.03357480 0.03361601 0.03365725 0.03369853 0.03373985 0.03378120 0.03382259 0.03386402 0.03390548 0.03788033 0.03792531 0.03797033 0.03801538 0.03806048 0.03810562 0.03815079 0.03819601 0.03824126 48 2 Gear Teeth Geometry Now it is time to determine the coordinates of the involute. According to Figure 2.1/5, when the parameter ȥ is used, the results for this are x = r b (sinȥ – ȥ cosȥ) (2.1/6) y = r b (cosȥ + ȥ sinȥ) (2.1/7) Between the parameter ȥ and the angle Įy the following relationship exists (refer to Figure 2.1/1): y y yȥ + inv Į Į ȥtanĮ = respectively = (2.1/8) Figure 2.1/5 Involute quantities If one uses the general equation for the radius of curvature of a curve N()3/221 + y = y′ ′′ (2.1/9) the following derivations are obtained with Equations (2.1/6) and (2.1/7) taking into account the relationships d1 d ( ) 1 = and = dd dȥ dȥ dȥ dȥ′′′ ′ ⋅⋅yyyyxx and the quantity of the radius of curvature already mentioned: Nbȥ | = |r Since ȥ = tanyĮ (Equation 2.1/8), the result is by = t a n Į| |rN (2.1/10) 2.1 Geometry of Spur Gear Teeth 49 With that, the quantity for N obtained from observation and used in the envelope construction is mathematically validated. The angle Įy can be determined according to Equation (2.1/5). In some cases, the arc length l of the involute is required, which lies between two given radii, ry1 and ry2. They are derived using the differential quantities d x and d y: 2 2d = d + d lx y (2.1/11) From Equations (2.1/6) and (2.1/7) one arrives at b bd = ȥ sinȥ d ȥ , d = ȥ cosȥ d ȥ ,xr yr 2 1b ȥ dȥ l = rψ ψ and after integration and application of the limits, the arc length l is ()22 21b 2 ȥȥrl = − (2.1/12) The angles ȥ1,2 result from Equations (2.1/5) and (2.1/8) with the given radii ry1,2, between which the arc length l of the involute is to be calculated (Figure 2.1/6). The length ǻg associated with the same given radii ry1,2 on the path of contact or associated rolling 22 11 21 13PP PP = (Figure 2.1/6) is 22 2 2 y2 b y1 b g = rr rr Δ− − − Calculating the volume displaced by a tooth (e.g., gear pump) first necessitates the determination of the area ( P 11, P12, P21) under the involute. Assuming a differential angle element d Q (Figure 2.1/6), when one applies the polar coor-dinates, one arrives at the elementary area (situated under the length of curve) ϕy2 yd 2 / 1 = dr A The quantities ry and Qy are expressed by the angle ȥy (Figures 2.1/1 and 2.1/6): 2 yb y yy y = 1 + ȥ = arc t an ĳȥ ȥrr − Figure 2.1/6 Geometric quantities to calculate the arc length of the involute and the tooth design area 50 2 Gear Teeth Geometry These relationships result in the area A (P 11 P21 M): ()2 12 ȥ2b 33 b2 21 yy6 2ȥȥ dȥȥ = r r A = ψ− Here, A is the area given by the points P11, P21, M, and Az represents the transverse tooth profile area Az (P11 P21 P12) that is drawn as the hatched area in Figure 2.1/6 and is enclosed by the points P11, P21, P12. It is situated between the given radii ry1 and r y2. () ()2 2 33 y 1 b z 21 y 2y 162 = ȥȥ ĳ ĳr rA −− − (2.1/13) Here the following applies: ȥ1,2 = tanĮy1,2, Qy1,2 = invĮ y1,2 and cosĮy1,2 = rb1,2/ry1,2. 2.1.1.2 Basic Rack The flank profiles can be approximated by their radii of curvature (see Figure 2.1/4). With a mating gear of increasing size, such as wheel 2, the radius of curvature NC2 also grows. As a limiting case, with an infinitely large diameter, the mating gear is a rack, and with db2 the radius of curvature of the tooth flank NC2 = 0.5 db2 tanĮ also grows beyond all limits. With that the flank profiles of the mating gear become lines. The simple straight-flanked profile given by the rack is very useful for indicating or determining the basic geometric quantities. This profile is called the basic rack . The operating pressure angle Į w is equal to the profile angle of the rack ĮP in the case of pairing a gear with a rack . The basic rack is standardised in ISO 53 (DIN 867), preferably for spur gears with modules m = 1 to 50 mm (Figure 2.1/7). It is structured in such a way that the wheels of a pairing have the same basic rack (symmetrical to the profile reference line). With that it is possible to pair any wheels with the same pitch. This is called the interchangeable gear property . If the tooth tips of the basic rack are elongated by the tip clearance c , this can also be considered as a tool profile. haP = 1m; hfP = haP +cP; NfP = (0.25…0.39) m dependent on c = 0.25 m; 0.4m Figure 2.1/7 Basic rack according to ISO 53; Type A, B, C, D Type A: Standard basic rack profile: cP = 0.25 m; hfP = 1.25m ; NfP = 0.38 m Type D: cP = 0.4 m; hfP = 1.4 m; NfP = 0.39m (full fillet radius) Type B: like A, but NfP = 0.3 m Type C: like A, but NfP = 0.25 m This tool profile is rolling in or cutting the counter-profile in the generating process. ISO 53 assigns the preferred tip clearances to the maximum root fillet radii. Concerning the tooth root capa-city, a minimum root fillet is also necessary. For gear cutting tools basic racks adapted to the manu- facturing process are given in DIN 3972. Regarding the root fillets, in the edition from 1952 they do not coincide with the spe-cification in ISO 53 (DIN 867). When using pinion-type cutters, the generation of a minimum root fillet is not mandatory be-cause with these tools often no tip radius exists. 2.1 Geometry of Spur Gear Teeth 51 When one observes the rack (tool) – gear (workpiece) rolling process from the point of view of the gear (i.e., the movement of the rack relative to the gear), the enveloping profiles depicted in Figure 2.1/8 result. The flank profile is produced as an enve-lope. Figures 2.1/9b, c, d show the prac- tical realisation of profile gene-ration suggested in Figure 2.1/8. With the addition of the clearance angle and rake angle, a rack-type cutter is created (Figure 2.1/9b), out of the form of which the rack is still clearly recognisable. If one arranges several rack-type cutters on a cylinder, shifted (one after the next) in a helical line, a hob results (Figure 2.1/9d). Since with a rack it is possible to pair two gears that have an arbitrary and different number of teeth, one can then consider one of these gears as the generating gear (tool), which hobs the gear teeth or cuts them (pinion-type cutter), as depicted schematically in Figure 2.1/9c. Section 10 con-tains more information about gear tooth production. Figure 2.1/9 Schematic description of shape cutting of gears in form cutting (a) and hobbing processes (b, c, d); 1: workpiece, 2: tool: a) form cutter; b) rack-type cutter; c) pinion-type cutter; d) hob Figure 2.1/8 Enveloping profiles of a tool with basic rack form 52 2 Gear Teeth Geometry With rack-type cutters, hobs and pinion-type cutters, on the real gear the **backlash** allowance has an effect on the root diameter df and thus also on the tip clearance c. In the case of pinion-type cutters, these quantities are also somewhat influenced by the profile shift and number of teeth, of both the gear being produced as well as the tool, since in the case of a pairing with a finite number of teeth no proportionality exists between thickness variation and centre distance variation (here shaper cutter - workpiece). A significant change of the root diameter can also occur because of dimensional deviations during heat treatment (e.g., case hardening). The tooth root curvature radius N f (size and course) that is actually produced is also dependent on the number of teeth z and the profile shift coefficient x (Section 2.3.3). If the tool tip radius is Na0 (e.g., Na0 = 0.25m n or 0.38 mn), tooth root curvature radii of Nf ≥ Na0 are produced on the gear in the case of rack-type cutters or hobs. If one uses pinion-type cutters, on the other hand, which do not always have a special tip radius, if an unfavourable profile shift occurs, then tooth root curvature radii Nf < 0.25 mn are possible, as opposed to the basic rack. This is a fact which is often ignored. In the calculation of load capacity (stress concentration effect), if one carefully observes the fillet being produced, potentially unfavourable effects will not remain unnoticed. Figure 2.1/10 Maximum values for profile modification according to ISO 53-1974. This standard has been revised by ISO 53-1998 (without profile modification). Note: According to a new conception, a sharp transition of the relief is unfavourable Figure 2.1/11 Basic rack of the gear for production with a protuberance tool (used in DIN 3990-3): ha0 = 1.4 mn; NfP = 0.4 mn; spr = 0.02 mn; Įpr = 8…10 (refer to Figure 2.1/7 for more data) Deviating from the basic rack specified by ISO 53 (DIN 867), in the sense of a modifica-tion, is permissible. Here the term modification is to be understood as a deviation which is within the order of magnitude of the elastic tooth deformation. Its purpose is to reduce the excitation of vibrations (internal dynamic tooth loads) resulting from variations in tooth deformation and/or reduce the risk of scuffing and/or minimise the effects of pressure angle and pitch deviations. For this, ISO 53-1974 specifies a limit, dependent on the module (Figure 2.1/10). Normally the deformations which actually occur do not exceed this quantity. A sharp transition of the modification into the involute should be avoided (contact pressure!). Since tooth deformation is dependent on the respective gear tooth data and the load, the specification of a fixed value for the profile modification quantity does not make sense. A deviation in the tooth root area from the profile shown in Figure 2.1/7 to avoid grinding steps is common and permissible. Because of variation in grinding stock and in deformation due to hardening, for most gears one must allow for an undercut. To ensure that there is no or no crucial reduction of the length of involute profile, a larger tooth root depth is necessary. Figure 2.1/11 depicts the profile on which the calculation of load capacity is most commonly based, with protuberance pr and undercut s pr for the gear teeth. 2.1 Geometry of Spur Gear Teeth 53 A different but likewise tried-and-tested, module-dependent determination of undercut is shown in Table 2.1/2. Reserves are available in a potential enlargement of the tool tip radius (while main-taining the undercut). Table 2.1/2 Cutter Basic Rack with Protuberance (Company Standard) (pr-spr) allowance per tooth flank s pr size of undercut pr protuberance Module m p mʌ sw0 pr-s pr 0.12 + 0.02 m spr 0.06 + 0.02 m pr 0.18 + 0.04 m ha0 1.45m Na0 0.25m q 2 6.2832 2.8011 0.160 0.100 0.260 2.900 0.500 0.272 2.5 7.8540 3.5652 0.170 0.110 0.280 3.625 0.625 0.323 3 9.4248 4.3293 0.180 0.120 0.300 4.350 0.750 0.375 4 12.5664 5.8575 0.200 0.140 0.340 5.800 1.000 0.479 5 15.7080 7.3858 0.220 0.160 0.380 7.250 1.250 0.583 6 18.8496 8.9140 0.240 0.180 0.420 8.700 1.500 0.687 7 21.9911 10.4422 0.260 0.200 0.460 10.150 1.750 0.791 Note: In newer designs Na0 is usually larger (0.35 m) and ha0 = 1.4 m chosen, and both pr as well as s pr avoided. Instead of the protuberance angle of 8 , 10 is often applied. It should be noted that dimensional changes during heat treatment do not only depend on the module, but also on the diameter of the gear, for example. So a module-dependent undercut re-presents a tool-based compromise. 2.1.1.3 Basic Quantities for Gear Teeth of a Cylindrical Gear The following equations apply equally to external and internal gear teeth. Here the number of teeth for external gear teeth is defined as positive and the number of teeth for internal teeth as negative . With this specified, according to the algorithm of DIN 3960, the computed diameter of the external tooth system is positive – as usual – and the computed diameter of the internal gear system is negative, as opposed to ISO 21171, which does not serve as a basis here. Figure 2.1/12 illustrates this according to DIN 3960. 54 2 Gear Teeth Geometry Figure 2.1/12 Transition of the positively defined diameters of the external gears into the negatively defined of internal gears: a) external gear ( d > 0); b) rack ( d ĺ ); c) internal gear (d < 0) From the perspective of the centre point of the segment of the external gear (Figure 2.1/12a), the reference circle can be viewed as the circle of curvature of the gear. If one enlarges this circle of curvature to d ĺ , the rack results (Figure 2.1/12b). Now if one imagines the segment to be deformed even more, in the sense used up to now, the centre point of curvature shifts to the other side of the assumed constant point of observation. This makes it useful to computationally define the wheel diameter and the number of teeth as negative, since the line of sight from the perspective of the fixed location does not lead to the concave, but now to the convex contour. Now a few basic quantities should first be explained or derived (refer to DIN 3960). They are necessary for either an understanding of the principles or for the illustration or calculation of load capacity. a) Pressure angle Į, reference diameter d With a growing distance from the base circle, the sloped position of the involute or the profile angle Į y also grows (see Figure 2.1/1); Įy results for a certain point of the involute with the distance ry from the wheel centre point for cos Įy = rb/ry; see Equation (2.1/5). Based on experience, in the middle range of the profile of the gear teeth, an angle of approximately Į y = 20 is good. That is why ISO 53 (DIN 867) standardises Įy = Į = 20 . The angle Į is called the pressure angle (see Figure 2.1/13). It is common to specify the diameter of the gear teeth in which the profile angle Įy corresponds to the pressure angle Į (= 20 ). This diameter is called the reference diameter d. For spur gear teeth, the reference diameter d is the diameter in which the involute is inclined to the connecting line to the centre point by Įy = Į (= 20 ) . Then d is calculated as the product of the number of teeth z and the factor m (module), which expresses the size for the gear teeth: d = z m (2.1/14) The reference diameter is purely a reference quantity and not directly measurable. 2.1 Geometry of Spur Gear Teeth 55 b) Reference circle pitch p , module m The distance between two adjacent teeth on the reference circle is the reference circle pitch p (Figure 2.1/13). It is ex- pressed as a multiple of ʌ. This multiple is the module m, introduced with Equation (2.1/14), which represents a basic quantity of gear geometry: ʌ p = m (2.1/15) With that the following definition applies: The module m is the number, which when multiplied with ʌ, results in the reference circle pitch p. Since with the module m all gear measurements grow proportionally, a second applies: The module m is a size factor for gear teeth. Figure 2.1/13 Basic quantities of external gear teeth In order to confine the variety of gears and to keep the effort for the gear tools to a minimum, the module was standardised (refer to Table 2.1/3). Even though only a number value is named when the module data is given, it must be noted that we are talking about a quantity with dimension. If only a number value is given, the unit mm applies. Table 2.1/3 Standardised Modules mn (in mm), ISO 54 (DIN 780), Preferential Modules: R I R I 1 1.25 1.5 2 2.5 3 4 5 6 8 10 12 16 20 25 32 40 50 R II 1.125 1.375 1.75 2.25 2.75 3.5 4.5 5.5 7 9 11 14 18 22 28 36 45 c) Base diameter db: The base diameter is decisive for the generation and thus also for the form of the involute (curvature) (refer to Figures 2.1/1 to 5). According to Figures 2.1/13 and 2.1/14, for the base diameter the following results: d dα cos = b (2.1/16) and with d , according to Equation (2.1/14) b = cos Į zm d (2.1/17) d) Base pitch pb, contact pitch pe The base pitch pb is the distance between two adjacent teeth on the base circle (arc length). Two adjacent teeth enclose the angle (2 ʌ/z). 56 2 Gear Teeth Geometry Figure 2.1/14 Basic geometric quantities of internal gear teeth Thus it is 2 z 2 = b bdp π With db, according to Equation (2.1/17) the result is b = ʌ cos Įmp (2.1/18) or with Equation (2.1/15) b cosĮ = p p (2.1/18a) If one specifies two points (e.g., I and II, Figure 2.1/13) on the circumference of a base circle at a distance of the base circle pitch, and follows their paths in the construction of the involute (rising of the rolling string with the points I and II), it becomes clear that in any position, the distance that separates these is constant (pII I= pb = II I). This constant distance is called contact pitch . From the Figures 2.1/13 and 14 it can be seen that the base pitch pb is equal to the contact pitch pe: eb = = ʌ cos Įm pp (2.1/19) Based on this reasoning, one of the important characteristics of involute teeth can be pointed out: The normal distance (contact pitch p e) of two adjacent, parallel tooth flanks is constant. Therefore, for the theoretical quantity pe along the involute, it does not matter from which point of the profile the vertical (normal) drops to the adjacent profile. This distinguishing property of involute teeth is used for the reference-free measurement. Only gears with the same base pitch are capable of being paired. Therefore, for the pairing, neither the module nor the pressure angle is decisive, but their product is. 12 cosĮ cosĮ (m = (m )) (2.1/20) e) Tooth height h , tip diameter d a, root diameter d f The tooth height h and the boundary diameter da respectively df of the gear teeth are also linearly dependent on the module m. The relevant quantities are shown in Figure 2.1/15. The addendum ha is determined with the help of the module m, the profile shift coefficient x (Section 2.1.2 spur gear pairing, clause f) and the tip shortening coefficient k [Equation (2.1/45)]: () aa P = / + + mm x khh (2.1/21) Note that haP/m = 1, according to ISO 53 (DIN 867). The dedendum hf results in 2.1 Geometry of Spur Gear Teeth 57 () ff P = / - mm xhh (2.1/22) with hfP = haP + c; c = c\*m. In the normal case c\* = 0.25; with protuberance c\* = 0.40…0.45 Figure 2.1/15 Tooth heights and diameters: a) pairing of external and internal gears (internal teeth equal the gap between the external teeth); b) external tooth; c) internal tooth In some special cases, such as with some older hobs and pinion cutters, the tip clearance factor is in the range of c\* = 0.16 to 0.35, and for most protuberance tools even more than 0.4. The tooth height h, as the sum of ha [with k according to Equation (2.1/45)] and hf, is ()aP2c +h = h m k∗ + (2.1/23) Based on the reference diameter d , with the addendum ha and the dedendum hf it is possible to determine the tip diameter da and the root diameter df of the gear: af f + 2 , = 2 a dd h dd h=− With haP = 1m according to ISO 53 (DIN 867) the outcome is a 2( 1 ) dz mm x k=+ + + (2.1/24) () f 21 dz mm x c∗=− − + (2.1/25) f) Tooth thickness s y on an arbitrary cylinder The tooth thickness sy is to be determined on an arbitrary cylinder with the diameter dy within the part of the tooth delimited by the involute, that is, outside of the area of the root fillet (flank - root circle). It is needed, for example, for calculations to avoid a too-small tooth tip thickness and to derive the profile shift modification. 58 2 Gear Teeth Geometry Based on Figure 2.1/16a, the tooth thickness sy in y y y = 2 ȥ2ds (2.1/26a) can be determined. Furthermore, from Figure 2.1/16a it is evident that y yb2 = 2 2 inv Į ȥȥ − (2.1/26b) and b2 = 2ȥ + 2 inv Įȥ (2.1/26c) With the tooth thickness s in the reference circle [refer to Equation (2.1/43)] one arrives at 20.5s = dψ (2.1/26d) The Equations (2.1/26a to d) result in the tooth thickness sy as the arc length on an arbitrary diameter dy with Į y from yb y cosĮ dd= in yy y invĮinvĮssdd =+ − (d y and d negative for internal gears!) (2.1/26e) g) Special tooth thicknesses With the tooth thickness s [Equation (2.1/43)] on the reference cylinder, which also depicts an arc length, with dy = d b and Į y = Įb = 0 the result is the base thickness : bb = + inv Įs sdd (d b and d negative for internal gears!) (2.1/27) This tooth thickness represents the distance between the right and left flank of a tooth on the base cylinder, when one imagines the involute extended up to there. Likewise, the tooth tip thickness s a is also dependent on the number of teeth and the profile shift. In cases of extreme profile shift and pairing of a gear that has a small number of teeth, or respectively in the external gear - internal gear pairing, as a result of only very minimal tip shortening or with a tip shortening of zero, a pointing or a too-small tooth tip thickness of the smaller external gear can occur. With surface-hardened gear teeth, a too-small tooth tip thickness can lead, among other things, to spalling at the tip. The tooth tip thickness s a results from Equation (2.1/26) with dy = da and Į y = Įa from cos Įa = db/da in aa a = + inv Įinv Įs sdd − (2.1/28) [s refer to Equation (2.1/43), da and d negative for internal gears; see definition in DIN 3960]. While a pointing of the teeth can occur in the case of external gear teeth, because of a too-large pro- file shift in certain pairings ( k = 0 or low tip shortening), for internal gear teeth there is an analogous risk that the root land becomes too narrow, or in limiting cases zero (refer to paragraph h). 2.1 Geometry of Spur Gear Teeth 59 a) External gear teeth b) Internal gear teeth Figure 2.1/16 a), b) Quantities to determine the tooth thickness sy on a cylinder with an arbitrary diameter dy 60 2 Gear Teeth Geometry Often the normal chord is relevant for measuring tooth thickness. The normal chordal tooth thickness ys on an arbitrary diameter is (Figure 2.1/16) yy y = sinȥ sd (2.1/29) with y y yȥs d= The chordal height is (Figure 2.1/16) ya y1 2( c o s ȥ) h = d d − (2.1/30) Similarly, the chordal tooth thickness on the reference cylinder results in sinȥ s = d (2.1/30a) with ȥ = s d The chordal height h up to the tooth thickness s is (Figure 2.1/16) a1 2( c o s ȥ) h = d d − (2.1/31) Noteworthy is the so-called constant chordal tooth thickness cs. As opposed to the other chordal tooth thicknesses, it is independent of the number of teeth. According to Figure 2.1/17, the constant chordal tooth thickness results in c =2 c o s Į sF H with b (inv Į)22s dFH = − and sb according to Equation (2.1/27) in 2 c cosĮ ss= [s is the tooth thickness at the reference circle, Equation (2.1/43)] The chordal height ch from the tip to the constant chordal tooth thickness is ca sinĮcosĮ2shh=− (2.1/33) h) Tooth space width The tooth space width ey on a cylinder with the diameter dy is the difference between pitch py and tooth thickness sy: ey = p y – sy (2.1/34a) 2.1 Geometry of Spur Gear Teeth 61 Figure 2.1/17 Constant chordal tooth thickness cs and chordal height ch (as the distance between the tip and the constant chord): a) external gear teeth; b) internal gear teeth With the pitch py = pT dy/d and the tooth thickness sy according to Equation (2.1/26), the result is yy yʌ 4 tan Į = inv Įinv Į2x + ed z− − (2.1/34) (dy and z negative for internal gears!) On special cylinders (e.g., tip) the tooth space width can be calculated by using the respective diameter (for dy) and angle (for Įy). Space width is a design limit only in the case of internal gear teeth (profile according to ISO 53). In the case of a large profile shift, theoretically the space width ef = 0 can occur in the tooth root, which represents an absolute design limit. 2.1.2 Geometry of Gear Teeth Pairing – Spur Gear Teeth a) Line of action Because of the properties of the involute (refer to the construction of the involute in Section 2.1.1.1), the normal of an arbitrary point of the tooth flank is always tangent to the base circle. Two touching tooth flanks have a common normal. As such, it must be tangent to both base circles of the paired gears and, according to the basic requirements of a gear tooth system or law of gears (Section 1.3), pass through the pitch point C (Figure 2.1/18). All contact points are located on this line because only in this way can the conditions named be fulfilled. For that reason it is called a line of action. The line of action is the geometric location of all contact points of the meshing tooth flanks. For involute teeth it is tangent to the base circles of the paired gears. 62 2 Gear Teeth Geometry Two tangents (lines of action) are possible on the base circles. The one on which flank contact takes place is dependent on the direction in which torque is transmitted. b) Centre distance a , operating pressure angle Įw The reference centre distance ad (Figure 2.1/18) is purely a calculation quantity. ad is equal to the sum of the reference circle radii d1/2 and d2/2: ()d1 2 = / 2ad d + (2.1/35) For an internal gear pair (internal gear - external gear), ad < 0 as well as a < 0, because d2 < 0 and respectively dw2 < 0 (definition in DIN 3960). Here ad corresponds to the actual centre distance a if the sum of the profile shift coefficients x1 + x 2 is equal to zero (see clause f of this section). Then also the operating pressure angle is Įw = Į. If one changes the centre distance of the gear pair, an operating pressure angle of Įw Į results. The following can be read from Figure 2.1/18: db 1 b 2 b 1 wb 2 w = / cos Į / cos Į / cosĮ + / cos Į a r + r ; a = r r From this the correlation between centre distance a and operating pressure angle Įw results: ()wd cosĮ = cos Į aa (2.1/36) In TGL RGW 229-75 the centre distances were standardised (excerpt Table 2.1/4). Table 2.1/4 Centre Distances a (in mm) According to TGL RGW 229-75; Range R 1 Preferred to R 2 R 1 50 63 80 100 125 160 200 250 315 R 2 71 90 112 140 180 224 280 c) Working pitch diameter dw In the transmission of motion, the working pitch circles roll off of one another slip-free. As such, their diameter ratio dw2/dw1 determines the ratio i of this pairing (slip-free friction gears): 21 w2 w1 2 1 12ȦȦ = / = / = / i = dd d d zz −− − (2.1/37) The sum of the working pitch circle radii rw1,2 = dw1,2/2 (Figure 2.1/18) results in the centre distance a: w1 w2 /2 + /2 a = d d (2.1/38) From Equations (2.1/37) and (2.1/38) the working pitch diameter dw1 is determined with w1 1 = 2 / ( )ai d − (2.1/39) and dw2 results in w2 w1 w11 12 = = 2 = / aid a ddi−−− (2.1/40) With Įw = Į the working pitch diameter dw is equal to the reference diameter d. Note: With an external gear pair i < 0; with an internal gear pair dw2 < 0; in Equations (2.1/39) and (2.1/40) i is Stageii= 2.1 Geometry of Spur Gear Teeth 63 Figure 2.1/18 Path of contact, centre distance a and working pressure angle Įw: a) external gear-external gear ( x1 + x2 = 0); b) external gear-external gear ( x1 + x2 > 0); c) internal gear-external gear ( x1 + x2 = 0); d) internal gear-external gear ( x1 + x2 < 0) d) Length of line of contact gĮ, transverse contact ratio İĮ Because of the tip circles, the tooth flanks are limited towards the outside. For this reason, contact of the flanks is possible only in a certain area of the path of contact. This area or respectively length, confined by the points A and E in Figure 2.1/19, is called the length of line of contact g Į. The distance between two adjacent tooth flanks on the line of contact is equal to the normal base pitch pe (see Figure 2.1/14). The relationship between the length of line of contact g Į and the contact pitch p e is called the transverse contact ratio İĮ: ĮĮ e İgp= (2.1/41) Whether or not a following flank pair already makes contact before the end of contact of the previous one, and in this way makes possible an uninterrupted transmission of motion, depends on how long the line of contact is or the value of the transverse contact ratio. The transverse contact ratio İ Į expresses the average number of tooth flank pairs currently in contact. 64 2 Gear Teeth Geometry The contact pitch pe can be determined according to Equation (2.1/19). According to Figure 2.1/19, the length of line of contact gĮ can be determined from the distances 1TE,A T2 and 2 1T T, taking into account the pairing of external gears and of an external gear and internal gear in ()2 Į 12 1 2 2zgT E T A T Tz=+ − Applying the geometric gear quantities for these distances results in () () () () 2 2 2 2 2 a1 b1 a2 b2 w 2 / 2 / 2 + / 2 / 2 sin Į||−− −z = a g dd ddzα (2.1/41a) with Įw according to Equation (2.1/36), da1,2 according to Equation (2.1/24) and db1,2 according to Equation (2.1/17). Note: The following applies for internal gear pairs: a < 0 and z2 < 0 [which is taken into account in Equation (2.1/41a) by z2 / Ňz2Ň]. In the case of tip rounding, instead of da1,2 the usable tip diameter dNa1,2 is to be used [ dNa = da – 2Nan(1 - sin Įan)]. Often İĮ is expressed by the tip and root profile overlap proportions İĮ1,2 in Į a1 a2İİİ=+ (2.1/42) Here, İĮ1 and respectively İĮ2 represent the tip and root path of contact related to the base pitch, which are determined by the beginning A or respectively the end E of the line of contact and their distances to the pitch point C (Figure 2.1/19a, c): Į1 Į2 ee İ = / , İ = / . EC AC p p The tip and root path of contact are () () () ( ) 2 2 2 2 a1 b1 w1 b1 / 2 / 2 / 2 / 2 −− − EC = dd d d () () () ( ) ()2 2 2 2 2 a2 b2 w2 b2 2z = / 2 / 2 / 2 / 2 ||z−− − AC dd d d [In the case of tip rounding, dNa is to be used instead of da; refer to the note for Equation (2.1/41a)]. The ratio of İĮ1/ İĮ2 also expresses the ratio of sliding velocity on the addendum of gear 1 to the sliding velocity on the addendum of gear 2, which must be observed during the design. Calculations show that for the specified tooth heights (ISO 53, DIN 867) and the pressure angle Į = 20 , for external gearings the transverse contact ratio İĮ is less than 1.98. Since a minimum of one pair of flanks is necessary for load transmission and since this should be guaranteed with a certain safety, the following applies for gears produced in practice: 1.1 ≤ İ Į < 1.98.1) However, this range for transverse contact ratio means that there is an alternation between the mesh of one and two pairs of flanks. Looking at Figures 2.1/19a to d, it is evident that there is a • single meshing (one pair of flanks in contact) in the area of B-D and • double meshing (two pairs of flanks in contact) in the areas of A-B and D-E. 1) Įİ< 1 is possible for helical gears with Ȗİ > 1, but mostly not best. 2.1 Geometry of Spur Gear Teeth 65 As such, in the tooth tip and tooth root areas there is double meshing, and in the range of the tooth centre there is single meshing. The larger the transverse contact ratio is, the smaller the single meshing range. The transverse contact ratio is important for noise and stress. The transition from single meshing to double meshing is identified by the points B and D. They are also called single meshing points (Figure 2.1/19e). The determination of İ Į by approximation is possible with the help of Figure 2.1/20. The elastic deformation of the teeth causes the flanks to make contact before and after the meshing points A and E . Pre-meshing and post-meshing are large for a small E-modulus (e.g., plastic gears), and for pairings of internal gears ( z2) with external gears ( z1), with a small difference of 21zz−. Figure 2.1/19 Meshing points and load or respectively stress along the line of contact a) external gear pair; meshing points A, B, D, E ; length of line of contact gĮ = AE b) external gear pair; load per tooth pair, tooth root stress and contact stress 66 2 Gear Teeth Geometry c) Internal gear pair; meshing points A, B, D, E ; length of path of contact g Į = AE d) Internal gear pair; load per tooth pair, tooth root stress and contact stress e) Meshing points of an external gear teeth pairing (double meshing, single meshing, double meshing) 2.1 Geometry of Spur Gear Teeth 67 Figure 2.1/20 Determination of the transverse contact ratio Įİ by approximation Figure 2.1/21 Profile shift and length of tip/root path of contact g a, gf of the rack-gear pairing; a) x = 0; b) x > 0 68 2 Gear Teeth Geometry e) Rack In the limiting case of an infinitely large gear ( z→∞), a straight-flanked tooth profile is produced (Section 2.1.1.2. and Figure 2.1/7). Aside from the importance of the rack as a basic rack, the gear- rack pairing is also used to transform a rotating motion into a straight motion or vice versa. The tooth form of a rack cannot be changed by the profile shift. This assertion applies increasingly with a rising number of teeth of a gear, right up to the limiting case of the rack. The distance change between rack and gear corresponds exactly to the profile shift xm of the gear. The tip shortening factor k results for zero [Equation (2.1/45)] and the operating pressure angle Į w also remains exactly Įw = Į at xm 0, just as the pitch circle of the gear paired with the rack corresponds to the reference circle. Compared to the pitch line of the rack, the profile reference line is shifted by xm. The profile shift drastically changes the tip and root path of contact and with that also the sliding velocity ratios (Figure 2.1/21). Figure 2.1/22 also gives the dimensions of the rack. Figure 2.1/22 Quantities existing on a rack, as limiting values of gear teeth f) Profile shift Figure 2.1/18 shows the great advantage of involute teeth, that the basic requirements of a gear tooth system / law of gears are still met (because i = −rw2 / r w1 = constant), even if the centre distance is changed. This means that portions of the involute outside and inside the reference circle can be used as the tooth flank. A shift in the utilised profile range compared to the normal position is called the profile shift. For that reason, Equations (2.1/21) and (2.1/22) for the addendum h a or respectively dedendum hf, as well as the Equations (2.1/24) and (2.1/25) for the tip diameter da or respectively the root diameter df contain the profile shift coefficient x . When gears are manufactured, the profile shift is realised by moving the tool (depicted by the rack) farther forward or back from the reference circle in the direction of the tip circle by the amount xm (Figure 2.1/23). 2.1 Geometry of Spur Gear Teeth 69 When the reference line of the basic rack, which is imagined as a tool, makes contact with the reference circle, this is an X-zero gearing (gears with no profile shift). When the profile reference line has a distance of xm 0 from the reference circle, this is an X-gearing (X-plus-gearing or X- minus-gearing). If the retraction from the reference circle goes in the direction of the tip circle, the profile shift is positive; otherwise it is negative. The circle which the profile reference line P-P comes in contact with after the profile shift is the X- circle d v: m x d d 2v + = Figure 2.1/23 Profile shift by shifting the tool: a) external gear x = 0; b) external gear x > 0; c) external gear x < 0; d) internal gear x = 0; e) internal gear x < 0; f) internal gear x > 0 g) Tooth thickness s at the reference circle In the rack-gear pairing, even with xm 0 (and constant profile angle of the rack), the reference circle (of the gear) remains the pitch circle. Only in this way the perpendicular on the flank of the rack can be tangent with the base circle of the gear. Because of the slip-free rolling, the tooth thickness s on the pitch circle, which in this case corresponds to the reference circle, is equal to the space width of the tool. From Figure 2.1/24 the following results for the tooth thickness s on the reference circle of the gear: 22 t a n Į s = p/ + m x In this equation, if the reference circle pitch p according to Equation (2.1/15) is used for the tooth thickness s on the reference circle, one arrives at ()ʌ22 t a n Į s = m / + x (2.1/43) 70 2 Gear Teeth Geometry The tooth thickness s is the distance of the profile of the right and left flank of a tooth on the reference cylinder (arc length!). h) Necessary sum of profile shift coefficients (x1 + x 2) An issue that needs to be resolved for centre distance alterations is how large the sum of the profile shift coefficients must be to retain zero-**backlash** tooth meshing.1) Since the working pitch circles roll off one another slip-free, for zero-**backlash** meshing the sum of the tooth thicknesses of the paired gears must be equal to the working pitch p w (refer to Figure 2.1/25): w1 w2 w ss p+ = (2.1/44a) Figure 2.1/24 Tooth thickness s on the reference circle a) External gear (- rack) b) Internal gear (- virtual rack) For sy = s w according to Equation (2.1/26), with s according to Equation (2.1/43), the result with dy = d w and ywĮ=Į is 1,2 w1,2 w 1,2w1,2ʌ 2 + 2 tan Į = inv Įinv Įx + dzs − (2.1/44b) The working pitch pw results from the circular pitch angle p/(d/2) for pw = p dw/d, and with d = d b /cosĮ, dw = db /coswĮ one arrives at w wcosĮ cosĮpp= (2.1/44c) 1) To avoid clamping, **backlash** is necessary. This is subsequently realised by a **backlash** allowance for the tooth thickness or respectively span. Whether this is similar to the addendum modification (e.g., hob cutter) or purely a tooth thickness allowance is done depends on the manufacturing method (Appendices 3 and 4). 2.1 Geometry of Spur Gear Teeth 71 Applying Equations (2.1/44c) and (2.1/44b) and p = mʌ in Equation (2.1/44a), the result is the sum of the profile shift coefficients: w 12 122invĮ - inv Į()tanĮ + = + xx zz (2.1/44) The working pressure angle Įw is determined according to Equation (2.1/36). For inv Į = inv 20 the result is inv 20 = 0.0149043. Equation (2.1/44) specifies which sum of the profile shift coefficients is necessary for zero-**backlash** tooth meshing in the case of a centre distance that deviates from the sum of the reference circle radii. As a result, the centre distance a ad has an angle of Įw Į, which is accounted for in Equation (2.1/44) by inv Įw. Table 2.1/1 has information for inv Į, (Įy = Į w). If (x1 + x2) 0, then the pairing is named X-gearing. On the other hand, however, if ( x1 + x 2) = 0 and 12xx=− , the gearing is named X-zero-gearing . With ( x1 + x2) = 0 and x1 = x 2 = 0 the name equal addendum teeth or zero-gearing is common . Figure 2.1/25 Tooth thicknesses sw1,2 on the working pitch circle: a) external gear pair; b) internal gear pair i) Tip shortening factor If one compares the increase of centre distance ǻa = a - a d 0 with the sum of the increase of the tip and root circle radii ( x1 + x 2) m, for external gear teeth one discovers that ( x1 + x 2) m > (a - a d). With constant tooth height this would make the tip clearance c smaller, and there would be a risk of meshing interference because of contact of the tip edge point in the root fillet of the mating gear (interference). For that reason, one shortens tooth tips by Ň(k ⋅m)Ň and with that, keeps the common tip clearance (usually c = 0.25 m) (refer to Equation 2.1/24). On the other hand, in the case of internal gear pairs, with ( x1 + x2) 0 one notices an enlargement of the tip clearance. Taking the risk of interference into consideration, for this one usually sets k = 0. Here, k is called the tip shortening factor . Corresponding to the stated considerations, it is calculated according to Equation (2.1/45):1) d 12()aakx xm−=− + (2.1/45) With (x 1 + x 2) 0, according to Equation (2.1/45) the tip shortening factor k for external gear teeth is always negative, which corresponds to tip shortening. 1) Deviating from this, in DIN 3960 (1987) k is defined as an absolute quantity, i.e., d1 2 n() ( )ka a m x x=− − + . 72 2 Gear Teeth Geometry However, with the internal gear - external gear pairing, with ( x1 + x2) 0 it always comes to k > 0. With internal gears one usually omits an increase of the tooth height and sets k = 0. The tip clearance can easily be checked with the centre distance and the tip and root diameters [Equation (2.2/42)]. j) Effects of profile shift; undercut Profile shift influences: • Tooth thickness and tooth form, • Radii of curvature of the tooth r oot fillet (flank / root circle), • Transverse contact ratio or respectively position of the single meshing points, • Operating pressure angle, • Sliding velocities, specific sliding, gear losses, and • Load capacity. With an increasing profile shift coefficient x, with external gear teeth the tooth root thickness grows, while the top land (with k = 0) shrinks. Figure 2.1/26 shows teeth with different profile shifts, which differ in terms of tooth thickness and in the use of different portions of the involute. Table 2.1/5 provides an overview of tooth forms, dependent on z and x . Figure 2.1/26 Change of tooth form caused by profile shift a) external gears; z = 10; tooth 1: x = +0.5; tooth 2: x = 0; tooth 3: x = –0.5 b) internal gears; z = –50; tooth 1: x = +1.5; tooth 2: x = 0; tooth 3: x = +0.5 For external gear teeth, an undercut may occur with small numbers of teeth and / or a negative profile shift (Figure 2.1/26a). Because of the undercut, the tooth root thickness can be considerably reduced, and a significant portion of the involut e (Figures 2.1/27a and d) can be cut off. Figures 2.1/27b and 2.1/27e serve to derive the condition of the undercut. Undercuts can only exist in external teeth and occur when the radii of the generated profile have two instead of one meshing point with the cutting edge of the tool that is generating the involute (Figure 2.1/27b, e). This is the case when, viewed from the pitch point C in the direction of the point T, the meshing points of the straight-flanked part of the too l, or respectively the involute part, still have meshing points beyond (below) T. When the point of intersection F coincides with T, this is a limiting case. From this, for tools with a basic rack-like shape (hob cutters, rack-type cutters), it can be concluded that an undercut can be avoided if the condition according to Equation (2.1/46) is met; () Fa0 a0 a01s i nĮ hh=− −N , \* Fa0h=Fa0m h ; usually \* Fa0h=1. 2.1 Geometry of Spur Gear Teeth 73 \*2 Fa00 sinĮ2zx h≥− (2.1/46) Here, h\* Fa0 describes the addendum coefficient for the straight part of the gear tool (rack). Table 2.1/5 Tooth Forms of External Cylindrical Gears, Dependent on the Number of Teeth and the Profile Shift, Basic Rack ISO 53 (DIN 867) with Tip Radius Na0 = 0.38 m, Hob or Rack-Type Cutter 74 2 Gear Teeth Geometry Gear teeth according to ISO 53 (DIN 867) can be manufactured without an undercut with x = 0 and z ≥ 17 if the tool addendum is ha0 = 1.25 m and the tool tip radius is Na0 = 0.38 m (h\*Fa0 = 1). For that reason, zmin = 17 is called the theoretical limiting number of teeth . With zƍmin < 14 and x = 0 the undercut becomes important (practical limiting number of teeth) . When pinion-type cutters are used, Equation (2.1/46) does not (exactly) apply. Figure 2.1/27 Transition involute-root fillet: a) root form circle radius rFf with external gear – rack-like tool (hob, rack-type cutter); no undercut; b) undercut with external gear – rack-like tool (hob, rack-type cutter); meshing beyond T; c) enlarged detail of the rack-like tool in the tooth tip area From Figure 2.1/27e the condition to avoid an undercut results in 2 2 a0 b0 0 w0 - sin Į rar ≤ (2.1/47) Here the centre distance of the generating gear pair a0 is 0 02() z m za = + and the transverse pressure angle at generation Įw0 is 0 w0 02 inv Į = tan ĮinvĮxx + zz+ + Overall, the issues among shaper cutters are somewhat more complicated because here the number of teeth and the profile shift of the tool (index 0) must also be taken into consideration. For that reason, Equation (2.1/46) is only an orientation. 2.1 Geometry of Spur Gear Teeth 75 Figure 2.1/27 d) Dedendum form circle radius rFf, e) Dedendum form circle radius rFf = ru, no undercut with external gear – shaper cutter; undercut with external gear – shaper cutter; meshing before T (inside TC) meshing beyond T (outside TC) Table 2.1/6 lists the theoretical limiting number of teeth for production with a hob or rack-type cutter for different pressure angles and whole depth factors for spur gears and helical gears (refer to Section 2.2). Table 2.1/6 Theoretical Limiting Number of Teeth zmin to Avoid an Undercut with x = 0 for Different Pressure Angles, Helix Angles and Addendum Factors of the Straight-Flanked Portion for Rack-Like Tools (Hob, Rack-Type Cutter) Fa0 nh m Pressure angle Įn 15 17.5 20 25 Helix angle 0 15 30 45 0 15 30 45 0 15 30 45 0 15 30 45 Number of teeth 0.8 23.9 21.6 15.9 9.0 17.7 16.1 11.8 6.8 13.7 12.4 9.2 5.4 9.0 8.2 6.2 3.7 1.0 29.9 27.0 19.8 11.3 22.1 20.1 14.8 8.5 17.1 15.5 11.5 6.8 11.2 10.2 7.7 4.7 1.2 35.8 32.4 23.8 13.5 26.5 24.1 17.8 10.2 20.5 18.6 13.8 8.1 13.4 12.3 9.2 5.6 1.4 41.8 37.9 27.8 15.8 30.9 28.1 20.7 11.9 23.9 21.7 16.1 9.5 15.7 14.3 10.8 6.5 Note: The real limiting number of teeth corresponds to the respective next larger integer value. It should be noted that the radial infeed of the tool to achieve the **backlash** allowance amounts to a negative component of the profile shift, which, in limiting cases, can shift the undercut limit for external gears. The component of profile shift coefficient ǻx, which is to be superimposed on the calculated nominal quantity x of the respective gear, leads to the generating profile shift coefficient x E (see Attachment 4 and DIN 3960) and is calculated according to 1,2 1,2( 2 tan Į)ΔsEx = m (2.1/48) The profile shift also has a significant influence on the radius of curvature Nf of the tooth root fillet, and thus also the stress concentration effect . With an increasing profile shift coefficient x up to x 1, the radius of curvature of the tooth root fillet decreases to the radius of the tool tip radius, to then begin to rise again. With increasing profile shift and with 76 2 Gear Teeth Geometry tip shortening to a constant tip clearance, the transverse contact ratio drops. The outer single meshing point advances closer to the tooth tip. Since, on the other hand, the tooth root becomes thicker and the radii of curvature of the tooth root fillet drop, it is not possible to immediately detect the profile shift’s influence on the load capacity of the tooth root (bending stress). However, numerical calculations show that with an increasing and favourably assigned profile shift, the nominal tooth root bending stress drops, and with that, a higher static capacity is achieved. With an increasing number of teeth this effect declines. In cases of high load cycle numbers, especially in high-cycle fatigue, the notch stress takes effect, and an effective increase in the load capacity of the tooth root with an increasing sum of the profile shift only occurs with smaller numbers of teeth (up to roughly z = 15). For external gear teeth, an increasing sum of the profile shift coefficients causes an increasing operating pressure angle Į w, but with an external gear-internal gear pairing, Įw drops. Since the radii of curvature at the pitch point amounts to NC1,2 = rb1,2 tan Įw, and these are decisive for the contact stress quantity (Section 6.5.1.2), with an increasing sum of the profile shift and favourable splitting, this kind of stress drops. ISO TR 4467 (DIN 3992) provides recommendations for the selection of the sum of the profile shift coefficients (x1 + x2) and its splitting on paired gears. In gear systems with external gears produced in practice, -0.5 ≤ (x1 + x 2) ≤ 1.5 is usually the case. In production using a typical hobbing process, there are no extra costs because of profile shift. Section 7.3 provides more specific information for determining the profile shift. 2.2 Geometry of Helical Gear Teeth 2.2.1 Generation and Basic Characteristics The teeth of a rack with helical gearing, which we will imagine here as a tool, are inclined by the angle towards the axial direction (Figure 2.2/1a). In Figure 2.2/1b the involute helicoid is depicted with the base cylinder and the generating plane. Figure 2.2/1a Generation of the helical tooth While rolling the pitch plane on the pitch cylinder, a helical tooth is generated also on the mating gear. The three-dimensionally curvilinear gearing generated in the process can also be imagined as being composed of many flat profiles made up of sections perpendicular to the axis (transverse sections). Figure 2.2/2 illustrates this. Different transverse sections, for example, A-A, B-B, and C-C , result in different projections of the section trace with the flank of the helical-toothed rack. On each section the involute can be determined according to the enveloping curve construction. 2.2 Geometry of Helical Gear Teeth 77 Figure 2.2/1b Involute helicoid The relative positions of the profiles on the intersecting planes are determined by the inclination of the generation profile (flank of a rack with helical gearing) in the axial direction. On each of these planes, the contact point of the paired flanks is determined by the normal NN to the tooth surface profile of the rack, which is tangent to the base cylinder of the wheel at point T (Figures 2.2/1 and 2.2/2). All normals that are tangent to the base circle are on one plane. Analogous to the line of ac-tion (Section 2.1.2), this plane is called the plane of action (Figure 2.2/1). It intersects the tooth flank of the rack and the gear in a straight line. Based on the considerations above, the paired tooth flanks have contact on it (law of gears; refer to Section 1.3). Figure 2.2/2 Helical gearing in three transverse sections (see points B1, B2, B3 in Figure 2.2/1) The line on which the paired flanks are in a line-like contact (in Figure 2.2/1, line B 1, B2, B3) is called the line of contact (refer to Figure 2.2/3; line B). If several flank pairs are meshing simultaneously, several lines of contact exist too. Figure 2.2/3 shows these lines and the field of action for a certain roll position on a helical gear. The field of action is the area of the plane of action delimited by the face width and the length of the path of contact. The position of the lines of contact in the field of action is dependent on the contact position, or respectively the rotating angle. 78 2 Gear Teeth Geometry Figure 2.2/3 Lines of contact ( B), field of action ( E) and tooth traces ( F) for helical gearing Even when, instead of a rack-gear pair, two helical gears are paired, the tooth flanks are in contact in straight lines – the lines of contact. This is recognisable when one imagines a rack whose contour has a thickness close to zero (for example, with a thin sheet) between the paired teeth. Analogous to a raised thread in the case of spur gearing, here the flank of a helical tooth can be imagined as being formed by a belt, cropped diagonally, uncoiled, and pulled taut. In Figure 2.2/1a, the line 13BB represents the end of this belt, which in 13TT merges into the base cylinder. When unwinding and winding up on planes perpendicular to the axis, each point on the line13BB , which represents the line of contact, defines an involute in transverse sections. In the transverse section – and only there – the tooth profile of helical gearing is an involute. If we imagine it is wound up, the line B1B2 or the end of the diagonally cropped belt, respectively, results in a helix. Figure 2.2/4 depicts how this line is formed and, using external gears as an example, illustrates the terms left-handed and right-handed. A gearing is right-handed if the flanks run upwards to the right on a gear laid on its end face. Figure 2.2/4 The tooth trace as a component of a helix: a) unwinding of a diagonally cropped belt from a reference cylinder Figure 2.2/4 b) Tooth traces on an arbitrary cylinder and unwinding of a tooth trace 2.2 Geometry of Helical Gear Teeth 79 In the generalisation in favour of internal gearing, it is necessary to be more precise. In order to recognise or determine the hand of lead the proper line of sight for both external and internal gearing is always from the helix (gearing) in the direction of the gear axis. Based on this rule, external and internal gearing with the same hand of lead and with the same quantitative dimensions, such as coupling teeth similar to nuts and bolts, can be paired (spline connection). Furthermore it can be said: In the case of right-handed gear teeth, for external gearing the helix angle has a positive sign and for internal gearing a negative sign. For left-hand gearing the inverse sign applies. Figure 2.2/5 illustrates this rule. Based on this sign rule, the following applies for cylindrical gear pairings (external and internal gear pairings): 12ȕ+ ȕ0= (2.2/1) The lead must be constant, even when the cylinder diameters of a gear are different. As such, on the tip and base cylinder, for example, tooth traces exist that are characterised by a helix with the same lead p z, but different lead or helix angles ( ȕy, ȕa or ȕb). The tooth traces on helical gearing are helixes. For a cylindrical gear pairing with helical teeth and parallel axes, only gears with the same quantitative helix angle , but different prefixes, can be paired. This means that for an external gear pair the paired gears have different hands of lead, and for an internal gear pair they have the same hands of lead, with the same quantitative helix angle for each. Figure 2.2/5 Example of the definition for the hand of the lead and the sign of the helix angle: a) left-hand external gearing ( < 0); b) right-hand external gearing ( > 0); c) left-hand cylindrical gear with internal gearing; d) rack; arrow direction: line of sight for definition of hand of lead d) 80 2 Gear Teeth Geometry Based on the fact that the lead of a gear in the form of a helix is constant on any cylinder of any diameter, it is possible to deduce the respective helix angle. According to Figure 2.2/4a, the lead pz can be deduced for a known helix angle ȕ on the reference cylinder with px as the axial pitch (Equation 2.2/10) in zx = tandp zpπ=β (2.2/2) If d is generally replaced by dy in Equation (2.2/2) and ȕ generally by ȕy, one arrives at y y z|| tan | | = d pπβ or, with p z expressed by Equation (2.2/2), for the helix angle ȕ y, in an arbitrary diameter d y, the following results: y y tan = tan d dβ β (2.2/3) The following applies in particular for the helix angle in the base diameter : b b tan = tan d dβ β (2.2/4) or respectively bttanȕ cosĮtanȕ = (2.2/4a) and for the helix angle in the tip diameter a a tan = tan d dβ β (2.2/5) If one uses y yn ybnz = cos Į cos ȕm d and nbn cos Į cos ȕzmd = , from the parity of the leads on different diameters of a helix, one arrives at bn bn z yn n yy|z| ʌʌ =cosĮ cos tan | cos Į cos ȕ tan |ȕ ȕȕ|z| mm = p|| and the helix angle ȕy on an arbitrary cylinder diameter dy in n y ynsinȕcosĮsin = ȕcosĮ⋅ (2.2/6) 2.2 Geometry of Helical Gear Teeth 81 and particularly the helix angle ȕb in the base circle n b sin = sin ȕcos ȕ Į ⋅ (2.2/7) Table 2.2/1 contains a summary of the equations for the helix angles ( y, b). Because of the helical shape of the tooth traces, for helical gears these do not immediately engage and disengage across the full face width. This has a beneficial effect on vi-bration characteristics and the ge-neration of noise. With a similar quality of production and operating conditions, compared to spur gears, helical gears are quieter. If the manufacturing deviations are not too significant, they also provide slightly more load capacity than do spur gears with the same main dimensions (d, b, m). Their disadvantage is the occurrence of axial forces. As a result of the me-shing of several flank pairs, they also react more sensitively to manu-facturing deviations than do spur gears. The advantages of helical gearing compared to spur gearing can only come to the fore when the gear is so wide that the mutual position of the tooth flank profiles differs sufficiently on both gear end faces. A dimension of this is the ratio İ ȕ of facewidth b to axial pitch px (refer to Figure 2.2/6a), the overlap ratio: ȕ xİbp= (2.2/8) Table 2.2/1 Helix Angle, Pressure Angle and Modules for a Helical Gear With the parameters for p x [Equation (2.2/10)], the following can also be written for the overlap ratio: ȕ nsinȕİʌb m⋅= (2.2/9) 82 2 Gear Teeth Geometry The axial pitch can be determined, in accordance with Figure 2.2/6a, with xnʌsinȕ pm= (2.2/10) In the case of spur gearing, because ȕ = 0, İȕ = 0 is also true. It is best to design the overlap ratio of helical gearing with integer values and the largest possible ( İȕ =1, 2, 3, ...). This ensures that the variation in the sum of the lengths of lines of contact is minimal. Because of limits to the face width and helix angle, or axial force respectively, which can considerably reduce the lifetime of the roller bearings, often it is also necessary to apply values of İ ȕ which are considerably below one. With helical gearing, th e lines of contact follow a diagonal path over the plane of contact (Figures 2.2/3, 2.2/7a, and 2.2/8). The total length Ȉ lj of the lines of contact is generally not constant, but instead dependent on the rotating angle. It is clearly visible in the decreasing variation in the sum of the lengths of lines of contact, which goes hand in hand with increasing overlap ratio. For spur gearing ( İ ȕ = 0) the variation is especially large. At İȕ = 1 it achieves a constant value and an initial minimum. With İ ȕ > 1 the variation initially begins to rise again, without reaching the value for İȕ = 0, and at İȕ = 2, it would show the next constant size. The variation of the lengths of lines of contact is a factor that influences the excitation of vibration and noise and demonstrates the advantage of designing with integer numbers for overlap ratio İ ȕ. With İȕ ≥ 2 (1.5), the influence that still exists is only minimal. Figure 2.2/7b shows the quantity jmin jmin b = cos ȕ lbZlΣΣ (2.2/11) derived from the minimum lengths of the lines of contact, as a function of transverse and overlap ratio. Figure 2.2/6 Helical gearing; a) helical external gear with rack (basic rack) 2.2 Geometry of Helical Gear Teeth 83 Figure 2.2/6 b) helical gear (with external gearing); c) helical internal gear The sum of the transverse contact ratio İ Į and overlap ratio İȕ is called total contact ratio İȖ: ȖĮȕ = + İİİ (2.2/12) A continuous transmission of motion is possible with İȖ ≥ 1. Concerning vibration considerations, İȖ ≥ 2.5 has proven favourable. 84 2 Gear Teeth Geometry Figure 2.2/7a Variation in the sum of lengths of the lines of contact ( Ȉlj = l1 + l2 + l3 + ...) Figure 2.2/7b Related sum of the minimum of the lengths of the lines of contact [Equation (2.2/11)] as a function of transverse contact ratio and overlap ratio [according to TGL 10545 (1963 edition)] 2.2 Geometry of Helical Gear Teeth 85 2.2.2 Basic Quantities for Gearing of a Cylindrical Gear – Helical Gearing The following section focuses especially on calculating and presenting the quantities that are special features of helical gearing. As can be seen in Figure 2.2/6a, helical gearing can be observed on a plane perpendicular to the axis – transverse section S-S – and on a plane perpendicular to the tooth traces in the reference cylinder, or the pitch plane, respectively – the normal section N-N. The tooth rack profile is specified as a basic rack in the normal section because this way the tools can also be used for any helix angle. It corresponds exactly to the profile already given for spur gearing. Figure 2.2/8 illustrates the relationships. The standardised basic rack ISO 53 (DIN 867) is used for the normal section. In order to differentiate between the two, the index n is used in the normal section and the index t in the transverse section. As such, one differentiates among others between in the normal section in the transverse section mn normal module mt transverse module n normal pressure angle t transverse pressure angle pn normal pitch pt transverse pitch Table 2.1/3 provides standardised values for the normal module mn = m according to ISO 54 (DIN 780, excerpt). In the transverse section and the normal section, the tooth flank profiles have different inclinations, that is, different pressure angles, as shown in Figure 2.2/8 for the tooth of the rack. The pressure angle in the transverse section Į t on the reference cylinder results from the standardised pressure angle Įn = 20 in the normal section with the relationships which can be understood from Figure 2.2/9: t nn tn t2 2 tanĮ = , tan Į = , = cos ȕs s ssHH in n ttanĮtanĮ = cosȕ (2.2/13) Since tooth flanks are involutes in sections perpendicular to the axis, helical gearing in the transverse section can be considered as spur gearing with the pressure angle Bt, the pitch p t = mtʌ and with a tooth height of h ≥ 2.25m n that corresponds to the basic rack. Similarly, in reference to an arbitrary diameter dy, the following is true (refer to Table 2.2/1): yn yt ytanĮtanĮ = cosȕ (2.2/14) DIN 3978 provides recommendations for the size of the helix angle ȕ on the reference cylinder. For (single) helical gearing, ȕ is usually in the range of 8 ≤ |ȕ| ≤ 20 . For double helical gearing (refer to Figure 1/21c), angles larger than 30 are common, since here the axial forces in the gear compensate for one another. The transversal pitch p t and normal pitch p n on the reference circle are (Figure 2.2/6a) ttpm=π (2.2/15) 86 2 Gear Teeth Geometry nnʌ pm= (2.2/16) Since pn = pt cosȕ (Figure 2.2/6a), with Equations (2.2/15) and (2.2/16) the transverse module results in tn cosȕ mm= (2.2/17) Figure 2.2/8 Transverse section and normal section on helical gear and rack Figure 2.2/9 Transverse pressure angle and normal pressure angle The reference diameter d is therefore (refer to Figures 2.2/6b and c) t dz m= (2.2/18) or respectively cos ȕnzmd = (2.2/19) 2.2 Geometry of Helical Gear Teeth 87 The base diameter d b is calculated by the reference diameter d and the transverse pressure angle Įt b t cosĮ dd= (2.2/20) or respectively after applying the quantities for the reference diameter in accordance with Equation (2.2/19) nt bcosĮ= cos ȕzmd (2.2/21) The tip diameter d a is calculated by the reference diameter and the basic rack that applies for the normal section, taking the profile shift ( xmn) and tip shortening ( kmn) into consideration: aP n na = + 2 + + hdx k dmm (2.2/22a) The root diameter d f is naP fn = 2 + hdd m x cm∗ −− (2.2/22b) where c\* is the tip clearance factor (c\* = c/m n) and can be found in the range c\* = 0.1 to 0.3, according to ISO 53 (DIN 867). Usually c\* = 0.25 is given preference. In exceptional cases, c\* is an amount up to more than 0.4 (as in the case of a protuberance). In accordance with ISO 53 (DIN 867), haP/mn = 1 applies for the standardised basic rack. Since the tooth tip and the tooth root are not directly involved in flank contact, these quantities are usually allowed a larger range of tolerance in the production drawings and – if the tip diameter d a is not meant as technological basis – the tip cylinder envelope surface is only roughly machined. The tolerance value for d a is often h9 (or H9 respectively) and rougher. The root diameter df is often not specified because it is automatically generated during the manufacturing process, for example hobbing. In the case of larger gears, through heat treatment (case hardening in particular), considerable “growth” or “shrinkage” can occur, which makes an in clusion of this diameter ( d f) in the design, as a condition for dimensional control, recommendable. Other things that have an influence on the root diameter are the **backlash** allowances because they are usually considered by tool advance or retraction. Additionally, in the case of production using the shaper cutter, the root diameter is also dependent on the number of teeth and the profile shift of the shaper cutter and the workpiece. For helical gearing, the pressure angle Į t > Į = 20 with ȕ 0 results in an undercut limit that is shifted in the direction of the lower number of teeth (Table 2.1/6). If it is possible to approximate the tooth form of helical gearing using equivalent spur gearing (virtual spur gearing), it can be simpler to specify certain geometric quantities, especially for the calculation of load capacity, because there is one parameter less (the helix angle). The approxi-mation succeeds with the help of the curvature of the ellipse created on the plane normal to the tooth flanks (Figure 2.2/10). For this, first the reference cylinder is selected. The large half-axis A of the ellipse is A = d/(2cos ȕ) and the small half-axis B is B = d/2. The radius of curvature Nn = A2/B is in point C. 88 2 Gear Teeth Geometry Figure 2.2/10 Radius of curvature of the equivalent spur gearing (virtual spur gearing) With the relationships given above for A and B the radius of curvature is n 2 = 2cosȕdN If one uses 2 'n = dn, with Equation (2.2/19), the virtual reference diameter results on the normal section in β cos z = 3n nmd (2.2/23) This can also be expressed in the normal section, dependent on the virtual number of teeth z v of the equivalent gear . Therefore the following applies: v3cosȕzz= (2.2/24) Here, zv is the number of teeth of the imagined equivalent spur gearing, which approximates the tooth form of the helical gear in the normal section. One arrives at an only slightly different resulting relationship if, instead of the reference diameter, the base diameter is used as the starting point. Derived similarly, the result is the number of teeth of the equivalent spur gearing , which is identified by z nx, in order to distinguish between it and the value given in Equati on (2.2/24), with nx v 2 b = cos cos ȕȕzzz ≈ (2.2/25) One also arrives at the number of teeth given in Equatio n (2.2/25) if one assumes that in the case of a profile shift coefficient equal to zero ( x = 0) the undercut limit of the helical gear is expressed by the number of teeth of the equivalent spur gear [2], that is, zmin = znx = 17 for gears that correspond to ISO 53 (DIN 867). Table 2.2/2 contains a list for calculating important geometric quantities for spur and helical gearings. Additional information for some diameter-dependent quantities: If one assumes that an arbitrary diameter dy is the product of the transverse module myt and the number of teeth yt yt = zm d (2.2/26) then for this the general module s yt yt = dmz (2.2/27) yn yt = cos ȕ mm (2.2/28) and the specific modules for the base cylinder d b can be defined: t b tt nbtcos Į cos ȕ = = cos Į = dmm mz (2.2/29) b t b bn bt ncos ȕcos cos ȕ Įcos ȕ = = mm m (2.2/30) 2.2 Geometry of Helical Gear Teeth 89 It is also possible to determine the pitch on an arbitrary diameter dy y yt yt t = = ʌdm ppd (2.2/31) yn yn yt y = cos = ʌȕ m pp (2.2/32) Particularly in the case of the base cylinder the pitches on the transverse and normal section, or normal base pitches respectively, are t bt n bte tcos Į cosȕ = ʌ = ʌ = p p mm (2.2/33) b bn t n bne ncos ȕ = ʌ = ʌ cosĮ = cos ȕmm p p (2.2/34) The tooth thickness syt on an arbitrary cylinder on the transverse section is [in analogy to Equation (2.1/26) and Figure 2.1/16] t yt y ty t = + inv Į invĮssdd − (2.2/35) with tt nʌ = + 2 tan Į2sm x bn n t yt t ytanĮcosĮ = ; = ; tan Į = cos ȕ cos ȕdm md and syt is the tooth thickness as arc length. From this the tooth thickness syn on the normal section results as the arc length yn yt y = cosȕ ss (2.2/36) y y tan = tan ȕ ȕd d The normal chordal tooth thicknesses on the transverse and normal sections on the Y-cylinder are yy yt ȥsind s= (2.2/37) with yt yytȥ = ds (d y for transverse section) y yn yn ȥsind s= (2.2/38) with y 3 yn yn yt y 2 y = and = ȥȥ ȕ cos ȕcosdd The height from the tip cylinder to the chordal dimension (on the diameter d y) is ()yy t ay 1c o sȥ 0.5 hh d =−+ (2.2/38a) Particularly for the normal chordal tooth thickness on the reference cylinder, the result is nnsinȥ sd= (2.2/39) with dn = d/cos2ȕ and ȥn = ȥt cos3ȕ. The height from the tip cylinder to the chordal dimension on the reference cylinder is ()an an 1c o sȥ 0.5 hh d =−+ (2.2/39a) 90 2 Gear Teeth Geometry 2.2.3 Geometry of Gear Pairing – Helical Gearing The profile shift as a measure of the tool advancing or retracting is expressed as the product of the coefficient x with the normal module mn; this means that the profile shift coefficient x refers to the normal module , or the normal section, respectively. In analogy to spur gearing, Equation (2.1/44), for helical gearing on the transverse section (index t) with the pressure angle Bt the result is the sum of the profile shift coefficients ( xt1 + x t2), which initially references the transverse module mt, assuming swt1 + s wt2 = p wt and cosĮwt = (ad/a)cosĮt in ()12 t1 t2 w t t t = invĮ invĮ 2 tanĮzzxx++− From the tool advancing and retracting, ( xt1 + x t2)mt, which can also be expressed using the quanti- ties which reference the normal section ( xn1 + xn2) mn, follows (x t1 + xt2) mt = (xn1 + x n2) mn. With mn/mt = cosȕ, tanĮt = tanĮn/cosȕ, and simplified xn1,2 = x1,2, the result is () () 12 wt t 12 ninvĮ invĮ + + = 2 tanĮzzxx−⋅ (2.2/40) By definition, in an pairing of an internal gear with an external gear, ( x1 + x2) > 0 means a reduction of the absolute value for the centre distance, whereas for an external gear pairing ( x1 + x2) > 0 means an enlargement. With respect to the definition, the following naturally also applies in the case of helical gearing for both external as well as internal gears: With a shift of the tool from the reference circle in the direction of the tooth tip, the profile shift modification is positive (x > 0); with a shift in the opposite direction, the profile shift is negative (x < 0). By definition, for external gearing, x > 0 means an enlargement of the tip and root diameter; for internal gearing, on the other hand, it means a reduction of the absolute value of these quantities. The tip shortening coefficient k is d 12 n ( + )aak = x x m−− (2.2/41) It results for the external–external gear pairing with (x 1 + x 2) 0 in k < 0, and for the internal– external gear pairing with ( x1 + x2) 0 in k > 0. So k < 0 means tip shortening. Nevertheless, in the case of the internal–external gear pairing, for a computed result of k > 0 as per Equation (2.2/41), with regard to the interference problems (refer to Section 3.1), usually k = 0 is used, and with that an enlargement of the tooth depth can be avoided and an enlarged tip clearance c is achieved. For (x1 + x 2) = 0 is of course a = a d and k = 0. The tip clearance necessary to prevent meshing interference can be checked in accordance with Equation (2.2/42), if the tip and root diameters and the centre distance a are given: f 2, 1 1,2 a1,2(/ 2 / 2 ) c a d d =− + (2.2/42) 2.2 Geometry of Helical Gear Teeth 91 In analogy to spur gearing (Figure 2.2/7a), the length of path of contact gB, which delimits the field of action on the transverse section, isĮ = AEg : 2 2 2 2 2 a1 b1 a2 b2 wt Į 2 + s i n Į22 | | 22dd dd z = a g z −− − (2.2/43) In the case of tip chamfering, in Equation (2.2/43) instead of da1,2 the usable tip diameters dNa1,2 are to be used. The transverse contact ratio İĮ that applies for the transverse section is Į Į etİg = p (2.2/44) with the transverse normal base pitch nt etʌ cos Įcosȕ = m / p and gĮ in accordance with Equation (2.2/43). The transverse contact ratio for the normal section can be calculated as a virtual quantity with the number of teeth of the equivalent spur gearing, whereby the Equations (2.1/41) and (2.1/41a) specified for spur gearing apply. However, it does not characterise the meshing behaviour of helical gearing. For the length of the path of contact of helical gearing in the transverse section, in principle the points of single tooth contact B, D can also be determined. However, they do not have the same meaning as in spur gearing, since because of the lines of contact running diagonally over the flank – Figures 2.2/7a and 2.2/11 – there is no sudden change of load between one tooth pair and two tooth pairs on these points. This is also represented in the total length of the lines of contact with İ β > 0 compared to İβ = 0 (Figure 2.2/7a). With a total contact ratio of İγ = İα + İβ ≥ 1, a smooth (continuous) transmission of motion can take place even with İα<1, since – as observed in the field of action – one flank pair is always in contact. For an undercut, the considerations analogous to those which are discussed in Section 2.1.2 for spur gearing apply. They result in Equations (3.3/7) or (3.3/9). Table 2.1/6 lists the results for a minimum number of teeth. Table 2.2/2 Geometric Quantities according to DIN3960 (1987) ; ( Note: For internal gear pairs the number of teeth and the diameters of the internal gear are negative. The centre distance of the external–internal gear pairing is negative.) Cons. No. Designation Symbol Equation 1 Virtual number of teeth, Number of teeth of equivalent gear zn n23 b = cos ȕ ȕ ȕcos coszzz ≈ 2 Transverse module mt n t = cos ȕmm 3 Normal pitch pn n nt = ʌ = cos ȕ mpp 4 Reference circle pitch pt n t tʌ ʌ = ʌ = = cos ȕm dmpz 92 2 Gear Teeth Geometry Table 2.2/2 (continued) Cons. No. Designation Symbol Equation 5 Axial pitch px n x ʌ = sin |ȕ|mp 6 Lead pz n z|z| ʌ = sin |ȕ|mp 7 Base pitch pbt n tt bt t ʌ = cos Į = cos Įcos ȕmpp 8 Normal base pitch pen n nn en et b n = cos = cos Į = ʌ cos Į ȕ pp p m 9 Transverse normal base pitch pet b tt et t ʌ = = cos Į = ʌ cosĮtdm ppz 10 Transverse pressure angle tĮ n ttan ĮtanĮ = cos ȕ 11 Helix angle on the base cylinder bȕ n b sin = sin ȕ cos Įȕ 12 Reference diameter d n cos ȕzmd = 13 Base diameter db b t = cos Į dd 14 Tip diameter da aP an n= + 2 + + hdd m x k m 15 Root diameter df aP fn nn = 2 + c hdd m x mm −− 16 Operating pitch diameter dw1 ww12 w1 2 112 = ; = 2 aad ddz z− + 17 Tooth thickness in the reference circle on the transverse section (arc) st ()tn t t tn = / 2 + 2 tan Į = ʌ / 2 + 2 tan Įsx mp mx 18 Tooth thickness in the reference circle on the normal section (arc) sn ()nn n n nn = / 2 + 2 tan Į = ʌ /2 + 2 t a n Įsp x m mx 19 Reference centre distance ad ()12 1n d2 11 = = 22 c o s ȕzm ddaz z++ 2.2 Geometry of Helical Gear Teeth 93 Table 2.2/2 (continued) Cons. No. Designation Symbol Equation 20 Centre distance a t d wtcosĮ cosĮa = a 21 Operating pressure angle Įwt 12 t wt t ( + )zzcosĮ = cos Į2m a 22 Sum of the profile shift coefficients x1 + x2 wt t 12 12 ninvĮ invĮ( + )zz2 tanĮ inv Į = tan Į arc Įxx = −+ − 23 Tip shortening coefficient k d 12 n ( + )a- ak = xxm− 24 Transverse contact ratio İ Į 2 2 a1 b1 Į et 2 2 2 a2 b2 wt 21 İ22 + sin Į|| 2 2dd= p dd za z − −− In the case of tip chamfering, instead of da1,2 the usable tip diameters dNa1,2 are to be used 25 Overlap ratio İȕ w ȕ n sin |ȕ|İ ʌb= m bw common face width 26 Base tangent length Wk kn nt nn k bt et bz = cosĮ ʌ + z invĮ 2 | z | + 2 sin Į or z = + cos ȕ|z| − − k Wm xm ks p W k number of teeth spanned (number of gaps spanned for internal gearing) Conditions: k b 2 2 k a b f b sin | |ȕ ( + 2 ) < + < cos ȕb > W Wdc d d 94 2 Gear Teeth Geometry Table 2.2/2 (continued) Cons. No. Designation Symbol Equation 27 Measurement over balls MdK dK M K dK M K M b K Kt b b Kt b n t b = + ( even) ʌ = cos + ( )2 = ; inv a = Ș cos Į cos ȕ ʌ 4 tan Į = invĮ Ș2zz d MD d z odd MDz d Ddd x− −− DM ball diameter Conditions: b dK aM aM M M MK t b b > ; < ; = cosĮ tan Į = tanĮ cos ȕddd dd M D d− 28 Normal chordal tooth thickness (on the reference cylinder) in the normal section ns nn n sinȥ sd= , n n n 2 nȥ = ; = ȕcosd sdd 29 Chordal height ns to tip cylinder ah ()aa n n 0.5 1 cos ȥ hh d =+ − 30 Constant chord (of a spur gear) cs 2c cosĮ ss= 31 Height over the constant chord to tip cylinder ch ca sinĮcosĮ2shh=− ⋅ 2.3 Supplemental Information concerning Geometry 2.3.1 General Information on Tooth Root Geometry In manufacturing using the form cutting process, the tool tooth corresponds with the tooth space (e.g., form cutter, broaching tool). Because of this technology, the tooth root fillet corresponds to the tool tip. For this reason, and because the tool usually has a constant tip radius, it is easy to calculate the tooth root thickness. Other issues are relevant in the case of the generating process. Here one can imagine the generating gear paired with a mating gear or a rack and in a process of rolling off, that is, in motion together like a simple gear mechanism. During the generating process, the mating gear or the rack, respectively, generates the tooth contour (Figure 2.1/9). Here the tool tooth form does not correspond to the tooth space form of the teeth being generated. Even the tooth root fillet curve generally has a different form and size than the tip radius of the generating tool. If Na0 stands for the tool tip radius and NFn for the (generally not constant) radius of curvature of the tooth root fillet in the normal section, then the following is true: NFn ≥ Na0. 2.3 Supplemental Information concerning Geometry 95 The borderline case NFn = Na0 exists when the centre point of the tool tip rounding M lies on the generating pitch circle or the rack pitch line in the case of a rack-shaped tool (rack-type cutter, hob). The equations related to tooth root thickness and to curvature of the root fillet curve will be derived in the following subchapters. A different approach can be found at Hirschmann [2/1]. 2.3.2 Tooth Root Geometry of External Gearing in the Case of Production Using Hobs or Rack Cutters, with an Arbitrary Tool Profile Angle 2.3.2.1 Tooth Root Thickness of External Gears Spur gearing will be used to make the deductions. For helical gearing, the number of teeth of the equivalent gear z1/(cos ȕ cos2 ȕb) has to be used as an approximation for z1. In the general case, it is assumed that the generating pitch circle is not equivalent to the reference circle; that is, the tool flank angle Į0 does not correspond to the pressure angle of the gearing in the reference circle [6.5/17]. The general assumption is Į 0 Į (2.3/1) That means that because of the necessary equivalence of normal base pitch pe0 = pe, or m0 ʌ cos Į0 = m ʌ cos Į, the rack-shaped tool has a different module than the generated gearing. This general assumption may seem meaningless for actual practice. However, one must keep in mind that for generating grinding, for the grinding disc often a different pressure angle is used than for the gear being produced. Even though a flank is generated which corresponds to the basic rack (e.g., ISO 53, DIN 867), the generating roll angle influences the curvature of the tooth root fillet, which can be important in the case of grinding steps, among others. Figure 2.3/1a depicts a relative position of the tool to the gear. Figure 2.3/1b shows the tool with a protuberance. The following is directly evident from Figure 2.3/1a: sF = 2X (2.3/2) where X is the horizontal coordinate of the contact point of the tool profile at the tooth root, and Y is the distance away from the gear centre point: w1 1 a0 sinĳ cosșsinȥhXrΔ =− + N (2.3/3) w1 1 a0 cosĳ sinșsinȥhYr Δ=− + N (2.3/4) The working pitch circle radius rw1 of the workpiece (1) and the height ǻh result in accordance with Equations (2.3/5) and (2.3/6) in 01 w1cosĮ 2c o sĮmzr=⋅ (2.3/5) 0a0 a0 1 1cosĮ 1 2c o sĮǻ − =− − + mhh x m z N (2.3/6) Also evident from Figure (2.3/1a) is 1ȥșĳ=+ (2.3/7) where ș is the tangent angle that can be specified on the root fillet (e.g., ș = ʌ /6), and the roll angle Q1 can be replaced by 111ĳȖȦ=+ (2.3/8) where Ȗ1 is the roll angle, if the relative motion of the tool P0 is congruent with P1 and P1ƍ (the centre point of the tool tip radius M is on the connecting line from the gear workpiece centre O1 to the point P0 on the working pitch line or working pitch circle). 96 2 Gear Teeth Geometry Figure 2.3/1 Geometrical dimensions for deduction of the tooth root thickness sFn: a) tool (hob) – gear (workpiece) Note: If B0 v B the tool thickness has to be adjusted to the pitch circle Figure 2.3/1 b) Protuberance tool with profile angle Į0 (additional marking with the index for the basic rack was omitted here); P-P profile reference line For 1Ȗ one arrives at a0 0 00pr w1 w1 1 w122Ȗ tanĮ cosĮ cosĮ1 s shs rΔ=+ Δ + −− N (2.3/9) with sw1 = sy1 in accordance with Equation (2.1/26), with yĮ=0Į, rw1 in accordance with Equation (2.3/5), and hΔ in accordance with Equation (2.3/6), sw0 = pw - sw1, and the thickness modification of the tooth sw1 on the generating pitch circle without radial infeed (lateral shifting). Particularly for Į0 = Į and spr = ǻsw1 = 0, the result is 1 a0 a0 a0 11 11 122t a nĮ22Ȗ tanĮ cosĮʌxh x zm m z m z+ =+ − − + ⋅ NN (2.3/9a) Furthermore the following is true: 1 w1Ȧ tanȥh rΔ = (2.3/10) The coordinates of the contact point of the tangent on the tooth root fillet (half tooth root thickness X and distance Y to the gear centre point) can be written if the angle Q1 is replaced by ȥ: 1 1 a0 w1 w1w1sinȖ cosȥȖ tanȥ sinȥ tanȥhh h Xr rrΔΔ Δ =+ − + − − N (2.3/11) 1 1 w1 w1w1 a0cosȖ sinȥȖ tanȥ sinȥ tanȥhh hYr rrΔ=+ + − − − Nƅƅ (2.3/12) With a given tangent angle ș the angle ȥ is to be determined iteratively from 1 w1ȥș +Ȗ tanȥh rΔ=+ (2.3/13) 2.3 Supplemental Information concerning Geometry 97 Figure 2.3/2 Tooth root thickness s Fn/mn at the 30 -tangent ( ș = 30 ) for a tool tip radius 'a0 = 0.38 mn and when generation by hob or rack-type cutter (Į w = Į = 20 ) is applied; gearing: ISO 53 (DIN 867) A good initial value for solving Equation (2.3/13) iteratively results from Equation (2.3/14): ()1 1 w1 w1 w1 w12ș+Ȗș +Ȗ 11 1 ȥ 2 1 / (2 ) 1 / (2 ) 4 1 / (2 )h hr r hr hrΔ ≈⋅ + ⋅ + +Δ +Δ +Δ (2.3/14) where by all angles are to be used in radian units. Another possible approach is to vary the roll angle Q1 in increments until the specified angle ș1 is reached [Equations (2.3/7) to (2.3/10) and (2.3/13)]. As an example for a tool tip radius of Na0 = 0.38 mn, Figure 2.3/2 depicts the tooth root thickness sF/m = sFn/mn, which is related to the module m (normal module mn). 2.3.2.2 Radii of Curvature of the Tooth Root Fillet in External Gearing, for Production Using Hobs or Rack Cutters The deductions below apply in the general form for generation with any kind of generating tool (hob, rack-type cutter, shaper cutter). The numerical evaluations in this section are for a production using hobs and rack-type cutters. Helical gearing is approximated using the normal section by subsequently substituting the number of teeth of the equivalent gear z nx = z/(cos ȕ b cos2 ȕ) for the number of teeth z. To calculate the curvature, one can use the Euler-Savary equation ([22] among others). For the variables in Figure 2.3/3, the following applies if the pitch circle 2 rolls off on 1 and MPis firmly connected with 2: 1211 Ȧcosȟ ll u = (2.3/15) Here u is the pole velocity and Ȧ the relative angular velocity. The minus sign in Equation (2.3/15) applies when, viewed from the standpoint of pole P, the points M and M0 are on the same side, as shown in Figure 2.3/3. The relative angular velocity Ȧ of the moved plane is 12 w2 w1dd dĳ dĳ dĳȦ ddd d dt t ttt t tuu r r == + = + (2.3/15a) and the following results: w1 w21 Ȧ1 u r r=+ (2.3/15b) 98 2 Gear Teeth Geometry With that one arrives at the Euler-Savary equation in the general form 12 w1 w211cosȟ11 ll rr = + (2.3/16) The radius of curvature N of the tooth root fillet results from the radius of curvature NM of the relative path of the centre point of the tool tip rounding and the tool tip radius Na0: Ma 0=+NN N (2.3/17) Figure 2.3/3 Geometric quantities for deducing the tooth root curvature radii in accordance with the Euler-Savary equation for two pitch circles rolling off one another With the quantities in accordance with Figure 2.3/3 and Equation (2.3/16) one arrives at M21ll=−N ; 12 W11 1 cosȟ llr=+ ; ȟ=ʌ2ȥ;− 2/s i n lh=Δ Ψ ; ww 1 w 211 1 ; =+rr r with rw2 = rw0 (tool) The result is the radius of curvature ' of the tooth root fillet : ()w2 a02 sinȥ sinȥh rhΔ+ +Δ=NN (2.3/18) When using a rack-type cutter or a hob in particula r then rw0 ĺ and rw1 is the pitch circle radius of the workpiece ( rw1 = rb1 / cos Į0). Then ȥ is to be determined according to the rela- tionship given in Section 2.3.2.1 [Equations (2.3/13) and (2.3/9)]. For the smallest radius of curvature of the tooth root fillet, at the lowest point in the root fillet with ȥ = ʌ/2, the result is min a0 w2 hh r ΔΔ=+ +NN (2.3/18a) The following is valid particularly for generation or production using hobs and rack-type cutters, and in the case of Į w = Į0: a0 a0 1 hh x mΔ= − − N 1 w12mz r= With that the following results: 1 1 1a0 a0 a0 min2 1 2h x mm z mmx−− =+ +− N N N (2.3/18b) Furthermore, if one simplifies with a0 a01h mm−=N and z1 = z, x1 = x. 2.3 Supplemental Information concerning Geometry 99 The result is Equation (2.3/18c) for orientation about the influence of number of teeth and profile shift: ()2 a0 min1 1 2zx mmx−=+ +−N N (2.3/18c) Equation (2.3/18c) clearly shows the influence of the individual parameters. Figure 2.3/4 gives an example of the calculation result for NFn/mn, with Na0 = 0.38 mn, ș = 30 , and Į0 = Į = 20 . The general tendency shows a minimum of the radius of curvature in the area of the profile shift coefficient x = 1.0 to 1.25. Here the lowest point of the centre point M of the tool tip rounding coincides with the pitch circle. The radii of curvature of the tooth root fillet depend very strongly on the distance ǻh of the centre point M of the tool tip rounding from the pitch circle (Figure 2.3/1a). Then ǻh > 0 results in intricate curves of the relative path of the centre point M of the tool tip rounding. In contrast, with ǻh < 0 elongated, rolling paths are produced. In the borderline case ǻh = 0, a point (Figure 2.3/5) results. Figure 2.3/4 Radius of curvature NFn/ mn of the tooth root fillet for Na0 = 0.38mn and ș = 30 using a hob or rack cutter ( Įw = Į0 = 20 ); gearing: ISO 53 (DIN 867) Figure 2.3/5 Curves of the relative path of the centre point of the tool tip radius, dependent on the distance of the points M a,b,c… from the pitch point C while rolling off a straight line on the pitch circle (hob, rack cutter) 100 2 Gear Teeth Geometry Sometimes one can make use of the change to the tooth root curvature caused by different roll angles (or pitch circles r w). In this way it is possible, for example, to influence the curvature of the grinding step curve during grinding by using other roll angles, or in cases where there is a risk of the grinder coming into contact with the root fillet (heat-treatment distortion) produced by protuberance tools, it is possible to prevent a pointed step. In the Niles method, among others, the dressing of the grinding disc to a different tool angle or roll angle can be easily achieved. Figure 2.3/6 gives an example for the number of teeth z = 40. Depicted here are the radii of curvature in relation to the tooth root thickness, for which the parameter decisive for the notch effect was used. These illustrations depict both the strong influence of the roll angle Į w and the tool tip radius Na0 on the notch parameter NFn/sFn, as well as the change to the curvature or the notch parameter, respectively. By changing the roll angle, it is possible to radically change the tooth root fillet if the tooth root surface is also ground (exceptional) and the rest of the contours of the profile are retained (e.g., Į, p e, db). Figure 2.3/6 Notch parameter 2 NFn/sFn dependent on the profile shift coefficient x with different roll angles Įw = Į0, tool tip radii Na0 and tangent angles ș at the root fillet; z = 40; Į = 20 ; gearing: ISO 53 (DIN 867) Often it is necessary or desirable to determine the key quantities of the tooth root fillet for other tangent angles ș. According to the given relationships, this can be easily achieved by specifying ș. The change to the notch parameter when the roll angle is changed from αw = 20 to αw = 26 (with Į = 20 = constant) can be seen by comparing Figure 2.3/7 with 2.3/8. These calculations are based on a tool without a tip radius ( Na0 = 0), as would occur in the theoretical borderline case of a grinding disc.1) Unfavourable or too-small root transition radii can be avoided by Įw = Į0 > Į. 1) In reality Na0 > 0 is given because the grinding disc has a certain grain size. 2.3 Supplemental Information concerning Geometry 101 Figure 2.3/7 Notch parameter (2 NFn/sFn) dependent on profile shift coefficient x, tool tip radius Na0 = 0, and roll angle Įw = Į 0 = 20 ; rack-shaped tool; gearing: ISO 53 (DIN 867) Figure 2.3/8 Notch parameter 2 NFn/sFn dependent on profile shift coefficient x, tool tip radius Na0 = 0 and roll angle Įw = Į0 = 26 ; rack-shaped tool; gearing: ISO 53 (DIN 867), ( Į = 20 !) 2.3.3 Tooth Root Geometry for Production with a Shaper Cutter – External and Internal Gearing 2.3.3.1 Tooth Root Thickness The following deductions are for external spur gearing. They can be applied to internal gearing if a negative number of teeth are used for the workpiece. The results can be used as an approximation for helical gearing if the quantity z1/(cos2 ȕb cos ȕ) is used for the number of teeth z1 of the workpiece and the quantity z0/(cos2 ȕb cos ȕ) is used for the shaper cutter ( z0). In Figure 2.3/9 the geometric quantities are shown in a specific position in the joint rolling motion. The same meaning applies for the symbols as introduced in the deduction with hobs or rack cutters. 102 2 Gear Teeth Geometry Fn2 sX= (2.3/19) w1 1 a0 sinĳ cosșsinȥhXr Δ=− + N (2.3/20) w1 1 a0 cosĳ sinșsinȥhYr Δ=− + N (2.3/21) 11 12ĳĲ Ȧ=+ (2.3/22) with 11Ĳ2ʌz = , 10 0 1ȦȦ zz = , and ()01Ȧȥ ș Ĳ 2 =− − . The working pitch diameter 2 rw1 of the workpiece is also dependent on the numbe r of teeth and the profile shift of the shape r cutter. In Equations (2.3/20) and (2.3/21) the roll angle 1ĳ is expressed as the su m of half the tooth space angle 12Ĳ and the difference angle1 Ȧ. Figure 2.3/9 Geometric quantities for shaper cutter and workpiece in a specific position of the joint rolling motion The angle ș of the tangent on the root fillet is dependent on the roll angle Q1 and auxiliary angle ȥ: 1ș=ȥĳ− (2.3/23) The other parameters can be seen in Figures 2.3/9 to 2.3/11. One possible approach is to vary the angle Q1 in increments. Each Q1 results in the difference angle Ȧ 1 in accordance with Equation (2.3/22), Ȧ0 = Ȧ1 z1/z0, the auxiliary angle ȥ in accordance with Figure 2.3/9, and from that a tangent angle ș (Q1) in accordance with Equation (2.3/23). Then Q1 is varied until the calculated angle ș corresponds to the desired angle within a specified error bound. With respect to convergence, the al gorithm in Appendix 1 turned out to be the most favourable (Newtonian approximation). The equations in this section also apply as approximations if hobs or rack-type cutters are used if a very large value, such as z0 = 10000, is used for the number of teeth z0 (of the shaper cutter). 2.3 Supplemental Information concerning Geometry 103 Figure 2.3/10 Coordinates of tip roundin g of the shaper cutter Figure 2.3/11 Geometric quantities on a shaper cutter 2.3.3.2 Radii of Curvature of the Tooth Root Fillet The relationship (2.3/18) is valid generally. For internal gearing, a negative value is to be used for the number of teeth of the workpiece. In the same sense it is possible to use the calculation with a shaper cutter that has internal teeth for a workpiece with external teeth. For the quantities ȥ , r w1, and ǻh the equations or instructions in Section 2.3.3.1 apply. Often shaper cutters have no tip rounding. In an unfavourable shaper cutter–workpiece pairing, during generation (production) the pitch circle can be close to the root circle. Therefore, it is already plain from Equation (2.3/18) that with ǻh → 0 and Na0 0 a tooth root fillet approaching zero results. Figure 2.3/12 illustrates this phenomenon. It becomes plain that very unfavourably small root fillets can be produced. This should always be kept in mind when designing gears that are subject to considerable strains and can be checked very easily by comparing the root diameter and the production pitch diameter of the workpiece. 104 2 Gear Teeth Geometry The actual profile shift of the shaper cutter is dependent on the regrind (total amount of grind-off). Since this usually is not known at the design stage, the calculations should be based on both extreme values x 0max and x0min. They should be determined using the manufacturer’s information, based on the maximum amount of grind-off and relief angle. Figure 2.3/13 shows an overview of the relative paths of the centre point M of the tool tip rounding. Figure 2.3/12 Dependence of the radius of curvature of the tooth root fillet of internal gearing on the profile shift, using a shaper cutter [gearing according to ISO 53 (DIN 867)] Figure 2.3/13 Relative paths of the centre point of the tool tip rounding dependent on the type of tool and the position of M 2.3.4 Comparative Study Related to Tooth Root Geometry The information in documentation or standards on the factors for calculating tooth root stress such as YF (form factor) and YFS (combined tooth form factor) mostly makes reference to the hobbing produ ction method. Since the type of tool (e.g., hob, rack-type cutter) to be used is usually also not specified on the drawing, the choice is left up to the technologist. With that it is often the case that during calculations a rack-shaped form of the tool (hob, rack-type cutter) is assumed, and manufacturing is done using a shaper cutter. This is often also unavoidable because of the form of the gear body, which, in the case of stepped gears, only permits a little step. 2.3 Supplemental Information concerning Geometry 105 Undoubtedly of interest is the difference between the decisive quantities related to tooth root stress for the two basic tools (hob, shaper cutter), such as tooth root thickness, radius of curvature, and notch parameter. Figures 2.3/14 to 2.3/16 provide an overview with the results of calculations. The basic profile of the rack-shaped tool has generally been based on a tool tip radius of Na0 = 0.38 m. The tooth tip of the cutting wheel is assumed as not rounded ( Na0s = 0) for the borderline case Na0s ĺ 0. In Figure 2.3/14 it can be seen that for production with the cutting wheel ( z0s = 28) the tooth root thickness is almost always larger than that which is generated by the hob. This deviation is generally negligible. In contrast, the deviation of the tooth root curvature radius NFn or the notch parameter 2 NFn/sFn can be significant (Figure 2.3/15). Figure 2.3/14 Deviation of tooth root thickness at the 30 tangent in production with a shaper cutter ( z0s = 28) instead of a hob Figure 2.3/15 Deviation of the notch parameter 2 N/sF at the 30 tangent in production with a shaper cutter ( z0s = 28) instead of a hob 106 2 Gear Teeth Geometry Based on the approximation that the stress concentration factor is 3 Fn1SY∝N deviations of YS result here of up to 20% (more generally, refer to [2/2]). Figure 2.3/16 shows the deviations of the notch parameter 2 N/sF = 2NFn/sFn for a number of teeth on a shaper cutter of z0s = 80 compared to the values for a number of teeth on a shaper cutter of z0s = 28. If the profile shift on the workpiece is large, especially large deviations and unfavourable notch parameters will resu lt. With tip rounding on the cutting wheel, which at z0s = 80 begins to approach the form of a rack, the root geometry could be considerably improved. Figure 2.3/16 Deviation of the notch parameter 2 N/sF at the 30 tangent in production with a shaper cutter with the number of teeth z0s = 80 compared to a shaper cutter with z0s = 28 2.4 Symbols and Symbol Explanation Az mm2 tooth profile area a mm centre distance of a cylindrical gear pair ad mm reference center distance (sum of refe- rence circle radii) a0 mm centre distance of the generating gear pair b mm face width C - pitch point c mm tip clearance c\* - tip clearance coefficient d mm reference diameter da mm tip diameter db mm base diameter df mm root diameter (nominal value) dw mm pitch diameter dy mm diameter of y-circle dNa mm usable tip diameter ES ȝm tooth thickness allowance (at reference cylinder) e mm space width at reference cylinder ef mm space width at root cylinder ey mm space width at y-cylinder eP mm space width of gear reference profile g mm path of contact gĮ mm length of path of contact (total) H mm height of a rack tooth extended to a pointed tip h mm tooth height (between tip line and root line) ha\* - addendum coefficient of a gear ha mm addendum of the tooth hFa0\* - addendum coefficient of the straight- line flank proportion of the tool haP mm addendum of the gear reference profile hf mm dedendum of the tooth hf\* - dedendum coefficient hfP mm dedendum of the gear reference profile 2.4 Symbols and symbol explanations 107 hg mm radial height of a tip relief defined for a special reference profile hl\* - coefficient of usable tooth height hw\* - coefficient of common tooth height of reference profile pair hwP mm common tooth height of reference profile and mating profile ch mm height over constant chord s i - gear ratio k - tip shortening factor m mm module (reference diameter pitch) mb mm module at base circle mn mm normal module mt mm transverse module my mm module at y-cylinder p mm pitch at reference cylinder pb mm pitch at base cylinder pe mm normal base pitch pn mm normal pitch pt mm transverse pitch, reference pitch pw mm pitch of working pitch circle px mm axial pitch py mm pitch at y-cylinder pz mm lead pr mm protuberance amount ra mm tip circle radius rb mm base circle radius rf mm root circle radius ru mm radius of circle at transition to undercut rw mm pitch circle radius ry mm radius of y-circle s mm tooth thickness at reference cylinder sa mm tooth thickness at tip cylinder sw mm tooth thickness at pitch cylinder sy mm tooth thickness at y-cylinder sP mm tooth thickness of gear reference profile spr mm amount of undercut ݏҧ mm chordal tooth thickness ݏҧc mm constant chord / m/s velocity x - profile shift coefficient x0 - profile shift coefficient of cutting wheel Zİ - overlap coefficient z - number of teeth znx - number of teeth of equivalent gear zv - virtual number of teeth of equivalent gear in normal plane z0S - number of teeth of shaper cutter Į pressure angle Įa profile angle at tip cylinder Įn normal pressure angle Įt transverse pressure angle Įw operating pressure angle Įwt operating pressure angle in transverse plane Į0 profile angle of the tool Įy profile angle at y-cylinder ȕ helix angle at reference cylinder ȕa helix angle at tip cylinder ȕb helix angle at base cylinder ȕy helix angle at y-cylinder Ȗ1 special pitch angle ǻ mm 1. tangential dimension of a tip relief defined for a special reference profile - 2. part of a symbol İa1, İa2 - partial overlap ratios at spur gearings İĮ - transverse contact ratio İȕ - overlap ratio İȖ - total contact ratio ș angle between tooth centre and tangent at root fillet N, ȡ mm radius of curvature, rounding radius Nan, ȡan mm tip rounding radius in normal plane Na0, ȡa0 mm tip edge radius at tool NC, ȡC mm curvature radius at pitch point NF, Nf, mm curvature radius at root fillet ȡF, ȡf ( NF mostly used as significant dimension in root stress calculation) F,∗Nf∗N - relative curvature radius Fȡ,∗fȡ∗ (F∗N=FN/mn; f∗N= fN/mn) NfP, ȡfP mm root fillet radius at gear reference profile Ȉlj mm total length of contact lines Q1 roll angle ȥ - parameter for calculating involute coordinates ȥ tooth thickness half-angle at reference circle ȥy tooth thickness half-angle at y-circle Ȧ s-1 angular velocity Ȧ1 auxiliary angle at manufactured gear Ȧ0 auxiliary angle at tool Indices B dimension at base cylinder (or for driven gear) F form circle (determining the max/min usable profile range) or special dimension at tooth root f dimension at tooth root max maximum value min minimum value N active circle (active profile range used by mating gear) n dimension in normal plane (also for equivalent spur gearing of a helical gearing) P dimension of gear reference profile pr dimension at protuberance 108 2 Gear Teeth Geometry t dimension in transverse plane or in tangential direction w dimension at working pitch cylinder resp. working dimension of a gear pair y arbitrary point or circle or diameter or cylinder (y-circle, y-cylinder) 0 dimension at manufacturing tool 1 dimension at pinion of a gear pair 2 dimension at wheel of a gear pair 109 3 Meshing Interferences 3.1 Overview The term meshing interferences denotes disturbances in the uniform transmission of motion due to non-compliance with the Law of Gears (Section 1.3). For involute gearing, this occurs if • flank parts become engaged that do not belong to the involute region, • the total contact ratio is İȖ < 1, • contact of the tooth flanks occurs again outside the path of contact (e.g., overlapping in the case of internal gear pairs). Such interference causes not only an increase in running noise but also damage (e.g., tooth breakage), increased wear and pitting. Apart from these types of meshing interferences, a radial mounting may well not be possible in the case of an internal gear pairing. If a standardised basic rack is present and the number of teeth in the pairing is already defined, then it is still possible to influence or eliminate the meshing interferences by • tip shortening, • changing the amount of profile shift, • using other manufacturing tools, • modifying the tooth root geometry, (e.g., position of grinding step). It is possible that preventing misalignments and deformations may remedy the situation in individual cases. Internal gear pairings are typical for cases of meshing interferences, particularly when the absolute magnitude of the difference in the number of teeth is less than 10. Interference occurs less frequently in externally toothed gears. Nevertheless, this must be generally taken into account. 3.2 Meshing Interferences of External Gears 3.2.1 Meshing Interferences Resulting from Too-Small Contact Ratio In principle, a gearing can be operated smoothly provided that at least one flank pair is always engaged. This is displayed in Figure 3.2/1 for the plane of action. The condition for at least one touching flank pair leads to the equation g Į > pet b tan b If this is then divided by the normal base pitch pet and Equations (2.1/41) and (2.2/9) are taken into account for the transverse contact ratio İĮ and overlap ratio İȕ, then, following the substitution, we obtain as the condition for smooth operation 1Įȕ >γε= ε+ ε (3.2/1) 110 3 Meshing Interferences For the transmission of major torques and powers, it naturally makes sense to design the gear for J >> 1. At the boundary condition J 1, it should be noted that it is possible for a gap to occur at a front side – particularly with gears under light loads as a result of positional deviations of the toothing (f Hȕ) and the boreholes ( fȈȕ, fȈį), and thus İĮ < 1 is also effective for İȖ < 1. Therefore a smooth operation is no longer possible. It is also frequently observed that a gear still displays a usable performance despite the fact that İ Į < 1 and İȕ = 0. This is due to the extension of the contact (ǻgA, ǻgE) as a result of the elastic deformation of the teeth (Figure 3.2/2), [3/1], [3/2]. The contact then ensues off the line of action. From a metrological perspective, it was possible to determine an increase in the transverse contact ratio up to ǻİ Į > 0.1 for steel gears in the usual load range. The increase in İĮ is even more marked with plastic gears. The wear on the head edges causes a gradual reduction of the load takeover in the off-line-of-action contact range. For spur gearing, in particular, it is possible for a too-small contact ratio to be present in specific circumstances. These include • a too-large profile shift modification in conjunction with a too-low number of teeth, • a too-large tip shortening, • stub gearing, • an undercut due to a too-low profile shift modification in conjunction with a too-low number of teeth or too-large undercut in conjunction with an unfavourably designed protuberance tool. In the case of an undercut and for the use of protuberance tools, the transition from the root fillet to the involute profile, which does not ensue tangentially here, should always be noted. Figure 3.2/1 Boundary condition for smooth operation Figure 3.2/2 Off-line-of-action contact ǻ gA and ǻgE (areas A A and E E) as a result of elastic tooth deformations and adjacent pitch deviations fA or fE (for calculation refer to Appendix 8, derivation <6.3/1> 3.2 Meshing Interferences of External Gears 111 Figure 3.2/3 Usable root circle; external gear pair Figure 3.2/4 Usable root circle; internal gear pair (a < 0; z2 < 0) The diameter in question du = 2ru (Figure 3.2/5) is to be compared with the usable circle diameter dNf, which marks the deepest flank contact of the gear that is not undercut. For a gear pair, the usable circle diameters (Figures 3.2/3 and 3.2/4) with dNf = 2rNf are given by 2 2 22 2 = 2 sin Į + wt Nf1 || a2 b2 b12za dd d dz −− (3.2/2) 22 22 2 = 2 s i n Į + wt Nf2 || a1 b1 b22za dd d dz −− (3.2/3) (index 2: internal gear in case of an internal gear pair; otherwise the allocation is freely selectable). Equations (3.2/2) and (3.2/3) are generally expressed for external gear pairs and internal gear pairs. For the path of contact g Į calculated according to Equation (2.2/44) to be correct, the following equation must apply: Nf1,2 u1,2 > dd (3.2/4) Equation (3.2/4) is only relevant for external gears, since no undercut occurs at internal gears – with the exception of an undercut due to protuberance tools. The transitional diameter d u of the undercut limit for the involute flank (Figure 3.2/5) was calculated by Hofer [3/3]. Relationships have also been provided by Schiebel [3/4] and Keck [3/5]. Here it is the intention to make use of computing technology to show another way of enabling the solution for gearings hobbed with protuberance tools. The equations derived in Sections 2.3.2.1 and 2.3.3.1 to calculate the tooth thickness with regard to the number of teeth and profile shift apply without restriction, that is, also to undercut gearings. If the tooth thickness of the undercut area generated from the tool tip s FK is calculated step by step for various tangential angles ș starting from ș = 0 for negative ș values and compared with the thickness formed by the involutes above the base circle, the radius of the beginning of the undercut can be determined on an iterative basis. 112 3 Meshing Interferences The relationships are illustrated in Figure 3.2/5. For a given tangential angle ș(i) to the root fillet, half the tooth thickness Xi and the vertical distance to the centre of the gear Yi can be determined in accordance with Equations (2.3/3) and (2.3/4). With X(i) and Y(i), it is possible to determine the radius r (i), which represents the distance of the point PFK(i) from the centre of the gear. The tooth thickness sFK(i) of the root fillet at the radius r(i) as an arc (refer to Figure 3.2/5) is expressed by qFK(i) (i) FK(i)2s =M P or with X(i) and Y(i) by FK(i) (i) (i)2s =r ϕ (3.2/5) where 22 (i) (i) (i)zr = X + Yz⋅ (3.2/5a) ()(i) i (i)tanĳX Y= (3.2/5b) The tooth thickness limited by the involute s E(i) is qE(i) (i) E(i)2s =M P (3.2/6) The thickness limited by the root fillet (FK), sFK(i), is compared with the thickness sE(i) belonging to the same radius r(i). Here, sE(i) is the tooth thickness limited by the involute, which is regarded as not undercut in the undercut area above the base circle. Thus sE(i) is calculated for ry = r(i) rb according to Equation (2.1/26). ǻs(i) = sFK(i) – sE(i) (3.2/7) After these calculations have been performed, a new angle is selected until ǻs changes in sign [refer to Equation (3.2/5)]: ș(i + 1) = ș(i) + ǻș < 0 (3.2/7a) e.g. with ǻș = –5 Following this, all the quantities are determined anew. From ǻs = ǻs(ș), then by means of iteration it is ultimately possible to determine the radius sought ru for ǻ s 0 with any precision required. Information on undercut prevention is contained in Sections 2.1.2 and 3.3.2.1 as well as Table 2.1/6 (minimum profile shift, limiting number of teeth). Figure 3.2/5 Transition radius r U to involute undercut curve ( rU = dU /2) 3.2 Meshing Interferences of External Gears 113 3.2.2 Meshing Interference in Non-involute Regions In borderline cases, meshing interferences can also occur in externally toothed gears by the action of the tooth tip of one gear in the tooth root area of the mating gear that no longer belongs to the involute part of the tooth flank (for internal gears refer to Section 3.3). This meshing interference can be avoided if the deepest meshing point present in the gear pairing – expressed by the usable circle diameter d Nf [Equations (3.2/2) to (3.2/3)] – is higher than the deepest point on the tooth surface still belonging to the involute, provided by the root form diameter dFf = 2rFf (see Figure 3.2/6): Nf Ff Ffdd d ≥ Δ+ (3.2/8) Here ǻdFf represents a safety distance (to be determined by experience), which must be selected taking into account the tolerance fields, deformations, and so on. When manufacturing using the hob or rack cutter, the form diameters result in accordance with Equation (3.2/6) as () t0 a0 Ff a0 E n b 22t0 t0 = sin Į 2 ( 1 sin Į ) / sinĮ + d dh xm d −−− − N (3.2/9) Figure 3.2/6 Root form radius r Ff of external gearing when manufacturing with a hob or rack cutter The generating profile shift coefficient xE is the gear profile shift coefficient corrected by the effect of the tooth thickness deviation E s, which plays an important role in generating tools with a tooth rack profile (hob, rack cutter, shaper cutter); refer to Equation (3.2/10).1) 1) Es′ is the tooth thickness deviation generated by radial infeeding of the tool with a tooth rack profile taking account of the gear and tool allowance envisaged. 114 3 Meshing Interferences sE nn 02t a nĮExxm=+′ (3.2/10) (Note: E s is a signed value, usually negative) For grinding steps , the analogous tip height of the grinding tool is to be inserted into Equation (3.2/9) for h a0. This is equal to the hob tip height minus the distance ǻ h of the grinding tool to the tooth root surface; ǻh hs (for h s refer to Figure 3.2/7). The form diameter thus derived must be greater than the usable circle diameter. When manufacturing using the shaper cutter, Equation (3.3/5) applies to the form diameter. Equation (3.2/2) or (3.2/3) can also be used provided that the tool data is used in the equation in each case and that d Ff = dNf is written. For more accurate investigations, the generating profile shift must be taken into account in accordance with Equation (3.2/10). Investigations into the interference in the case of external gearings have been undertaken for | i| = 1 by Erney and Scieniczei [3/6], [3/7]. In this process, the interference limits present for large magnitudes | x 1 + x2| are expressed by the operating pressure angle for a uniform division of the total profile shift. For an “overhung” mounted pinion (e.g., a central pinion in conjunction with planetary gears), it is necessary to take into consideration that – for a short time at least while starting up – the **backlash** may be entirely used up (Figure 3.2/7), because the pinion tooth will drop into the gap of the counter gear teeth by its own weight. Figure 3.2/7 Momentary approximation with “overhung” mounted gears (e.g., planetary gears) In this case, the centre distance corrected (reduced) by ǻa instead of the value for the centring of the overhung mounted pinion is to be used in Equations (3.2/2) and (3.2/3). In addition, the roll angle must be redetermined and taken as a basis for the limit considerations. The centre distance alteration (approximation) is given by ()s1 s2 n wncos Į 2 sin ĮEEa −+Δ≈ (3.2/11) To avoid meshing interferences in this case (grinding steps), not only Equation (3.2/8) but also the following condition from Equation (3.2/12) should be satisfied (Figure 3.2/7): 3.3 Meshing Interferences for Internal Gears 115 s < ( )ca h −Δ (3.2/12) The conditions specified for the grinding steps may, of course, be dispensed with in conjunction with the usually present undercut of the tooth root surface (protuberance tools). Then, however, it is necessary to check whether the transitional diameter du = 2ru (Figure 3.2/5) is less than the usable circle diameter dNf. Failing which, the reduction of the path of contact and profile contact ratio must be noted. Meshing interferences can be counteracted if the path of contact resulting from the action of the paired gears is set as the minimum length for evidence of the profile positioning and profile length for the gear pairing (profile) and entered on the drawing. The path of contact g Į is then generally (|z2| |z1|) given by () ( ) () () () ( ) () () 2 2 2 2 y1 b1 w1 b1 2 2 2 2 2w2 b2 y2 b2 2 / 2 / 2 / 2 / 2 + / 2 / 2 / 2 / 2 ||g = d d dd zdd d d zα −− − −− − (3.2/13) Here dy is to be inserted for the diameter: external gearing: internal gearing: dy = dNf, if d Nf du; else dy = du dy = dNf The operating pitch diameter d w can be calculated using Equations (2.1/39) and (2.1/40). Together with gĮ [from Equation (3.2/13)], Equation (2.2/44) is also applicable for the profile contact ratio. 3.3 Meshing Interferences for Internal Gears 3.3.1 Preliminary Considerations Observance of the meshing interferences is a key factor in the development and generation of internal gear pairs, because they are more a problem of internal than external gears and other inter- ferences like overlapping of the tooth tips can appear additionally. The greatest region of meshing interference of internal gear pairs is to be found in gear ratios of | z2|/z1 close to one. For this reason, a minimum difference in the number of teeth of (| z2| z1)min = 10 has usually been complied with in conjunction with Xzero-internal gear pairs where h\*aP = 1 and Į = 20 in order to avoid meshing interferences. The minimum difference in the number of teeth (| z 2| z0)min for the generating gear pair (internal gear – shaper cutter) should be up to two times greater than for the internal gear pair (internal gear - pinion) (| z 2| z1)min in order to avoid meshing interferences between the shaper cutter and the internal gear during generation. This recommendation results from the requirement that the addendum of the shaper cutter, which is equal to the addendum of the mating pinion increased by the tip clearance, may not exhibit any new meshing interference when generating the internal gear. The restriction or elimination of the meshing interference region beneath the aforementioned difference in the number of teeth for internal gear pairs with a gear ratio close to one is possible by means of a profile shift ( X zero- or X-internal gear pairs). In the process, the internal gear should be corrected in such a way that a standard shaper cutter can be used. If it is not possible to achieve satisfactory solutions, then a specifically designed shaper cutter should be fallen back on. Recent investigations [3/8] have revealed that internal gears with i = 1 ( |z2| z1= 0; a 0), are also possible. 116 3 Meshing Interferences The use of state-of-the-art computer software technology [3/9 and 3/10] allows the time expenditure required for the designing of internal gear pairs to be reduced to a minimum – with regard to the requirements imposed – and the reliability of the results to be significantly enhanced. Detailed information on the various meshing interferences caused by non-compliance with the Law of Gears has already been provided in Section 3.1. According to this list, the following specific meshing interferences can occur with internal gear pairs. Flank parts become engaged that do not belong to the involute area: • tooth root meshing interference on the pinion: Section 3.3.2.1 (the tooth tip of the internal gear comes into contact with the tooth root fillet of the pinion), • tooth root meshing interference on the shaper cutter of the generating pair: Section 3.3.2.2 (the tooth tip of the internal gear is cut off from the tooth root fillet of the cutting wheel), • tooth root meshing interference on the internal gear: Section 3.3.2.3 (the tooth tip of the pinion comes into contact with the tooth root fillet of the internal gear). Tooth flank parts come into contact outside the plane of action: • tooth tip meshing interferences: Section 3.3.3.1 (tooth tip interference of pinion and internal gear or cutting away of the internal gear tooth tips in the generating), • radial mounting interference: Section 3.3.3.2 (the pinion is so large that it comes into contact with the teeth of the internal gear outside the plane of action in the radial mounting), • infeed meshing interference: Section 3.3.3.2 (the shaper cutter selected is so large that it cuts away the teeth of the internal gear in the infeed direction). In addition to these interferences, “passive” meshing interferences may also occur, which result not in a cutting away but rather in an undesired contact of flank parts. For generating gear pairs, they occur during the return stroke of the shaper cutter (Section 3.3.3.3). In borderline cases, particularly with extremely elastically designed internal gears and very small differences in the number of teeth (| z 2| z1), the risk of interference has to be checked, taking into account the elastic deformation, primarily of the gear body. It is usually the case that only by experimenting is it possible to obtain definitive information in this regard. Table 3.3/1 (at the end of Section 3.3.4.3) shows the meshing interferences occurring with internal and internal generating gear pairs. At this point, it should be pointed out that for other types of generation, such as broaching (small internal gears) or hobs (large internal gears), the meshing interferences possible when generating with a shaper cutter do not usually occur. Internal gears with reference diameters of |d | 800 mm are also often produced by hobs. Specially manufactured generating tools are required for this purpose. These are more economical to produce compared to face milling manufacturing and the use of a shaper cutter for production. With regard to the normal base pitch, gears with a greater accuracy than from the single dividing process are achieved. To be able to make use of these advantages, it is necessary that the spiral-shaped blades arranged on the generally cylindrical base body of the generating tool be shaped in such a way that an exact involute profile is created in the generation (Section 10.2.4.5, “Hobbing”). The following sections deal with the computational analysis of the individual meshing interference possibilities (refer to [4] and [3/11] to [3/14]). Elastic deformations are not taken into account in this calculation. 3.3 Meshing Interferences for Internal Gears 117 3.3.2 Meshing Interferences in the Theoretical Plane of Action - Tooth Root Meshing Interferences Tooth root meshing interferences occur on th e internal gear pairing if the edge of the tooth tip of the pinion (externally toothed gears) comes into contact with the non-involute tooth root part of the internal gear, or conversely, the edge of the tooth tip of the internal gear comes into contact with the non-involute tooth root part of the pinion. This interference always occurs if one of the involute connecting points (as depicted in Figure 3.3/1 for F 1S,Z at the pinion root) – or both connecting points F 1S,Z, F2S – lie within the length of the path of contact EAof the gear pairing. Figure 3.3/1 Meshing interferences in the plane of action ( dFa0(1) tip form diameter of the shaper cutter for generating the pinion toothing; dFa0(2) tip form diameter of the shaper cutter for generating the internal toothing) (Note: For the sake of a simple representation, the line of action in the manufacture and the gear pairing was assumed to be the same.) If F2S is situated within the section EC, then the tip point of the pinion engages in a non-involute flank section of the internal gear. This results in tooth root meshing interferences on the internal gear. In the same manner, meshing interferences occur on the tooth root of the pinion if the tip form diameter of the internal gear d Fa2 intersects the plane of action outside the section 1SCF (generation using a shaper cutter) or 1ZCF (generation using a toothed rack-shape tool/hob or rack cutter) or even in the always non-involute area between 12TT (such as the position of the tip meshing point A in Figure 3.3/1). The last-named meshing interferen ce in [4] is also called an involute meshing interference. 118 3 Meshing Interferences 3.3.2.1 Tooth Root Meshing Interference on the Pinion Tooth root meshing interference on the pinion occurs if the tooth tip of the internal gear makes contact within the tooth root transition curve of the externally toothed gear (Table 3.3/1, serial no. 2.1; end of Section 3.3.4.3). This type of interference occurs primarily if the number of pinion teeth z 1 is small compared to the number of internal gear teeth | z2|. If the pinion involute is limited by the point F1Z,S in the direction to the tooth root fillet, then meshing interferences will occur in accordance with Figure 3.3/2, if the tip circle d Fa2 of the internal gear (tip circle or, for rounded or chamfered tooth tips, the diameter of the “involute-tip rounding” transition) intersects the plane of action beneath this point (within the section 1Z.S 1FT ). Meshing interference is therefore present in the case of Figure 3.3/2. Figure 3.3/2 Tooth root meshing interference on the pinion The generating tool begins to create the involutes at F1Z,S, and it is only from this point that a involute contact is possible. In order to prevent tooth root meshing interference on the pinion, the usable root diameter d Nf1 from Equation (3.2/2) (identifying the start of contact in the tooth root region of the pinion) must be greater than or equal to the effective root form diameter d Ff1 from Equation (3.2/9). 3.3 Meshing Interferences for Internal Gears 119 Nf1 Ff1dd ≥ (3.3/1) The geometrical relationships of the tooth root meshing interference on the pinion can be seen in Figure 3.3/2. The condition specified in Equation (3.3/1) is fulfilled if the tip form circle radius |d Fa2|/2 (frequently identical to the tip circle radius |d a2|/2) is greater than or equal to the section 21 Z , SOF . We thus obtain the relationship Fa2 2 1Z,S|| 2 | |dO F ≥ (3.3/2) () Fa2 Ff1 2 22 2 wt b2 b1 + 2 | | sin Į + a dd dd ≥− (3.3/3) The effective root form diameter on the pinion dFf1 is determined by the type of generation of the pinion. When using a hob to generate the gear, the geometrical relationships ( d Ff1(Z) = d Ff) depicted in Figure 3.2/6 and specified in Equation (3.2/9) apply. If Equation (3.2/9) is inserted into Equation (3.3/3), then – to avoid tooth root meshing interference on the pinion when manufacturing the pinion with a hob or rack cutter – we obtain for the permissible tip form diameter of the internal gear 2 a0 E n a0 n0 2wt t0 n 1b2 Fa2t0 (1 sin Į)| + 2 | | sin Į + sinĮ 2 sinĮm hx|a m d dd −−−≥− N (3.3/4) Figure 3.3/3 Root form diameter dFf1 = dFf1(S) on the pinion when generating using a shaper cutter ( S) 120 3 Meshing Interferences For externally toothed gears, generated by the generating process using a shaper cutter (number of teeth z0, base diameter db0, tip form diameter dFa0), we obtain F1S as the point of intersection of the effective shaper cutter tip form circle dFa0 (the effect of any tip rounding present is to be deducted from the tip diameter da0) with the line of action. The diameter of the involute connecting points is calculated from Figure 3.3/3 as 2 22 20w t 0 Ff1(S) Fa0 b0 b1 = 2 sin + a dd d d α− − (3.3/5) If Equation (3.3/5) is inserted into Equation (3.3/3), then the relationship for the permissible tip form diameter of the internal gear – to avoid tooth root meshing interference on the pinion when manufacturing the pinion with a shaper cutter – is 2 22 2wt 0 wt0 b2 Fa2 Fa0 b 0| + 2 | | sin + 2 sin |a add d d ≥α α − − (3.3/6) If an undercut is present when generating the pinion, then the tooth tip of the internal gear is able to engage the undercut tooth area of the pinion (no full utilisation of the involute flank part). This does not involve an uneven movement in the sense of a meshing interference. Notwithstanding this, it should be checked whether the profile contact ratio is still sufficient. The involute transition diameter d u from Section 3.2.1 [Equations (3.2/4) to (3.2/7a)] caused by an undercut is to be used for the root form diameter dFf1 in the calculation. An undercut is present if the number of pinion teeth z1 is less than the limiting number of teeth z1g. When manufacturing the pinion with a hob or rack cutter, then to avoid undercut ()\*2 11 g E t FaP02cos sinĮ zz h x≥= − Ƣ (3.3/7) applies with xE taken from Equation (3.2/10). For a rounded tool tip (rounding radius a0∗N= a0N/mn), then ()\*\* \* FaP0 a0 a0 n 1s i n hh = −−Ʊơ (3.3/8) where \* a0 a0 n / hh m= is the associated tool addendum. When manufacturing with a shaper cutter, the relationship to avoid undercut is () 2\*2 200 FaP0 t0 011 g 0 22 wt0 t0 + 2 cos ȕ + cosĮ = tanĮ cosĮzx h z z z z − ≥− (3.3/9) where ( h\*FaP0 + x 0)mn is the radial distance between the reference circle and the start of the tip rounding of the tool. 3.3 Meshing Interferences for Internal Gears 121 3.3.2.2 Tooth Root Meshing Interference on the Shaper Cutter of the Generating Gear Pair Just as in the case of internal gear pairing, it can also happen with generating gear pairs that the tooth tip of the internal gear touches the shaper cutter in the non-involute region of the tooth root fillet ( z0 is small compared to |z 2|). In this case, the prematurely acting tip edge of the internal gear is cut away. In order to avoid the associated shortening of the tooth height (decrease in contact ratio İ Į) on the internal gear, then, in analogy to the tooth root meshing interference on the pinion [Equation (3.3/3)], it is necessary for the condition for the tooth tip form diameter d Fa2 of the internal gear to be fulfilled as follows: ()22 220w t 0b2 Ff0 b0 Fa2| + 2| | s i n Į + |d a d dd ≥ − (3.3/10) The root form diameter dFf0 of the shaper cutter is dependent on its manufacture. Precise values should be requested from the manufacturer. When generating the shaper cutter using the generating process with a tool containing an infinite number of teeth (rack cutter, hob), the d Ff0 is calculated from Equation (3.3/11). To distinguish from the other tool sizes (index 0), the diameters of the cutting tool receive here the index S: () 2\*nE FaP0 2t0 Ff0 FfS S bSt02 = = sin Į + sinĮ mxh dd d d − − (3.3/11) 3.3.2.3 Tooth Root Meshing Interference on Internal Gear Tooth root meshing interference on an internal gear occurs if the crucial tip circle of the pinion (tip form diameter d Fa1) makes contact within the tooth root transition curve of the internal gear (Table 3.3/1; serial no. 2.2; end of Section 3.3.4.3). To avoid this meshing interference, the condition Nf2 Ff2| | | |dd ≤ (3.3/12) must be fulfilled where d Nf2 is taken from Equation (3.2/3) and dFf2 is taken from either Equation (3.3/15) or (3.3/17). The required effective root diameter | d Nf2| is determined by the tip form diameter dFa1 of the pinion. The requirement from Equation (3.3/12) is thus fulfilled if the condition (Figure 3.3/4) Fa1 1 2 2 dO F≤ (3.3/13) is fulfilled. Using the information in Figure 3.3/4, we obtain 22 2 2wt Fa1 Ff2 b2 b1 + 2 | | sin Į d a d dd − ≤− (3.3/14) The root form diameter dFf2 and thus the position of F2 is dependent on the type of generation of the internal gear. It should be noted that the tip form diameter dFa0 is used as the tip circle diameter. 122 3 Meshing Interferences Figure 3.3/4 Tooth root meshing interference on internal gear (pairing with existing tooth root meshing interference |dNf2| > |dFf2|) If the internal gear is manufactured using a shaper cutter, then the root form diameter d Ff2(S) is derived from Figure 3.3/5 as 2 22 20w t 0 Ff2(S) b2 Fa0 b0| = + + 2 | | sin Į |add d d − (3.3/15) Substituting this root form diameter or involute transition diameter into Equation (3.3/14), we obtain the permissible tip form diameter of the pinion as 2 22 20w t 0 w t 0 Fa1 b1 Fa0 b0 + + 2 sin 2 sin Į aa dd d d ≤− − ơ 3.3/16) When manufacturing internal gears with a hob, it can generally be assumed that the blades of the tool are ground such that each blade only generates a small involute region. The juxtapositioning of these segments produce the flank profile. The fillet radius in the root is determined by the blades with fully formed tip rounding. This last statement also applies when other forms of hobs, such as barrel-shaped hobs with special cutting geometry and cylindrical hobs with helical teeth, are used. It is therefore possible for the root form diameter from Figure 3.3/6 to be determined for this manufacturing method. 3.3 Meshing Interferences for Internal Gears 123 Figure 3.3/5 Root form diameter dFf2 = dFf2(S) on an internal gear with generation by shaper cutter(S) Figure 3.3/6 Root form diameter dFf2 = dFf2(Z) on an internal gear with generation by hob ( Z) 22 22a0 a0 Ff 2(Z) f 2 b2 b2| = + ( | | - 2 - + 2 ) |dd d d NN (3.3/17) 124 3 Meshing Interferences If Equation (3.3/17) is inserted into Equation (3.3/14), then – to avoid tooth root meshing inter- ferences on the internal gear when manufacturing internal gears using a hob – we obtain for the per-missible tip form diameter of the pinion () 2 222a0 a0 wt Fa1 f2 b1 b2 + | | 2 + 2 2 | | sin Į a dd d d ≤− − − NN (3.3/18) 3.3.3 Meshing Interferences outside the Regular Meshing Region Meshing interferences outside the regular meshing region occur if flank overlapping occurs outside the plane of action. One type of these meshing interferences is caused by tooth tip overlapping. Tooth tip overlapping (or tooth tip interferences) may occur not only with internal gear pairings during the rolling process (Figure 3.3/7; Section 3.3.3.1) but also with a radially mounted pinion or with generating gears during the radial infeeding (Figure 3.3/9; Section 3.3.3.2). This characteristic is particularly exhibited in association with small differences in the number of teeth (| z 2| z1 = 1 to 10). Another type is to be found in the so-called passive meshing interferences. This type may occur with generating gears during the return stroke and result in unintentional flank contacts (Figure 3.3/13; Section 3.3.3.3). 3.3.3.1 Tooth Tip Interference Tooth tip interferences occur if the tooth tips of the pinion and internal gear overlap during the rolling process outside the plane of action (Table 3.3/1; serial no. 2.3; end of Section 3.3.4.3). From Figure 3.3/7, an interference-free action is provided for the pinion and internal gear if the point of intersection of the tip circle D (point of intersection of the circles d a1 and da2, or alternatively, the point of intersection of the circles dFa1 and dFa2 for tooth tip chamfer or tip rounding) is first reached by the internal gear tooth tip B and subsequently by the decisive tooth tip of the pinion A during the rotary motion of the pinion and internal gear. If the gears therefore rotate from the starting position marked until the tooth tip of the pinion A coincides with the point of intersection of both tip circles, then the pinion with the number of teeth z1 requires a time of t1 = (ș1D + ȥ)/71. Point B of the internal gear with the number of teeth z 2 reaches this point after a time of t2 = (ș2D ǎ)/72. If this encounter at D is to be avoided, then t1 must be t2. From this ensues the condition 1D 2D 12 + ψ −ν θθ≥ ωω (3.3/19) or with u = 71/72 = |z2|/z1 the relationship W from Equation (3.3/20) as the condition to be fulfilled to avoid tip meshing interferences: 1D 2D 1 ( )uW+ψθ≥ ⋅− νθ= (3.3/20) 3.3 Meshing Interferences for Internal Gears 125 The angles 1D, 2D, ȥ and can be readily calculated from the depiction of Figure 3.3/7. For the angle 2D Equation (3.3/21) is obtained: 2 22 a2 a1 2Da2 + 4 = arc cos 4 | | | |a dd da − θ ⋅ (3.3/21) and for the angle 1D Equation (3.3/22): 2 22 a2 a1 1Da1 4 = arc cos 4| | a dd da −− θ (3.3/22) Figure 3.3/7 Geometry of tooth tip interference The angles ȥ and in radian measure are derived as at1 wt inv Į inv Į = ψ− (3.3/23) and wt at2 inv Į inv Į = ν− (3.3/24) Thus Įat1 is the transverse pressure angle at point A and Įat1 the transverse pressure angle at point B. b1at1 a1Į arccos d= d and b2at2 a2Į arccos d= d (3.3/25) If the condition required in Equation (3.3/20) cannot be fulfilled, then it is still possible to avoid this kind of meshing interference by changing the number of teeth (increasing ŇuŇ), or by changing the profile shift (reducing x1 + x2) or by shortening the tip on the pinion or internal gear (reducing the transverse contact ratio). 126 3 Meshing Interferences Reference values for the selection of the sum of the profile shift coefficients can be taken as a function of the sum of the number of teeth from Figure 3.3/8, or alternatively, from a similar figure in DIN 3993 [3/15]. Figure 3.3/8 Limits for the selection of the sum of the profile shift coefficients ( x1 + x2) for x 1 = x2 and k = 0 to avoid tooth tip interference [curves for W = 1 according to Equation (3.3/20); diagram applicable to zn > 25] The limit curve plotted in Figure 3.3/8 applies to W = 1 according to Equation (3.3/20), as well as the definitions that x1 = x2 and k = 0. The region with no tooth tip meshing interferences is further restricted by the geometry thresholds | da2| |db2|, by reaching the space tip limit for the internal gear as well as by the beginning of the undercut limit for the pinion. For the number of teeth on internal gears |z n2| < 25, the space tip limit and the design limit |d a2| = |d b2| will therefore be exceeded even before the limit curve is reached. The fact that gearing and axis position deviations as well as elastic deformations of the internal sprocket affect the position of the limits also has to be taken into consideration. For the sake of reliability, the limit curves for W = 1.002 are specified in [3/11], [3/13], and [3/14] and in DIN 3993 [3/15]. The limit curves are shifted to lower values Ȉx, thus avoiding tooth tip meshing interference with a high level of reliability. Insignificant offsets of the profile shift sum permitted are possible as a result of a changed division of Ȉx. For details, refer to the diagrams contained in DIN 3993 [3/15] (refer to Section 3.3.4.2). For given gearing parameters, it is also possible to check for tooth tip meshing interferences with a representation in analogy to Figure 3.3/12 (Section 3.3.3.2) or according to the information supplied in Table 3.3/1, serial no. 2.3. In borderline cases, however, it is necessary to undertake a control according to Equation (3.3/20). Tooth tip meshing interferences may also occur when generating internally toothed gears using a shaper cutter , since this type of generation represents a new internal gear pairing. The geometry of the generating tooth tip meshing interference is the same as in Figure 3.3/7, except for the substitution of the pinion geometry values by those of the shaper cutter to be undertaken. In addition, in Equations (3.3/19) to (3.3/25) index 1 also has to be replaced by index 0. 3.3 Meshing Interferences for Internal Gears 127 3.3.3.2 Infeed Meshing Interference and Radial Mounting Interference Infeed meshing interferen ce occurs when generating internal gears using a shaper cutter, if the shaper cutter chosen is so large that it damages the internal gear teeth in the infeed direction. For internal gear pairings, radial mounting interference occurs if the pinion selected is so large that it comes into contact with the internal gear teeth during the radial mounting. Figure 3.3/9 depicts several damaged tooth tips (hatched areas), created as a result of positioning the tool radially ( infeed meshing interference ). This interference is not identical with the tooth tip interference caused in the generation, when the shaper cutter cuts an equal piece off each internal gear tooth (Section 3.3.3.1). Figure 3.3/9 Infeed meshing interference on the internal gear when generating using a shaper cutter The geometrical relationships of the radial mounting interferences can be seen in Figure 3.3/10, which depicts a pinion tooth and a tooth space in the internal gear in a symmetrical position of the direction of radial mounting. If a shaper cutter (index 0) is used instead of the pinion (index 1), then the geometrical relationships depicted are in accord with the infeed meshing interference associated with generation (existing radial mounting interference in analogy to Figure 3.3/9). In the initial position specified, the tip edge point A is not able to come into contact with the tip edge point B during the radial mounting. If, however, a position of the tip edge point B changed by the angle Έ 2G is considered (in Figure 3.3/10 this position is denoted by B ), for which the tangent passes through the tip edge point B parallel to the direction of mounting, then – for a correspondingly small difference in the number of teeth ( Ňz2Ň z1) – the tip edge point A comes into contact with the tip of the internal tooth at B (the beginning of the danger region). If the angle in question Έ2G is further increased, then the tangent specified leans to the left at point B and the corner of the tip leans inwards (related to the direction of mounting). The possibility of the tip edge point of the pinion cutting the tip edge point of the internal gear exists for dimensions other than that specified in Figure 3.3/10. The tooth tip of the pinion is no longer able to penetrate the tooth tip circle of the internal gear outside the angle Έ 2D because mounting point A no longer engages into the tooth space of the internal gearing. 128 3 Meshing Interferences Figure 3.3/10 Geometry of radial moun-ting interference (example without radial mounting interference; ǻj > 0) A, B : Initial position A ′, B′: Start of threatened region (limit G) Aj, Bj: Intermediate position within the threatened region A′′, B′′: End of threatened region (limit D) To avoid radial mounting interferences, the following condition can be used for the threatened region (from Έ 2j = Έ2G to Έ2j = Έ2D): 2j 1j j 2j 1j a2 a1 = = | / 2 | sin ( / 2 ) sin > 0 dd −δ − δ νν Δ (3.3/26) where į1j = į1G + (į 2D į2G)() () 21zz j k ⋅ (3.3/27) į2j = į2G + (į 2D į2G)(j/k) (3.3/28) j = 0, 1, 2, ... k number of steps or divisions (3.3/29) The limiting angle for the start of the threatened region is calculated as į1G = Ȥ + (į2G ĳ)|z2|/z1 (3.3/30) b22G at2 a2arccos d= d δ= α (3.3/31) where at1 1 tta t 1 1 a1 + 4 tan = + inv inv 2s x = z dπαχα − α (3.3/32) 3.3 Meshing Interferences for Internal Gears 129 at2 2 tta t 2 2 a2 4t a n = + inv inv || 2 | |e x = z dπ− αϕα − α (3.3/33) and Įat1 and Įat2 are obtained from Equation (3.3/25). The end of the threatened region is reached when į1D = į1G + (į2D į2G) |z2|/z1 (3.3/34) 2 22 a2 a1 2D2 + 4 = arc cos 4| | | |aa dd ad − δ ⋅ (3.3/35) For rounded or chamfered tooth tip edges, the tip form diameter d Fa1 or d Fa2 is to be used for the tip diameters da1 and da2. To check for radial mounting or infeed meshing interferences, the entire threatened region must be examined; that is, j is to be calculated for the angle region į2G < į2j < į2D, or alternatively, for 0 j k. Following the determination of the auxiliary angles Įat1, Įat2 according to Equation (3.3/25), as well as Ȥ and ĳ according to Equations (3.3/32) and (3.3/33), the limiting angles į1D, į2D, į1G and į 2G and thus the function ǻj according to Equation (3.3/26) can be determined. The principal progression of this function is depicted in Figure 3.3/11. Figure 3.3/11 Qualitative progression of ǻ j from Equation (3.3/26) to check radial mounting or infeed meshing interferences a) without meshing interferences ( ǻjmin > 0) b) with meshing interferences ( ǻjmin < 0) When the curve touches or intersects the abscissa, radial mounting interferences or infeed meshing interferences caused in generation will occur. In such cases, the drive must be redesigned with a lower sum of profile shift m(x 1 + x2). For ra1 Ňra2Ň, a radial mounting is not possible, and a check of radial mounting interferences is therefore not required. In addition to this calculational method, the existence of radial mounting interferences (or infeed meshing interferences with generating gears) can be checked with the graphic technique depicted in Figure 3.3/12 (in accordance with DIN 3993 [3/15]). This requires that the tip form diameters of the pinion and internal gear intersect in the non-hatched area 1 or the hatched area 2 for im pact between the tooth tip edges to be avoided. If the point of intersection occurs in the hatched area 3, then it is necessary to recalculate using the above method. If the point of intersection occurs in field 4, then this generally relates to gears with meshing interferences. The example in Figure 3.3/12 corresponds to an internal gear pairing without radial mounting interferences and without tooth tip meshing interferences (with an extremely large operating pressure angle Į wt1). 130 3 Meshing Interferences Figure 3.3/12 Graphic for checking for meshing interferences No impact of the tooth tip edges outside the contact region. No risk of radial disengagement. No impact of the tooth tip edges. Radial disengagement only possible to a limited extent (recalculation required). Impact of the tooth tip edges and radial disengagement can only be analysed by computer. Impermissible region; meshing interferences 3.3.3.3 Passive Meshing Interference on Internal Generating Gear Pairing When generating internally toothed gears using a shaper cutter, then it is possible that not only the previously mentioned meshing interferences but also passive mesh ing interferences on the return stroke of the tool may occur. These meshing interferences do not occur when manufacturing with a hob. Passive meshing interferences after Muletarow [3/12] particularly occur if the rolling-off movement is maintained during the return stroke of the shaper cutter. The meshing interferences possible according to Equation [3/12] are indicated in Figure 3.3/13. It should be noted that although these interferences hardly result in changes of form on the internal gear, they do, however, cause abrasion wear and heating up of the tool and thus lead to additional loads on the rolling machine. As the size of passive meshing interferences not only depends on the geometry of the internal generating gear but also the kinematics of the machine tool, the precise calculation of the smallest permissible difference in the number of teeth z min = (Ňz2Ň z0) is only possible in the context of the tool machine data. 4 3 2 1 3.3 Meshing Interferences for Internal Gears 131 Based on practical experience and calculations, diagrams are provided in DIN 3993 [3/15], from which – on specification of the number of teeth of the internal gear ( z 2), number of teeth of the shaper cutter (z0), profile shift coefficient of the internal gear (x 2) – reference values for the profile shift of the shaper cutter can be inferred in order to avoid passive meshing inter-ferences. Figure 3.3/14 shows the principal struc-ture of these diagrams. The limiting line denoted by c lies beneath the limiting lines for infeed and tooth tip meshing interferences. By this means, the last-named gene-rated meshing interferences are also avoided if the conditions determined by the line c are fulfilled. Figure 3.3/14 Principal progression of meshing interference limits for the pairing of an internal gear with a shaper cutter. Lower limit for x0 (shaper cutter): a Undercut on the shaper cutter b Internal gear tip circle goes through the beginning of the generation path of contact along the shaper cutter base circle Upper limit for x 0 (shaper cutter): c Generation meshing interferences along the shaper cutter flank and the internal gear tooth tip during the return stroke of the shaper cutter during simultaneous rolling movement d Tooth tip thickness on shaper cutter 0.2 mn e Tooth root space width on internal gear (for special information see DIN 3993) Fig. 3.3/13 Passive meshing interferences ( Muletarow [3/12]) 132 3 Meshing Interferences 3.3.4 Information on Designing Internal Gear Pairs and on Tool Selection 3.3.4.1 General Aspects for the Design of Internal Gear Pairs It is good practice to undertake the design of an internal gear pair and the attendant determination of the toothing geometry for the pinion and internal gear according to the following considerations: 1. The pinion should (as in the case of the wheel) exhibit favourable load-bearing capability cha- racteristics. For this reason, Section 7.3 should be taken into account when selecting the profile shift coefficient x1. 2. A meshing interference-free operation of the internal gear pair (pinion / internal gear) and the generating gear (shaper cutter / internal gear) must be ensured. In order to fulfil the second requirement, all existing meshing interferences must be checked taking into account the following information once the number of teeth and the profile shift are known. a) Tooth root meshing interference on the pinion of the gear pairing or on the shaper cutter of the internal generating gear pairing As already described, these interferences occur if z1 or z 0 is very small compared to the number of teeth on the internal gear | z2|. If z1 is less than z0, then this meshing interference has to be checked both on the pinion of the internal gear pairing and the cutting tool of the generating gear. Starting roughly from (z 1 z0) 20, checking the internal generating gear pairing is generally sufficient. In the event of interferences occ urring, this can be remedied by Changing the gear pair data while leaving the tool data unchanged: • Change the distribution of the sum of the profile shift so that x 2 is less than ( x2new – x2old < 0) and making x1 correspondingly greater, • Reduce the sum of the profile shift by decreasing x2 (x2new – x2old < 0). Both methods are frequently required in order to keep changes in load-bearing capabilities low. Changing the tool data for the manufacture of the pinion while maintaining the gear pair data: • Select a shaper cutter with a smaller profile shift (regrind if necessary), • Select a shaper cutter with a greater number of teeth, • Select a tool with a greater ha0. Changing the shaper cutter data for the manufacture of the internal gearing to avoid tooth root meshing interferences on the shaper cutter: • Select a shaper cutter with a greater profile shift, • Select a shaper cutter with a greater number of teeth. b) Tooth root meshing interference on the internal gear This type of meshing interference occurs if the number of teeth on the shaper cutter is very small compared to the number of teeth on the pinion because the root fillet radius is large and the root form circle diameter of the internal gear ŇdFf2Ň becomes smaller. When using V-internal gear pairings with a negative sum of the profile shift ( Vminus-internal gear pairing), this interference becomes less important. A check should always be made to see if z 1 is significantly greater than z0 and a 0- or Vzero-internal gear pairing is being used. When using a shaper cutter with rounded or chamfered tooth tips, checking the meshing inter-ference in accordance with Equation (3.3/16) is recommended in every case. 3.3 Meshing Interferences for Internal Gears 133 For z0 > z1 and Įwt0 = 20 ... 28 this type of meshing interference does not generally occur. A sufficiently long involute is formed along the internal gear. If it is not possible to fulfil the requirement in accordance with Equation (3.3/16), then the problem can be remedied by using a shaper cutter with a greater number of teeth or with a smaller profile shift (possibly by re-sharpening) for the internal gear manufacture. Decreasing the profile shift x 2 on the internal gear may also prove successful ( x2new – x2old < 0). c) Tooth tip interference outside the plane of action This type of meshing interference (Section 3.3.3.1) can occur both on the internal gear pairing and on the generation internal gear pairing. With differences in the number of teeth, ( Ňz 2Ň z1) < 10, this interference is particularly prevalent. It can usually be avoided by using Xminus-internal gear pairing (negative sum of the profile shift). For internal generating gear pairings, tooth tip meshi ng interferences ar e to be feared if the differences in the number of teeth ( Ňz2Ň z0) is 12 due to the greater tooth addendum of the cutting tool. d) Radial mounting interference This type of interference is to be checked for if – for reasons of construction – the pinion has to be radially mounted (or brought into contact) with the internal gear. This occurs in analogy to tooth tip meshing interferences – particularly for small differences in the number of teeth ( Ňz 2Ň z1) – and is to be checked, as required, against Equation (3.3/26). If the radial mounting is free of meshing interference, then no tooth tip meshing interferences will occur. Conversely, however, this statement does not apply. e) Infeed meshing interference This type of interference only occurs on the internal gear of the internal generating gear pairing. Checking ensues in analogy to the radial mounting interference in accordance with Equation (3.3/26) (for index 1, index 0 is to be used) and is always recommended for differences in the number of teeth ( Ňz 2Ň z0) < 18. Infeed meshing interferences can usually be avoided by Xminus-internal generating gear pairs (reducing the sum of the profile shift). f) Passive meshing interference Passive meshing interferences only occur on internal generating gears and lead to unwanted flank contact between the shaper cutter and the internal gear during the return stroke with simultaneous rolling movement. Under these manufacturing conditions, the danger of passive meshing interferences can be reduced or avoided if the transverse pressure angle at generation, Į wt0, is selected as large as possible. For this reason, the shaper cutter with the greatest number of teeth should be used for manufacturing the internal gears. Diagrams for the selection of a suitable shaper cutter are available in DIN 3993 [3/15]. Return-stroke interferences are generally avoided if multi-skiving processing is used, as required by DIN 3993. Multi-skiving processing involves the dividing of the total depth of cut into individual depths of cut with radial infeeding per internal gear rotation. Passive meshing interferences are partly determined by the characteristics of the machine tool, and in individual cases it is necessary to request the adjustment data from the machine tool manu-facturer. A remedy may also be provided by the “radial infeed without rolling” method, that is, wor-king only with a radial infeed until the total depth of cut is reached and only then activating the roll motion. 134 3 Meshing Interferences Summarising on a) to f), it can be stated that – in addition to the possibilities described on avoiding meshing interferences – it is also possible to remedy or reduce meshing interferences by tip shor- tening on the internal gear or pinion and by increasing the pressure angle Į. In addition, the increasing of the helix angle ȕ also usually results in a reduction of the meshing interferences. It should be noted that İĮ 1.1 is a requirement for obtaining good generating conditions . In Table 3.3/2 (at the end of Section 3.3.4.3), directions for changing the individual gearing parameter are stated, which may contribute to a reduction of meshing interferences. 3.3.4.2 Profile Shift to Avoid Meshing Interferences on Internal Gear Pairs The aspects described above are crucial to the choice of the profile shift: • avoiding meshing interferences in operation and assembly, • a high degree of load capacity and favourable performance of the gear pair, • avoiding meshing interferences when generating internal gears. Provided that the scope of DIN 3993 [3/15] is complied with [ z 1 (10) 14; z2 ( 23) 40; (z1 + z 2) 1; Ȉx = 1.5 to +0.5; |ȕ | 30 and haP0 = 1.25 m (values in brackets are extreme cases)], it is possible to select meshing interference-free internal gear pairings from this standard even without detailed recalculation. For the specified numbers of teeth z 1 (pinion) and z2 (internal gear), the region of the permissible sum of the profile shift to avoid tooth tip meshing interferences can be taken from Figure 3.3/8 (refer to DIN 3993). The division of the sum of the profile shift into pinion and gear can ensue in accordance with diagrams whose schematic principle is depicted in Figure 3.3/15 (DIN 3993 contains illustrations allocated to the number of teeth of the internal gear). Figure 3.3/15 Example of the geometrical limits of the internal gear / pinion pairing. Lower limits for x1: Upper limits for x1: a undercut on the pinion d transverse contact ratio JĮ = 1.1 b internal gear tip circle goes through the beginning of the path of contact along the base circle of the pinion e tooth tip thickness on the pinion 0.2 mn [either (d) or (e) becomes the upper limit depending on the number of teeth] c tip edge contact of internal gear and pinion f tooth root space width on the internal gear 0.2mn If Figure 3.3/16 is also considered, then the required space width on the tooth root cylinder and the trivial internal gear limit ŇdaŇ > ŇdbŇ are therefore complied with. 3.3 Meshing Interferences for Internal Gears 135 Figure 3.3/16 Geometrical limits for cylinder gears with internal toothing as a function of the number of teeth znx of the equivalent spur gear and the profile shift coefficient x (without tooth tip shortening) ( efn = tooth root space width on the internal gear) The tooth tip thickness on the pinion and the tooth root space width on the internal gear should not be less than 0.2 m. It is possible that the desired tip clearance between the pinion tooth tip and the internal gear tooth root, c 0.1 ... 0.3m, is not always achieved due to the internal generating gear pair, which is usually in the form of a V-gearing. The existing tip clearance c should be determined with Equation (3.3/36) from the geometry values of the gear used: c = (|df2| da1) / 2 |a| (3.3/36) If no measurements are available, then the centre distance of the generating gear pair and the tip diameter of the shaper cutter must be known in order to be able to perform the calculation of d f2 using Equation (3.3/37): |df2| = 2| a0| da0 (3.3/37) For Xzero- and Vzero-internal generating gears, an overlapping of the basic racks of the internal gear and shaper cutter is available. In these cases, the root diameter df2 can be determined with Equation (3.3/38) from the internal gear and basic rack data: En a P 0 f2 2 = + 2 2 xm h dd − (3.3/38) 136 3 Meshing Interferences where xE is taken from Equation (3.2/10). If the desired tip clearance is not achieved, then corrections are possible, either by tip shortening or selecting a tool with a greater tip height. It is also possible to increase the tip clearance by greater profile shift coefficients Ňx1+x2Ň, if the tip height modification factor k is set to zero. 3.3.4.3 Shaper Cutter Selection to Generate the Internal Gears Except for broaching with usually low numbers of teeth and hobbing with larger ring gear diameters ( ŇdŇ 800 mm), then the manufacture of the internal gears should be undertaken using the shaper cutter. The shaper cutter and the internal gear to be generated form an internal generation gear that can exhibit meshing interferences just as much as the internal gear pairing itself (internal gear / pinion). The difference is that the possibility of meshing interferences occurring is greater for internal generation gears, given the same difference in the number of teeth, | z 2| z1 = |z2| z0. The reason for this lies in the addendum of the shaper cutter, which compared to the pinion is greater by the amount of the tip clearance. Two important aspects are critical in the selection of the shaper cutter: • A standard shaper cutter should be used (DIN 1825 [3/16], DIN 1826 [3/17], DIN 1828 [3/18]). • Generation meshing interferences must be avoided (this also includes the passive meshing interferences during the return stroke of the tool). These requirements can be fulfilled if the selection of shaper cutter (standardised shaper cutter are preferred) ensues with the help of diagrams contained in DIN 3993 [3/15]. From the perspective of avoiding meshing interferences, it is recommended that the shaper cutter with the greatest possible number of teeth be selected, since a large shaper cutter forms a longer tooth flank region as involute. This decreases the risk of tooth root meshing interference on the internal gear. It should, however, be pointed out that increasing the number of teeth on the shaper cutter results in sharper tooth root fillets on the internal gear, and, therefore, unfavourable effects on the load-bearing capacity of the tooth root must be expected. The upper limit of the number of teeth on a shaper cutter for manufacturing an internal gear or the lower limit of the number of teeth on a internal gear for a given shaper cutter is generally determined by the condition • avoidance of passive meshing interferences and thus • avoidance of infeed and tooth tip meshing interferences. The lower limit for z0 (for a given z 2) or the upper limit for z2 (for a given z0) is provided by the condition • avoidance of tooth root meshing interferences on the cutting tool. If the last-named limit for z0 is fallen short of, or for z2 is exceeded, then parts of the involute flank region on the tooth tips of the internal gear will be cut away. A requirement on the selection of a shaper cutter based on the diagrams in DIN 3993 [3/15] is that a shaper cutter with a tooth addendum of ha0 = 1.25 m be used. 3.3 Meshing Interferences for Internal Gears 137 Table 3.3/1 Types of Meshing Interferences Section Basic design conditions Meshing interferences 3.3.2.1 Schematic diagram Condition |da| > |d b| ef > 0 İĮ 1.1 c 0.2 m san1 0.2 m dNf1 > dFf1 Name Tip diameter Tooth space width Profile contact ratio Tip clearance Tooth tip thickness (pinion) Meshing interferences between the internal gear tooth tip and the pinion root (tooth root meshing interference on the pinion) Serial no. 1 1.1 1.2 1.3 1.4 1.5 2 2.1 138 3 Meshing Interferences Table 3.3/1 Types of Meshing Interferences (continued) Section Meshing interferences 3.3.2.2 3.3.3.1 3.3.3.2 Schematic diagram Condition |dNf2| < |dFf2| The point of intersection D of the tip circle must lie above the contact straight lines g. Otherwise a further check is required, as in the figure. The point of intersection D of the tip circles must lie above the straight line e , which passes through the points of intersection of the basic circles. Otherwise a further check is required, as in the figure. Name Meshing interferences between the pinion tooth tip and the internal gear tooth tip (tooth root meshing interference on the internal gear) Tooth root meshing inter- ferences outside the plane of action Built-in radial interferences on the drive or infeed meshing interferences in association with generating gears Serial no. 2 2.2 2.3 2.4 3.3 Meshing Interferences for Internal Gears 139 Table 3.3/2 Change of Direction of Gearing Parameters to Avoid Meshing Interferences for Specified Numbers of Teeth in an Internal Gear Pairing Type of meshing interference Possibilities for avoiding meshing interferences Pinion Internal gear Shaper cutter Remarks x1 da1 x2 |da2| x0 z0 Tooth root meshing interferences on the pinion ↑ ↓ ↑ Ȉx = constant; tool data unchanged con- stant ↓ ↑ Ȉx = reduced; tool data unchanged ↑ ↓\* ↑\* Gear pairing data unchanged on the internal gear con- stant ↓ ↓ Ȉx = reduced; tool data unchanged ↓ ↓ ↑ Gear pairing data unchanged on the shaper cutter ↑ ↑ ↑ Tooth root meshing interferences on the internal gear pairing ↓ ↓ ↓ ↑ on the generating internal gear pairing ↓ ↑ ↓ Radial mounting interference ↓ ↓ ↓ ↑ Infeed meshing interference ↓ ↑ ↓ ↓ Passive meshing interferences in association with the return stroke with simultaneous rolling motion ↓ ↓ as large as possi-ble In borderline cases, consultation with the machine manufacturer ↓ Reducing ( x2new – x2old < 0) or ↑ increasing ( x2new – x2old > 0) the respective geometrical gearing geometry in order to rectify or reduce existing meshing interferences. \* Arrows refer to the shaper cutter for the manufacture of the pinion (otherwise generally to the shaper cutter for internal ge ar generation). 140 3 Meshing Interferences 3.4 Symbols and Symbol Explanations A - point at start of action a mm centre distance a0 mm centre distance of generating gear pair b mm tooth face width c mm tip clearance d mm reference diameter da mm tip diameter db mm base diameter db0 mm base diameter of shaper cutter df mm root diameter (nominal value) du mm diameter at transition into undercut trace dw mm pitch diameter dy mm diameter of Y-circle dFa mm tip form diameter dFa0 mm tip form diameter of shaper cutter dFf mm root form diameter dFf0 mm root form diameter of shaper cutter dNf mm useable root diameter E - point at end of action Es ȝm tooth thickness deviation (at reference cylinder) ea mm space width at tip cylinder ef mm space width at root cylinder F1,F2 - intersection of line of action and root form circle of gear 1 and 2 fA,E mm elastic tooth deformation fHȕ ȝm helix slope deviation fȈȕ ȝm axial skew fȈį ȝm axial inclination gĮ mm length of path of contact (total) ha0 mm addendum of tool haP mm addendum of gear reference profile haP0 mm addendum of tool reference profile hFaP0 mm tip form height of tool reference profile hS mm height of grinding step i - gear ratio j - counting variable k - tip height modification coefficient, number of steps m mm module 0 - centre point (cylinder; circle) pet mm base pitch (in transversal plane) ra mm tip circle radius sa mm tooth thickness at tip cylinder sFK mm tooth root thickness of undercut region sE mm tooth root thickness at transition of involute to undercut trace T - point of contact of line of action with base circle t s time u - gear ratio x - profile shift coefficient x0 - profile shift coefficient of shaper cutter xE - manufacturing profile shift coefficient z - number of teeth zg - limit of number of teeth zn - number of teeth of equivalent gears used for calculating profile shift coefficients z0 - number of teeth of shaper cutter Į pressure angle Įa profile angle at tip cylinder Įn normal pressure angle Įn0 generating normal pressure angle Įt0 generating transverse pressure angle Įw operating pressure angle Įw0 operating pressure angle of generating gear pair ȕ helix angle ȕb helix angle at base cylinder į auxiliary angle įD auxiliary angle įG auxiliary angle İĮ - transverse contact ratio İȕ - overlap ratio İȖ - total contact ratio tangent angle D auxiliary angle auxiliary angle Na0, ȡa0 mm radius of tool rounding ĳ auxiliary angle Ȥ auxiliary angle ȥ auxiliary angle 7 1/s angular velocity ǻa mm centre distance deviation ǻdFf mm root form oversize, difference between dNf and dFf Ȉx - sum of profile shift coefficients Indices min minimum value n normal plane (also for equivalent spur gearing of a helical gearing) t transverse plane or in tangential direction S shaper cutter Z rack-type tool 0 tool or generating gear pair 1 pinion of a gear pair 2 wheel of a gear pair \* head pointer, which is used for expressing a partition or multiple of normal module (e.g., haP = h\* aP mn) 141 4 Special Involute Gearing 4.1 Types and Possibilities Even though the basic rack is standardised (ISO 53, DIN 867) for general mechanical engineering and heavy machine construction, deviations from it are common. Apart from fine-pitch gearing design (module m = 0.1 to 1 mm), for whose special needs there is a standardised profile (DIN 58400; see also ANSI/AGMA 1103-H07) with a modified root geometry, special demands such as • very quiet running, • high load capacity with respect to fatigue, • high load capacity in cases of impact loading or high overload capacity in cases of low numbers of load cycles, and • achieving extremely small numbers of teeth in particular lead to special gearing. Below, special involute gearing will be discussed (refer to [4/4] for special non-involute gearing). Deviating from the standardised basic rack always results in higher tool costs. In the case of mass production, such as in the motor vehicle industry, these additional expenditures are irrelevant, just as they are in the case of expensive large gearing for an excavator, for example. Pressure angles that differ from Į = 20 , other tooth depth factors, modified selections of tooth thickness that are independent of profile shift, and unsymmetrical basic racks (Figures 4.1/1 to 4.1/4) are used. In the borderline case, gears can be realized this way with one tooth (Figure 4.1/8). Often extra-depth gearing is used to achieve very quiet running. With a transverse contact ratio of F B = 2, a smaller variation of gearing stiffness is achieved than with the standard profile and thus less excitation of vibrations, which results in smaller internal dynamic tooth forces and lower gear noise. An overlap ratio of F C with an integer value improves it even more. If the advantages of extra-depth gearing related to noise and load capacity are to be achieved, the flank modification must be determined using methods which allow the spatial mesh conditions to be precisely taken into account. Pressure angles of Į < 20 – usually Į = 15 or Į = 17.5 – are used. As tooth addendums, instead of h aP = 1.0 m for the standardised profile (DIN 867), values up to roughly haP = 1.3 m are common. Because of the larger dedendum and the potentially smaller pressure angle, without a profile shift it is not possible to achieve such small numbers of teeth as in standardised gearing. a) Į = 15 b) Į = 20 c) Į = 28 Figure 4.1/1 Gearing with different pressure angles (standard tooth depth; z = 30; x = 0; ȕ = 0 ; Na0/m = 0.2) 142 4 Special Involute Gearing a) Gearing according to ISO 53 (DIN 867) b) Extra-depth gearing haP/m = 1.3 ; h fP/m = 1.55 Figure 4.1/2 Standard gearing and extra-depth gearing ( z = 30; x = 0.35; ȕ = 0 ; ࣁa0/m = 0.2) Figure 4.1/3 28 -gearing (according to Figure 4.1/1c) compared to 20 -standard gearing (according to Figure 4.1/1b) Figure 4.1/4 Example of gearing with different pressure angles for the working (main load) flank and non-working flank (refer to [4/5] to [4/10]); z = 30; β = 20 ; haP/m = 1.0; hfP/m = 1.25; x = 0.6; ɲn left = 40 ; ɲ n right = 10 With an addendum of ha = hfN = 1.3 mn, Į = 15 and ȕ = 0 , the theoretical limiting number of teeth is zg = 39, and with ȕ = 15 and otherwise the same values, zg = 35. That is why often Į = 17.5 is realized ( zg = 29 with ȕ = 0 or zg = 28 with ȕ = 15 and ha = 1.3 mn), or in cases of small numbers of teeth, an undercut is avoided by applying a positive profile shift. For a helix angle of ȕ = 10 , Figure 4.1/5 shows the necessary tooth addendum ha/mn for a trans- verse contact ratio of İĮ = 2.0, dependent on the number of teeth, gear ratio, and profile shift. Figure 4.1/5 Extra-depth gearing: addendum factors h a/mn for transverse contact ratio İĮ = 2, pressure angle Į = 17.5 and helix angle = 10 4.1 Types and Possibilities 143 If larger pressure angle s are used, such as Į = 28 , the load capacity can be considerably improved. If the load cycle numbers are low (for instance, N < 104), in the case of normalised and quenched and tempered gearing, the tooth root notch has hardly any impact on the yield limit of the tooth profile, and the larger tooth root thickness takes full effect. Its influence prevails over the increasing bending lever arm, which develops as a consequence of the smaller transverse contact ratio in cases of larger pressure angles. It is different altogether in cases of gearing with a hard layer . Since in the case of increasing pressure angles the radii of curvature of the root fillet become smaller (because of the smaller space width), the notch effect increases. For the load limit, the maximum local tooth root stress (crack) and the load in the overall tooth profile (plastic deformation) must be considered. Different influences take effect when the pressure angle increases. These mainly include the increasing size of the tooth root thickness, the increasing bending lever arm, and the increasing notch effect, which somewhat but not completely cancel one another out. Higher load capacity is usually achieved with this special design (e.g., Į = 28 ), as opposed to gearing with the standardised basic rack in accordance with ISO 53 (DIN 867). In Figure 4.1/3 these are shown in comparison to 20 -gearing. Figure 4.1/6a illustrates the transverse contact ratio İ Į dependent on the number of teeth and the gear ratio for spur gearing, and Figure 4.1/6b illustrates the tooth tip thickness sa. Figure 4.1/6 a), b) 28 -gearing \* a aP1; 0 ; hm h x == = 0β=D a) Transverse contact ratio b) Tooth tip thickness 144 4 Special Involute Gearing If there is only one main direction of loading, a high flank and root load capacity can be achieved with unsymmetrical teeth, by designing the working flank with a large pressure angle, such as Į = 40 , and the non-working flank with a small pressure angle, such as Į = 10 . With that, for the loaded flanks one is already very close to the pressure angle Į = 45 , which represents the optimal value with respect to flank load. As an example, Figure 4.1/4 shows gearing with different pressure angles for the working and the non-working flanks. The profile shift is also envisaged in order to avoid undercut on the non-working flank, but this could have also been realized through different basic racks (tools!) for gear and mating gear. The utilization of every last reserve with respect to load capacity or the realisation of extremely low numbers of teeth leads to complementary profiles ([4/1] to [4/3]). For the benefit of geometric and load capacity advantages, one deliberately does without a common basic rack here (no identical tools for the gear pairing) and may well design the gearing for the purpose of mating ability for gear and mating gear with complementary basic racks. Figure 4.1/7 Complementary basic racks Figure 4.1/7 shows general complementary basic racks. Determining them is a real optimisation task. Here the following quantities must be involved: h aP/hfP, the size of the tooth depths and the ratio of addendum to dedendum ( ha1, hfN2; ha2, hfN1); s1P/s2P, the ratio of tooth thicknesses or distribution of the pitch on the tooth thicknesses of the pairing at the pitch circle, respectively; Į IP, ĮIIP, the pressure angle of working and non-working flank Įn = ĮIP or Į n = ĮIIP; and NfP, the root fillet radius. Figure 4.1/8 shows a gear with a pinion with only one tooth (complementary profile). Figure 4.1/8 Pinion with one tooth (z = 1) (Model collection, Institute of Electromechanical and Electronic Design, TU Dresden) 4.2 Design Because of the ability to change all quantities of the basic rack and to specify complementary basic racks in the case of negligible tool costs for gear and mating gear, there is no obvious need for profile shift. 4.2 Design 145 The design begins with preset values that have been specified. If required, it is necessary to iteratively approximate the global optimum using new starting values. In addition to low noise, usually extremely small numbers of teeth or the maximum utilisation of load capacity reserves (e.g., pitting load capacity) are of primary importance for the design. As follows, a few relationships are given which are considered when designing complementary profiles. It is assumed that the determination of the centre distance a from the contact stress is a viable starting point. With the chosen or stipulated numbers of teeth and the chosen position of the length of the path of contact on the line of action, taking the limiting conditions (e.g., tooth tip thickness) into consideration, it is then possible to determine the distribution of tooth thicknesses on the pitch circle. A sufficient load capacity of the tooth root must be verified in a subsequent repeat of the calculations, and if necessary a new design chosen with a new number of teeth or a new distribution of the pitch circle pitch in the tooth thicknesses. From the equation for contact stress t HH E ȕB,D H 1İ1 FuZZZ Z Z Kbd uσ+=⋅ ⋅ with Hv H Į Hȕ K KKK K=$ , ,21t1 tdMF= ()11 bb d d= ,1 wd d= 12 1 w+=uad , 1 2z z u = for 21zz≥ results the required centre distance based on the main transmission direction (“working flank” Įn = ĮIP; Figure 4.1/7): 2 H132 HP 12 11 2ıZK M uuabd u++=⋅ ⋅ (4.2/1) with HE ȕB,Dİ ZZZZZ Z= (4.2/1a) For the dimensioning, the following must be defined: HPı Permissible contact stress b/d1 For case-hardened gearing b/d1 ≤ 1; for quenched and tempered gearing b/d1 ≤ 1.2 (running in) KH According to the overload relevant for the fatigue load capacity ( KA). If this is already included in M1 as equivalent torque, initially KH = 1.3 may be used for calculations (consideration of KHȕ). Z According to Equation (4.2/1a) with the selected αn = αwn = α IP and ȕ for Z H according to Equation (6.5/9b); Zİ = 1; ZE appropriate for the chosen material according to Equation (6.5/9a); ZB,D 1.1; Zȕ according to Equation (6.5/9f). Note: In subsequent re-calculation, ZB,D must be included [refer later to Equations (6.5/9d) and (6.5/9e)], whereby, particularly with an extremely small number of teeth and a larger shift of the field of action vis- -vis the pitch point, an important influence is taken into consideration. With the normal pressure angle αn, the chosen helix angle β, the transverse pressure angle α t resulting from this, and the given gear ratio u = z2/z1 = 12ȦȦ− , the base circle radii rb1,2 for αwt = αt are t b1cosĮ 1aru⋅=+ (4.2/2a) b2b 1rr u= (4.2/2b) 146 4 Special Involute Gearing This results in the base pitch or normal base pitch pb = pe, if the number of teeth is either already determined or chosen as a start. 1 et bt b12ʌ p pr z== (4.2/3) Here, the module does not have the usual meaning of the standardised size factor. In the transverse section it is et t (ʌcosĮ)tmp= , and in the normal section it is nt cosȕ mm= . In the case of helical gearing, for İĮ theoretically one could also use İĮ < 1 as a starting point. With a transverse contact ratio of İĮ specified, the length of the path of contact gĮ is defined (e.g., extra- depth gearing, İ Į = 2): ĮĮ etİ gp= The length of the path of contact g Į is to be distributed on the portions ga1 and g a2 on the two sides of the pitch point (Figure 4.2/1a)1). In the case of speed-reducing ratios, especially with small numbers of pinion teeth, one strives for a higher sliding speed on the pinion tip than on the gear tip. Because of this, the contact is shifted more towards the area of pulling sliding and away from the base circle of the pinion. The overlap ratio (an integer value is advantageous) results from the centre distance a and the face width b in ()b12 1 ȕ ttanȕ 1İ2ʌcosĮbz z z a+= (4.2/4) In the case of speed-increasing ratios, on the other hand, a higher sliding speed on the gear tip than on the pinion tip is deemed favourable. With regard to the required distance from the lowest meshing point on the pinion root to the base circle, however, a compromise is often necessary. If one chooses the ratio g a1/ga2 (e.g., with ga1/ga2 = 1.2 in the case of speed-reducing ratios) and with that defines the tip paths of contact ga1 and ga2 based on ga1 + ga2 = gĮ, then it must be verified whether the conditions b2 a1 t <t a nĮ2dg and b1 a2 t <t a nĮ2dg (4.2/5) are met. Otherwise a new distribution or new design is necessary. To achieve low contact stress even with u > 1, in extreme cases one will not only try to choose a large pressure angle, but also try to push the length of the path of contact in the direction of the centre of the line of action (Figure 7.3/1, centre of 12TT). For gear 2 then g a2 < 0 is possibly true, and very large sliding ratios ( / g// t) can exist, which must be taken into account. Besides that, tooth thickness proportions may exist that are unfavourable for the tooth root stress. With the definition of the position of the path of contact or the ratio ga1/ga2, respectively, the tip diameters da1 and da2 are also determined: ()22 a1,2 b1,2 b1,2 t a1,2 2t anĮ dr r g =+ + (4.2/6) The working depth hw is a1 a2 w22ddha=+− (4.2/6a) 1) Often ga1 with ga and g a2 are simply marked as gf. 4.2 Design 147 a) b) c) Figure 4.2/1 Minimum and maximum tooth thickness in cases of free distribution of the pitch on the working pitch circle ( z1 = 10; u = 2.5; ȕ = 0 ; α = 25 ; İĮ = 1.15): a) gear pairing b) minimum tooth thickness sw1min and maximum tooth thickness sw1max on the working pitch circle of gear 1 c) minimum tooth thickness sw2min and maximum tooth thickness sw2max on the working pitch circle of gear 2 After choosing the tip clearance c1,2, one arrives at the root diameter d f1,2: f 1,2 a1,2 w 1,2 22 dd h c =− − for example with (4.2/7) 1,20.252whc=⋅ The working pitch diameters dw1,2 (which do not differ from the reference circle diameters d1,2 in the case of this “free” design) are given because of the centre distance a and the gear ratio u = z 2/z1 = 12ȦȦ− : w12 1adu=+ (4.2/8a) w2 w1dd u= (4.2/8b) 148 4 Special Involute Gearing With Įwt = Įt, the pitch pwt on the working pitch circle is bt wt t t cosĮppp== The sum of the tooth thicknesses on the pitch circles is equal to the pitch circle pitch: wt1 wt2 wtss p+= (4.2/8c) The value pwt can be divided into swt1 and swt2 according to the respective, pertinent aspects, for example, for a maximum in the tooth bending safety factor, for specifically influencing tooth stiffness, or to comply with the minimum tooth tip thickness, especially in the case of small numbers of pinion teeth. As follows, it is deduced how large s wt must be to ensure that the tooth tip thickness does not fall below a minimum value; for example, san1,2min = 0.1 mn: an1,2 an1,2minss ≥ The minimum tooth thickness swt on the pitch circle follows from the minimum tooth tip thickness. To make the deduction, the general case is assumed that the right and left flanks have different pressure angles. The right (I) and the left flank (II) of a tooth then also have the same pitch circle, but different base circles. The tooth thickness s at on the tooth tip in the transverse section results from the tooth thickness san on the tooth tip in the normal section in an at a cosȕss= with ȕ a according to Equation (2.2/5). With the general assumption of different pressure angles for the working and non-working flanks, the minimum tooth thickness on the pitch circle, which falls within the tooth thickness on the tip circle, is anm i n w wt min atI atII wtI wtII aa2invĮ invĮ invĮ invĮ2c o s ȕs dsd =+ + − − (4.2/9) with bI,II atI,II aI,IIcosĮd d= In the case of basic racks that have the same profile angle for the working and non-working flanks (symmetrical gearing), the result is anm i n wt min w at wt aainvĮ invĮcosȕssdd =+ − (4.2/9a) with dw according to Equation (4.2/8a,b); at a b cosĮ dd= ; wt tĮ=Į; () aatanȕ tanȕ dd= A minimal tooth thickness on the pitch circle for each gear of the pairing ( swtmin1,2 ) results. The gearing can only be realized if wtmin1 wtmin 2 wtss p +≤ (4.2/9b) This condition is necessary but not sufficient; that is, compliance with other conditions, such as no interference, no undercut (for the smaller of the profile angles) and sufficient space width, must be verified. 4.2 Design 149 If a tool tip radius קa0 is specified, when a hob or rack cutter is used, the condition according to Equation (4.2/10) must be satisfied for the gearing to be produced: () [ ] []f nI wn wn nI nII nI nI a01 nII nII nII a01tanĮ tanĮ cosĮ (1 sinĮ)t a nĮ cosĮ (1 sinĮ)t a nĮsp h≤− + − − − −− −N N (4.2/10) with wn wt cosȕ, pp= w wn min wt min cosȕ, ss = ()fw f1 2, hd d=−wȕȕ= In the case of basic racks that have the same profile angle for the working and non-working flanks (symmetrical gearing), the result is [ ]wn wn f n a 0 n n n 2t a nĮ2c o sĮ(1 sinĮ)ta nĮ sp h≤− − − − N (4.2/10a) It is also possible to use Equation (4.2/10) to calculate the potential tip radius in the case of a known or specified swn. In the case of symmetrical gearing it is ()wn wn f n a0 nn n2t a nĮ 2c o sĮ 1s i nĮtanĮps h−−⋅≤−− N (4.2/10b) An adequate load capacity of the tooth root must be verified in a subsequent repeat of the calculations (Section 6.5.2.1), whereby the modification of the factor YFa and (in the case of precise analyses) YS must be considered. When calculating the tooth root thickness, in the case of different working and non-working flanks, the quantity X [refer to Equation (2.3/3) and Figure 2.3/1a] must be determined for each half of the tooth, defined by half the tooth thickness on the pitch circle. It is then sF = XI + XII (4.2/10c) In precise analyses and in the case nPI nPIIĮĮ≠ , the cross section which results from the 30 tangent on the root fillet of the flank under the most load, with the distance Y to the gear centre [ Y according to Equation (2.3/4)] applies as a reference cross section. With that, for the non-working flank an angle results which deviates from 30 . In the case of large deviations from ĮP = 20 , the approximate equations and numerical results from Section 6.5.2.1 (likewise from ISO 6336-3, DIN 3990-3) only roughly apply for taking the stress concentration into consideration. Therefore, for more precise examinations, calculations with numerical methods (BEM or FEM) are recommended. In an example, Figures 4.2/1a to 4.2/1c show the limits of the tooth thickness when dividing the pitch of the working pitch circle p wt. In the case of a design that is far from the typical data, the real increase in load capacity should be experimentally tested, since otherwise, obscured influences may not be neglected anymore. Table 4.2/1 lists the geometric relationships for designing extra-depth gearing with complementary profiles. Compared to standard gearing, extra-depth gearing (also with complementary profiles) has larger minimum tooth numbers. With more precise analyses (among others with the PC programme LVR [6.4/33] ) it was ascertained that, in cases where the module is the same, extra-depth gearing has a higher load capacity than gearing with a standardised profile according to ISO53 (DIN867). Here the flank modifications are also important. Extra-depth gearing is more susceptible to scuffing than standard gearing. Fewer losses are achievable through smaller tooth depths with larger pressure angles. However, tooth root stress and contact stress increase, especially because of the decreasing transverse contact ratio (refer to Appendix 10.1). 150 4 Special Involute Gearing Through unsymmetrical gearing, the contact stress of the working flank can be reduced. However, it must be noted that in the case of this gearing, the tooth root stress also influences the stress in the adjacent tooth root contour more than usual and in even larger magnitude. The alternating stress (double amplitude) that exists with this, as well as the influence on the strength value, must be taken into account in general (refer to Appendix 10.2 and [4/5], [4/6], [4/7], [4/8], [4/9], [4/10]) . Table 4.2/1 Geometric Relationships for Designing Symmetrical, Special Involute Gearing; Among Others, Extra- Depth Gearing with Complementary Profile ( BnI = BnII) Note : For unsymmetrical special gearing (Figure 4.1/7), the calculations are analogous to these equations; however, swt min is according to Equation (4.2/9). Cons. No. Name Equation 1 Initial data a; z1; z2; ȕ; Įn; b; ga1/ga2; san min (İα); (İȕ preferably integer values) 2 Pitch diameter ( dw = d) w1 122 1adzz=+; w2 w2 2 da d=− 3 Transverse pressure angle (Įwt = Į t) wt wn tanĮ tanĮ/c o sȕ = 4 Base diameter b1w 1 w t cos dd=α ; b2w 2 w t cos dd=α 5 Transverse normal base pitch et b1 1p dz=π2 b2z dπ = 6 Length of contact, limiting value 12 w tsinĮ; TT a=wt tĮĮ= 7 Length of path of contact ĮĮ et Į İ ;(e.g. high contact ratio gearing İ 2) gp== 8 Test A (Is extra-depth gearing achievable?) 12 gT Tα< ? (if not: new initial quantities, e.g., z 1,2 larger) 9 Partial lengths of contact, limiting values ()wt b1 1 tan2 α =d C T ; 1 2 1 2z z C T C T⋅ = 10 Parts of the paths of contact C Tg ggg a aa 2 2 111≤+=α; 1 2a ag gchosen C T g g g1 a1 2 a ≤ − =α 11 Minimum tooth tip thickness an1mins , an2mins chosen; e.g., n anmin 0.2 sm = z d mn β = cosw 12 Tip diameter () ()22 11 a 1 1 22abdd g TC =+ + 4.2 Design 151 Table 4.2/1 Geometric Relationships for Designing Symmetrical, Special Involute Gearing; Among Others, Extra- Depth Gearing with Complementary Profile ( BnI = BnII) (Continued) Cons. No. Name Equation 12 () ()22 2b 2 a 2 2 22 add g TC =+ + 13 Minimum tooth thickness on the pitch circle (due to s an min) an1min wt 1 m i n w 1 at 1 wt a1 a1invĮ invĮcosȕssdd =+ − an 2min wt2 m i n w 1 at2 wt a2 a2invĮ invĮcosȕssdd =+ − cos Įat = d b /da tan ȕa = (da /d w) tan ȕ 14 Test B (Is san min achievable?) wt 1 m i n wt2m i n w tss p +≤ ? z d pπ =w wt 15 Dividing the pitch of the working pitch circle in tooth thicknesses wt 1 wt 1 m i n ss≥ , wt wt 2 wt 1 wt2 m i n sp s s=−≥ 16 Test C (Reasonable tooth root thickness values?) Approximate calculation of the stress ratio on the diameters of the active flank root dNf ; ()2 Nf t1 Nf t 2 F2 F1ss σσ≈ typical range: F1 F2σ σ= 0.7 ... 1.3 () Nf t1 Nf 1 w t1 w1 w t Nf 1 inv inv sd s d αα =+ − ()Nf2 wt w2 wt2 2 Nf 2Nft inv invα − α + = d s d s () ()122 Nf1 b1 a2 22 ddT C g =+ − () ()22 Nf 2 b2 1 a1 22dT C g d =+ − Nf 2 b 1 ,2 Nf1 ,2 cosĮ dd= 17 Test D (Is strain permissible?) Check the strain [if necessary re-distribute the length of path of contact (g a1, ga2) or the tooth thicknesses ( swt1, swt2) or re-determine the number of teeth] 152 4 Special Involute Gearing 4.3 Symbols and Symbol Explanations a mm centre distance of a cylindrical gear pair b mm tooth face width c mm tip clearance d mm reference diameter da mm tip diameter df mm root diameter (nominal value) Ft N nominal tangential force at reference circle in transverse plane gĮ mm length of path of contact ha mm addendum of the tooth haP mm addendum of the gear reference profile hfN mm useable dedendum hfP mm dedendum of the gear reference profile hw mm working depth of a gear pair KA - application factor KH - load factor KHĮ - transverse load factor (contact stress) KHȕ - face load factor (contact stress) Kv - internal dynamic factor m mm module (reference diameter pitch) N - number of load cycles p mm pitch at reference cylinder p b mm pitch at base cylinder pe mm normal base pitch pw mm operating pitch ra mm tip circle radius rb mm base circle radius rf mm root circle radius Indices b dimension at base cylinder n dimension in normal plane (also for dimension of equivalent spur gearing of a helical gearing) r w mm pitch circle radius sa mm tooth thickness at tip cylinder sNf mm tooth thickness at useable root cylinder sP mm tooth thickness of gear reference profile sw mm tooth thickness at pitch cylinder M Nm torque u - gear ratio ZE - elasticity factor ZH - zone factor Zβ - helix angle factor Zİ - contact ratio factor z - number of teeth Į pressure angle Į P profile angle of gear reference profile Įwt operating pressure angle in transverse plane ȕ helix angle at reference cylinder ȕ a helix angle at tip cylinder İĮ - transverse contact ratio Na0, ȡa0 mm tip radius at tool NfP, ȡfP mm root fillet radius at gear reference profile ıH N/mm2 Hertzian pressure, existing stress ıHP N/mm2 allowable flank pressure Ȧ s-1 angular velocity t transverse plane I working flank II non-working flank Circular Spline FlexsplineWave Generator1. Start 2. 1/4 Input rotation 3. 1/2 Input rotation 4. 1/1 Input rotation Harmonic Drive AG develops, designs, manufactures and supplies, providing its customers with technically advanced, innovative products, which are characterised by outstanding accuracy, high power density, compact size, light weight and long life. We provide individual and professional advice, our word counts – we urge you to challenge us!Our inspiration Harmonic Drive AG | Hoenbergstra e 14 | 65555 Limburg/Lahn | T +49 6431 5008-0 www.harmonicdrive.co.ukHigh precision Harmonic Drive Gears and Actuators „made in Germany“ are used around the world in all important key industries, including robotics, machine tools, medical technology, aerospace and automotive. QUICKLINK www.harmonicdrive.co.uk/0020 PERFECTLY CALCULATED AND ACCURATELY MANUFACTURED. HELICAL GEARS AND ANY SOPHISTICATED TOOTHED COMPONENTS ... 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By location of damage: • damage as a result of flank stress • damage as a result of root stress (including gear body influence) By type of damage: • damage as a result of maximum stress or contact stress due to exceeding the temperature limit • damage due to exceeding the fatigue strength And relevant loads should be noted: • maximum value • total load cycle course Although other types of damage due to corrosion, current flow, contamination, and so on are also possible, these – as well as damage caused by the machining process (grinding cracks, grinding burn, scale formation, forging defects, hardening cracks) – are not further considered here (DIN 3979 and ZFN 201, a group standard of ZF Friedrichshafen AG). The individual types of damage are briefly described below as to appearance and cause (refer also to ISO 10825: Gears – Wear and Damage to Gear Teeth – Terminology; ANSI/AGMA 1010-E95: Appearance of Gear Teeth – Terminology of Wear and Failure). Damage as a result of flank stress a) Pitting Pitting is the result of material fatigue in the area close to the surface. Fine cracks appear, into which oil can seep. When rollover occurs, the openings are sealed and ultimately particles are blasted out. The indentations resulting thereby are known as pittings (Figure 5.1/1). Pitting mainly occurs below the working pitch circle in the region of negative sliding and is invariably the result of repeated rollover. These chippings are absolutely inadmissible if the parts of the flank surfaces destroyed progressively increase with time. For hardened materials, the pitting fatigue strength is reached with load cycle numbers of N ≥ 50 10 6. The underlying reason for this material fatigue is the contact stress. Although primarily charac-terised by the Hertzian contact stress, the shear stress as well as the stress caused by alternating temperature as a result of sliding and the bending stress at the tooth flank are effective components. Finally, roughness and waviness affect the contact geometry. Also as a result of the hydroelastic effects, the pressure distribution in the contact zone will change somewhat. Overlaid by the aforementioned influences, a spatial stress gradient is created by the surface contact, which – starting from the surface – continues below the surface. The starting point of pitting, originating in the formation of cracks, can be the surface (Figure 5.1/2). But often it also starts inside the material. The determining factor is the relation between the effective stress (equivalent stress) and the strength profile below the surface. 156 5 Load Capacity: Introduction, Initial Values Figure 5.1/1 Pitting in a case-hardened gear; 20MnCr5; N 106; ıH 1600 N/mm2; m = 6 mm; CHD 1 mm; b = 16 mm; TU Dresden test result Figure 5.1/2 Formation of cracks on the surface of a case-hardened gear with pitting (approx. 100-fold magnification, data as in Figure 5.1/1) An equivalent stress results from the three stress components of the Hertzian contact, which – calculated in accordance with the maximum distortion energy theory postulated by von Mises – possesses its maximum beneath the surface. Particularly with extremely large surface-hardened or nitrated gears, from a mathematical perspective the maximum stress often is below (or in the transition of) the hardened layer, so that here the crack can originate inside the material. Pitting is inadmissible if it proceeds progressively, whereby – if not checked – it can result in impermissible vibrations, impermissible noise and ultimately tooth breakage. Initially occurring pitting that then stops expanding (non-progressive pitting) is conditionally permissible. It is frequently the case that fatigue fractures originate with pittings, particularly with case-hardened or nitrated gears, where a case-by-case assessment, also taking into account the importance of the system, is required. Therefore in borderline cases, such as surface-hardened gears in the aviation industry, even a single pitting is inadmissible. Figure 5.1/3 Spalling on a case-hardened helical gear; ZF Friedrichshafen AG group standard, ZFN 201 (1990) Figure 5.1/4 Pitting and micro-pitting on a case-hardened gear; TU Dresden test rig tests (data as in Figure 5.1/1) A relative pitting proportion ( VGes = V1Ges + V2Ges) from pinion and wheel of 2% in total of the active flank surfaces for quenched and tempered (or normalised) gears, and of 0.5% in total of flank surfaces for gears with a hardened surface layer, is approximately applicable as a permitted limit (maximum 4% pitting surface admissible on a single tooth flank). Typical pitting on a case-hardened gear is depicted in Figure 5.1/1. Figure 5.1/2 shows the formation of cracks as a result of pitting. Often large-scale spallings can result from small amounts of pitting in association with hard surface layers. These spallings tend to occur with smaller core hardness and too-small hardening depths. 5.1 Damage Cases 157 b) Micro-pitting (grey staining) Material fatigue is also the cause of micro-pitting – but in thinner layers than occur with pitting. The roughness is of paramount importance. It enables the contact for smaller radii of curvature, which magnify the stresses close to the surface. Cracks appear at shallow depths if the endurance limit in the thin surface layer is exceeded by the local stresses. The endurance limit and the properties of the surface layer are in part significantly influenced by the lubricant as a result of tribochemical changes. A thin oil film and (probably) unfavourable residual stress conditions as well as a decrease in hardness directly on the surface are conducive to micro-pitting. The extremely fine, closely set or immediately contiguous cracks cause the surface to appear matt grey in colour. The matt grey appearance of the micro-pitting is recognisable in Figures 5.1/4 and 5.1/5. A detailed image taken from a scanning electron microscope depicts a scale-like structure (Figure 5.1/6). The risk of micro-pitting is not ranked as high as that of pitting since the cracks are shallower in depth. Experience shows that they can be avoided (or reduced) by low roughness, coatings, higher oil viscosity (or better cooling) as well as by special additives (for calculation refer to ISO/TR 15144). Figure 5.1/5 Micro-pitting on a helical gear; photo from ZF Friedrichshafen AG group standard, ZFN 201 (1990) Figure 5.1/6 Scanning electron microscope image of a region of micro-pitting (approx. 1000-fold magnification); photo from ZF Friedrichshafen AG group standard, ZFN 201 (1990) c) Scuffing Scuffing occurs if localized welding created as a result of tooth contact is separated by sliding. If this occurs with low influence of temperature, it is known as cold scuffing . This may occur at low speeds and with the use of low-viscosity oils. The more common form, however, is warm scuffing , mainly simply called scuffing. Local increases in temperature may occur on contact with high sliding speeds and contact stress. Once the stress limit has been exceeded, this results in a breakdown of the lubrication film, characterised by local vaporisation or simple destruction of the otherwise insulating protective lubricant film. Following the destruction of the lubricating and protective film, single weld bridges are formed as a result of the high local temperatures. These bridges are then immediately separated again due to the relative motion of the paired flanks. This gives rise to strong surface roughening running radially in the sliding direction (Figures 5.1/7 and 5.1/8), in part revealed by annealing colours due to the high local temperatures. The strain is mainly expressed in terms of the maximum temperature occurring in the meshing. The deficiencies in the current theory are revealed in the assumption that bulk temperature and contact flash temperature have the same effect, that is, that the simple sum is calculated, or alternatively, an inadequate evaluation ensues and, in addition, that the scuffing temperature computed is dependent on the speed. 158 5 Load Capacity: Introduction, Initial Values Figure 5.1/7 Scuffing; TU Dresden test rig tests Figure 5.1/8 Scanning electron microscope image of a scuffing zone; photo from ZF Friedrichshafen AG group standard, ZFN 201 (1990) The effect of the lubricant protecting the surface against premature destruction is based on the hydrodynamic (or hydroelastic) effect, the adhesion to the gear material and a tribochemical effect, particularly from oil additives [EP additives (extreme pressure additives)]. As a consequence of the stochastic character of the roughness and waviness, the variation of the scuffing load capacity values is very large, particularly for gearing without running-in. Scuffing is not associated with fatigue. The apparently time-dependent process is due to the time required until the stationary end temperature of the gear is reached. d) Wear a) Schematic representation of the progress of wear (adhesive and partly abrasive wear) b) Wear of an induction-hardened tooth flank (adhesive wear, lack of lubrication); C: pitch cylinder; for progress of wear refer to a) The most common types of wear are • adhesive wear and • abrasive wear. Both of them only represent a genuine limit of load capacity and lifetime in extreme cases. After the greatest wear (decreasing in the direction of the pitch point) has initially occurred on the tooth tip and the tooth root, then, from a specific extent of the wear, it progresses equidistantly, ultimately resulting in strong scouring. This wear progression or wear situation is depicted in Figures 5.1/9a), b). c) Idle mark (brown-red line, line of contact of the mesh position, tribo-wear) Figure 5.1/9 a) , b), c) Tooth flank wear 5.1 Damage Cases 159 Typical types of wear Adhesive wear is typical for very slow operating gears. As a consequence of the weakly formed hydroelastic effect, an intensive strain of the sliding partners occurs with a high intensity of wear. Factors furthering wear include high contact stress and low oil viscosities in association with low rolling speed. The character of wear is identified by fatigue wear with a partly low-cycle nature. Abrasive wear arises as a result of foreign matter penetrating the mesh, for example, impurities or wear particles in the lubricant. Idle marks occur as a result of small, vibration-like relative movements of gear tooth flanks in contact when at a standstill. The cause of the marks is also attributable to tribo-wear or tribo-corrosion on account of the brownish discolouration. When at an advanced stage, pitting can occur on these marks. Damage as a result of root strain caused by exceeding the maximum (or fatigue) tensile strength, the deformation or crack limit and plastic deformations in the tooth cross section (above the tooth root) Overload fractures, cracks and permanent deformation arise from exceeding the quasi-static strength due to the maximum value of the stress occurring. The greatest stress can also occur as a result of (amongst other things) blockages caused by defects in other components or start-up procedures. For extremely sharp notches and brittle surface layers, the damage begins with the crack initiation. For quenched and tempered materials and in cases where notches are not too sharp, also for case-hardened materials, notches in general do not lower the load capacity with regard to the damage discussed here. Although the stress on the compression side of the loaded tooth is larger than the stress on the tension side, the crack starts on the tension side due to the material properties. Gears with a very elastic rim are an exception, where th e radial component of the tooth force may cause a shifting of the endangered cross section. Figure 5.1/10 depicts the typical appearance of the rupture cross section of a case-hardened gear ( brittle fracture ). Figure 5.1/10 a) Overload fracture on a helical gear (approx. 1.5-fold magnification), brittle fracture; a case-hardened gear, photo from ZF Friedrichshafen AG group standard, ZFN 201 (1990) Figure 5.1/10 b) Overload fracture of a case-hardened gear; scanning electron microscope image; surface layer to core zone transition; intercrystalline and trans-crystalline fracture (uneven plane of fracture, beginning of honeycomb fracture to the core area). Centre of image: rod-shaped mini-sulphide inclusion; approx. 800-fold magnification; photo from ZF Friedrichshafen AG group standard, ZFN 201 (1990) For mild as well as quenched and tempered steels, in general ductile fractures occur, characterised by plastic deformation and the formation of bulges at the opposite side of the start of the fracture (Figure 5.1/11). In the event that the overload does not result in a fracture, then a plastic deformation may nevertheless occur, even with case-hardened gears. Figure 5.1/13b) depicts the change in the involute test diagram of a case-hardened gear before a crack appears. 160 5 Load Capacity: Introduction, Initial Values For fatigue fractures at operational loads, the typical lines of rest are characteristic for a part of the rupture surface (Figure 5.1/14). The formation of these lines of rest may not occur if the amplitude of the stress remains constant (Figure 5.1/15). Figure 5.1/11 Ductile fracture (bulge fracture) on a spur gear (approx. two-fold magnification); photo from ZF Friedrichshafen AG group standard, ZFN 201 (1990) Figure 5.1/12 Flank fracture, TU Dresden, machine elements test [6.5/108/] Fractures starting from the region of the pitch circle may also occur (Figure 5.1/12). This is ultimately caused by Hertzian strain, primarily as a result of peak loads. This induces localised plastic deformations with unfavourable residual stress changes in or below the hard surface layer. Microcracks form based on this – and frequently as a result of inclusions. As a result of the bending stress due to the superimposed Hertzian strain, the fracture, known as flank fracture , ensues either transversely towards the tooth centre or downwards in the direction of the tooth root. a) Before loading b) After loading c) Zones of plastic deformation (FEM) Figure 5.1/13 Permanent deformation of a case-hardened gear; a), b) determined from involute test diagrams ( m = 4 mm); 1000-fold magnification; c) FEM calculation result a) b) Figure 5.1/14a) Fatigue fracture on a helical gear; ZF Friedrichshafen AG group standard, ZFN 201 (1990) Figure 5.1/14b) Fatigue fracture on a four-tooth helical pinion 5.1 Damage Cases 161 Figure 5.1/15 Fatigue fracture, case-hardened spur gear ( m = 6 mm, b = 100 mm, 16MnCr5), pulsator test, TU Dresden; no typical lines of rest as load amplitude is constant For disc-like gears, the crack begins on the tension side corresponding to a tangential angle at the root fillet of ș = 30 to 50 . For an elastic design of the gear, the crack runs steeper into the gear body the thinner the rim of the wheel is. In borderline cases, the fracture runs through the wheel rim nearly vertical to the root cylinder. For internal gears with a large number of teeth and a particularly elastic mounting of the wheel rim, the start of the fracture changes to the compression side of the tooth (Figures 5.1/16 to 5.1/18). The reason for this is the effect of the radial component of the tooth force. It subjects the tooth under load to a bending (compression) stress, which substantially affects the amount of resulting stress, superimposing the bending (tension) stress. The resulting stress is reduced on the tension side and increased on the compression side. Offset by a larger circumferential angle, a bending tension stress still occurs on compression side of the tooth as a result of the bending moment in the geared rim, this giving rise to a reverse load. a) cracks on compression side b) broken-off segment (start of crack made visible by the penetration of oil) Figure 5.1/16 Geared rim fracture of a quenched and tempered internal gear with clutch toothing, m = 20; z = –74 Figure 5.1/17a) Fractures on internal gears; path of crack in the wheel rim; Figure 5.1/17b) Fractures on internal gears; path of crack in the toothing; 162 5 Load Capacity: Introduction, Initial Values Figure 5.1/17c) Fractures on internal gears; cracks starting from tooth root surface to “tension and compression side” a) very thick geared rim; start of crack on the tension side; fatigue tooth breakage; b) thin geared rim; start of crack on the compression side; fatigue tooth breakage; c) very thin geared rim; start of crack on the compression side; fatigue rim fracture Figure 5.1/18 Course of cracks (fatigue cracks) for internal gears; Oil holes in the tooth root often represent the starting point of fatigue fractures. For case-hardening steels with case-hardened and extremely high strains, they should be completely avoided. Their impact (double notch!) on the root bending stress can be minimised with the help of a precise analysis (e.g., Finite Element Analysis). The risk of gear body fractures from keyways can often occur in association with tribo-corrosion (Figure 5.1/19). It is also possible that “smooth”, thin wheel rims with external teeth (e.g., planetary gears, sun pinions) can result in gear body fractures (Figure 5.1/20). Figure 5.1/19 Fatigue fractures of a gear rim with a keyway; TU Dresden test rig tests Figure 5.1/20 Fatigue fracture of a gear with thin gear rim; photo from ZF Friedrichshafen AG group standard, ZFN 201 (1990) 5.2 Options for Verifying the Load Capacity 163 Figure 5.1/21 depicts an induction-hardened gear (the hardened layer is recognisable as a dark area on the teeth). The clearly visible hard surface layer ends in front of the tooth root surface, whereby a significant decrease of the root strength occurs compared to hardening of the complete contour. The cracks occurring run into the wheel rim. Pitting may be the cause of fatigue fractures starting then above the tooth root. Grinding cracks can represent a particular risk to fracture and pitting safety (Figure 5.1/22), which under no circum-stances are admissible in association with significant strain. Figure 5.1/21 Fatigue fracture of an induction-hardened gear (small root fillet radius, unfavourable residual stress at the hardened layer transition, material notch and geometric notch) Figure 5.1/22 Grinding cracks; photo from ZF Friedrichshafen AG group standard, ZFN 201 (1990) (most common cause: too high C content of the hardened layer) 5.2 Options for Verifying the Load Capacity The statement that no gear failure takes place during the operational time specified would un- doubtedly represent the ideal case. Due to the variation of the load, the strength values and the manufacturing tolerances, however, it is not possible to absolutely avoid damage or the occurrence of inadmissible operating conditions. To comply with the real world, it would be appropriate to specify the probability P lim that during the operating period required, the strength is greater than the stress – or alternatively, express this by the probability that the actual service life is greater than the service life required. If a given statistical distribution of the strength ı lim and the stress ı is assumed, then the following equation (Figu- re 5.2/1) is obtained for the probability Plim that the strength is greater than the stress: + ı lim lim ılim ı =(ıı ) ( ı) dı dı∞ −∞ −∞ >= PP ff (5.2/1) where fı lim represents the density function of the strength and fı the density function of the stress. The strength can frequently be described by the Weibull distribution or normal logarithmic distribution , and the stress by a normal distribution. In calculation guidelines, it is often the case that only the strength value is allocated a survival probability, while the fact that the power or torque to be transmitted also possesses a statistical distribution (e.g., normal distribution) is not pointed out. The user is thus often misled into also allocating the strength probability to the associated safety calculated – this being, of course, wrong in general cases. 164 5 Load Capacity: Introduction, Initial Values Unfortunately, the statistical distributions of the determining parameters are not sufficiently known to be able to provide a comprehensive introduction of the concept of probability in relation to the verification of load capacity to date. Usually, only very uncertain information is available – particularly for loads – and it may appear paradoxical indeed that this “element of uncertainty” is faced by an “element of safety” in association with the practical calculation. Figure 5.2/1 Example of calculation of the probability of survival in association with a statistical distribution of strength and stress The safety S can be calculated from various parameters, such as service life values, stresses or forces. It should be noted that the safety inc ludes not only the load reserve but also the element of uncertainty of the calculation method and, in part, the assumptions made. If the safety is calculated from service life values , then we obtain vorh L gef = LSL (5.2/2) vorh = calculated gef = required where Lgef is the required service life and Lvorh is the calculated service life. If the service life values are expressed by the load cycle N, then we obtain the safety SN as vorh N gef = NSN (5.2/3) By using the S-N curve equation (Woehler curve equation), then in the range of limited life, the parameter Nvorh can be replaced by q lim vorh lim vorhı = ıNN and we obtain the result q lim lim N gef vorhı = ıNSN (5.2/4) where ılim represents the strength and ıvorh the stress present. The S-N curve exponent for the tooth flank pressure and the tooth root stress is in the range q = 6 to 12. For a small variation in the stress present, such as by a factor of 1.1, the safety with a curve exponent of q = 10 changes significantly by a factor of 1/2.59. 5.3 Load Frequency Function 165 From Equation (5.2/4) and the example given above, it is clear that there is often a large reserve included in the service life safety. For this reason it is usually better to calculate the safety based on the pure relationship of strength and stress: lim ıı ıS= (5.2/5) It would be a mistake in this regard to interpret Sı as an expression of the power reserve. The stress is not only a function of the tooth load acting but also – amongst other things – of the load distribution, which is itself, in turn, a function of the tooth load. If, for the sake of simplicity, we take as a basis the face load distribution in the range of 1 ≤ KHȕ ≤ 2, we obtain the tooth root stress, simplified with KFȕ KHȕ as FF 0 F ȕ F0 1 tı=ı ıK KF= and 3 1 4 2 t3 Hȕ Fȕ ; 1 K K K KFKK K = + + = ≈ ()Flim ı 1t 2 4ı 1SKF K K=++ (5.2/6) In the equation KF KH , K2 characterises the parts affected by elastic deformation (shafts, bearings) and the term with K3/Ft the part caused by gear manufacturing deviations (for details of K H refer to Section 6.4.4). From Equation (5.2/6), it emerges that for K4 0, a doubling of Ft does not imply a halving of S ı. If the power reserve is to be expressed in terms of the safety, it is then expedient to form the quotient from the force corresponding to the endurance limit load and the load present as ttlim FtFFS= (5.2/7) Ftlim is to be determined – taking into account the load distribution, the additional dynamic load and further influences – so that the endurance limit (or strength) is just achieved for the load cycle number required. Here, Ft is the load acting. It is also possible to make use of the moments (torques) instead of the loads. While the most commonly used method for the verification of the load capacity Sı as per Equation (5.2/5) characterises the “damage safety”, SFt expresses the power/load capacity reserve as per Equation (5.2/7). 5.3 Load Frequency Function 5.3.1 Load Characteristic along the Tooth Characteristic of the tooth loading during gear rotation is the fact that it can be described as a pulse train, since each tooth is only subjected to load for a short time during the meshing while remaining free from load for the remaining period of the rotation. The tooth is therefore always subjected to a dynamic load – usually a pulsating load – even at constant external operating load. For idler gears of gear trains, the alternating action on both flanks results in an alternating load on the tooth. The magnitude of the load pulse is determined by the tooth force acting during contact, this essentially being the result of the torsional moment acting on the gear if additional internal dynamic forces are neglected. The development of the torsional moment on the wheel over time is randomly “sampled” by the tooth on each rotation of the wheel for the duration of the mesh. Just as the torsional moment varies, so ensues a variation in the size of the individual impulse (Figure 5.3/1). Different pulse trains ensue for each tooth due to the differing time of the sampling of the course of the moment. 166 5 Load Capacity: Introduction, Initial Values The pulses on the individual teeth are approximately the same if the load on the wheel only scarcely changes during a rotation. In this process, influences on the pulse size due to internal dynamics (particularly variations in the pitch deviations) are neglected. Figure 5.3/1 Load characteristic resulting from the external tooth load course on an individual tooth 5.3.2 Recording the Load Course in the Load Spectrum The load actually present during the operation on the tooth cannot be used as an unordered pulse train for a usual verification of load capacity. It is only possible to consider the real load in the calculation in the form of a load spectrum. This load spectrum is the expression of the change in load experienced by the tooth during its service life. Here, the large number of load pulses of various sizes is summarised into a limited number of load levels; that is, the load characteristic is classified. Both single-parameter and two-parameter classification methods are used for the classi-fication of given random load courses. In this process, either the exceeding of class limits, the reaching of local peak values, the ranges between such extreme values, or closed hystereses in the Rainflow method are counted and summarised in classes. The damage potential of a stress-time function is best recorded using the Rainflow method, whereby all the results of the single-parameter counting methods can be determined from the Rainflow matrix. The procedure for classification is described in detail in ASTM Standard E 1049 [5.3/1] and DIN 45667 [5.3/3]. The scientific foun-dations and application possibilities are comprehensively described by Buxbaum [5.3/2] and Haibach [5.3/4]. A comparison (with examples) is provided by Westermann-Friedrich and Zenner [5.3/5]. Since the load on the tooth does not represent a continually variable development but rather a pulse train, the classification ensues based on the maximum load occurring in each meshing. Figure 5.3/2 Example of a load spectrum as a total frequency graph The numbers of load cycles determined for the individual load levels are then entered in order of size in a frequency graph by decreasing load levels (Figure 5.3/2). In this clear presentation of load spectra in the form of frequency graphs, it is nevertheless accepted that the chronological sequence of load pulses differing in intensity is disregarded. In addition, the differences in the load frequency as a function of the operating speed are not recor-ded in the classification. This frequency depen-dence, however, can be disregarded. Neither is it included in the respective strength limit. 5.3 Load Frequency Function 167 The load spectrum to be used in the calculation of load capacity is the one occurring on the most severely stressed tooth. It may be considered as “representative”. A tooth subjected to significantly greater load than the other teeth can only exist if the development of the load torque varies in con-junction with the rotational frequency of a gear. In such cases, the same tooth will be engaged again and again when the maximum load torque is acting. Typical sources of such periodic rotational speed load variations are diesel motors (on the drive side) and piston compressors, as well as presses (on the output side). For stochastic or periodic load developments not involving the rotational frequency of the gear, on the other hand, approximately uniform load spectra ensue over the operating period. The same, however, is also true for the development of a periodic rotational speed load if the phase relation- ship between the load variation and the rotating angle does not remain constant. For such cases, over longer periods during the operating time, ever different teeth will appear as the tooth is subjected to the greatest load. Such differing phase relationships arise, for example, when shifted transmissions or shift clutches are used, as when the excitation source of the load variation is not rigidly combined with the gear and thus results in differing phase relationships according to the time of the shifting or coupling. For non-periodic additional loads resulting from starting and braking processes as well as load cycle pulses, the same tooth invariably affected by the maximum additional load is scarcely expected to appear, but rather different teeth will alternately experience the maximum load pulse arising from these additional loads. In these cases, the maximum load pulses are approximately uniformly distributed on all of the teeth, unless the triggering of load cycles or braking and starting processes is directly associated with a specific angularity of the gear. 5.3.3 Experimental Determination of Load Spectra The direct determination of the tooth load acting for the purpose of classification is often a difficult process. Measurements in a closed transmission under real operating conditions are either associa-ted with an extremely large measuring technique overhead or come to grief due to the lack of space available to install the measuring technology. It is therefore common practice to use the course of the torsional moment on the gear shafts for the load spectrum. It is then possible to extrapolate to the load spectrum present for the total operating life from the torque distributions for various operating conditions. When creating the load spectrum, it is necessary to take into account the ope-rating conditions measured according to their load cycle proportion of the total number of load cycles. The torque characteristic should actually be sampled and classified for each and every tooth of the gear in the cycle of its meshing frequency. This way, an individual load spectrum is obtained for each tooth. The most hazardous of these load spectra should then be used in the calculation of load capacity. Since, however, it may be assumed that all of the teeth of a gear are subjected to the same load spectrum in the lifetime, it is possible to simplify the classification. Under the above con-ditions, the individual spectra of all of the teeth can be summarised; that is, the course of the torque sampled in the meshing frequency only needs to be classified in a single spectrum. This simpli- fication is only inadmissible if the torque characteristic is coupled to the wheel position. There then usually exists one tooth subjected to the maximum stress, which is always engaged when the torque reaches its maximum value within a revolution. In this case, the maximum torque occurring in the course of one revolution is to be classified. If the torque characteristic is positive over the entire period recorded, then one load spectrum (i.e., one classification grading) is sufficient for the calculation of the tooth root and the tooth flank load capacity. If, on the other hand, the torque characteristic also includes negative values, then the tooth root stress and the contact stress must be classified separately. For the contact stress, a load 168 5 Load Capacity: Introduction, Initial Values spectrum is created from each of the positive and negative values; the calculation of load capacity then ensues with the more hazardous of the two spectra. For the tooth root stress, the negative values are also classified and recorded separately from the positive values. In this case, it is neces- sary to properly adhere to the sequence of the torque reversals for the respective load cycle number or period – at least initially over the entire load cycle period – and create sections of approximately constant load in between the torque reversals as far as possible in the sequence of the loads. Only in this manner is it possible to satisfactorily take into account any damage occurring in reverse operating mode, where in borderline cases a pure reverse load (Figure 5.3/3a) on the one hand and long-period loads with one direction of action (Figure 5.3/3b) on the other hand may be present. Figure 5.3/3 Load spectra with alternating direction of torque; a) change of torque direction after each load cycle (pure alternating load); b) change of torque direction after a large number of load cycles in one load direction (reverse operating mode) A stress spectrum should be created from the load spectrum to allow both the action on the tooth (intermediate gears, alternating stress) and the mean stress to be taken into account. Although in reverse operating mode, in particular, the calculation principles and guidelines are still considerably insufficient, essential data should not, however, already be blurred when determining the load. For idler gears, it should be noted that both flanks are equally loaded and that the tooth root is subjected to an alternating load consisting of positive and negative loads in approximately equal proportions. In contrast to idler gears, the teeth on distribution gears are only subjected to stress on one flank. It is thus sufficient to perform only one classification and multiply the number of load cycles by the number of meshing gears. The torque developments of all of the output gears only have to be mea-sured and classified for non-uniform load splitting (e.g., for the main and auxiliary drives). The load spectrum is thus derived by adding these individual spectra. The load spectrum of the distribution gear is then derived by adding these individual spectra. Calculated determination It is possible to determine the load spectrum not only experimentally but also from the calculated distributions of tooth forces. Provided that the individual operating conditions can be precisely simulated in a vibration calculation, then the classification can be based on the calculated tooth force trace. The individual operating conditions should be taken into account in the load spectrum 5.3 Load Frequency Function 169 according to their shares in time. To classify the force distributions on idler and distribution gears as well as in association with the alternating load on the individual gear concerned, the same approach applies as with the classification of the measured torque distributions. Empirical load spectrum assumption Irrespective of measurements or calculations for the specific application, it is also possible to derive load spectra from the results gained under similar application conditions. This is particularly true for the draft design calculation for gear pairings. If there is no data available from analogous application conditions, then the operating conditions to be expected and the load level and time duration in the resulting load spectrum can only be estimated. For frequently alternating loads, this estimate is nevertheless still more suitable for the purpose of gathering the actual operating conditions than is the product of the nominal load and the application factor. When estimating the load spectrum, it is possible to draw on typical load spectrum forms and use the applicable equival ence factors in the calculation of load capacity (refer to Section 5.3.5). 5.3.4 Considering Load Spectra in the Calculation of Load Capacity When using the nominal load and an application factor to describe the load variation in the calcu-lation of load capacity, the operational loads actually existing are only roughly considered. The load present can be more precisely taken into account in a lifetime calculation. This is based on the load spectrum as the characteristic factor for the load present, the S-N curve (Woehler curve) for the stresses to be withstood and a corresponding damage accumulation hypothesis. The fact that each load cycle causes damage serves as the basis for the lifetime calculation. The amount of damage primarily depends on the stress level and may be zero for low stresses. The calculated lifetime then identifies the capacity of the material to accumulate the individual damage components before a malfunction occurs. In this manner, it is therefore possible to define a damage function s, the value of which is initially zero, which then increases with each load pulse by the damage amount s k and reaches the value 1 on malfunction (in part, the malfunction is also to be expected for a value of s k < 1 ([6.5/132], taken from the habilitation treatise of Eulitz , TU Dresden, 1999 [6.5/133]): () ( ) N k s k s k s ..., , 2 , 1 , 0 , 1k = + = + (5.3/1) where () 0 0=s (5.3/2) () ( )() kkı 1, ss k s k=+ (5.3/3) Each new part-damage is dependent on the pulse amplitude ı(k + 1) and the already existing overall damage s(k). If, on reaching the overall load cycle number Ng, the damage function s (Ng) is less than 1, then the service strength is ensured and the reciprocal value 1/s (Ng) provides information on the safety present. This general approach for the part-damage in Equation (5.3/3) can be used in all damage accu-mulation hypotheses. If in this approach the effect of the already present damage is neglected, that is, the sequence of loads is ignored, then the following approach for the damage function s (N g) results: () () ==m 1 jj j g N s N s (5.3/4) where m is the number of load steps of the load spectrum, N j is the number of load cycles of the jth load step, and s j is the part-damage caused by all load cycles of the jth load step. 170 5 Load Capacity: Introduction, Initial Values If it is also assumed that the part-damage caused by the load cycles in a given load step are all equal and due to the effect of the class mean value ıj of the jth load step, then the following relationship is obtained: () ( )m gj j j j1ı sN f N == (5.3/5) If the load only ensues in the jth load step, then damage will occur when () ( ) jj j jı 1 sN f N == (5.3/6) The part-damage from one load cycle in the jth load step fj(ıj) is thus determined by ()jj j1ıfN= (5.3/7) For the damage function s (Ng) from Equation (5.3/5), we thus obtain the relationship known as the Palmgren-Miner Rule : () Smm Sjj S33 S22 S11 g ... ...NN NN NN NN NNN s + + + + + + = (5.3/8) The load cycle numbers NSj are the points on the S-N curve, which can be determined from fatigue tests with a constant load corresponding to the stress ıj. From Equation (5.3/8), the frequency distri- bution of the stress spectrum is simply placed into the relationship for the S -N curve (Figure 5.3/4). Figure 5.3/4 Stress spectrum and S-N curve (Woehler curve) in comparison In order to evaluate the lifetime, it is necessary to determine the resulting stress ıj of the jth load step for the mean tooth force Fj or the torque Mj acting in each class. By this means, we derive a stress spectrum from the load spectrum for both the contact stress and the tooth root stress. The following relationships apply for the conversion of the tooth load spectrum (in analogy to the torque spectrum) for the jth load step in accordance with ISO 6336 and DIN 3990: contact stress: tj İ HE B , D Hj ȕ Hvj Hȕ jHĮj 11ıF uZZZ Z Z K K Kdb u+=⋅ (5.3/9) tooth root stresstj Fj F Sȕ Fvj Fȕ jFĮj :ı nFYYYK K Kbm= (5.3/10) 5.3 Load Frequency Function 171 Here the application factor KA is not included in the calculation, as that effect is already included in the load spectrum. The calculation of the rotational speed-dependent dynamic factor KHv = K v or KFv = Kv ensues for the mean speed. Load level rotational speeds should only be used if it is possible to unambiguously assign load steps to specific rotational speeds. The stresses ıHj and ı Fj resulting for the load spectrum in the individual load steps can be combined into an equivalent stress ıH or ıF, which – as one constant load level – causes a damage equivalent to the damage caused by the load spectrum. This stress can then be set in the load capacity calcu-lation in relation to the existing strength to determine the safety factor. For the calculation of the equivalent stresses, the gradient for limited life strength and the load cycle number N lim at the fatigue reference point of the S -N curve with ılim must be known. The S-N curve exponent q, which identifies the gradient in the region of limited life strength, and the load cycle number limit Nlim differ for the contact stress and tooth root stress criteria. The following equations then apply for the calculation of the equivalent stresses1): 1)contact stress HHk Ej qqHH j j=1 eqNı= ıN (5.3/11) FFEj 1) FF j j1 eqtooth root stress : ıık qqN N = = (5.3/12) where m eq j j1and NN k m === for =<m 1 jlim jN N (5.3/13a) or lim eqN N= for =≥m 1 jlim jN N (5.3/13b) and j EjN N= for =<j 1 ilim iN N (5.3/14a) or j1 Ej lim i i1and NN N k j− ==− = for =≥j 1 ilim iN N (5.3/14b) Flank: N lim = NHlim; Root: N lim = NFlim; qH, qF, Nlim from Table 7.4/27; m = total number of load steps. The load spectrum is therefore cut off at Nlim when the total load cycle number Ng exceeds this limit load cycle number, as also depicted in Figure 5.3/5. Figure 5.3/5 Formation of the equivalent load spec- trum by cutting off at the limit load cycle number Nlim 1) ISO 6336-6 and DIN 3990-6 apply the pure Palmgren-Miner Rule, whereby only negligibly deviating results are achieved that do not comply better with practical experience. 172 5 Load Capacity: Introduction, Initial Values This approach is based on the consideration that all of the load cycles after the limit load cycle number cause no damage as long as the corresponding stress resulting from this load cycle is below the endurance limit. This occurs when the safety factor in the load capacity is assumed as greater than one, at which the equivalent stresses and thus the stresses in the neglected load steps are below the endurance limit. A series of factors recorded in Equations (5.3/9) and (5.3/10) is not dependent on the load acting in each case; only the stress factors K HĮ, ..., Kv are dependent on the load. On the basis of this load dependency present, the existing safety factor for a given load spectrum does not directly corres- pond to the reciprocal value of the Miner sum s(Ng) or the safety calculated with the equivalent stress. The precisely existing safety factor in relation to the load acting is represented by the value S, for which s(Ng) = 1, when the tangential force acting in each load level Ftj is multiplied by S in Equations (5.3/9) and (5.3/10). For the practical calculation, this means that the existing safety can only be calculated on an iterative basis. New load levels are determined by multiplying the force Ftj by the safety factor S\* = 1/ s(Ng). With these new load levels, K...j factors and ultimately the stresses ıHj and ıFj of every level are determined, as a basis for then calculating the Miner sum s(Ng). If s(Ng) > 1.05, then the calculation has to be repeated with a lower value for S \*, while if s(Ng) < 0.95, it is necessary to repeat the calculation with a greater value for S\*. If the Miner sum falls within the range 0.95 < s(Ng) < 1.05, then the calculation can be stopped and the current value of S\* corresponds to the existing safety Svorh. The safety factors thus determined apply solely to the form and the total shape of the load spectrum used for the calculation. The comparison of the stress spectrum with other damage accumulation hypotheses takes place not with the S-N curve itself but with modified S -N curves that either do not have a genuine endurance limit value ( Corten-Dolan ), or partly include consideration of the effects of the load cycles passed on the component strength (extended damage accumulation hypothesis) or the change in the slope of the S-N curve due to the previous stress cycles (consequential Woehler curve concept). These damage accumulation hypotheses are only conditi onally applicable to the problem of the gear load capacity, since either the corresponding strength values from multiple-step tests are scarcely available or the effort involved is not worth making. The lifetime calculation based on Equations (5.3/8) to (5.3/10) is only suitable for the recalculation. For the dimensioning of the gearing, a greater iterative calculation effort arises that is due to the load dependency of the K ...j factors. If the differences between the K...j factors of the individual load levels are neglected and these factors are calculated for all of the steps with the same force Ft, then the spectrum of the individual forces Ftj can be substituted by an equivalent tangential force F tH to determine the contact stress and FtF for the calculation of the root stress 1)2): H Hk Ej 0.52tH tj j=1 eq1):= contact stressqqNFFN (5.3/15) FFk Ej 1) tF tj j1 eqtooth root stress :qqNFFN = = (5.3/16) where NEj and Neq are taken from Equations (5.3/13) and (5.3/14); qH, qF from Table 7.4/27. 1) ISO 6336-6 and DIN 3990-6 apply the pure Palmgren-Miner rule, wher eby only negligibly deviating results are achieved that do not comply better with practical experience. 2) Differing equivalent tooth-circumference forces result for the pinion and wheel due to the different load cycle numbers. In general, it is sufficient to determine the equivalent circumference tooth forces for the gear with the greater load cycle number. The load cycle numbers actually present should only be used separately for pinion and wheel if the safety for the wheel with the lower load cycle number is not sufficient. 5.3 Load Frequency Function 173 In addition to the cutting-off of the load spectrum practiced in Equations (5.3/13) and (5.3/14) when the load cycle number limit Nlim is exceeded, DIN 3990-6 (ISO 6336-6) still limits the number of load levels to be taken into account by the requirement that the respective load level exceeds 50% of the specified nominal tooth force. Lower load levels are not considered. However, the resulting equivalent forces F tH and FtF are thus influenced by the arbitrary choice of a nominal torque. In light of this rough definition, DIN 3990-6 (ISO 6336-6) proposes an enhanced selection criterion. The arbitrary 50% limiting value of F tnenn is replaced by the 70% limiting value of the fatigue-resistant tooth force F tH∞ or F tF∞ arising for the operating conditions as well as the geometrical and material parameters. This additional selection criterion is intended to prevent the equivalent tooth forces being increased by load components that do not intrinsically lead to damage. This is because for such load components, the safety factors calculated are below the values actually present for the load spectrum. The numerical calculations show that these restrictions are unnecessary, since HF0.5qq tH tF (0.5 . (0.5 )) respFF ∞∞ only result in a hardly perceptible influence. If it is intended to determine the sustainable load by specifying the form of the load spectrum in the design, then it is sensible to define load spectrum factors instead of equivalent forces. These factors are also determined with Equations (5.3/15) or (5.3/16) if the load level forces F \* tj = Ftj /Ftnenn related to the nominal force are used instead of the load level force Ftj. Instead of the equivalent forces FtH or FtF, this results in the relative equivalent forces F\*tH = FtH /Ftnenn or F\*tF = FtF /Ftnenn. Using these factors, it is then possible – under the operating conditions as well as the gearing and material parameters selected – to convert the fatigue-resistant sustainable circumferential force F tH∞ or FtF∞ into the fatigue-resistant circumferential forces Ftnenn sustainable for the respective load capacity criterion, where Ftnenn = F t ∞ /F\*t. The sustainable load level forces Ftj are then determined by Ftnenn via F\*tj. The essential difference between considering the load spectrum by means of the equivalent stresses and the consideration using equivalent tooth forces is reflected in the fact that the load-dependent stress factors K Hv, KFv, KHĮ, KFĮ, KHȕ and KFȕ are determined with different loads. When using the equivalent stresses, they are precisely determined for the load levels specified, while with the method using the equivalent tooth forces F tH and FtF, they are only approximately determined with the equivalent loads. 5.3.5 Typical Load Spectra and Characteristics Load spectra differ also in their form for the various operating conditions encountered in practice. For the most part, however, these differences are small, and it is possible to characterise several basic types to which the specific load spectra can be assigned. In principle, it is possible to distinguish between “heavy” and “light” load spectra. Both of these basic types are depicted in Figure 5.3/6. Heavy load spectra are distinguished by a certain number of high-amplitude load cycles mainly coming from high loads in start-up processes (e.g., in cement mill drives). With light load spectra, in contrast, such high-amplitude load cycles scarcely occur. To determine the equivalent load F tH and FtF, it is possible to make use of the load spectrum forms depicted in Figure 5.3/7. A load spectrum form should be chosen that best corresponds to the load spectrum specified or the load assumption made. The equivalence factors H and F associated with the selected load spectrum form and the S-N curve exponents corresponding to the material used can then be taken from Table 5.3/1. The approximate equivalent tangential forces can be calculated with the equivalence factors: tH t1 Hȝ FF= (5.3/17) tF t1 Fȝ FF= (5.3/18) where Ft1 is the greatest force in the highest load level of the spectrum. 174 5 Load Capacity: Introduction, Initial Values Figure 5.3/6 Basic types of load spectra; a) for high, short-duration and otherwise balanced loads; b) for loads whose class frequencies are subject to a more or less normal distribution Figure 5.3/7 Load spectrum forms for the determination of equivalent forces as per Equations (5.3/15) and (5.3/16) using the equivalence factors ȝH or ȝF from Table 5.3/1 5.3 Load Frequency Function 175 Table 5.3/1 Equivalence Factors for the Typical Load Spectra in Figure 5.3/7 Load spectrum number in Figure 5.3/7 Equivalence factors H or F for the Woehler curve exponent qH /2, qF qH /2; qF 3 qH /2; qF 6 qH /2; qF 9 qH /2; qF 12 1 0.503 0.596 0.664 0.709 2 0.342 0.437 0.500 0.562 3 0.191 0.290 0.388 0.445 4 0.749 0.776 0.799 0.817 5 0.640 0.664 0.685 0.701 6 0.453 0.485 0.501 0.539 7 0.257 0.311 0.359 0.406 8 0.778 0.782 0.784 0.786 9 0.534 0.548 0.555 0.562 10 0.327 0.355 0.359 0.406 11 0.182 0.192 0.243 0.335 12 0.045 0.052 0.108 0.188 13 0.794 0.818 0.836 0.841 14 0.630 0.723 0.774 0.800 15 0.565 0.634 0.691 0.715 16 0.500 0.580 0.632 0.661 17 0.398 0.485 0.541 0.562 176 5 Load Capacity: Introduction, Initial Values 5.4 Symbols and Symbol Explanations b mm face width d1 mm reference diameter of gear 1 Ftj N nominal tangential force in jth load stage FtF∞ N 70% of enduring capable tangential force, bending FtH∞ N 70% of enduring capable tangential force, pitting fj(ıj) - partial damage of a load cycle in jth load stage KA - application factor KFvj - dynamic factor for root stress in jth load stage KFĮj - transverse load factor for root stress in jth load stage KFȕj - face load factor for root stress in jth load stage KHvj - dynamic factor for contact stress in jth load stage KHĮj - transverse load factor for contact stress in jth load stage KHȕj - face load factor for contact stress in jth load stage mn mm normal module NEj - equivalent number of load cycles in jth load stage Neq - reference number of load cycles Ng - total number of load cycles Nj - number of load cycles in jth load stage Nlim - number of load cycles at fatigue point of S-N curve NSj - number of damage load cycles in jth load stage qF - exponent of S-N curve of tooth breakage qH - exponent of S-N curve of pitting sk - damaging value of the kth load impulse s(k) - damage sum after k load cycles (damaging function) sj(nj) - partial damage of jth load stage u - gear ratio z2/z1 YF - form factor YS - stress correction factor Yȕ - helix angle factor for root stress Yİ - contact ratio factor for root stress ZB - single pair tooth contact factor for gear 1 ZD - single pair tooth contact factor for gear 2 ZE 2N/mm elasticity factor ZH - zone factor Zȕ - helix angle factor for pitting Zİ - contact ratio factor for pitting F - equivalence factor for load spectrum for root stress ȝH - equivalence factor for load spectrum for pitting ıFj N/mm2 root stress in jth load stage ıHj N/mm2 contact stress in jth load cycle ılim N/mm2 endurance limit Indices H flank; Hertzian pressure F tooth root vorh = calculated gef = required 177 6 Load Capacity and Running Performance of External and Internal Gearing 6.1 Velocities 6.1.1 Sliding Velocity The tooth flanks roll and slide on one another. The vel ocities that occur here are deduced as follows. In the contact point Py of a tooth flank pairing, the resulting velocities of the flank 1 or 2, according to the angular velocities ω1 or ω2 and the distances ry1 or ry2 to the gear centres, amount to (Figure 6.1/1a, b) 1y 1 y1Ȧr υ= (6.1/1) y2 y2 2Ȧr =−υ (6.1/2) These velocities can be broken down into two components positioned perpendicular to one another, in the direction of the normals ( n) and in the direction of the tangent ( T) to the flanks. In the direction of the normals, /y1 and /y2 must have a component /n of the same size, because otherwise a flank separation would occur: ny1 ny2 n==υυυ (6.1/3) ny 1 y 1 y 2 y 2 cos cos =α =αυυ υ (6.1/3a) or n1 b 1 2 b 2ȦȦ== −rr υ (6.1/3b) The tangential components (rolling velocities ) /Ty1, /Ty2 are Ty1 n y1 Ty2 n y2tanĮ tanĮ= =υυ υυ (6.1/4) They are important for the hydrodynamic effect and scuffing load. With /n = ω1rb1 = –ω2rb2 and tan αy = Ny/rb the following is true: Ty1 1 y1 Ty2 2 y2Ȧ Ȧ= =−υ υN N (6.1/5) 1) 1) The following is true for the gear (1) - rack (2) pairing: υTy2 = υn tanαt = ω1d1/2 sinα t = const. 178 6 Load Capacity and Running performance of External and Internal Gearing The smallest radius of curvature of gear 1 is N1min = N1A =A T1 (Figure 6.1/1a, b) () ()22 2 1min wt a2 b2 2sinĮ /2 /2zad dz =− − N (6.1/5a) and the largest N1max = N1E = E T1 (Figure 6.1/1a, b) is () ()22 1max a1 b1 /2 /2dd=−N (6.1/5b) For an arbitrary point P y within the length of path of contact, the following applies: 1min y1 1max≤≤NN N (6.1/5c) y2 wt y1sinĮ a=⋅ −NN (6.1/5d) Figure 6.1/1 Velocity components of the teeth at the contact point Py: a) external gear pair, b) internal gear pair Particularly for the meshing points A, B, C, D, E (transverse section), according to Figures 6.1/1a, b and 2.1/19a–d, the following applies: A 11 B 11 C 11 D 11 E 11 2 ;; ; ; ;TA TB TC TD TE== ===NNNNNN (6.1/5e) () ()22 2 A1 wt a2 b2 A2 wt A1 2sinĮ /2 /2 ; s i n Įzad daz=− − =−NN N (6.1/5f) 6.1 Velocities 179 () ()22 B1 a1 b1 et B2 wt B1 /2 /2 ; s i ndd p a=− − = − α NN N (6.1/5g) wt C1 C2 wt C1 21sinĮ;s i n Į1/aazz== −+NN N (6.1/5h) D1 A1 et D1 wt D1 ; s i n Į pa =+ = −NN N N (6.1/5i) () ()22 E1 a1 b1 E2 wt E1 /2 /2 ; s i n Į dd a=− = −NN N (6.1/5j) The sliding velocity /gy constitutes the difference of the velocities /Ty1,2: gy Ty1 Ty2 =−/// (6.1/6) /Ty1, /Ty2 according to Equation (6.1/4) With Equation (6.1/5), taking Equation (6.1/3b) into account, whereby the radii of curvature of the tooth surface profiles are in Figure 6.1/1 y1,2 1,2 yTP=N , from Equation (6.1/6) one arrives at or respectively gy 1 y1 2 y2 b1 gy 1 y1 y2 b2Ȧ Ȧ Ȧ r r=+ =− / /NN NN (6.1/7) With the distance C P qy y= of the meshing point Py from the pitch point C (refer to Figure 6.1/1), that is, y1 b1 wt y tanĮ rq=+N and y2 b2 wt y tanĮ rq=−N , ultimately the sliding velocity results from Equation (6.1/7) or respectively () ()gy 1 y gy y 1 2Ȧ 11 / ȦȦ=− =−qi q/ / (6.1/8) Note: i = – z2 / z1 = ω1 / ω2 External gear pair i < 0 (reversal of direction of rotation!); () gy 1 yȦ 11 / =+qi/ Internal gear pair i > 0 (N y2 < 0; z2 < 0; same direction of rotation!); () gy 1 yȦ 11 / =−qi/ From Equation (6.1/8) this follows (refer to Figure 6.1/2): The sliding velocity (absolute amount) grows linearly with the distance of the meshing point from the pitch point. At pitch point C itself it is equal to zero. Only there does pure rolling take place, whereas in all of the remaining area there is rolling and sliding. The sliding velocity is important for power loss (efficiency, heating) and scuffing strain. Figure 6.1/2 shows the trace of sliding velocity for external and internal gearing. For a constant transverse contact ratio, the result is a minimum in sliding velocity if the pitch point C is situated in the centre of the length of path of contact. From the beginning of contact (Point A, tooth root region of the driving gear/tooth tip of the driven gear) right to the pitch point, due to the larger velocity component of the driven gear, which is perpendicular to the line of action, its tooth tip flank pushes away the oil film in front of it. The motion conditions in this region AC are called “pushing” or “driving” sliding . In contrast, in the region CE of the length of path of contact, the flank of the driving gear pulls the oil film into the narrowing lubrication gap; this is called “ pulling” sliding. In the region of the pushing sliding, especially in the region of the meshing point A, there is increased frictional resistance, aggravated by pre-meshing due to the deformation of the teeth. 180 6 Load Capacity and Running performance of External and Internal Gearing a) External gear pair b) Internal gear pair Figure 6.1/2 a), b) Sliding velocity /g, specific sliding ζ1,2 of the tooth flanks lengthwise to length of path of contact gα E A = 6.1.2 Specific Sliding For the wear, the local heating, the micro-pitting and the pitting (the last of these in cases of smaller gears), for a certain point of the tooth flank it makes a difference which path of the counter-profile slides over it in each contact, which is characterised by specific sliding. For this we will consider Figure 6.1/3. The path (2 b H) is the width of contact caused by Hertzian flattening. The time Δt, which a point of the flank 1 needs to complete the path (2b H), is H Ty12btΔ=/ (6.1/9) During this time, the path Sg1 slides completely over this point of the flank 1: 6.1 Velocities 181 ()H g1 Ty1 Ty2 Ty12bS=−/// (6.1/10) Similarly, for flank 2 the following results: ()H g2 Ty2 Ty1 Ty22bS=−/// (6.1/11) The paths Sg1,2 are also known as sliding paths . When one forms the relationships Sg1,2/(2bH), one arrives at quantities which are known as specific sliding ζ1,2. For an arbitrary point y the following is true: Ty1 Ty2 y1 Ty1ȗ−=// / (6.1/12) Ty2 Ty1 y1 Ty2ȗ−=// / (6.1/13) with /Ty1, /Ty2 according to Equation (6.1/4). If one uses the specification factors for /T1,2 in accordance with Equation (6.1/5) in the Equations (6.1/12) and (6.1/13), one arrives at y2 1 y1 y1 2ȗ 1z z=− ⋅N N (6.1/14) y1 2 y2 y2 1ȗ 1z z=− ⋅N N (6.1/15) As examples, in Figures 6.1/2a and b, the specific sliding ζy1,2 for external and internal gear pairings are added over the length of path of contact. Figure 6.1/3 Tangential velocity components of the gearing in the Hertzian flattening region (velocities perpendicular to the line of action) 6.1.3 Sliding Factor The sliding factor K g is defined as the ratio of sliding velocity /g to the circumferential velocity /wt of the working pitch circles: 182 6 Load Capacity and Running performance of External and Internal Gearing g g wtK=/ / (6.1/16) The smaller this factor, the more favourable is the design in terms of heating and scuffing. When the special specification factors (refer among others to Figure 2.1/19) are applied, the result for the end of contact (Point E) with CE g a= is a1 ga w1 221gzKdz =+ (6.1/17) and the beginning of contact (Point A) AC g f= f1 gf w1 221gzKdz =+ (6.1/18) Furthermore, if one uses (Figure 2.1/19) 1e t a e t //CE p g p==J or 2e t f e t //ACp g p==J , one arrives at (6.1/19) (6.1/20) The quantity (1/ z1 + 1/z2) is proportional to the degree of loss and references Kg as a parameter for loss and heating. In the case of internal gearing, the degree of loss is small (favourable) because z2 < 0. 6.1.4 Sum of Velocities The sum of velocities /Ȉy is the sum of the velocities /Ty1 and /Ty2, that is, the sum of the com- ponents positioned perpendicular to the line of action (see Figure 6.1/1a, b): yT y 1 T y 2 Σ=−/// With Ty1 1 y1Ȧ=/N Ty2 2 y2Ȧ=−N / and 1 21 2ȦȦz z=− the result is 1 Ȉy 1 y1 y2 2Ȧz z =+ /NN (6.1/21) with Ny1, Ny2 according to Equation (6.1/5c–j); refer to Figure 6.1/1.ga 1 wt 12 gf 2 wt 12112ʌ cosĮ 112ʌ cosĮ Kzz Kzz =+ =+ J J 6.2 Tooth Stiffness 183 For the pitch point C in particular 1w t ȈC 1 22Ȧ sinĮ 1a z z= +/ (6.1/21a) /Ȉy has significance for the hydrodynamic effect. 6.2 Tooth Stiffness 6.2.1 Basics The gearing stiffness has an influence on various quantities that play a role in the gearing’s load- bearing capacity. It has an effect on the • load distribution over the face width ( KHβ, KFβ), • load distribution over the gear circumference ( KHα, KFα), • stimulation of additional internal, dynamic loads ( Kv), • resonant frequency of the gear pairing ( ω0). For tooth stiffness, the relationship between the load in effect and the elastic tooth deformation which it causes is looked at. In determining tooth stiffness as a parameter, various approaches are used, as appropriate for the problem to be examined. To determine the resonant frequency of the gear pairing, only the mean value of gearing stiffness, which fluctuates during meshing, is important. It results from the superimposition of the single stiffnesses of the tooth pair coming into contact. Due to the changing meshing conditions, the resulting gearing stiffness fluctuates. These fluctuations play a role in the stimulation of internal, dynamic tooth loads. In order to ascertain the load distribution on the meshing tooth pairs, it is not sufficient anymore to only consider the stiffness of the overall gearing. The single stiffnesses of the tooth pairs in the respective meshing position must be considered. Finally, it is necessary to also analyse these single stiffnesses over the face width, if one wants to determin e the load distribution over the face width. 6.2.2 Parts of Tooth Stiffness In order to ascertain the existing stiffness, the tooth deformation that exists in the direction of the effective load must be determined. Tooth deformation can be broken down into a series of individual parts, which are shown in Figure 6.2/1. The main part of this deformation usually comes from the bending deformation of the tooth. This bending part originates in the elasticity of the tooth itself (Figure 6.2/1a), which can be compared to a fixed bar that bends under a load. A second bending deformation part is added to that from the fixing elasticity that exists (Figure 6.2/1b), when the bar additionally tilts. Furthermore, because of lateral load (Figure 6.2/1c), shear deformation takes place in the tooth cross sections. Now if a meshing tooth pair is considered, a deformation part comes about because of the Hertzian flattening of the tooth flanks under load (Figure 6.2/1d). In contrast with the parts which have already been named, this deformation part is non-linear; the amount of flattening does not increase proportionally with the effective load. As consequence of the tooth load component acting in the radial direction, there is likewise a com-pression of the tooth, which produces a minor radial deformation part in the direction of meshing (Figure 6.2/1e). Another deformation part is cau sed by the contact elasticity (Figure 6.2/1f). Because of the surface roughness that exists on the tooth flanks, minor deformations take place on the surface when tooth contact occurs, which diminish due to increasing wear from running in. In 184 6 Load Capacity and Running performance of External and Internal Gearing the case of helical gearing, the longitudinal tooth curvature (Figure 6.2/1g) that exists also has an influence on tooth deformation. As an approximation, the elasticity parts already talked about can be viewed as being constant across the face width. If deformation is more precisely ascertained across the face width, compared to the tooth centre these parts show minor differences on the tooth edge. In the case of helical gearing, the helical shape of the tooth edges affects the deformation on the end faces (Figure 6.2/1h). Larger differences in tooth deformation across the face width occur in the case of gears of a webbed design due to the deformation parts which are caused by rim elasticity (Figure 6.2/1i). Figure 6.2/1 Parts of the resulting tooth deformation: a) bending deformation of the tooth b) bending deformation due to fixing elasticity c) shear deformation d) flattening due to Hertzian contact stress e) compression in radial direction f) deformation caused by contact elasticity due to surface roughness g) influence of tooth curvature on the deformation across the width h) influence of the ends of the teeth on the deformation across the width i) influence of rim elasticity on the deformation across the width The main deformation parts are • bending deformation of the tooth, • bending deformation due to fixing elasticity, • shear deformation of the tooth, and • deformation due to Hertzian flattening of the tooth flanks. 6.2.3 Calculation of Tooth Stiffness If a plain deformation state is assumed as an approximation, for spur gearing these deformation parts can be determined with the calculation equations according to Weber/Banascheck [6.3/6]. In the calculation of bending and shear deformation, these assume the elementary Euler–Bernoulli beam theory and supplement the resulting tooth deformation with the parts owing to Hertzian flattening (refer to Section 6.4.2.3). For helical gearing, such a calculation is only feasible if one breaks the tooth into multiple small slices, as shown in Figure 6.2/2. Due to the diagonal line of contact running over the tooth, the deformations which occur for each of these tooth slices are different because the application of the load is at a different height on the flank in each case. The summation of the tooth stiffnesses of the individual slices then results in the effective tooth-pair stiffness in the respective meshing position. Still not taken into consideration in this method are the interactions (support effects) between the tooth slices. 6.2 Tooth Stiffness 185 In order to precisely determine the tooth stiffness across the face width, it is possible to utilise the method for deformation influence numbers described in Section 6.4.2.3, for example. From the load distribution calculated using this method, with the help of the influence numbers one arrives at the sought-after tooth deformation across the face width. Load distribution and deformation ultimately lead to the sought-after tooth stiffness. These days, the finite element method is increasingly being used for the deformation calculations. If one compares the trace of tooth pair stiffness from the beginning to the end of contact, compared to spur teeth pairs, helical teeth pairs demonstrate a distinctly more drastic drop in stiffness from the beginning to the end of contact. The reason for this is the very short length of the line of contact for the beginning and end of contact in the case of helical gearing. The stiffness progress for spur and helical teeth pairs is shown in Figure 6.2/3. Figure 6.2/2 Slice model for calculating gearing stiffness in the case of helical gearing [6.3/7] Figure 6.2/3 Tooth pair stiffnesses across the length of path of contact (ISO 53; DIN 867; m = 4 mm; a = 100 mm; b = 50 mm; z1 = 17; x1 = 0.5): a) spur gearing ( z2 = 32; x2 = 0.036), b) helical gearing with β = 10 ( z2 = 31; x2 = 0.1847), c) helical gearing with β = 16 ( z2 = 30; x2 = 0.093) 6.2.4 Approximate Calculation of Tooth Stiffness according to ISO 6336-1 (DIN 3990-1) In the calculation of load-bearing capacity in ISO 6336-1 (DIN 3990-1), tooth stiffness is calculated approximately using a polynomial approach according to Sch fer [6.3/16] (based on the equations of Weber/Banascheck [6.3/6]). The theoretical stiffness value cth which is determined is adapted to the actual conditions through a series of correction factors. Taking the contact ratio into account, the calculated single stiffness cƍ ultimately produces the resulting mean stiffness of the gearing: the meshing stiffness. For the relationship between the meshing stiffness c γ and the single stiffness c ƍ, with β < 45 , for spur and helical gears the following empirical equation is used: ()/ ȖĮ 0.75 0.25 cc=+J (6.2/1) which includes the influence of contact ratio. The single stiffness cƍ is the maximum value for the tooth stiffness of a tooth pair in the transverse section. In ISO 6336-1 (DIN 3990-1), the following equation is given for the calculation: // th M R B cosȕ cc C C C= (6.2/2) 186 6 Load Capacity and Running performance of External and Internal Gearing The factors contained in Equation (6.2/2) are calculated as follows: a) Theoretical tooth stiffness cƍth For the calculation, a series expansion is given based on the analyses of [6.3/6]: 1 1 th n1 n2 n1 22 2 21 2 n21 0.15551 0.257910.04723 0.00635 0.11654 0.00193 0.24188 0.00529 0.00182xxcz z z xx xxz=+ + −−′ −− + + (6.2/3) with the number of teeth of the equivalent gear 12 n1 n2 33z;cosȕ cosȕzzz≈≈ . (6.2/4) Equation (6.2/3) applies for: • Standardised basic rack (ISO 53, DIN 867) with Į = 20 , hf = 1.2 mn, ha = mn • Profile shift coefficients x1 ≥ x2 and -0.5 (x1 + x2) ≤ 2.0 • External gearing and as an approximation for internal gearing with stiff gear rim if zn2 = is set • A line load of Ft/b = 300 N/mm (in the range of 100 N/mm Ft/b ≤ 1600 N/mm, the deviations lie between +5% and -8%) • Steel/steel pairings Here, /c,Ȗcand / thcrefer to the direction of the normals of the tooth load in the transverse section and express the tooth load required to deform the gearing by 1 m, at a face width of b = 1mm. If other materials are used with the elasticity of material E\*, the following applies: // th th\_Stahl/Stahlȟ=cc (6.2/5) with Stahl ȟ /∗=EE 12 122∗=⋅+EEEEE (6.2/6) ξ = 0.74 for steel/cast iron, ξ = 0.55 for cast iron/cast iron b) Correction factor C M In view of the tooth stiffness values that are frequently lower in measurements than those calculated according to Weber/Banascheck [6.3/6], in Equation (6.2/3) a reduction of the underlying stiffness value based on [6.3/6] is undertaken with the factor CM = 0.8. The reason for this reduction is above all the contact elasticities of the tooth flanks, which are not included in the calculation. c) Shape factor CR The shape factor C R accounts for the flexibility of gear rims and webs. For solid disc gears the following is true: C R = 1. For gears with a webbed design the following equation applies: R ns R 5ln 1 5s mb bC e =+ (6.2/7) with bs the web width, b the face width, sR the gear rim thickness, and the limiting conditions as follows: if bs/b < 0.2 ĺ bs/b = 0.2 if bs/b > 1.2 ĺ bs/b = 1.2 if sR/mn < 1 ĺ sR/mn = 1 6.3 Load on the Tooth 187 d) Basic rack profile factor CB The factor CB takes into account the deviations of the basic rack vis- -vis the standardised basic rack according to ISO 53 and DIN 867: ()f B n1 0.5 1.2 1 0.02 20 ĮhCm =+ − ⋅− − (6.2/8) For the standard basic rack, CB = 1 is true. In the case of unequal dedendums in the paired gears, the mean value CB = 0.5 ( CB1 + CB2) must be used. For pairings of solid disc gears made of steel with a basic rack according to ISO 53 and DIN 867, one can use the following as guide values in rough calculations: Single stiffness (one tooth pair, single tooth contact, transverse section) cƍ = 14 N/(mm m) Mesh stiffness (mean stiffness across one pitch, transverse section) c γ = 20 N/(mm m) Here a transverse contact ratio of 1.2 < Jα < 1.9 and a line load of KAFt/b ≥ 100 N/mm are assumed. 6.2.5 Symbols and Symbol Explanations of Section 6.2 b mm tooth face width bs mm web thickness CB - basic rack factor CM - correction factor CR - gear blank factor cƍ N/(mm m) tooth pair stiffness per unit face width (single stiffness) cƍth N/(mm m) theoretical single stiffness cγ N/(mm m) mean value of mesh stiffness per unit face width E N/mm2 Young’s modulus Ft N nominal transverse tangential load at reference cylinder ha mm addendum of basic rack of cylindrical gears hf mm dedendum of basic rack of cylindrical gears KA - application factor KFα - transverse load factor (root stress) KFβ - face load factor (root stress) KHα - transverse load factor (contact stress) KHβ - face load factor (contact stress) Kv - dynamic factor mn mm normal module sR mm rim thickness x1, x2 - profile shift coefficient of gear 1 and gear 2 z1, z2 - number of teeth of gear 1 and gear 2 zn - virtual number of teeth of a helical gear Į pressure angle ȕ helix angle Jα - transverse contact ratio ξ - elasticity ratio ω0 rad/s angular velocity of the gear pair 6.3 Load on the Tooth 6.3.1 General Information In a precise approach, the load-time function of all teeth of the gear pairing is to be determined, and with that the probability of survival ascertained. Strictly speaking, it depends among other things on the number of teeth (the probability of breakage rises with an increasing number of teeth) and the overall development of stress during contact. But since neither the strength’s dependency on frequency nor the static distribution of strength exists for a specific load function, and because usually only a rough value is known for the size of the load in the first place, one has no choice but to make do with approximations, which are described as follows. Here one can differentiate between the following typical cases: 188 6 Load Capacity and Running performance of External and Internal Gearing I Load spectrum given Ia) Load-time trace obtained from measurements or simulations of operation Usually the additional internal dynamic loads are not yet included and must still be added on. These load spectrums can be used for random trials or more precise analyses. Ib) Load-frequency trace obtained from measurements and ordered by decreasing size Since there is usually a very frequent alternation between heavy and light load during operation times, an influence based on order usually does not have any practical effect, which means that it is permissible to order the load spectrum by decreasing size. The internal dynamic tooth loads are generally still to be added. Ic) The effective power to be transmitted or the effective torque to be transmitted is known from project planning specifications in terms of size and duration. Both the additional internal as well as the external dynamic loads are to be added to this spectrum. Factors ( KA, Kv) or results of special analyses can be used for this. Section 5.3 contains more detailed information about cases Ia) to Ic). II Nominal power and nominal number of revolutions are given In this case, the additional external and internal dynamic loads are accounted for by factors. These factors register the additional load vis- -vis a reference load, for example, the nominal load with regard to maximum strain ( KAS) and fatigue stress ( KAB), for the load cycle number corresponding to the operating time of the system. The following section will go into the determination of tooth loads in cases where the external load is given and into the ascertainment of additional forces. 6.3.2 Loads on the Tooth from Effective Power 6.3.2.1 Resolution of the Loads on the Tooth In the calculation of load-bearing capacity for gearing, the tangential load in effect on the reference circle is used. As one of the three load components that are produced in the resolution of normal tooth load (Figure 6.3/1), it marks the load that exists on the tooth. On the driving gear it works against the direction of rotation, and on the driven gear in the direction of rotation of the respective gear. The resolution of the normal tooth force active perpendicular to the tooth flank is usually done as assumed here (independent of pairing), on the reference circle. The normal load can always be shifted up to this diameter on the flank. The tangential load F t active in the transverse section on the radius d/2 produces the balance to the nominal torque Mnom affecting the gear shaft, and with that it is determined by nom t2=MFd (6.3/1) Here the losses resulting from flank friction, which are slight anyway, are ignored. From this load the torque M nom results, which is to be used in the calculation of load-bearing capacity, which is described in Section 6.3.2.2. The two other load components can be calculated from the tangential load as follows: at rt taxial load tan ȕ radial load tan Į= =FF FF (6.3/2) 6.3 Load on the Tooth 189 Figure 6.3/1 Resolution of normal tooth load Fbn on the reference circle Between the three load components and the normal tooth load Fbn active in the normal section or the normal tooth load Fbt (Figure 6.3/1) defined in the transverse section, with the use of the helix angle βb (sinβb = sin βācosαn) which exists on the base cylinder, the following relationships apply: Normal tooth load in the transverse section bt bn b cosȕ FF= Axial load ab n b sinȕ FF= (6.3/3) Tangential load tb t tb n b t cosĮ cosȕcosĮ FF F==⋅ Radial load rb t tb n b t sinĮ cosȕsinĮ FF F== ⋅ Additional friction loads emerge due to the sliding of the tooth flanks taking place out of the pitch point and the friction that exists here. The line of action of these friction loads is always a tangent on the tooth flank through the line of contact of the flank pair. The direction of action is dependent on the meshing position. From the beginning of contact to the pitch point, a pushing-sliding of the addendum flank of the driven gear against the dedendum flank of the driving gear takes place. In this meshing region, the torque produced by friction load acts on the driving gear in the direction of rotation and on the driven gear against the direction of rotation. After passing through the pitch point C, in which there is only pure rolling and thus the friction loads are nil, the directions of action of the friction loads reverse. Here pulling-sliding of the addendum flank of the driving gear takes place along the dedendum flank of the driven gear. Thus the friction load generates a torque, which acts against the direction of rotation on the driving gear and in the direction of rotation on the driven gear (Figure 6.3/2). In each mesh position, the friction load and normal load produce a resulting tooth load, which hardly differs from the normal tooth load. From the equilibrium conditions for the torques affecting the driving or driven gear, it is possible to deduce the effect of the friction loads on the normal tooth load. Using the identifiers used in Figure 6.3/3 for a pair of spur gear teeth, the following applies for single tooth contact: 190 6 Load Capacity and Running performance of External and Internal Gearing () ( ) 1b b 1 b b 1 w sgnȗȝ tanĮȗ MF r F r=− − (6.3/4) Equation (6.3/4) after solving F b results in [for ()sgnȗ, refer to Equation (6.3/7)]: () ( )1 b b1 w b1 1s g nȗȝ tanĮȗ /MFrr=−− (6.3/5) Figure 6.3/2 Direction of action of the friction loads, dependent on meshing position Figure 6.3/3 Friction loads, dependent on the meshing position: a) pushing-sliding, ζ > 0 (single tooth contact), b) pulling-sliding, ζ < 0 (single tooth contact) 6.3 Load on the Tooth 191 Figure 6.3/3c) Double meshing Figure 6.3/4 Relative friction torque loss ȴM2 in a pair of spur gear teeth across the length of path of contact The torque balance for gear 2 is () ( ) 2b b 2 b b 2 w sgnȗȝ tanĮȗ MF r F r=− + (6.3/6) For a tooth pair (single tooth contact), from Equations (6.3/5) and (6.3/6) one arrives at () ( ) () ( )wb 2 21 wb 11s g nȗȝ tanĮ+ȗ/r 1s g nȗȝ tanĮȗ /rMM i−=−− (6.3/7) with the sign function sgn( ζ) = 1 with ζ > 0 (running-in side, “pushing-sliding”) sgn( ζ) = -1 with ζ < 0 (running-out side, “pulling-sliding”) In the case of double meshing, the torque balance for gear 1 is to be ascertained from the sum of the parts resulting for meshing points I and II (Figure 6.3/3c): 11 I1 I IMMM=+ (6.3/8a) with () ( )1I bI b1 bI b1 w sgnȗȝ tanĮȗ MF r F r=− − (6.3/8b) 192 6 Load Capacity and Running performance of External and Internal Gearing () ( )1II bII b1 e bII b1 w e sgnȗȝ tanȗ MF r p F r p=−− − + α (6.3/8c) and ζ = ζ I for the position of tooth pair I The total normal load Fb is b bI bII F FF=+ (6.3/9) Corresponding to the load distribution, which is considered to be known here, with bI F bF FΔ= (6.3/10) the loads acting on the pairings can be replaced by FbI = ǻ FFb and FbII = (1 –ǻF)Fb. Finally in the case of double meshing, the moment M2 = M 2I + M2II for spur gearing can be ascertained with a friction coefficient I = ȝ(ζ) and ȝII = ȝ(ζ + p e), which is dependent on the meshing position: () ( ) () () () () () ()FI w b2 2b b 2 Fe II w e b21s g nȗȝ tanĮȗ / 11 s g n ȗȝ tanĮȗ /r MF r pp r Δ− − = +− Δ − − − − (6.3/11) In the case of helical gearing it would be necessary to divide the gearing into small slices in the axial direction and for each of these parts to ascertain the part friction load for each contact according to the load distribution, to apply the equations above correspondingly, and from the sum to establish the total normal load and total torque for each section. With ȝ = 0.1 and Į = 20 as well as ȕ = 0 , the result with z 1 = 14 and z2/z1 = 1 or at the beginning of contact for single tooth contact, which is assumed as a borderline case, is a normal tooth load of F b 1.074 Ft/cos Įw or 1.038 Ft/cos Įw, and at the end of contact Fb 0.996 Ft/ cos Įw or F b 0.965 Ft/cos Įw. Thus in the pushing area the normal tooth load is increased; in the area of pulling-sliding it decreases. However, judging by the amount, the effect of friction load on the normal tooth load is minor. In the preceding numerical example, for simplicity’s sake it was assumed that the tooth load is only transmitted from one tooth pair. If one considers the real meshing conditions, the existing friction loads compensate for their effect to some extent in regions of multiple meshing, and in the case of helical gearing, the friction loads acting along the line of contact even cancel one another out to some extent in single tooth contact. With Equation (6.3/7) it is possible to determine the torque losses M V2 of the output gear occurring from the beginning to the end of contact, which result from friction on a tooth pair. In the case of single tooth contact, the losses ǻ M 2 with regard to the acting nominal torque Mnom2 = Pnom /Ȧ2 result in () ( ) () ( )wb 2 V2 2 nom2 wb 11s g nȗȝ tanĮ+ȗ/11s g nȗȝ tanĮȗ /−Δ= = −−−r MMM r (6.3/12) In Figure 6.3/4 the trace of the torque losses occurring on a tooth pair of a gear pair are shown with z 1 = 19, z2 = 40, x1 = 0.6, x2 = 0.05 for different friction values ȝ = 0.05 ... 0.25. In order to determine the losses of the gear pair during a mesh period, it is necessary to also consider the load-sharing conditions in the case of multiple meshing. In more precise analyses to determine the dynamic factor K v for the calculation of load-bearing capacity, the friction loads are accounted for in the vibration calculation by a correction of the trace of the tooth stiffness acting in the direction of the normal tooth load. This trace then exhibits a jump at pitch point C, with which the vibration stimulation by the pitch circle impulse – the change of direction of action of the friction loads at the pitch point – is also registered [6.3/1], [6.3/2]. 6.3 Load on the Tooth 193 6.3.2.2 Definition of Effective Power for Tooth Loading The load on the toothing is produced by the torque acting on the gear transmission during operation. The individual progress of each of these loads, which result from work resistance on the transmission output and from input torque, can diffe r widely. The dimensioning of the gearing is based on a continuous load M nomKAB that is equivalent for the operating time and the maximum load (impact load) MnomKAS. The purpose of the factor KAB is to register load spectra and additional external dynamic loads. With KAS, maximum loads (impacts) are registered. For nominal torque Mnom, usually the nominal torque of the driven work machine is used, which is taken from the design data of the machine. If it is not possible to determine the exact operating loads of the work machine, the gearing can be dimensioned according to the nominal power of the motor. Sometimes this can lie clearly above the necessary input power, which means that the gearing is oversized for the corresponding drive task. However, this design according to the nominal input power provides the assurance that it is possible to fully utilise the capacity of the drive system right to the limits of available input power, without premature failure of the gearing. But the actual operating conditions of gear transmissio ns are only seldom marked by virtually constant continuous loads from nominal torque. Often there are larger fluctuations in load due to intermittent work processes. These are characterised by frequent start-ups and shutdowns of the drive system as well as specific load cycles, which are run through repeatedly at virtually the same level of load. Such operating conditions are accounted for in th e calculation by load factors related to the nominal load. With these load factors, the nominal load is converted into a continuous load, which would cause the same damage effect on the gearing as the given load cycles (refer to Section 5.3). In addition to the load fluctuations which result from the variable work resistance, further load fluctuations arise from the vibration phenomena in the drivetrain that exist during operation, which are superimposed on the work loads. Registering these load fluctuations in the calculation of load - bearing capacity is explained in Sections 5.3, 6.3.4 and 6.3.5. 6.3.3 Additional External Dynamic Loads There are many cases of applications wherein the gear transmissions are subjected to additional load fluctuations which are superimposed on the load steps of a work cycle, or the constant nominal load respectively. These load fluctuations acting on transmission input and output generate additional dynamic loads in the gearing, which are known as additional external dynamic loads. The causes of these load fluctuations can be a) vibrations caused by frequent start-up and braking operations as well as changes to the load level, but which die down again relatively quickly, or b) vibrations in the drive system. The effects of the additional loads acting on the gear transmission from the outside are minor if tech- nical measures are implemented to isolate the gearing from vibrations from the rest of the drive system. Substantial isolation can be achieved under certain conditions with the use of elastic shaft couplings. Aside from that, naturally it is also possible to reduce the additional loads by manipulating the exciting mechanisms themselves. Regardless of that, any play that exists in the drive system is often the source of unwanted, major additional loads during start-up and braking operations. Vibrations in the drive system that occur in a stationary position must always first be prevented by removing their cause. But in the case of a large number of drive systems it is impossible to com-pletely prevent the intermittent excitation of vibrations, such as in the case of piston engines or universal joint shafts. Here it is very important to ensure that the drive system is not operated in speed ranges which are in the direct vicinity of the resonance speeds which result for the main periodic excitations that exist. By exerting an influence directly on the parameters of the drive system (especially torsional stiffnesses), it is often possible to shift the resonant frequencies of the system in such a way that the frequency ranges are not in the vicinity of the operating speed. 194 6 Load Capacity and Running performance of External and Internal Gearing In practice, additional external dynamic loads can hardly be completely avoided. However, their causes and their effects on the gear transmission can be considerably reduced by exerting an influence on the constructio n of the entire drive system. In order to ascertain the load-bearing capacity of the gears, the additional external dynamic loads must be quantified. Through an application factor K A they can be registered together with load spectrum influences (refer to Section 5.3). Different application factors are also to be taken into account for the two kinds of damage, between which a basic distinction must be made. When determining static safety against tooth breakage or permanent deformation or cracks due to maximum impact loads, the application factor K AS and, concerning fatigue safety, the application factor KAB are to be used. With KAS the maximum size of temporary impact loads is registered. With KAB, aside from the influence of the load spectrum, the additional loads that are constantly present during operation and decisive for fatigue strength are also taken into account. Where possible, the additional loads should also be determined experimentally in each concrete individual case. Direct measurements of the existing additional dynamic loads on the gearing involve extensive measurement effort, which is not possible in every case. It is much easier to measure the torque fluctuations in the transmission input and output shafts and to equalise these with the tooth load fluctuations. This is permissible because in the gearing, which is relatively torsionally stiff compared to the rest of the drive system, the torque fluctuations are reflected virtually unaltered. Larger deviations are only to be expected in gearing which is likewise excited to strong internal vibrations, due to external or additional internal excitations. This is the case when the gearing does not act any longer solely as a transmission link of the rotary motion. In the gear transmission in this case, internal vibrations are superimposed on the rotary motion introduced from the outside. This usually occurs in cases of higher speeds, where additional internal dynamic loads must also be taken into account. In cases where the gearing is coupled into the drive system very stiffly, deviations from the quasi-static transmission behaviour that are due to vibrations in the gear mechanism can already be seen at lower speeds. In general cases, based on the results of measurements, it is possible to determine strain in other positi ons through inference , with the help of the relationships for the forced vibrations in the torque system. When designing gearing, because of experimental analyses on similar drive systems (e.g., vehicle transmission technology), in some areas of application it is possible to utilise the available data for typical additional dynamic loads in the drivetrain. However, often no reference values obtained from experimentation are available, which means that the additional external dynamic loads must be determined in advance. To do this, vibration calculations are necessary for the drive system, which will only provide reliable results if the model is precisely prepared and one has sufficient knowledge of the excitation functions. Information on model formation and vibration calculation can be found among others in Liebig [6.3/17]. Figure 6.3/5 depicts a model for the vibration calculations. Also depicted is the load characteristics information for input, output, and gearing ascertained at different drive speeds, which results from periodic external excitation of the input. For the underlying relatively stiff coupling between gear transmission and the rest of the drive system, clear differences between the additional dynamic loads on the input or output and in the gearing are visible. If the excitation functions as well as the load spectrum are not precisely known, or if the effort required for calculations is not justified, the only possibility remaining is to register the additional external dynamic loads in an empirically determined application factor K A. This application factor can be determined with the help of empirical values which are available for certain operating conditions and can be found in standards for load-bearing capacity calculations. Such values, taken from ISO 6336-6 and DIN 3990-6, are listed in Table 6.3/1 and are supplemented in Tables 6.3/2 to 6.3/4 by the allocation of working characteristics of some drive and work machines. However, if not specifically stipulated by the client or not calculated based on the load spectrum, they should only be used for calculations in draft designs. 6.3 Load on the Tooth 195 Figure 6.3/5 Torsional vibration model of a drive system with the load characteristics information for various operating speeds Table 6.3/1 Application Factor KA= KAB Fatigue Loading (ISO 6336-6, Annex B) Working characteristics of the driving machine Working characteristics of the driven machine uniform light shocks moderate shocks heavy shocks uniform 1.00 1.25 1.50 1.75 light shocks 1.10 1.35 1.60 1.85 moderate shocks 1.25 1.50 1.75 2.00 heavy shocks 1.50 1.75 2.00 2.25 or higher Table 6.3/2 Examples for Driving Machines with Various Working Characteristics Working characteristics Driving machine uniform electric motor (e.g., DC motor), steam or gas turbine with uniform operation a) and small, rarely occurring starting torques b) light shocks steam turbine, gas turbine, hydraulic or electric motor (large, frequently occurring starting torques b)) moderate shocks multiple-cylinder internal combustion engines heavy shocks sing le-cylinder internal com bustion engines a) Based on vibration tests or on experience gained from similar installations. b) See service life graphs Z NT, YNT for the material in ISO 6336-2 and ISO 6336-3 196 6 Load Capacity and Running performance of External and Internal Gearing Table 6.3/3 Industrial Gears; Examples of Working Characteristics of Driven Machines 1) Working characteristics Driven machine uniform steady load current generator; uniformly loaded conveyor belt or platform conveyor; worm conveyor; light lifts; packing machinery; feed drives for machine tools; ventilators; lightweight centrifuges; centrifuge pumps; agitators and mixers for light liquids or uniform-density materials; shears; presses and stamping machines a) ; vertical gear, running gear b) light shocks non-uniformly (i.e., with piece or ba tched components) loaded conveyor belts or platform conveyor; machine tool main drives; heavy lifts; crane slewing gear; industrial and mine ventilators; heavy centrifuges; centrifugal pumps; agitators and mixers for viscous liquids or substances of non-uniform density; multi-cylinder piston pumps; distribution pumps; extruder (general); ca lenders; rotating kilns; rolling mill stands c) (continuous zinc and aluminum strip mills; wire and bar mills) moderate shocks rubber extruders; continuously operating mixers for rubber and plastics; ball mills (light); wood-working machines (gang saws, lathes); billet rolling mills c),d) ; lifting gear; single-cylinder piston pumps heavy shocks excavators (bucket wheel drives); bucket chain drives; sieve drives; power shovels; ball mills (heavy); rubber kneaders; crushers (stone, ore); foundry machines; heavy distribution pumps; rotary drills; brick presses; de- barking mills; peeling machines; cold strip c), e); briquette presses; breaker mills a) Nominal torque = maximum cutting, pressing or stamping torque b) Nominal torque = maximum starting torque c) Nominal torque = maximum rolling torque d) Torque from current limitation e) K A up to 2.0 because of frequent strip cracking Table 6.3/4 High-Speed Gears and Gears of Similar Requirements: Examples of the Working Characteristics of Driven Machines1) Working characteristics Driven machines uniform centrifugal compressors for air conditioning installation, for process gas; dynamo- meter test rig; base or steady load generator and exciter; paper machinery main drives moderate shocks centrifugal compressors for air or pipelines; axial compressors; centrifugal fans; peak load generators and excite rs; centrifugal pumps (all types other than those listed below); axial-flow rotary pump; paper industry: Jordan or refining machine, machine auxiliary drives, stamper medium shocks rotary-cam blower; rotary-cam compressor with radial flow; piston compressor (3 or more cylinders); ventilator suction fans, mining and industrial (large, frequent start-up cycles); centrifugal boiler-feed pumps; rotary cam pumps, piston pumps (3 or more cylinders) heavy shocks piston compressor (2 cylinders); centrifugal pump (with water tank); sludge pump; piston pump (2 cylinders) 1) The items are categorised based on information from machinery manufacturers, taking information from various standards and guide lines into account: ISO 6336-6, ANSI/AGMA 6011-I03. 6.3 Load on the Tooth 197 The American standard AGMA 421.06 contains comprehensive information for special machines, drives and mechanisms. Important to note is that not only additional external dynamic loads but also lifetime influences are registered in the application factor K AB (refer to Section 5.3), which means its values are even less capable of being applied elsewhere, and confusion often arises. Aside from that, in more meticulous analyses of additional dynamic loads, the superimpositions of additional external and internal dynamic loads must also be accounted for (refer to Section 6.3.5). If load spectra exist, and if the equivalent loads F tH, FtF are determined from these [Equations (5.3/15) and (5.3/16)], then it is possible to calculate the application factors in this case for the fatigue load (contact stress KAB(H), tooth root stress KAB(F)) and also for the known maximum loads (K AS(H,F) ): Fatigue load: KAB(H) = FtH /Ft or KAB(F) = FtF /Ft (with Ft = F tnom) Maximum load (e.g., crack): K AS = KAS(H) = K AS(F) = F tmax /Ft (with Ft = Ftnom) 6.3.4 Additional Internal Dynamic Loads 6.3.4.1 Causes and Traces Additional internal dynamic loads are caused by vibrations in the gear transmission, which are independent of external load fluctuations and mainly occur during high-speed operation. They manifest themselves in continuous loads because the excitation of vibrations occurs with each meshing of the teeth. The vibrations are excited by • fluctuations in gearing stiffness due to the alternating number of tooth pairs in contact, • manufacturing deviations from the theoretical flank of the gearing, • load-based deviations of circular pitch due to tooth deformation, which lead to an impulse due to off-line-of-action tip contact (impact of contact), • friction load reversal at the pitch point, • variable roller bearing stiffness due to an alternating number of bearing roller elements, or • imbalances on the gear wheels. Figure 6.3/6 Typical trace of additional internal dynamic loads over speed (refer to Pagel [6.3/7]) Fluctuations in gearing stiffness and the gearing deviations manifest them-selves as main excitation sources. The excitation caused by fluctuations in stiffness occurs in the angular mesh frequency ȍ z (ȍz = 2%nz) and multiples of this mesh frequency. The profile form deviations and profile position deviations to be considered as gearing deviations likewise have these exciting frequencies, whereas the de-viations of circular pitch lead to excitations with frequencies that lie below the mesh frequency. The char- acteristic trace of additional internal dynamic loads is depicted in Figure 6.3/6. 198 6 Load Capacity and Running performance of External and Internal Gearing At the position of the main resonance, with ȍz/Ȧ = 1, at which the angular mesh frequency ȍz is equal to the natural angular frequency Ȧ assigned to the gear pair, the maximum additional dynamic loads occur. In the range up to main resonance, further rises in additional dynamic loads can be detected in pre-resonance positions. Here, multiples of the mesh frequency resonate with the resonant frequency of the gear pair. In pre-resonances of very high order ( ȍ z/Ȧ << 1), such increases in tooth loads hardly exist. Above the main resonance, the additional internal dynamic loads sink again down to a value which is determined by the size of the existing stiffness fluctuations and gearing deviations. The formation of the additional internal dynamic loads can be described in connection with the speed as follows: In the case of very slow rotary motion, the load is quasi-static; this means that the fluctuations in gearing stiffness and the gearing deviations are compensated for in meshing by the relative motion of the gear wheels in such a way that always only the moment acting from the outside is transmitted by the gearing. The adapting trace of the tooth stiffness deformation of the deviation-free gearing in cases of uniform load is precisely the reciprocal value of the trace of the gearing stiffness. In cases of higher speeds, because of their inertia the gear wheels are no longer capable of compensating for the rapid change of stiffness and deviations in the gearing. A phase shift of the trace of the tooth stiffness deformation occurs vis- -vis the quasi-static development, which generates additional tooth loads. In the main resonance, this phase shift then amounts to roughly 90 (360 is for the mesh period length), and the maximum excitation of vibrations occurs in the gear wheels, which also causes maximum additional dynamic loads in the gearing. Above the resonance, the gear wheels are only slightly excited to vibrations; because of their inertia, they can hardly even vibrate in the high exciting frequency. Because they are then moving almost completely without any superimposed vibrations (corresponding to the input rotational speed) and because the tooth stiffness deformation during meshing resulting from the relative motion of the gear wheels is virtually constant, the size of the additional loads results directly from the amplitudes of the stiffness fluctuations and the gearing deviations. As a result of the vibration phenomena, not only different maximum additional loads occur for different speeds (Figure 6.3/7), but also load traces are created that differ from one another (Figure 6.3/7). In the quasi-static trace (speed n 0), the change between single and double meshing is recognisable with the corresponding load increase or decrease, respectively. When the speed approaches the main resonance in the subcritical range, clear vibrations are superimposed on the quasi-static trace. The vibration amplitudes reach their maximum in the range of the main resonance. In the case of very high speeds, the trace for the supercritical range above the main resonance approaches a constant de formation for the entire duration of meshing. To register the additional internal dynamic load, in the calculation of load-bearing capacity (ISO 6336-1, DIN 3990-1) the dynamic factor K v is used, which is defined as follows: v z max stat dyn max stat /1 / K FF F F== + (6.3/13) with () z max z zmaxȥ Fc = (6.3/14) K v dynamic factor F z max maximum tooth load during a mesh period F dyn max maximum additional dynamic load during a mesh period F stat static tooth load owing to nominal load c z gearing stiffness ȥz tooth deformation resulting from relati ve vibrations of the gearing and gear deviations 6.3 Load on the Tooth 201 With this definition of the dynamic factor, only the load exaggeration during a mesh is registered, but without its actual effect on the strain on the tooth. In the calculation of load-bearing capacity, however, the maximum increase in contact stress or tooth root stress due to additional dynamic load is of interest. Here, consideration must be given to which increases in pressure and stress the additional dynamic loads actually cause. In the case of contact stress, the different flank curvatures during a mesh period must be noted. In the case of tooth root stress, differences in the stress concentration and in the nominal bending stress must be accounted for, which stem from the changeable load application angles and bending moment arms during meshing. Consequently, in more precise analyses, the dynamic factor K Hv for contact stress and KFv for tooth root stress is to be derived from the maxima of contact stress or tooth root stress, respectively, during a mesh [6.3/23]. Here the maximum in each case is to reference the nominal value (contact stress at the inner point of single tooth contact or tooth root stress for load application at the outer point of single tooth contact). Here, K Fv corresponds to this relationship for the tooth root stress; for contact stress, K Hv results from the squaring of this relationship. Exactly how precisely the additional load actually present on the tooth is registered with that is described in Section 6.3.5. Calculation guidelines for the dynamic factor are given in Section 6.3.4.8. Figure 6.3/7 Trace of tooth loading over the tooth flank (spur gearing) for various speeds (refer to Rettig [6.3/13]) 6.3.4.2 Mechanical Spare Model General model structure A multi-body system is used as a mechanical spare model of the gearing. In this system, the gear wheels act as a discrete mass, and the gearing, the gear shafts and the housing act as elastic elements. Simple vibration models for gear drives only take into account the rotational degree of freedom defined by the rotary motion of the wheels, as depicted in Figure 6.3/8. In the first research on gearing dynamics it was possible to prove that this simple torsional vibration model was quite suitable for registering the real vibration behaviour of gearing. For a more precise vibration analysis, the torsional model is enlarged to include the three degrees of freedom of translation, axial motions and radial motions of the gear wheels. With that one takes into account the coupling between bending and torsional vibrations which exists over the gearing stiffness. Such a complex vibration system of a single-stage helical gear transmission is depicted in 202 6 Load Capacity and Running performance of External and Internal Gearing Figure 6.3/9. The stiffness of the shafts, the bearings and the housing here are summed in the axial and radial directions, and the connection to the drive system is realised through the torsional stiffness of the input and output shafts. Even more exact analyses also incorporate into the model the degree of freedom of the tilting movements of the gear wheels, whereby gyro effects resulting from this are also considered. Principally in the analysis of gearing dynamics, for the sake of an effective engineer-like work method, the spare model should only be as complicated and comprehensive as necessary. As such, it is possible to do without the inclusion of the tilting degree of freedom of the gear wheels, because as far as their size goes, the additional loads which they generate are negligible. Figure 6.3/8 Torsional vibration model of a gear pair Figure 6.3/9 Complete vibration model of a helical single-stage cylindrical gear transmission (according to K c kay [6.3/5]) Including the translational degrees of freedom is only necessary when the distribution of parameters in the gear transmission cause strong bending and axial vibrations to be expected. This is the case when radial and axial stiffnesses are present in the order of magnitude of the gearing stiffness, and the resonant bending frequencies lie in the vicinity of the mesh frequency. For the torsion-bending model, the soft bearing characteristic usually applies ([6.3/1]); that is, the stiffness of the bearing is clearly smaller than the stiffness of the gearing. Then the torsional model likewise applies for the approximate calculation if the resonant frequency of the gear pair is adjusted with a factor K m [Equation (6.3/20)]. So in comparison to gearing, very stiff bearings make the direct transition to the torque system possible. Very soft bearings cause an increase in the resonant frequency of the torque system, in cases where the very low resonant bending frequencies are disregarded. So for the natural angular frequency Ȧ, one stage of the gearing (gear pair) decoupled from the rest of the drive system applies: 6.3 Load on the Tooth 203 mȖ mm eq redȦ==Kc Kc mm (6.3/15) with 1T 2T eq 1T 2T=+mmmmm (6.3/16) eq red=mmb (6.3/17) bccm Ȗ= (6.3/18) 12 1T 2T 22 b1b 2șș; mmrr== (6.3/19) and eq eq m 12 m1 1=+ +=mmKmm K (6.3/20) Here the following apply: ω natural angular frequency of a decoupled gear pair cm mean gear stiffness cȖ mean gear stiffness (meshing stiffness) with reference to face width; Equation (6.2/1) meq reduced equivalent mass mred reduced equivalent mass with reference to face width ș1, ș2 moment of inertia of the gears m1, m2 masses of the gears In a comparison of the natural angular frequency calculated for a stiff bearing with measured values, however, a good congruence can also be ascertained in cases of a relatively soft bearing. This is because the calculated gearing stiffness is often too high, and with that the shift in resonant frequency actually effected with the correction factor K m in Equation (6.3/15) is already caused by the too-high gearing stiffness. This often gives the impression that even in cases of a soft bearing the pure torsional model would be fully valid [6.3/1]. The separation of the gearing from the drive system, which is an approach often practiced, is a model simplification, which is not permissible in every case. Relatively stiff input and output shafts make it necessary to include parts of the drive system in the gearing vibration model. Research to determine the minimum model for registering the influence of the drive system on the internal dynamics of the gearing have been undertaken by B rner [6.3/3] and have indicated a method for reducing the model, based on the analysis of energy distribution in the normal mode of vibration. Special spare models For numerous gear transmissions, the model of the isolated gear pair with solid disc gears, on which the Kv calculation (Section 6.3.4.8) is based, does not apply. However, often it is possible to form relatively simple spare models, in the form of a gear pair, in order to make the calculation procedure for the dynamic factor K v applicable for these gear pairs too. Such special cases are described as follows: a) Pinion shaft with reference diameter shaft diameter The disc model only inadequately registers the pinion inertia effect that is actually present. Co-vibrating shaft regions raise the pinion inertia and lead to a drop in the resonant frequency. The torsional stiffness of the shaft lying in the order of magnitude of the tooth stiffness is also to be included in the model. Through this, the resonant frequency of the gear pair rises, which is mainly determined by the vibrations of the then additionally restrained pinion. In their impact on the resonant frequency, both effects of the rise in inertia and of stiffness approximately compensate for one another, so the simple disc model can be used for such pinion shafts. 204 6 Load Capacity and Running performance of External and Internal Gearing b) Two-stage and multi-stage gear transmissions Here precise calculations of tooth load require the analysis of the fu ll model of the gear transmi ssion because the way the stages of the gearing affect one another must be included. Nevertheless, in cases where the torsional stiffness of the shaft is low (e.g., when the distance between the wheels of the intermediate shafts is relatively large), the stages of the gearing can be calculated separately as decoupled gear pairs. But if the step transmission ratios are large (starting at approx. i > 2.5), it is likewise possible to calculate the first stage of the gearing as a decoupled gear pair. In this case it is mainly t he clearly smaller pinion that will be excited to vibrate; the wheel hardly vibrates at all, and so this way it also does not generate any noteworthy reactions in the second stage of the gearing through the intermediate shaft. In two-stage gear transmissions, a spare model can likewise be determined for the second stage of the gearing in cases of large step transmission ratios and a stiff intermediate shaft. Here the moment of inertia of the wheel of the first stage and the moment of inertia of the pinion of the first stage translated to the rotation axis of the wheel must be combined as equivalent pinion moments of inertia. The wheel of the second stage of the gearing is used as the driven gear of the equivalent gear pair. The gearing stiffness of the spare model results from the series connection of the intermediate shaft stiffness and the gearing stiffness of the second stage. The moment of inertia of the pinion of the second stage is completely neglected in the spare model. This model formation is based on the fact that the dynamic of the second stage is decisively dominated by the relative vibra tions of the driven wheels of the first and second stages. This approach is less suitable in cases of gear transmissions with three and more stages, because here the resonant frequencies, which can be attributed to the second, third and higher stages, result from the relative vibrations of all driven wheels, which are quite strongly coupled via the gearing stiffness and the intermediate shaft stiffness. Given this relatively stiff coupling, a suggestion for the formation of a spare model comes from Schmidt, Tondl [6.3/4]. Here, for pinion inertia, the pinion of the stage and all the moments of inertia that are upstream and that are translated to the pinion rotation axis are combined, and likewise all the moments of inertia that are downstream from the driven wheel of the stage are translated to the wheel rotation axis and assigned to this. For stiffness, the existing gearing stiffness of the stag e of the gearing is used. c) A meshing wheel with several small pinions If the differences in the number of teeth are very large (approx. from i > 2.5) and thus also the differences between the moments of inertia of the pinion and wheel, each pinion-wheel gear pair can be calculated separately, without taking into account other pinions. This is related to the fact that the vibration behaviour in each pair is decisively determined by the pinion vibration, and hardly at all from the low wheel vibrations. d) Idler gears in gear chains [6.3/24] The vibration models for gear chains demonstrate a strong interconnection between the individual gears via the gearing stiffness. That is why the vibration calculation should be done here in a multiple-mass model. Despite that, it is still possible to estimate the dynamic factor using the methods described in Section 6.3.4.8. The important thing here is to use the proper resonant frequency when determining the speed factor N (related rotational speed). To determine the dynamic factor K v by approximation, the relatively simple calculation of resonant frequencies with the multiple-mass model consisting of all gear wheels and the gearing stiffnessess is quite helpful, without carrying out the more involved vibration calculations. So to generate the related rotational speed N, all significant resonant frequencies (refer to Section 6.3.4.8) must be used. For a gear chain which has only one intermediate wheel, the two significant resonant frequencies can be calculated as follows: ( )22 1Ȧ 0.5 4A AB =− − (6.3/21) ( )22 1Ȧ 0.5 4A AB =+ − (6.3/22) with m12 m23 m12 m23 1T 3T 2Tcc ccAmm m+=++ (6.3/23) 1T 2T 3T m12 m23 1T 2T 3TmmmBccmmm++= (6.3/24) with m1T, m2T, m3T on the moment of inertia of the gear wheels referencing the plane of action, according to Equation (6.3/19) cm12, cm23 mean gear stiffness of the input gear–intermediate gear pairing and idler gear–output gear 6.3 Load on the Tooth 205 As an approximation, it is possible to calculate both resonant frequencies using Equation (6.3/15) with eq 1T 2T 3T4 121= ++m mmm for the first resonant frequency (6.3/25a) eq 1T 2T 3T1.33 121= ++m mmm for the second resonant frequency (6.3/25b) and ()mm 12 m 23 0.5=+cc c (6.3/25c) whereby under the condition of m1T m2T m3T and cm12 cm23 the resonant frequencies ascertained in this way are relatively exact. e) Epicyclic gears Due to the branching of power and the gear wheels interconnected across several gearing stiffnesses, the vibration behaviour of planetary gears is very complex. The use of simple spare models leads to very imprecise calculations; however, some simple estimates of the dynamic factor with such models are usable. The generally applicable torsional vibration model for a set of planetary gears is depicted in Figure 6.3/10; here the symmetry due to arrangements of multiple planetary gears is accounted for. Figure 6.3/10 Vibration model of a set of planetary gears, based on the calculation of torsional resonance frequencies For the depicted model, the following motion equations apply for ascertaining the resonant frequencies: () ()ss k ss bs s s pp pp b p ss h h hh k hh bh h h pp 2 tk tt ps t b h h b s s s ppșĳ ĳȥ 0 șĳ ȥ ȥ 00 șĳ ĳȥ 0 șĳĳȥ ȥ 0crcnn rc c crcnn cma r c r cnn++ = +− + = ++ = +− + + = (6.3/26) with the tooth deformations of the ring gear (internal) tooth gearing hb h h b h t b p pȥĳĳ ĳ rrr=− − (6.3/27a) sun gear (external) tooth gearing sb s s b s t b p pȥĳĳĳ rrr=−+ (6.3/27b) and c gearing stiffness ck coupling torsional stiffness ș moment of inertia rb base circle radius m mass as centre distance sun gear-planet Q rotating angle np number of planet gears with the indices s sun gear h ring gear p planet t planet carrier 206 6 Load Capacity and Running performance of External and Internal Gearing If the following two conditions apply: • the ring gear is restrained stiffly ( ckh very large) or its moment of inertia șh is very large, • the planet carrier is restrained ( ckt very large) or its moment of inertia șt is very large, then the resonant frequencies of the set of planetary gears can be calculated directly. In this case, the vibration angles Q h and Qt are negligibly small, and it is only necessary to take Qs and Qp into account. Then the following applies: ( )22 1Ȧ A AB =− − (6.3/28) () ()2 22 pȦȦ appears 1 times =−Cn (6.3/29) ( )22 3Ȧ A AB =+ − (6.3/30) with ps sh \*\* spnc ccA0 . 5mm +=+ (6.3/31) \* p\* sh s p m mc c nB= (6.3/32) \* ph s mc cC+= (6.3/33) and \* 2 bșmr= (6.3/34) To assess the excitation of vibrations and noise on sets of planetary gears, it is necessary to observe the existing meshing order on the central gears and the phase relationship between meshes on the planetary gear, as explained in Appendix 5 . 6.3.4.3 Mathematical Calculation Model To calculate gear transmission vibrations, a mathematical model must be created using the mechanical spare model; the motion equations must be set up. These motion equations are differ-rential equations of the second order, which have the following properties: • forced-excited through tooth system deviations and fluctuating external moments , • parametric-excited (rheonomous) through fluctuations in tooth stiffness, • non-linear in the inclusion of load-dependent bearing and tooth stiffnesses, • coupled by the mutual influence of arranged degrees of freedom. In matrix notation (the matrix dimension equals the number of degrees of freedom) the differential equation system takes the following form: () () (),, Mq D q C q q r q p t q q h++ = + + (6.3/35) with M mass matrix (diagonal matrix) qխ acceleration vector D damping matrix r vector of forced excitation C(q) stiffness matrix p(t,q,qլ) vector of non-linearities q position vector (coordinate vector) h vector of the static load (pre-load) qլ speed vector To simplify the vibration calculation, the dependency on the rotating angle of individual parameters is converted to a dependency on time. In a good approximation one can assume that the changes of the position vector due to vibration motions are distinctly smaller than the changes due to the motion of a rigid body induced at the input. 6.3 Load on the Tooth 207 Another simplification is to neglect the load-dependency of the gearing stiffness. The reasons for load dependency are the existence of off-line-of-action tip contact, which leads to an enlargement of the contact ratio and whose size is dependent on the load, and the load-dependent stiffness part, which is produced by Hertzian deflection. If one assumes that the load fluctuations resulting from the vibrations are relatively minor compared to the nominal load (the aim in the design of the gear transmissions), then the path of stiffness ascertained for the nominal load can be used as a load-independent quantity in the calculation. Moreover, in view of the difficulties involved in determining the damping’s dependency on the rotating angle (refer to Section 6.3.4.4), only a mean damping constant is factored into the calculation. The differential equation system (6.3/35) now takes shape as follows: () () (),, Mq Dq C t q r t p t q q h++ = + + (6.3/36) For a further simplification one disregards the non-linearities ()q q t p , ,. These are produced especially when the **backlash** of the tooth and roller bearing clearance take effect, when the static pre-load that exists is cancelled out due to the vibrations. That only happens in cases of very large vibration amplitudes, which should be avoided in applications in practice. That is why a linearisation by a deletion of the vector ()q q t p , , in Equation (6.3/35) or Equation (6.3/36) is permissible for a practical calculation in the area of uninterrupted flank contact. The linearised model (Figure 6.3/11) only loses its validity if K v exceeds the value 2.0 (in direct vicinity of resonance). Figure 6.3/11 Vibration model of a single-stage gear transmission (torsion-bending-vibration system) 6.3.4.4 Parameters of the Vibration System Masses and moments of inertia The masses and moments of inertia can be precisely calculated and are also measurable. If there are no disc-like solid gears, in cases of thin webs and hubs it is sufficient to only include the gear rims in the calculation. Using the designation for diameter given in Figure 6.3/12, the following applies for the calculation of the moment of inertia: ()44 mg e a r ș0.03125ʌ 1 =− db QN (6.3/37) with af m2+=ddd (6.3/38) 208 6 Load Capacity and Running performance of External and Internal Gearing i m=dQd (6.3/39) For the equivalent mass of a cylindrical gear pairing in Equation (6.3/15) that is to be used, according to Equation (6.3/16) and using the names from Figure 6.3/12 (assuming hub and web are negligible), the following then applies: () ()2 2 m1 m1 eq b1 44 2 1 gear1 1 2 gear2 2ʌ 81 1 11 =⋅ + −−ddmd Qb Qb uNN (6.3/40) with eq red=mmb, N density, 21=uz z (6.3/41) Only for the exact calculation of vibrations is it necessary in the case of small pinions and the inclusion of bending vibrations to also take the shaft masses proportionally into account in the gear masses. The proportion which is apportioned to a gear stretches right to the shaft cross section where the half point of the torsional elasticity is reached, which is in effect between the gear and the neighbouring mass, or in cases of free shaft ends, over the entire shaft end. Figure 6.3/12 Definition of the gear dimensions for the calculation of moment of inertia Gearing stiffness The gearing stiffness of a gear pair is comprised of the sum of the stiffnesses of all the tooth pairs in a state of meshing. In very strong simplifications, these tooth pair stiffnesses over the length of the path of contact are considered as a constant, so the resulting gearing stiffness is characterised by a rectangular path. More precise analyses, however, also take into account the different contact conditions during the meshing of a tooth pair, which lead to a curved tooth pair stiffness path, which has its maximum roughly in the middle of the length of the path of contact, near the pitch point. The extent to which the stiffness drops from this maximum value in the direction of the beginning of contact and end of contact depends on the tooth geometry of the driving and the driven gears. Determining tooth pair stiffness is described in Section 6.2. In further considerations this path will be assumed as a known quantity. At the beginning of contact and at the end of contact in the theoretical tooth pair stiffness path, a jump occurs from zero to the stiffness value at the beginning of contact, or from the stiffness at the end of contact to zero, respectively. But particularly at the beginning and end of contact, the path of stiffness used in the calculation is often corrected in different ways, in order to register the existing off-line-of-action tip contact and the tip edge wear. The reason this off-line-of-action tip contact of the driving gear occurs is that the tooth pair currently meshing is deformed under load, and so between the tooth flanks of this tooth pair and the flanks of the teeth without any load on them that are right before meshing there is a distance larg er than the normal base pitch (Figure 6.3/13). Simply put, here one assumes zero-deviation gearing. As a result of this enlargement of the gap, the tooth tip of the driven gear touches the flank of the driving gear already before the theoretical beginning of contact. In the course of the rest of the rotary motion, the tooth pair currently experiencing tip edge contact is under more and more load, and with that the enlargement of the pitch due to deformation is gradually reduced, until at the theoretical beginning of contact the distance between the tooth flanks of the neighbouring tooth pairs under load again corresponds to the normal base pitch. Similar processes take place at the end of contact. If one uses the unloaded tooth pair as a reference level, then the deformation of the loaded tooth pair can be interpreted as a load and rotating angle dependent tooth system deviation, in the form of a flank bending back [6.3/1]. If one uses the loaded tooth pair as a starting point, then on the unloaded tooth pair there is a load and rotating angle dependent deviation of circular pitch in the tooth tip region, in the form of the protruding flank. 6.3 Load on the Tooth 209 Figure 6.3/13 Explanation of tooth mesh start impact by Niemann [6.3/14]. The teeth get the deformation f and respectively f ƍ under load (drawn enlarged), so the unloaded tooth 3 gets prematurely into contact with tooth 3 ƍ. The abrupt change in tooth load that exists, particularly in the case of spur gearing during the change from single and double meshing, is also known as mesh impact. Through tip relief, this abrupt change can be converted to a relatively gradual load takeover. In terms of amount, the tip relief must roughly correspond to the amount of the tooth flank bending back. Its length determines the time for the load takeover by the tooth pair coming into contact. Just as the tip relief amount decreases to zero from the beginning of contact onwards right to the end of tip relief, likewise the tooth deformation on the tooth pair coming into contact rises, and with that also the tooth load acting on it. Since tip relief is designed according to tooth deformation at a specific operating load, it also only brings about a clear improvement in quiet running for this load. For the vibrations excited by the fluctuations in stiffness, now the path of gearing stiffness resulting from the superimposition of tooth pair stiffnesses is of interest. The size of the fluctuation here plays a decisive role. To evaluate the excitation of vibrations, a Fourier analysis of the path of stiffness is appropriate here, whereby for the gearing stiffness the following applies: () () () zm s , Ȟ c,Ȟ Ȟ=1sinȞ cosȞ∞ =+ Ω + Ω cc C tC t (6.3/42) With the Fourier coefficients Cs,v and Cc,v the excitation intensities țv of each vth harmonic can then be determined: 22 s,Ȟ c,Ȟ Ȟ m+κ=CC c (6.3/43) Figure 6.3/14 depicts two typical traces of gearing stiffness for spur and helical gearing. Noticeable is the significantly more balanced path in the case of the helical gearing, because here at least two tooth pairs are always meshing. A similar path can be found in the case of spur gears with extra-depth gearing. In Figure 6.3/14 the spectra of the excitation intensities belonging to the stiffness traces are also given. Clearly recognisable is the fact that the excitation intensities of the helical gearing are far below those of the spur gearing. Figure 6.3/14 Tooth stiffness traces and spectra of the exciter intensities for spur and helical gearing (examples) 210 6 Load Capacity and Running performance of External and Internal Gearing Figure 6.3/15 Expanded model of the gearing stiffness to incorporate the load distribution over the wheel width Among others this is based on the lower additional internal dynamic loads in the case of helical gearing. Also remarkable in Figure 6.3/14 is the fact that with increasing order the excitation intensities clearly decline, and with that, with increasing order the pre-resonances they produce become less significant. The strong decline of the excitation intensities also makes it possible to reflect the traces of stiffness with a sufficient level of precision using a low number of Fourier coefficients. Unaccounted for in the path of gearing stiffness are non-linear influences, which are produced by a gaping of the tooth flanks during meshing, which are not touching on the entire face width. Because of the gaping, changes to the effective gearing stiffness can be approximately registered by breaking down the gearing stiffness into several separate stiffnesses, arranged over the face width, connected in parallel (Figure 6.3/15). Tooth system deviations As exciters of vibrations, attention must be given to the deviations from the theoretical flank due to manufacturing, which present themselves in the form of profile form deviations and profile position deviations and deviations of circular pitch. Registering tooth system deviations as changes in stiffness (parameter excitation) is only exact in a quasi-static operation. Registering them exactly in the calculation is only possible in the form of a position excitement (forced excitement). Attention must be given to the meshing conditions when determining the effective tooth deviation function. Depending on the position of the contact point, the deviations on the respective point on the tooth surface on the pinion and wheel are to be added, and aside from that attention must be given to the distribution of load in the case of multiple meshing. The resulting disturbance function of excitation caused by flank deviation fz only results with the inclusion of the effective tooth pair stiffnesses in regions of multiple meshing: ()z.i z,i 1 z z,i 1= == n i n icf f c (6.3/44) with cz,i tooth stiffness of the ith in the meshing tooth pair f z,i tooth system deviation of the ith in the meshing tooth pair n number of meshing tooth pairs whereby it is understood that because of the static initial load no separation of individual tooth flanks in the region of multiple meshing takes place due to the tooth system deviations, which would make the calculation of fz even more complicated. In order to determine the exact tooth deviation function, it is necessary to know the development of tooth deviations of the individual tooth flanks. However, often one only has knowledge of the maximum permissible tooth tolerance for a certain gear tooth quality or of measured maximum deviations. But particularly the development of the deviation over the tooth flank or the gear circumference has a considerable effect on the excitation function. Decisive for the size of the excited additional loads is the effective excitation intensity. This is formed from the ratio of the deviation fluctuations to the static deformation of the tooth. This means that gearing that has a high rate of capacity utilisation is less excited to vibrate by tooth system deviations than gearing that has a low rate of capacity utilisation, which rather tends towards “rattling” (separation of unloaded tooth flanks) or “hammering” (separation of loaded tooth flanks) [6.3/19], [6.3/20]. Deviations whose progress is almost in -phase in cases of two neighbouring t ooth pairs act as a particularly strong source of excitation. They lead to an almost purely periodic excitation, in which there is hardly any compensation from the deviations in regions of multiple meshing. Such deviations can be produced as a result of vibrations during the manufacture of gearing, if methods are used with tool-based constant pitch (hob, grinding worm). Gearing damping The damping in effect during meshing particularly influences the resonance amplitudes of the vibrations excited by the gearing and their stability behaviour (refer to Section 6.3.4.6); outside of such resonances and corresponding pre-resonances, in contrast, it is irrelevant. In the calculation the damping is usually assumed as viscous, that is, proportional to the deformation speed and constant throughout a mesh period. In an attempt to register damping as exactly as possible, its dependency on the mesh position is additionally taken into account. For gearing damping, constant tooth pair damping corresponding to the number of tooth pairs currently meshing is added together, which is then dependent on the mesh position. Even the determination of the constant gearing damping represents a 6.3 Load on the Tooth 211 problem which is not yet fully solved. Until now, an elementary calculation based on the conditions in the lubrication gap has not yet been carried out and would not be applicable as a sole, decisive influence. In research done on gearing dynamics, gearing damping parameters are usually derived from a comparison of measured and calculated additional internal dynamic loads at resonance points. However, the principle of parameter identification (refer to Hardtke [6.3/8] for example) used here, that certain parameters are ascertained based on a predefined model with the help of measured values, has the disadvantage that the predefined model structure and the stipulated model parameters have an influence on the target variable, damping. More than anything else, damping parameters ascertained in this way are dependent on the accuracy of the excitation functions used in the calculation. Aside from that, the gearing damping can be ascertained directly in subsidence tests, which only inadequately register the conditions during running, however. In the calculation, the damping is often introduced through Lehr’s damping ratio D. This has the advantage that even in cases where the actual value is unknown, which is very often the case, a cautious assumption (i.e., small Lehr’s damping ratio) very likely keeps the calculation result on the right side. With that, the damping coefficient dz can be ascertained as a parameter active on the plane of action, based on the simple torsional model of the gear pair: zz Ȗ red 2=dD b c m (6.3/45) with b face width c Ȗ mesh stiffness d z gearing damping coefficient D z Lehr’s damping ratio for the gearing m red equivalent mass referring to the plane of action, according to Equation (6.3/17) When the gearing damping results ascertained in individual tests and calculations are compared, very large differences become apparent. One reason for this is that the gearing damping is dependent on the respective operating conditions in effect. Table 6.3/5 lists a few rates of Lehr’s damping ratio for the gearing, provided by different authors (according to Donath [6.3/9]). Table 6.3/5 Information on Rates of Lehr’s Damping Ratio for the Gearing in Technical Literature according to [6.3/9] Lehr’s damping ratio Dz Molerus [6.3/18] Peeken [6.3/19] Friedrich [6.3/20] Pagel [6.3/7] Linke [6.3/1] Bosch [6.3/21] 0.01 ... 0.1 0.03 ... 0.1 0.014 ... 0.025 (at a standstill) 0.025 ... 0.4 (while running) 0.02 ... 0.14 (dependent on speed) 0.05 0.08 For more on damping refer to VDI 3830: Material and building element damping. Newer research by Gerber [6.3/2] shows that the mean damping of gearing for both spur gearing as well as helical gearing is proportional to the transverse contact ratio and increases with the load. Here the following applies: zĮ ze zĮ ze . dd b z w DD==JJ (6.3/46) with dze mean damping coefficient of a tooth pair D ze mean Lehr’s damping ratio of a tooth pair ĮJ transverse contact ratio (with ȕ = 0 effective transverse contact ratio) Despite the problems in determining damping, it must be pointed out that gearing damping only has an effect within the resonance speed regions, which are to be avoided in practical applications. With that, imprecise damping assumptions cannot lead to completely distorted results in practically oriented vibration calculations. 212 6 Load Capacity and Running performance of External and Internal Gearing 6.3.4.5 Vibration Calculation While ascertaining the additional internal dynamic loads, it makes sense to conduct the motion equation (6.3/34) in three steps: • ascertainment of the static behaviour under nominal load, • determination of the resonant frequencies and natural modes, and • ascertainment of the time behaviour and the amplitude-speed behaviour. Registering the static behaviour provides initial information about the strain on the individual coupling elements and possible weaknesses in the construction. Here the static position vector q0 marks the operating point in the vicinity of which the vibrations occur and is determined by m0Cq h = (6.3/47) The aim of the second step is to determine the resonance speed regions in order to make assertions about the anticipated additional loads already, without yet having carried out the subsequent vibration calculations. Put simply, with knowledge of the significant resonant frequencies, it is also possible to conduct the calculation of the additional loads in accordance with Section 6.3.4.8, whereby, however, major differences from the additional loads which actually exist must be allowed for. The resonant frequencies are determined with the help of Equation (6.3/48): ()2 i imȦ 0 CM v−= (6.3/48) In the actual vibration calculations, for a nominal rotational speed the course of the additional dynamic loads over a vibration period and the maximum value of the additional load in this vibration period are determined. With these maximum amplitudes it is then possible to create the speed-dependent tooth load trace (as in Figure 6.3/6, for example), whose value is produced at the end of the tooth load calculation. Finally, for a speed in question, the ratio of the maximum tooth load to the mean tooth load effective during a vibration period is stated as the dynamic factor K v. Analytically, the equation system of motion Equation (6.3/36) is not solvable in closed form. Two ways to find the solution are numerical simulation methods or partial analytical approximation methods. Numerical simulation methods enable the exact acquisition of the parameters for the respective vibration status, such as the stiffness changes in the case of separating flanks or the resulting tooth system deviations. In the numerical simulation, Equation (6.3/35) is converted into a corresponding equation of state xAx b=+ (6.3/49) with the state vector q xq= (6.3/50) whose solution is calculated with numerical integration methods. The approximation methods determine the rotating angle or time-dependent model parameters only approximately in the form of a Fourier series with a low number of harmonics. A Fourier series approach is also used for the solution components of q. As such, usually only linearised models (without **backlash** of the tooth) can be calculated. Compared to numerical integration, often clear benefits in terms of calculation time can be achieved, with sufficient precision, nonetheless. The following are used as approximation methods: 6.3 Load on the Tooth 213 • calculation based on perturbations, • methods of the slowly fluctuating phase and amplitude, • approach with infinite Fourier series, • successive approximation with integral-differential equations. Applying approximation methods to complicated structures and non-linear models demands a major effort of analytical incisiveness and preparation of the equation system. The methods of integral-differential equations have already been applied to the complicated structure of planetary gear transmissions by Hortel [6.3/10], for example. For complex models and the standardised depiction of coupled mechanical and electrical assemblies, and for determining control and regulation systems, multi-body simulation lends itself well and is described in Appendix 5. 6.3.4.6 Special Features of Vibration Behaviour Non-linear vibration behaviour At this point, only the influence of the “major” non-linearities owing to the existing tooth back- lash and bearing clearances are to be explained. Minor non-linearities of the stiffness of a tooth pair, which is to a small extent load dependent, and the non-linearity due to the load-dependent size of the off-line-of-action tip contact remain disregarded because they have no significant influence on the vibration behaviour. As far as the stiffness values are determined for the nominal load (= operating point), then this load influence has been sufficiently registered. In contrast, the effective stiffness of the gearing sinks abruptly to the value zero if the tooth **backlash** takes effect in the case of flank separation under load. Only when the loaded flanks come into contact again or the non-working flanks impact one another will the gearing stiffness take effect again. For the vibration process this means that in the gearing in the state of separated flanks there are no reactive loads in effect anymore, and with that no acceleration loads are caused through the deformation of the tooth stiffness. In cases of strong vibrations, where tooth **backlash** takes effect, gear transmissions with **backlash** would not exhibit such large vibration amplitudes as gear transmissions without **backlash**. In the tooth load trace over speed, the influence of tooth **backlash** makes itself evident in a decrease in resonance amplitudes (from roughly K v > 2.0) and a drop in the resonance curve towards lower speeds. This decrease can be plausibly explained by the fact that with **backlash** in effect, the mean value of the gearing stiffness in effect during a mesh period, and thus also the natural frequency of the resonance, decreases. The resonance shifts to lower speeds, and does this even more so the longer the **backlash** is in effect during a mesh period. However, this drop in the resonance curve brings with it the creation of unstable areas in this curve, in which the amplitude can jump. As such, different tooth load traces can occur when the resonance point is crossed with rising or falling speeds. At an increasing speed the abrupt change from the lower resonance curve segment to the upper only takes place at a higher speed than the jump from the upper to the lower resonance curve segment at a decreasing speed. Figure 6.3/16 shows a measured tooth load trace over speed, with the influence of the tooth **backlash** clearly evident (gear transmission rattling, refer to [6.3/11]). The processes actually taking place when tooth **backlash** takes effect can only be precisely registered if not only the non-linear spring characteristic curve is taken into account, but also when impact effects that occur when the tooth flanks strike are included. Especially in the case of the so-called “rattling” of unloaded gears that are also running in car shift gearboxes, the impact impulses must be taken into account because deformations of the gearing hardly occur, and only the application of impact theory can lead to solving the vibration problem [6.3/11], [6.3/19], [6.3/20]. 214 6 Load Capacity and Running performance of External and Internal Gearing Figure 6.3/16 Measured tooth load trace of a single-stage gear transmission with non-linear resonance behaviour owing to tooth **backlash** [6.3/1] (refer to Appendix 5) Unstable vibration behaviour In resonance regions, parameter excitation caused by fluctuation of the gearing stiffness can even lead to unstable vibration behaviour, that is, to constant growth in the vibration amplitudes. In practice, however, this uninterrupted growth in amplitude is prevented when non-linearities take effect. Such critical vibration conditions exist in cases where parameter and combination reso-nances occur with stiffness fluctuations of very high excitation intensity, and the vibration damp-ing is very low at the same time. These resonances, which are especially possible in parametric-excited vibration systems, exist when the following applies: jk zF HGȦ+ȦȞ1, 2, 3, ...; , 1, 2, ...Ȟjk n Ω= = = (6.3/51) with ȍz angular mesh frequency ν resonance order = number of the excitation harmonics ω natural angular frequency nFHG number of degrees of model freedom With j k one speaks of combination resonances, and with j = k of parameter resonances. Theoretically speaking, for multiple-mass models there is a multitude of such resonance points. However, only the following need to be considered; for these the following apply: • ν = 1, because the largest excitation intensity exists here (mesh frequency), • Ȧ j and Ȧk are natural angular frequencies, which are mainly allotted to the kinds of gearing whose mesh frequency one accounts for with ȍz. 6.3 Load on the Tooth 215 Since usually only one resonant frequency is significantly determined by the gearing – namely the torsional resonance frequency, which is dominated by the gearing stiffness – combination resonances do not rise to any level of importance. The vibrations here are sufficiently far away from instabilities. The vibration behaviour mostly approaches the unstable state when the fol-lowing parameter resonance exists: zT2Ȧ Ω= (6.3/52) with ȍz angular mesh frequency ȦT torsional circular natural frequency, which is determined by the gearing for which ȍz applies Here, instabilities can be detected in calculations only in the case of spur gearing and low damping because in the case of helical gearing the excitation intensities are too small [6.3/12]. In its existence, the unstable vibration behaviour is restricted to a very narrow speed range at the point of parameter resonance. The higher the damping, the smaller is this range. In practice, this means that the low speed fluctuations that always exist in real operating conditions are sufficient to prevent the constant growth of the vibration amplitudes. Moreover, in practice the endless growth of vibration amplitudes is interrupted by non-linearities taking effect when the separation of tooth flanks takes place due to large vibration amplitudes. Even though unstable vibration behaviour can hardly ever occur in real conditions, the parameter resonances determined by Equation (6.3/51) must still be taken into consideration because even in the case of stable vibration behaviour additional increases in vibration amplitudes occur. However, when one looks at gear transmissions used in practice, even in cases of very high-speed running the critical main resonance is usually far above the range of operating speed. The parameter resonance which must be heeded only occurs at double the main resonance speed, and so it is completely meaning-less for many applications. 6.3.4.7 Determination by Experiment In contrast with the process of determining additional external dynamic loads, for additional internal dynamic loads it is not possible to equate the dynamic loads in the gearing with the additional loads on the input or output shaft. The vibrations excited by the gearing are generally restricted to the gear transmission. In comparison with gearing, the cause of this decoupling of vibrations can be found in the considerably more elastic transmission input and output shafts. Only in isolated cases, in which the torsional stiffnesses of the shafts of input and output lie in the order of magnitude of the equivalent stiffness of the gearing c\* = r b2cz formed with the base circle radius, do the vibrations excited by the gearing also have an effect on the rest of the drive system. However, even in this case, there are clear differences between the dynamic load in the shafts and that in the gearing. In order to ascertain the additional internal dynamic loads, it is necessary to measure the loads directly in the gearing. These measurements can only be accomplished with a substantial amount of effort and are mainly carried out on test rigs which have been set up specifically for this. Such measurements are hardly ever carried out for gear transmissions working in drive systems. Two paths are taken to make direct measurements. With strain gauges one measures the strains at the tooth root and makes a comparison of static and dynamic load. Here the measurements can be taken only on single teeth, which may even only have to bear a portion of the additional load because of the tooth system deviations, and as such lead to too-low measurement values for the additional dynamic load. In the second method, the additional load is ascertained continuously for each mesh by measuring the accelerations on the gear wheels. In the case of torsionally soft transmission input and output shafts, it is only necessary to measure the circular accelerations on the input or output gears each because for the dynamic tooth load F dyn the following applies: 216 6 Load Capacity and Running performance of External and Internal Gearing gear dyn bș−= FrQ (6.3/53) with șgear moment of inertia of the measuring gear r b base circle radius of the measuring gear ĳ circular acceleration on the measuring gear In the case of torsionally stiff transmission input or output shafts, the dynamic moment Mdyn in this shaft must also be ascertained, without influencing the torsional stiffness of the shaft through special torque measurement shafts. The dynamic tooth load then results, in accordance with Equa- tion (6.3/53) in dyn gear dyn bș−= MFrQ (6.3/54) By integrating the measurement signals it is possible to ascertain the tooth deformations and tooth loads from the measured circular accelerations. When analysing the measured values, it is important to ensure that the trace of the tooth loads is ascertained over the mesh period and that here the effects on the load of the individual tooth cannot be registered, owing to the distribution of load in regions of multiple meshing. 6.3.4.8 Ascertaining the Dynamic Factor Kv according to ISO 6336-1, DIN 3990-1 Basics To ascertain the additional internal dynamic loads identified with the dynamic factor Kv, in addition to the exact calculation of vibrations, simple equations are also available in standards for calculating load-bearing capacity, such as ISO 6336-1 and DIN 3990-1. However, it must be emphasised from the outset that the equations given in ISO 6336-1 and DIN 3990-1 only permit a rough approximation of the additional loads which actually exist. The equations mainly register the speed dependency as well as the influences of gear tooth quality (size of the tooth system deviations) and contact ratio. A condition for the applicability of these calculation methods is the validity of the model of the decoupled gear pair on which these calculations are based. Particularly in the case of multi-stage gear transmissions, and gear transmissions and epicyclic gears that are stiffly coupled into the drive system, this condition is often not met. In these cases, however, mechanical spare models can be of use, the formation of which is described in Section 6.3.4.2. In addition to a reference to the method for precise analysis (method A), which is described in Sections 6.3.4.2 to 6.3.4.6, also to be found in ISO 6336-1 and DIN 3990-1 are three further calculation methods. Method B also requires that the main proportions of the gear pair, the circumferential velocity, the precision of the gear transmission, the tooth strength and the construction material are known. Method C, which compared to method B is simplified, can be used to calculate industrial gear transmissions. When the factor vz1/100 is less than 3 m/s (subcritical range), the effort involved in applying method B is hardly worth it; in this case method C is adequately precise. For method C it is merely necessary to know the gear tooth quality, the circumferential velocity v, the number of pinion teeth z1, as well as the load over the face width (line load) KAFt/b. Finally, there is yet a more simplified method D, in which, compared to method C, the load over the face width (line load) is specified. This method is for rough estimates of dynamic tooth loads in industrial gear transmissions and gear transmissions with similar demands, when the line load is unknown. Method B according to ISO 6336-1, DIN 3990-1 As a vibration model, the elementary spring-mass system is used, which is formed from the moments of inertia of the pinion and the gear, coupled through the gearing stiffness (Figure 6.3/8) . It is assumed that this model is valid. For calculation method B the following data are necessary: • precision of the gear pair (circular pitch deviations and profile deviations), • moment of inertia of pinion and wheel, • tooth stiffness, and • transmitted load. 6.3 Load on the Tooth 217 Basically the trace of additional internal dynamic loads over the speed is generated by the stringing together of individual straight sections for the subcritical range, the main resonance range and the supercritical range, whereby this can only ever be a rough approximation of actual development. In Figure 6.3/17 this trace ascertained for a gear transmission using the standard is compared to a curve actually measured on this gear transmission. The aim of this is to again underscore the fact that the Kv calculation is only a rough approximation . The calculation differentiates between three speed ranges, which are identified by the parameter N, the referenced speed (according to ISO 6336-1, DIN 3990-1: referenced speed). Here N is the ratio of input rotational speed (pinion speed) n1 to the resonance speed nE1 (main resonance): 1 E1nNn= (6.3/55) with 1 E1 1Ȧ 2ʌnz= (6.3/56) and Ȧ according to Equation (6.3/15); the following then applies: 1r ed 11 E1 Ȗ2ʌnmNn znc== (6.3/57) Figure 6.3/17 Comparison of the Kv trace (---), ascertained using ISO 6336-1, DIN 3990-1, method B, with the curve (żżżxxx) measured on the gear transmission [6.3/15] and the calculated trace ( ņņņ), using a model shown in Figure 6.3/8 218 6 Load Capacity and Running performance of External and Internal Gearing Here mred is to be formed in accordance with Equation (6.3/17), and the instructions regarding the formation of a mechanical spare model in Section 6.3.4.2 must be heeded. The determination of the mesh stiffness cȖ is described in Section 6.2. For gear pairings made of steel ( ࣁ = 7.83ā10-3 g/mm3), in which the gear widths b1, b2 are equal to the face width b, whose pinion is a solid gear, and for whom the model of the decoupled gear pair fully applies, the resonance speed (number of revolutions) can be read from Figure 6.3/18 or calculated using the following equation: 4 E1 22 t1Ȗ8.8 10 11 ncmz I⋅ =+ in s-1 (6.3/58) with ()22 4 2 1 IQ u=− (6.3/59) Q in accordance with Equation (6.3/39) and cȖ in N/(mm āȝm); refer to Section 6.2.4 for an approximation. Now the following three ranges are defined over the speed N: a) Subcritical range N 0.85 In this range, additional pre-resonances can occur with N = 1/2, 1/3, 1/4, .... In the cases of N = 1/2 and also N = 1/3 in particular, the actual tooth loads exceed the calculated values (Figure 6.3/17). This risk is small in cases of precise helical and spur gearing, the latter with commensurate profile correction (gear tooth quality 6 and finer, in accordance with ISO 1328, DIN 3961). Because of the low vibration amplitudes, resonances at N = 1/4, 1/5, ... are irrelevant. In the case of line loads of KAFt/b < 50 N/mm, there is a particular risk of vibrations (in some circumstances resulting in the separation of the meshing tooth flanks), especially for spur or helical gears with coarse gear quality and at higher circumferential speeds (influence of tooth system deviations). b)Range of the main resonance 0.85 < N ≤ 1.15 This region should generally be avoided, especially for spur gears without profile correction or helical gears of the gear tooth quality 7 or coarser, in accordance with ISO 1328 and DIN 3961. Helical gears of fine gear tooth quality and with İȖ > 2.5 can operate satisfactorily in this range. Spur gears of the gear tooth quality 6 or finer, in accordance with ISO 1328 and DIN 3961, need to have a commensurate profile correction. Thorough tests must be carried out. Figure 6.3/18 Diagram to determine the resonance speed nE1 according to ISO 6336-1, DIN 3990 -1; I 2 and cȖ Equation (6.3/59) 6.3 Load on the Tooth 219 c) Supercritical range N > 1.15 In this range, the same restrictions for operation apply as in the main resonance range. Here resonance peaks can occur at N = 2, 3, .... In most cases, however, the vibration amplitudes are small because, in general, the exciter loads with lower frequencies than the mesh frequency are small. In this range, it is also necessary in the case of some gears to take into account dynamic loads due to bending vibrations. For the given speed ranges, according to ISO 6336-1 and DIN 3990-1 the following equations apply: a) Subcritical N 0.85: () vv 1 p v 2 f v 3 k 1 KC B C B C B N=+ + + (6.3/60) b) Main resonance 0.85 < N ≤ 1.15: vv 1 p v 2 f v 4 k 1 KC B C B C B=+ + + (6.3/61) c) Supercritical N ≥ 1.5: vv 5 p v 6 fv 7KC B C B C=++ (6.3/62) In the intermediate range 1.15 < N < 1.5, Kv is interpolated between the values for the frequency range Kv (N = 1.15) and the supercritical range Kv (N = 1.5): ()() ( )()v vv1.15 1.51.5 1.50.35vKN KNK KN N=− === + − (6.3/63) For the subcritical range, the linear rise in Kv stipulated in accordance with Equation (6.3/60) only roughly corresponds to actual conditions. Particularly in the clearly subcritical range, larger additional loads are to be expected than calculated with the linear trace. In this range there are pre-resonances of higher orders lying very close together, which may only have low excitation intensities, but still cause higher additional loads than those calculated with Equation (6.3/60). A parabola-shaped trace of the Kv factor in the subcritical range corresponds better to reality. The following equation can be used for such a trace for 0 < N < 0.85: () vv 1 p v 2 f v 3 k 10 .85 =+ + +K CB CB CB N (6.3/64) In the equations for Kv the effects of the tooth system deviations and the profile correction are registered by the dimensionless parameters Bp, Bf and Bk. Here, Bp and Bf represent the ratio of the effective tooth system deviation s fpe eff or ff eff to the static tooth deformation; Bk is to register the size of the excitation of tooth stiffness, taking the mitigating influences of flank relief and the wear from running-in into account: p Atpb eff'(/ )f BcKFb= (6.3/65) f AtfĮeff'(/ )f BcKF b= (6.3/66) a k At'1/cCBKF b=− (6.3/67) For gear tooth quality 7 and coarser, according to ISO 1328-1 and DIN 3961, Bk = 1 applies. The values for the effective deviation of base pitch f pe eff and the effective profile form deviation ff eff are ascertained by deducting estimated “running-in” amounts ( yp and yf): f p pb eff Įeff f Į pb; f fyf f y=− = − (6.3/68) fpb, ffĮ For these values of transmission variations of the pairing, according to probability considerations the values of the gear of the pairing with the larger deviations can be used. yp, yf In cases where neither tests nor operating data on running-in characteristics of the basic materials exist, the running-in amount yĮ can be used. Here the following applies: For construction steel and quenched and tempered steel: y p = 160 ā fpb/ıHlim y f = 160 ā ffĮ/ıHlim for / 5 m/s: without restriction for 5 m/s < / ≤ 10 m/s: yp, yf 12800/ıHlim for / > 10 m/s: yp, yf ≤ 6400/ıHlim 220 6 Load Capacity and Running performance of External and Internal Gearing For case-hardened and nitrided steels: yp = 0.075 ā fpb yf = 0.075 ā ffĮ with the restriction yp, yf 3 ȝm Ca Design value for profile correction (tip relief). In cases of gears without the specified profile correction, the value Cay resulting from running-in is to be inserted into Equation (6.3/67): 2 Hlim ay118.45 1.518 97Cσ =− + (6.3/69) With the factors Cv1, Cv2, ..., Cv7 the effects of the individual tooth system deviations and the fluctuations in mesh stiffness are registered in Equations (6.3/60) to (6.3/62). Here the following apply: C v1, Cv5 for the effects of the deviations of circular pitch C v2, Cv6 for the effects of the profile form deviations C v3, Cv4 for the effects of the fluctuating mesh stiffness C v7 for the load portion from the tooth deformation, which results with practically constant circumferential speed because of the fluctuation of tooth stiffness The factors Cv1, ..., Cv7 can be taken from Figure 6.3/19 or determined with the help of the information in Table 6.3/6. Table 6.3/6 Values of Cv for the Calculation of the Dynamic Factor Kv in Accordance with ISO 6336-1, DIN 3990-1, Method B 1 < JȖ ≤ 2 JȖ > 2 Cv1 Cv2 Cv3 C v4 Cv5 Cv6 0.32 0.34 0.23 0.90 0.47 0.47 0.32 Ȗ0.57 0.3İ − Ȗ0.096 1.56İ − Ȗ Ȗ0.57 0.05 İ 1.44İ − − 0.47 Ȗ0.12 İ1.74 − 1 < JȖ 1.5 1.5 < JȖ 2.5 JȖ > 2.5 Cv7 0.75 0.125 sin [ ʌ (JȖ - 2)] + 0.875 1.0 Figure 6.3/19 Diagram of Cv factors for calculating the dynamic factor Kv according to ISO 6336-1, DIN 3990-1, method B 6.3 Load on the Tooth 221 Applicability in the case of multiple-mass models The given calculation methods are principally only for decoupled gear pairs. For approximate calculations of the dynamic factor Kv in the case of multiple-mass models, however, which take into account the couplings between the gearing stages and the coupled masses, the method B (ISO 6336-1, DIN 3990-1) can also be used. To form the referenced speed N, all of the resonant frequencies which are significant for the gearing are then to be included. To do this, the resonant frequencies determined in the multiple-mass model are analysed to see how strongly a deformation of the gearing takes place in the normal modes of vibration. The resonant frequencies which are considered as significant are those in which, because of the deformation, considerably larger forces or moments are generated in the gearing than in the other elastic elements. For each of these resonant frequencies, the referenced speed N is to be generated and, with the help of Equations (6.3/60) to (6.3/63), the dynamic factor determined. The largest dynamic factor determined this way is then to be used in the calculation of load-bearing capacity. Here the same resonance effects are ascribed to all the significant resonant frequencies, whi ch does not apply in reality, however. However, the omissions of additional resonance phenomena which occurred in the direct application of method B on the isolated gear pair are avoided. A distinguishing feature of such frequency analyses is that often only one or two resonant frequencies emerge as significant. 6.3.5 Resulting Load – Practical Approach, Omissions During operation, additional internal and external dynamic loads are superimposed on the effective load-bearing capacity and lead to the resulting load owing to operating conditions. In the calculation of load-bearing capacity this superimposition is registered approximately with the factors K AB, KAS and Kv, separately. With the factors KAB and KAS the nominal load is converted into the maximum effective load for continuous operation or for momentary impacts owing to additional external dynamic loads. This load is then assumed as an almost constant external load during meshing, when the maximum of the tooth load in the course of a mesh is calculated with the help of the factor K v. Thus the dynamic factor Kv used this way expresses only the change to the load introduced from the outside, owing to the internal dynamics. However, to take the influence of the internal dynamics into consideration in the most precise way, it is not sufficient to only register the tooth load maximum during a mesh. Due to the mutability of the flank curvature radii throughout the mesh and the likewise non-constant load application angle, bending moment arm, and stress concentration factors, the tooth load increase has different effects on the increase in contact stress and tooth root stress. Moreover, the maximum increase in tooth loads usually lies in the double meshing region, but because of the distribution of load, the stress maxima usually do not occur in this meshing region. The condition arrived at in the calculation of load-bearing capacity, that the maximum external load K AFt remains virtually constant during a mesh, is not met in every case. In cases of gear transmissions that are coupled very stiffly into the drivetrain, high-frequency external excitation leads to an influence on the tooth load trace during a mesh period because of this excitation. In such cases, in an exact analysis that goes beyond the rough approximation in accordance with Section 6.3.4.8, it is also necessary to include the external excitations when ascertaining the tooth load trace over a mesh period. Here the phase relationship of the excitation must be taken into account, which, in interaction with the additional internal dynamic loads, leads to the largest resulting load. In order to make a global assessment of the influence of the gearing on the excitation of vibrations, the excitation load L A was defined [6.3/22]. However, it is not suitable for characterising the dynamic behaviour of the gear transmission at a specific speed. 6.3.6 Symbols and Symbol Explanations of Section 6.3 A rad2/s2 constant in calculation of natural frequencies a mm centre distance B rad 2/s2 constant in calculation of natural frequencies b mm tooth face width, gear face width C Nm/rad, stiffness matrix N/ ȝm C rad2/s2 constant in calculation of natural frequencies C a ȝm amount of tip relief Cay ȝm amount of tip relief due to running-in Cc,v N/ȝm cosine coefficient of v th order of tooth stiffness Fourier series Cs,v N/ȝm sine coefficient of vth order of tooth stiffness Fourier series Cv... - constant in calculation of dynamic factor 222 6 Load Capacity and Running performance of External and Internal Gearing c N/ȝm translative stiffness at gearing and bearing c Nm/rad rotatory stiffness (for example shaft rotational stiffness) cz,i N/ȝm tooth pair stiffness of i-th tooth pair in contact (in considered mesh position) cȖ N/(mmāȝm) mesh stiffness (mean mesh stiffness over one mesh period per tooth face width) cƍ N/(mm āȝm) (mean) tooth pair stiffness per tooth face width D Nms, Ns/ ȝm damping matrix D - Lehr’s damping ratio d Ns/ȝm damping constant, translative d Nms damping constant, rotatory d mm diameter F N load, force f ȝm tooth deviation (from true involute) ffĮ ȝm profile form deviation fp ȝm single pitch deviation h Nm, N vector of static load (pre-loading) i - gear ratio KA - application factor (common, according to DIN 3990) KAB - application factor for permanent load KAS - application factor for shock load Km - inertia conversion factor Kv - dynamic factor M kgm2, kg inertia matrix m mm module m kg inertia N - resonance ratio (according to ISO 6336-1, DIN 3990-1) nE1 min-1 resonance speed of driving gear n1 min-1 speed of driving gear p Nm, N vector of non-linearities pe mm base pitch Q - diameter ratio q rad, ȝm vector of degrees of freedom (position vector) qլ rad/s, ȝm/s velocity vector, (first deviation of position vector with respect to time) qխ rad/s2, acceleration vector (second deviation ȝm/s2 of position vector with respect to time) r Nm, N vector of enforced excitation r mm radius (without index: for reference circle) M Nm torque t s time u - number of teeth ratio /i - mode of i-th natural frequency (i-th natural mode) / m/s circumverential speed x variable state vector yf ȝm running-in wear as profile form deviation yp ȝm running-in wear as pitch deviation α pressure angle β helix angle ǻF - force ratio, load ratio ǻM2 - relative friction loss moment at gear 2 JĮ - transverse contact ratio Jȕ - overlap ratio JȖ - total contact ratio ȗ mm distance between mesh point and pitch point θ kgm2 inertia moment κ - parameter intensity μ - friction coefficient N, ȡ g/mm3 material density Q rad rotational angle ȥz ȝm tooth deformation ȍ rad/s angular frequency of excitation ȍz rad/s angular frequency of gear mesh ω rad/s natural angular frequency ȦT rad/s natural angular frequency of torsional system Indices an input ab output b base cylinder dyn dynamic eff effective G gear unit eq equivalent f profile form (in case of tooth deviations) h internal gear k coupling m mean max maximum N nominal n normal section p planetary gear pb base pitch r radial red reduced s sun gear stat stationary T torsional t transverse section t planet carrier V loss w operating z gearing v order number of elements of Fourier series 1, 2 driving, driven gear 6.4 Load Distribution on Gears 223 6.4 Load Distribution on Gears 6.4.1 The Basic Problem For spur and helical gears, the load relationships in meshing are characterised by the load distribution on the individual tooth pairs engaged ( transverse load distribution ) as well as the distribution of the load either in the direction of the face width or along the line of contact (face load distribution ) (Figure 6.4/1). This can be influenced and is a crucial factor. The early scientific investigations into the distribution of load and contact stress include the work of F. Karas [6.4/35]. Figure 6.4/1 Load and stress distribution; discretisation of the load for influence coefficients methods (the forces Fj act normal to the tooth flank surface) The load distribution of the theoretically deviation- free engagement for real gears is in practice generally influenced by manufacturing deviations and elastic deformations as well as by the adjacent components (shaft, gear body, bearing, housing) and the bearing clearance, which results in a corresponding change to the transverse and face load distribution. For helical gears in addition, the line of contact runs diagonally across the tooth, and several tooth pairs are usually simultaneously engaged in the load transmission. As a result of the diagonally running line of contact, tooth sections nearer to the tooth tip and the tooth root (i.e., tooth lengths of differing stiffness) are engaged in the meshing and cause a non-uniform load distribution, even with deviation-free meshing along the line of contact. At the beginning of contact (tooth tip on the face side) or at the end of contact (tooth root on the opposite front face) or for gears with a large overlap ratio, specific tooth sections remain temporarily free of load and exercise a significant stiffening effect in the direct vicinity of the loaded tooth sections. If several tooth pairs of a gear pairing are engaged in the meshing, then these are involved in the power transmission in the sense of springs acting in parallel. The individual tooth pair takes up a specific part of the total force according to its spring stiffness. The load distribution can be calculated using the deformations. An explicit, precise solution is not possible. In addition to various kinds of experimental possibilities, there are also particularly two fundamental procedures of solution known, which are successfully used in accordance with the respective requirements: a) Methods allowing the curve of tooth load along the line of contact and the gearing stiffness to be calculated b) Approximate solutions in the form of load distribution factors, expressing the non-uniformity of the load at the most severely stressed poi nt in the line of contact. 224 6 Load Capacity and Running Performance of External and Internal Gearing 6.4.2 General Calculation Approach Calculation approaches to determine the load distribution have to reflect the load-deformation relationships in the meshing in relation to the deviations from the line of contact arising from the manufacturing deviations and flank modifications. One calculation model has proved itself, the basic idea of which is known from a series of pub-lications [6.4/1] to [6.4/8]. Basically, it is assumed that all the effects acting on the load distribution can be superposed. The overall size of the deformation is thus created by the superposition of the individual effects. When load is applied to a tooth pair, what is known as a total deformation is created, which is expressed in a change in rotation angle of both gears. The total deformation f z (s) in a given transverse section s of the gear (refer to Figure 6.4/15) in the direction of the normal tooth force thus results from the sum of both parts of the pinion tooth (I) and wheel tooth (II): z zI zIIf(s) f (s) f s () =+ (6.4/1) This sum of the individual deformations must be constant in each transverse section when the teeth (I and II) engage on a line (line of contact); they should theoretically neither penetrate each other nor diverge. zz zI zII. f(s) f (s) f (s) f c o n s t =+ = = (6.4/2) If a given deviation of line of contact fk (s) (also known as a gap) is superposed on the elastic deformations, then it is necessary to extend Equation (6.4/2). With the approximation fz, which is constant over the entire face width, we obtain zz zI zII kf(s) f (s) f (s) f (s) g (s) f c o n s t. =+ + − = = (6.4/3) The part g(s) is the residual gap of the tooth pair left over following the elastic deformation. If all the forces, deformations, stiffnesses and deviations of the line of contact are recorded in the direction of the plane of action in the normal section, then – based on the aforementioned correlations – it is possible to describe the load-deformation relationships of the meshing in terms of a system of linear equations. The coefficient matrix is a system of interaction influence coefficients of the deformation. The solution vector contains a number of individual forces reflecting the load distribution and the total deformation of the gear pairing. Representations are also known of the stress-deformation relation- ship from a matrix of supporting value pairs of an approximation polynomial [6.4/9], a system of non-linear integral equations [6.4/10], [6.4/11] or a tridiagonal matrix, consisting of length and coupling stiffnesses [6.4/29], [6.4/30]. The load distribution determined along the contact lines is the basis for subsequent calculations of the local stresses. Numeric, analytical and part-analytical methods are applicable as calcula-tion procedures. 6.4.3 Influence Coefficients Method 6.4.3.1 Fundamental Relationships Because of the possibilities arising for a simple and transparent inclusion both of all elastic defor-mations in gears and of an equally simple treatment of any deviations in the line of contact (including flank modifications), the influence coefficients method has proven favourable [6.4/7], [6.4/29], [6.4/30], [6.4/31]. This method is based on the principle that the deflections and displacements of the individual tooth (or tooth pair) consist of the effects of successive individual 6.4 Load Distribution on Gears 225 loads applied along the line of contact. For this purpose, the lines of contact are divided into lengths of equal size and assigned with unit forces. The transition from a given transverse section t in Equations (6.4/2) and (6.4/3) to the respective centre of a discrete length j or i of the line of contact is then accomplished. The deformation at a point i of the line of contact, arising from a force F n (j) acting at the point j, can be calculated using zij n = ( j )ij fa F (6.4/4) The system of equations for a tooth subjected to load is given as follows: 11 n 1 j n 1n n z1 i1 n i j n in n zi n1 n n j n nn n zn nn n bn( 1 ). . . ( j ). . . ( n ) ( 1 ). . . ( j ). . . ( n ) ( 1 ) ... ( j ) ... ( n ) ( 1 ) ... ( j ) ( n )aF aF a F aF a F a F aF aF aF FF F F⋅+ + ⋅+ ⋅ = ƒ ⋅⋅ ⋅ ⋅⋅ ⋅⋅⋅ ⋅ ⋅+ + ⋅+ ⋅ = ƒ ⋅⋅ ⋅⋅⋅ ⋅⋅⋅ ⋅ ⋅+ + ⋅+ ⋅= ƒ ++ + = (6.4/5) The following relationship applies for the tooth pairing: zij zijI zijII=+ fff (6.4/6) A system of equations is obtained for the individual pairing, equivalent to the individual tooth. On inserting the conditions from Equation (6.4/2) or (6.4/3), a row of linear system equations for the length i of a line of contact with n lengths has the form ()n ij n ki z i j=1 ( ) + = eF j g ƒ− ƒ (6.4/7) where eij is the sum of the deformation influence coefficients from the pinion tooth and wheel tooth: eij = a ij I + aij II (6.4/8) At the same time, the sum of all individual forces Fn(j) must be equal to the total force acting along the line of contact of the pth tooth pair. ()n n bn j=1 ( ) = jFp F (6.4/9) If several tooth pairs ( zk, number of meshing tooth pairs) are invo lved simultaneously in the load transmission (İ Ȗ > 1), then the system of Equations (6.4/7) and (6.4/9) can be accordingly expanded. The following equation applies to each point i for each meshing tooth pair p with individual loads n p: pn n k z ij i i j=1 ( ( ,) = ( + ( )) ) ep p j p gp F −ƒƒ (6.4/10) 226 6 Load Capacity and Running Performance of External and Internal Gearing where e(p)ij influence coefficient (p th tooth pair, point i, stress at j) Fn (p,j) load ( pth tooth pair, load at j) fz total or maximum deformation (approach of the tooth flanks) fk (p)i deviation from line of contact ( pth tooth pair, point i) g (p)i residual gap (p th tooth pair, point i) Since the sum of all the individual forces Fn(p,j) is equal to the total tooth force F bn, the following equation applies: k p z n nb n j=1 p=1 () = p, jFF (6.4/11) The precision of the load curve is highly dependent on the section face width. This should lie in the magnitude of 0.5m , and even less for special investigations. A solution of this contact problem becomes generally possible by means of an iterative method by initially assuming a flank contact over the entire line of contact [ g(p) i = 0] and setting this to zero [Fn(p,j) = 0] once the system of equations has been solved for a recalculation in the areas of negative forces. The method normally converges after one to two iterative steps. Considerable experience has been gained for a practical solution of the system of equations. The choice of effective algorithms results in short computing times [6.4/30], [6.4/33]. For planetary gear applications refer to [6.5/115] and Baumann, Fr.: Theoretical examinations on load sharing and load distribution ..., diss. TU Dresden 2010. 6.4.3.2 Determining the Deformation Influence Coefficients Methods, influences When a tooth is subjected to load, each part of the tooth experiences a corresponding deflection. If the validity of Hooke’s Law is assumed, the loads and deflections are linearly associated with each other. If several loads are applied along the lines of contact, then each and every one of them induces a deflection at each point of a line of contact of the meshing tooth pairs. The resulting de-flection of this point under the effect of all the loads is equal to the vectorial sum of all the individual deflections. The linear relationship between the external stress at one point and the deflection at this (or another) point can be represented by constant coefficients, the deformation influence coefficients e ij. The first index denotes the point at which the deflection occurs, while the second index denotes the external stress, as a result of which the relevant share of the deflection is caused. The symmetry of the influence coefficients e ij = eji can be derived from the independence of the sequence of loads applied to the final state of the deformation energy. The deflection at point i in the direction of F (i) by the force F(j) is equal to the deflection at point j in the direction of F(j) by a force of equal size F(i). The number of deformation influence coefficients required to solve the system of Equations (6.4/7) to (6.4/9) is thus reduced from k 2 to (k +1)k/2. The deformation influence coefficients theoretically contain the effects of all the elastic defor-mations within the gear to be investigated. The most important components are • tooth deformations, • gear body deformations, • shaft deformations, • bearing deformations, and • housing deformations. 6.4 Load Distribution on Gears 227 These components can be individually determined as to the linearly assumed stress-deformation behaviour and cumulatively superposed. For parts of the tooth deformation (Herztian flattening) and deformations of the shaft bearing, this linearity is not initially present. In such cases, an approximate linearisation can ensue in the area used. A depiction of all the components of the total deformation within a gear by means of deformation influence coefficients is possible in principle. Following an assessment of the cost/benefit ratio, however, the following model has proven itself to be favourable and largely satisfactory for the observed cases. Tooth and gear body deformations are recorded by means of deformation influence coefficients. Shaft, bearing and housing deformations can be determined for the respective load (constant along the face width), and taken into account as a resulting deformation of the line of contact in the further calculation. A recalculation for a changed load distribution will improve the result. While shaft deflection and shaft torsion can be determined with a simple me thod, housing deformations on the other hand may, generally speaking, only be ascertained using finite element analysis. Calculating the bearing deformation or deflection for roller bearings can ensue according to [6.4/12], [6.4/13], and approximately also according to [6.4/14], [6.4/32], and for plain bearings according to [6.4/15], [6.4/16]. For precise analyses of the deformation, the load distribution of adjacent gear stages has to be taken into account, or alternatively, the interaction involved in multiple meshing has to be considered , which – for planetary gear, in particular – can be of substantial influence [6.4/36] to [6.4/38]. Calculation models The actual tooth deformation, in turn, is considered as the sum of various deformation components. Deformations arising as a result of bending and shear load as well as Hertzian flattening of the contact points are regarded as critical. With a solid gear body, pressure deformations can be neglected. For systematic investigations, a finite element analysis is necessary to determine these deforma- tions. The high degree of calculation effort required, however, makes a simplification by means of suitable approximate solutions appear desirable. To begin with, for this purpose, an approximation of the complicated geometric form of the tooth is obtained by using a tooth model. The helical-shaped torsion of the helical tooth around the gear body is neglected, and the end faces of the helical tooth perpendicular to its longitudinal axis are assumed. The helical tooth is regarded as an equivalent spur tooth with a width b r and a normal tooth thickness sn. In this regard, rcos=βbb (6.4/12) In terms of the tooth pairing, the effects of the elastic, acute-angled and the stiffer, obtuse-angled face side of the tooth overlap as a series arrangement of springs, so it may be assumed that the end faces of helical gears are generally more elastic than for spur gears. For small helix angles ( β < 10 ), this effect can be neglected, while for helical gears with β > 10 , it should be included in the calculation of the influence coefficients; see Equations (6.4/15) to (6.4/21). The equivalent spur tooth is divided into sections of constant width in the face width direction. The division in the tooth height direction, which then simultaneously allows a depiction of the line of contact running diagonally across the tooth for helical gears, is best effected by a projection of the plane of action meshed at constant intervals on the active tooth flanks of the pinion and wheel. Based on this model, it is possible to perform the calculations of the individual tooth deformations described in the following section in a simple way. 228 6 Load Capacity and Running Performance of External and Internal Gearing Calculating the tooth deformation The method described in detail below considers the deformation at the position of force application of adjacent sections by using a related, generalised decay function E and approaches the actual deformation by the flexibilities q(y) on calculated tooth segments. By evaluating a large number of FEM calculations carried out in a targeted manner, a general, relative decay function E was determined [6.4/18], [6.4/19], which can be used as the basis for an approximate determination of the influence function of the toothing in each case (Figure 6.4/2). For the infinitely wide tooth, the equation is ()\*\* 3 \*2 \* \*\* \* ) = 0.146 cos 0.027 0.333 1.545 1 ||with = ; for > 6 is ) = 0 E ( x x x x xxx E ( xm∞ ∞ ⋅⋅ − ⋅ + ⋅ + (6.4/13) where x\* is the relative distance to the position of force application. The decay function \*()xE∞ records the principally uniform course of the related tooth deformation in the vicinity of the position of load application for the infinitely wide tooth. The reference value here is the flexibility q at the point y = y F of a tooth slice of equal tooth geometry and a face width b = 1 m. The differences in sizes of the absolute deformation or flexibility as a result of the special geometry (e.g., provided by the number of teeth and profile shift) are taken into account by the specific flexibility q(y = yF) of the respective gear. Different meshing positions, that is, different bending level arms, are therefore recorded by the corresponding flexibility q(y = y F). For helical gears, the lines of contact run diagonally across the tooth. Figure 6.4/2 General decay function of tooth deformation per Equation (6.4/13) This provides for the meshing of tooth segments of different flexibility by means of different bending level arms. The influence function to be determined in this manner (including diagonally across the tooth) is calculated by drawing on the flexibility q(y) by applying load at the point y F for the respective tooth segments not lying beneath the po sition of force application (Figure 6.4/3). (For helical gears, the direction of action of the force and deformation expressed by α′ differ slightly at points i ≠ j.). 6.4 Load Distribution on Gears 229 Figure 6.4/3 Tooth deformation calculation model per Equation [6.4/20] The effect of the finite face width b or the edge on the influence function E∞ of the infinitely long tooth is taken into account by mirroring the defor- mation behaviour on the face side of the tooth thought of as an infinite tooth strip. In this process, the defor-mations lying outside the face width b are superposed on the deformations lying inside the face width (Figu-re 6.4/4a) [6.4/17]: E = E ∞ + E∞ (6.4/14) The result can be further improved by introducing modification functions W f: \*\*\* \* \*\* \* \* VR VR V R f = ( = ) + ( = + ) ( , )EW xxx x xx x x EE∞∞ − (6.4/15) where VR VV max R6\*\* \* xx = ; = ; = xx xmm and xV, xR are taken from Figure 6.4/4b, and Wf from Equation (6.4/16). Figure 6.4/4 a) Determining the edge deformation using the mirroring method; b) definitions at the pointed side of the tooth 230 6 Load Capacity and Running Performance of External and Internal Gearing For helical gears, the effect of the end-face slope is taken into account by the approximation Wf as follows: Wf = 1 + f G fL (6.4/16) For helical gears, a further function fL allows the principal decay function of the edge influence to be represented, and a size function fG allows the increase or reduction of the deformation in the proximity of the tooth edge for the respective helix angle compared to a spur gearing to be realised. The edge influence of the deformation has decayed at a distance LR from the edge. This region can be specified as a function of the helix angle and related to the normal module, giving the (empirical) equation: \* R R = = 2.1 tan LLm⋅π⋅ β (6.4/17) For\*\* VR 0 < x L≤ and \*\* RR 0 < x L≤ (within the related decay region \* RL), the size xrel for the decay function fL (xrel), for which the deformation influence coefficient is to be determined, can be calcu- lated from Equation (6.4/18) as follows: R VR rel R2\*\* \* \* ( + ) / xx L = xL− (6.4/18) For xV, xR refer to Equation (6.4/15); xrelmin = 0. The decay function fL is also determined empirically and differs for the acute and obtuse end face: for ȕSt > 0: 54 3 2 rel rel rel rel rel rel L( ) = 8.26 15.6 + 10 2 + 0.333 x xx x x x−− ƒ (6.4/19) for ȕSt < 0: rel rel 10 2 7L() = .x x⋅ ƒ (6.4/20) We then obtain the size function f G: 32 GS t St St St ( ) = 0.645 + 1.454 + 1.176 + 0.0435ƒβ ββ β (6.4/21) where Stβ = ⏐ȕ⏐ for load on the acute end face and Stβ = -⏐ȕ⏐ for load on the obtuse end face (Stβ in radians). We then obtain the influence coefficients: ij ij =E q a ⋅ (6.4/22) E deflection influence function, Equation (6.4/15) aij influence coefficient (describes the influence of the force Fnj acting at the point j on the deformation at the point i) qij flexibility of a tooth slice with width ǻb = 1m from Equation (6.4/23) at the point i as a result of the load on the bending level arm of point j Flexibility The flexibility q(y) can be interpreted as the sum of the influence of the normal tooth force on the tooth and the contiguous part of the gear body (Figure 6.4/3). In the calculation, we differentiate between the bending deformation ( B) of the tooth, the shear deformation (S) of the tooth and the deformation of the part of the gear body contiguous to the tooth ( R). The deformation caused by Hertzian flattening (H) is considered separately: q = q B + qS + q R (6.4/23) 6.4 Load Distribution on Gears 231 The amount of bending flexibility qB is determined based on the equation of the bending line η of a board-shaped beam with variable cross section (refer to [6.4/20]). For regions of the beam where y < yF, we obtain 22 F bn( ) ( 1 ) () cos () = = () ′ ′′ −α − η′′ −By y y vqyIyE F (6.4/24) where I(y) equatorial moment of inertia of plane around the longitudinal axis of the beam at the point y of the face width section of the tooth b E modulus of elasticity α′ load application angle () ()22 F 1 B 1 cos () = d y + ' C()yy vy qEI y ′α− − (6.4/25) 2 BZ() = d y + C ′ yqq (6.4/26) The constants C1 and C2 are obtained from the boundary conditions at the clamping point qB(y = 0) = 0; qBƍ(y = 0) = 0. For regions of the beam where y > y F, then by making use of the deformation and tilt at the position of force application y = y F, we obtain () () () ()BF BF F F B > = = + = yy y y q qy qy y y ′ − (6.4/27) The amount of elasticity of the gear body directly adjacent to the tooth is obtained from the approach adopted by Weber and Banaschek [6.4/20] for the face width section ǻb subjected to the individual load as follows: () ()2 2 2 FF RF 2 FFcos = = 5.2 + + 1.4 1 + 0.294 tanyyyqyEb ss ′α′ ⋅α Δ (6.4/28) For regions where y yF and y < yF, the flexibility of the gear body has the following effect: () ()RR F F = = yyyqq yy (6.4/29) The flexibility qS due to the shear deformation is determined for the regions of the tooth where y < y F according to Equation (6.4/30) as follows: 2 3 S dycos () = + C'( )′ χα yqGA y (6.4/30) Ȥ shear distribution number ( Ȥ = 1.2 for rectangular cross section) G′ shear modulus for a board-shaped beam G′ = E/2(1 + ν) ν Poisson ratio A(y) tooth cross-sectional area at the point y of the face width section b The constant C3 is obtained from the boundary condition qs (y = 0) = 0 at the clamping point. For regions of the beam where y > yF, we obtain SF SFq(y y ) q(y y ) >= = (6.4/31) 232 6 Load Capacity and Running Performance of External and Internal Gearing Following the calculation of the flexibilities according to the position of the line of contact on the tooth, it is therefore possible to determine the helical tooth influence functions from the general influence functions. In this process, the bending level arms y or y F are derived from the meshing of the tooth flanks. In contrast to the bending deformation, shear deformation and deformation of the part of the gear body contiguous to the tooth, with which a strong interaction influence is present between the individual tooth sections, it may be assumed for the deformations due to Hertzian contact stress that only the respective tooth section subjected to load experiences a deformation. No interaction influence therefore needs to be considered, so that – in regard to these influences – the tooth can be regarded as consisting of decoupled slices of finite width and considered as single spur gears (tooth slices). The approach adopted by Weber and Banaschek can also be used here as a calculation method for the amounts of deformation. For pinion and wheel, respectively, the influence coefficients are calculated in analogy to Equation (6.4/22), whereas in this case, however, the influence function E = 1 is set (no interaction influence): ij Hj a( H ) q = (6.4/32) where qH is the flexibility ( H, Hertzian deformation) of a tooth slice of width ǻb (ǻb according to the meshing) at the point i as a result of the load at the point j. The Hertzian contact stress causes a flattening of the tooth surface along the line of contact and hence a deflection in the direction of the tooth load. The size of the flattening is dependent on the radii of curvature N1, N2 of the tooth flanks and the zones h1 and h 2 affected by the Hertzian contact stress. The quantities h1 and h2 are assumed from [6.4/20] as the distance from the contact point to the centre line of the tooth in the direction of the normal tooth force. Since for helical gears the line of contact runs diagonally across the tooth, these values change for every point in the line of contact. For helical gears, the conditions of the approximation resulting from the amount of Hertzian defor-mation can be represented by a model of two cones in contact with their tips facing away from each other. According to the division of the tooth into tooth slices, the cones are approximated in a cascading manner by cylindrical slices. The flexibility for a given tooth slice corresponding to the model is therefore calculated as per [6.4/20]: 2 H H2 ( 1 ) 2 ( ) () = l n ( ) 2 ( 1 )hy v vyqbE y vb −− Δπ − (6.4/33) The flattening half-width of Hertzian contact deformation bH(y) is obtained as 2 Hmax H4 ( ) (1 ) () = Eyybσ− νN (6.4/34) where 1211 1= +NNN, (6.4/35) that is, equal to the sum of the individual curvatures. The deformations due to Hertzian contact are not linear in nature but degressively load-dependent. By means of an initial value corresponding to a mean gear load of ı Hmax ≈ 800 N/mm2 for all of the tooth sections, the relationship between load and deformation can, as a first approximation, be linearised. An improvement of the load distribution calculated in this manner is achieved by an appropriate iteration. 6.4 Load Distribution on Gears 233 Notwithstanding this, many calculations show that an iteration calculation is only required in the event of extremely non-uniform load curves. Gear body deformations for rim gearings The rim gearings of external cylindrical gears differ from solid disc gearing in that the gear body is formed from web(s), wheel rim and hub (web gears). Savings in material and weight as well as the reduction in the moment of inertia are the reasons for such gear body designs. Nevertheless, it should be noted that the occasionally significantly higher gear body elasticity has an influence on the deformation and endurance behaviour. This influence must therefore be taken into account in the considerations on load distribution. For web gears, both the variable flexibility along the face width and an axial tilt resulting from the axial load of the helical gearing affect the load distribution, this being due to the fact that specific regions of the gearing can yield more than is the case with a comparable solid gear. For double-web gears, the axial tilt is less pronounced than for single-web gears because the deformation is hampered by the web coupling on the wheel rim. With regard to the load distribution, double-web gears exhibit behaviour more similar to solid gears. The interaction influence coefficients reflecting the gear body deformation for elastically designed rim gearings (single- or double-web designs) are taken from deformation calculations. There are various methods available to deal with such relatively complicated structures, which include spatial FEM and shell theory. Irrespective of the method, simplified models, which take into account the stiffening effect of the gear by increasing the gear rim thickness by a value of (1.5 m) while envisaging the transition between web(s) and hub as a lateral fixing, have proven their value. Deformation influence coefficients taking into account the gear body effect of a wheel rim gearing can be seen in Figure 6.4/5 in comparison to the corresponding solid disc gearing. Despite relatively simple models, the inclusion of finite element (or similar numeric) methods in the load distribution calculation are run-time intensive and require large volumes of data to be dealt with for the totality of the load components. For this r eason, approximate solutions are desirable in this regard. Figure 6.4/5 Deformation influence numbers for solid and elastically designed gear bodies Approximate solutions to include the gear body elasticity in the influence coefficients methods for external gears An analysis of the total deformation of the gear body in association with individual loads with the aim of deriving part deformations from simple calculation models initially provides the following principal deformation components : 234 6 Load Capacity and Running Performance of External and Internal Gearing a) From the individual radial load (Fr): This results in both a rim deformation and bending deformation of the web. The transition from rim to web can be regarded as a rigid ring, so for a one-sided relative rim load due to web deformation, the cross section of the rim is rotated around an axis perpendicular to the gear axis. The side of the rim subjected to load also undergoes deforma- tion in the manner of a pipe clamped at the web. The local radial compression of the web is insignificant in compa-rison to the other deformations. b) From the individual tangential load ( Ft): This results in a displacement of the cross section of the rim in the circumferential direction across the rigid ring. In addition, a warping of the web will also occur. The half of the rim subjected to load will also be twisted in the man- ner of a pipe clamped at the web. The tangential deformation of the web (torsion of a slice) is of an order of magni-tude, which is significant when determining the overall stiffness . c) From the individual load with a bending moment (bending load in the tooth root): The deformation of the rim as a result of a tooth bending moment basically occurs only on the half of the rim sub- jected to load (in contrast to the radial and tangential deformation). As the web only experiences an insignificant deformation from a tooth bending moment, the deformation of the rim can also be compared to the deformation of a pipe clamped on one side. These deformation components converted into the direction of engagement (components of an overall influence coefficient) for the normal tooth force Fn acting can be compared in Figure 6.4/6. d) From the individual axial load (refer to [6.5/115]): For helical gears, a deformation will occur in this direction under the influence of the axial load. Depending on the design, the gear body deformation for rim gearings will be greater than for solid gears, resulting in the face load dis- tribution being markedly influenced by these load components. As a model, the calculation of the web deformation as a result of axial load and thus of the displacement of the gear rim (including the gearing) can be based on a circular plate clamped at the hub with connecting ring (rim). Figure 6.4/6 Deformation components when loading a web gear with an individual load (gear rim thickness sR = 50 mm, width of rim bS = 80 mm, m = 20 mm, b = 400 mm, z = 60) The web will also experience a deformation due to a one-sided radial or circumferential load. These deformation components have a considerable influence on the load distribution – particularly in association with an extremely lopsided contact pattern – and cannot be neglected. Recording the gear body elasticity in a gear body influence coefficient The gear body geometry can be described by the following parameters: R mean wheel rim radius s R gear rim thickness b s width of rim b width of wheel d N hub diameter m n normal modulus of rim gearing For single-web gears, the determination of the crucial deformation components ensues in accordance with Table 6.4/1. If the deformation components are determined with a unit load and then converted into the direction of engagement and summed up, then we obtain the gear body influence coefficient a(i,j)RK,, which can then be superposed on the tooth deformation influence coefficients. As an approximation, the part deformations can also be initially determined via a load that is constant across the face width and taken into account as the line-of-contact deviation in the further calculation. Here a recalculation for a changed load distribution will also improve the result. 6.4 Load Distribution on Gears 235 Table 6.4/1 Approximate Calculation of Gear Body Deformation in Association with an Individual Load Deformation component from Calculation model Solution 1. Individual radial load (rim deformation) Clamped pipe Approximate solution after Kuptschik [6.4/19, 6.4/21] 2. Individual tangential load Clamped pipe Based on comparative calculations [6.4/19] 3. Individual axial load Circular plate clamped at the hub with connecting ring Approximate solution after Rao [6.4/22] 6.4.3.3 Calculating the Load Distribution and Analysing Special Features The recording of all the deformation influences, the determination of the line-of-contact deviation components and – not least of all – the solution of the comprehensive equation system require a computational processing of the method described. The method described to calculate the load distribution for spur gears and helical cylindrical gears is the basis of the calculation software [6.4/33], [6.4/34], mirroring the cur-rent state of technology in this field. (The examples listed here have been calculated with the program LVR [6.4/33].) From the solution of the contact problem derived using the equation system described here, we initially obtain the load (individual forces) along the line of contact for all meshing positions in the field of action (Figures 6.4/7 and 6.4/8) as well as the total deformation in the direction of engagement for each meshing position. Figure 6.4/7 Load distribution over the field of action, gearing with no line-of-contact deviation (z1 = 42, z2 = 61, mn = 3 mm, β = 15 , b = 60 mm, Fβy = 0 μm) FaFr 236 6 Load Capacity and Running Performance of External and Internal Gearing The total deformation is regarded as the rotational distance on the base circle, around which both wheels converge. The deformations can vary for each meshing position and result in a deviation from the ideal rota-tional distance ( transmission error ), which is zero for ideal involute nominal flanks with no deformations. Figure 6.4/8 Load distribution over the field of action, gearing with line-of-contact deviation ( z1 = 42, z2 = 61, mn = 3 mm, β = 15 , b = 60 mm, Fβy = 25 μm) After reference to the load, the total deformation itself is a measure of the flexibility (or overall stiffness) of the gear and – taking into account all of the mesh positions – provides the stiffness curve (Figure 6.4/9). Figure 6.4/9 Specific tooth stiffness for an uncorrected, normal spur gear (z1 = 42, z2 = 63, mn = 3 mm, β = 0 , b = 60 mm, Fβy = 0 μm) Premature tip edge contact and tip edge load bearing Due to the load, the tooth pairs engaged will deform (total deformation), and this results in a reduc- tion of the distance of the teeth by the amount of deformation of the teeth subjected to load before and after meshing. As the distance of the tooth tip from the mating flank is very small both before and shortly after the end of the theoretical meshing, this results in these teeth coming into contact (Figure 6.4/10). This will extend the length of the path of contact, which, in turn, will lead to a load-dependent increase in the contact ratio. For spur gears, this situation is manifested by an off-the-line-of-action contact of the entire tooth tip, while for helical gears it results in a tapering off of the line of contact along the tooth face at the tip. In the calculation method, the tip edge contact (or load-bearing tip edge) outside the length of the path of contact is obtained by determining the size of the ease-off present directly before the theoretical beginning of contact between the tooth tip and the mating flank as a tip relief in the calculation. The region of possible flank contact is increased beyond the limits of the theoretical field of action (extension of the length of the line of contact), and the equation system is accordingly expanded. The off-the-line-of-action contact as a result of the deformations of the tooth systems and gear body is effective even for spur gears and helical gears with a small helix angle. Pitch deviations also result in a variation of the length of contact due to their fluctuating magnitude over the gear circumference. 6.4 Load Distribution on Gears 237 a) b) Figure 6.4/10 Distance of the tooth tip from the mating flank outside the tooth meshing: a) line of action and path of contact AE; b) direct beginning of pre-meshing The effect of the additional tooth tip contact becomes particularly noticeable in the magnitude of the load and the transmission error. A significant reduction in the line load in the entire field of action is possible by extending the lines of contact. The greatest difference from the theoretical load occurs for gears with a large number of teeth and small helix angles. The transmission error is to a great extent dependent on the contact ratio. Integer values for the contact ratio are frequently striven for in order to keep the transmission error to a minimum. It is therefore important to note that deviation from the theoretical value will occur due to the load-dependent off-the-line-of-action contact. Figures 6.4/11a and 6.4/11b depict load distributions for gears with different helix angles. The bearing edge load tip regions, whose beginning is denoted by an arrow, are clearly visible (refer to Appendix 8). Figure 6.4/11 Effect of the helix angle on the tooth tip contact and the load distribution (z1 = 42, z2 = 61, mn = 3 mm, b = 50 mm, Fβy = 0 ȝ m) for a) β = 5 , b) β = 12 Edge influence In the designing of gears, the pinion is often made wider than the wheel to ensure the support on the entire width of the wheel in the case of axial offset as a result of assembly errors. Protruding gears are also commonly encountered with pinion shafts, where technical reasons necessitate an outlet of the gear cutting tool in the shaft. Such tooth ends protruding on the face stiffen the gearing in the edge region, thus affecting the load distribution. Calculations for a helical gear (Figures 6.4/12 and 6.4/13) show this effect. A comparison is made between a gear with pinion and wheel of equal width and a gear whose pinion is so wide that a length of one module protrudes on both the left and right sides. The stiffening effect results in a significant increase in load in the edge region. 238 6 Load Capacity and Running Performance of External and Internal Gearing For spur gears, this increase has to be determined on both sides. For helical gears, the influence of the acute or obtuse end face on the stiffness in the edge region is a further effect to be considered. For example, the acute end of a wheel tooth is so elastic at an edge of the field of action that the stiffening effect of the protruding pinion toothing is to all intents and purposes compensated for. On the other hand, however, the stiffening effect of the obtuse end of the tooth on the opposite side in-creases the peak load caused by the protruding pinion toothing. It should be noted that this effect is dependent on the respective direction of rotation (right or left flank engaged). For gears of equal width, the stiffness-redu-cing effect of the acute tooth end is predominant in the region of the face sides, compared to the stiffness-enhancing effect of the mating tooth, so a reduction in load ensues here for helical gears (Figure 6.4/12). As far as practical applications are concer-ned, the increases in load and thus stress described can be compensated for by tooth flank modifications. Figure 6.4/12 Load distribution of a helical gear pair where the pinion and wheel are of equal face width (z1 = 14, z2 = 65, mn = 4 mm, β = 30 , b = 80 mm, Fβy = 0 μm); the numerical value of the maximum line load specified in the figure applies to the evaluation range Figure 6.4/13 Load distribution of a helical gear pair where the pinion and wheel are not of equal face width ( z1 = 14, z2 = 65, mn = 4 mm, β = 30 , b1 = 100 mm, b2 = 80 mm) 6.4.4 Considering the Load Distribution in the Tooth Flank and Tooth Root Load Capacity Calculation by Means of Load Distribution Factors Consideration of the actual load conditions across the tooth flank due to bearing clearance, production-related deviations and deformations of the toothing and the elements surrounding the toothing, based on the methods described in Section 6.4.2, is particularly important when designing and assessing flank modifications (e.g., helix angle modification, width and height crowning) while also being the requirement for further investigations, such as on the influence of the gear body design and helix angle. 6.4 Load Distribution on Gears 239 In the common, mostly standardised methods [6.4/23] to [6.4/25] of calculating the load capacity of cylindrical gears, the irregularity of the load along the line of contact is taken into account by means of load distribution factors . These factors express the overload on the point along the line of contact subjected to the highest load (in comparison to the nominal load). The maximum value of the load is thereby obtained from the load distribution factor and the mean load F/b and F/L . To be precise, the maximum load occurring at a point is applied to the entire face width (or line of contact), where a different fracture probability applies. Strength values and factors are subsequently made use of in an attempt to reduce this overvaluation of the increase in load. Although load distribution factors can, of course, be used from the methods described above, approximate solutions corresponding to the character of the relationships to determine the tooth root and tooth flank stress would appear desirable. The following main assumptions then underlie the basic relationships reflecting the load- deformation relationships: • Adjacent tooth sections (both in the face width and the circumferential direction) have no influence on each other. • The specific tooth stiffness along the face width remains constant (for helical gears, the influence of the face side is taken into account by using a separate approximation, as appropriate). • The progression of the deviations across the face width ensues linearly. Figure 6.4/14 Important toothing deviations in the field of action In all of the standards, a differentiation is made between the transverse and face load distribution factors. Based on the simple rela- tionships for spur gears, in which pitch deviations generally cause a changed distri-bution of the overall load on the teeth engaged (transverse load distribution), while helix angle deviations (or deformations and manu-facturing deviations of the components adja-cent to the gearing acting in this manner) generally result in a non-uniform load along the line of contact (face load distribution), this distribution is adopted in the standards. This adoption ensues despite the fact that the relationships for helical gears are far more complicated. For helical gears, for example, a helix angle deviation causes not only a lopsided load but also a simultaneous transverse load distribu-tion different from the deviation-free meshing (gaps across the entire face width; Figure 6.4/14). By separating the resulting line-of-contact deviation into a constant component f ka and a helix component fkb, an exact distinction is made between the transverse and face load distributions, enabling the determination of the transverse and face factors as follows. 240 6 Load Capacity and Running Performance of External and Internal Gearing 6.4.4.1 Transverse Factor KHĮ The transverse factor K HĮ can be defined as the ratio of the mean specific load of a tooth pair j for meshing with deviation to the mean specific load for deviation-free meshing :1) mf HĮ m0( ) = ( )j qKj q (6.4/36) The tooth pairs engaged, zk, are considered in the field of action. The individual tooth pairs are denoted by the running index i and the tooth pair to be calculated by the index j. The tooth pair i moves in an error-free state along the line of contact l(i). The entire length of the line of contact L of the gear pair in a given meshing position is given by i=1kz = ( ) L li (6.4/37) The tooth flanks of the gear with deviations move across the entire length of the theoretical line of contact. Assuming a constant specific stiffness c ′, the mean specific tooth load of the tooth pair j is calculated using the deviation components as per Figure 6.4/9 from Equation (6.4/31) as follows: l( ) kb zk a mf () = 0() ( ) = ( ) ( ) d ( )() ( )ξ′ ƒ−−ξξƒƒ j jj cj jj j qlj l j (6.4/38) [ ]zk a k b mf ( ) = ( ) 0.5 ( ) ′−−ƒƒ ƒ jcj j q (6.4/39) Although it is also possible to consider the more general case, where the deviations reach magni- tudes of fk > fz, resulting in the flanks not touching across the entire line of contact, this, however, leads to significantly more complex relationships. In this regard, reference is made to the derivation possible with the introduction of further simplifications. For gears with f k > f z, these relationships can then be used to estimate the load ratios. The sum of the individual tooth pair loads must be equal to the sum of the total normal tooth force, as follows: kl( )z kb bn zk a i=1 () = 0( ) = ' ( ) ( ) d ( ) ( )i iiFci i iliξ ƒ−− ξ ξƒƒ (6.4/40) kz kb bn zk a i=1 ( ) = ' ( ) ( ) ( ) ( )2iFc l i i l i l iƒ −− ƒƒ (6.4/41) For faulty meshing, this results in the tooth deformation fz: kkzz zz 0 k a k b i=1 i=1( ) ( ) = + ( ) + 0.5 ( ) ƒƒ ƒ ƒ li liiiL L (6.4/42) and the tooth deformation for deviation-free meshing from bn z0 = ƒ′F cL (6.4/43) where c′is the specific tooth stiffness of a tooth pair in the individual field of action taken from Equation (6.2/2), Section 6.2.4. 1) Although this definition does not comply with ISO 6336-1 and DIN 3990-1, it nevertheless appears more suited to more precise analyses. 6.4 Load Distribution on Gears 241 For deviation-free meshing, the tooth load of this tooth pair is obtained from Equation (6.4/44): z mm 0 00( j ) = = c qq ′ƒ (6.4/44) After inserting Equations (6.4/39), (6.4/42) and (6.4/44) into output Equation (6.4/36), then following a few transformations, we obtain the final relationship in the form of Equation (6.4/45) [transverse factor KHĮ for the jth tooth pair, which is exactly KHĮ (j)]: Kz H ka kb ka kb i=1 z01 ( ) = 1 ( ) + 0.5 ( ) ( ) + 0.5 ( ) α +− ƒƒ ƒ ƒ ƒ liKi i j jL (6.4/45) 6.4.4.2 Face Load Factor KHȕ The face load factor K Hȕ is defined as the ratio of the maximum specific load of a tooth pair j for faulty meshing to the mean specific load for faulty meshing.1) This mean specific load is considered as the product of the mean specific load for deviation-free meshing and the transverse load distribution factor, as follows: maxf maxf Hȕ HĮ mf m 0 ( ) ( ) = = ( ) jj qqKjK qq (6.4/46) For meshing with tooth deviations, the maximum tooth load is calculated by assuming a constant specific stiffness c ′ and including the error components as per Figure 6.4/14 from the equation [ ]zk a maxf ( ) = ( ) ′ −ƒƒ j jc q (6.4/47) With the help of Equations (6.4/42), (6.4/44) and (6.4/45), it is thus possible to write the face load (distribution) factor as follows: []Kkb H z zk a k b k a 0 i=11 0.5 ( ) = 1 ( ) + ( ) + 0.5 ( ) ( )β− ƒ − − ƒƒ ƒ ƒ jKliii jL (6.4/48) A non-uniform load distribution along the line of contact also causes increases in local stresses in the region of the tooth root. The load increase on the flank does not affect the root stress in full size due to the support of the loaded region by unloaded or less loaded sections of the tooth. In line with the relationships of a plate carrier clamped on one side, a relationship dependent on the face width and tooth height is indicated, taking this context into account [6.4/23], [6.4/24], [6.4/25]. The face load factor for the calculation of the tooth root stress K Fȕ will (therefore) be less than the face load factor K Hȕ used for the contact stress. In contrast to this, however, the full effect of the increase in the mean load of a tooth pair resulting from the transverse load factor K HĮ is felt on the tooth root bending stress. The breakdown of the factors KĮ and K ȕ according to the type of deviation thus meet the understanding of this relationship. 1) Although this definition does not comply with ISO 6336-1, DIN 3990-1, it nevertheless appears more logical for detailed analysis. The differences are compensated for finally in the product of KHĮ and K Hβ. 242 6 Load Capacity and Running Performance of External and Internal Gearing The load distribution factors for the tooth root bending stress are calculated as per Equations (6.4/49) to (6.4/51): FĮ HĮ =K K (6.4/49) ()N Fȕ Hȕ = KK (6.4/50) () () 2 2 / = / + ( / ) + 1bhN bh bh (6.4/51) where b is the face width; h is the tooth height. When determining the load distribution factors, it should be noted that, for the contact stress and the tooth root stress, different mesh positions lead to the maximum stress. It has been possible to determine that the maximum flank pressure is often reached in the meshing position, in which the line of contact intersects the pitch circle cylinder on the face side , as well as the maximum tooth root stress in the meshing position, in which the line of contact intersects the tip circle cylinder on the face side . The tooth pair associated with this line of contact is thus simultaneously the vulnerable tooth pair j to be calculated. The load distribution factors are accordingly determined in these meshing positions. 6.4.4.3 Simplified Load Distribution Factors The specified factors K Į and Kȕ apply in this form both to spur gears and helical gears. The intro- duction of further simplifications arising at the transition from helical to spur gears and from consi-deration in the individual meshing region enables faster and more straightforward relationships to be derived. Since these also form the bases for the relationships specified in the standards [6.4/23], [6.4/24], [6.4/25] for load distribution factors, we shall look at them in more detail. For spur gears in the single meshing region , the following relationships apply: ka kb k = = 1; ( 1 ) = = ( 1 ) = 0 ; ( 1 ) = ij l b L ƒƒ ƒ (6.4/52) Thus for the transverse and face load factors of spur gears with contact over the entire face width, we obtain HĮ = 1 K (6.4/53) k Hȕ z0= 1 + K0.5 ƒ ƒ (6.4/54) Experience gained in the application of load distribution factors has resulted in a modification to the theoretically derived factor C = 0.5 used in ISO 6336-1 and DIN 3990-1. For practical calculations, the face load factor for the region 1 < KHβ < 2.5 can be determined from Equation (6.4/55). Additional external and internal dynamic loads, taken into account with the factors KA and K v (refer to Sections 6.3.3 and 6.3.4), amplify the tooth deformation and are therefore included in this relationship. The effective line-of-contact deviation F βy is used for the ease-off amount fk. 6.4 Load Distribution on Gears 243 Hȕ Av z 0 = 1 + yCFKKKfβ for KHβ ≤ 2.5 1)1) (6.4/55) where KA from Section 6.3.3 Kv from Section 6.3.4 fz0 from Equation (6.4/43) Fβy from Section 6.4.4.4 C • C = 0.5 according to ISO 6336-1 and DIN 3990-1 following this derivation. • As the load exaggeration due to KHȕ > 1 only affects part of the face width while the tooth stiffness falls away somewhat in the edge region, C = 0.4 can be set on an empirical basis. As the peak load (acting on the tooth end face) for a non-uniform face load distribution has a lower probability of reducing the lifetime than a uniformly distributed line load of equal value (pitting only occurring on the face sides is usually admissible), the influence of the peak load in addition to the effect of the running-in wear has been discounted. The effective line-of-contact deviation F ȕy is reduced by 20%. Both these relationships and the calculations for the load distribution factors specified in Sections 6.4.4.1 and 6.4.4.2 assume a linear load-deformation behaviour in the tooth mesh; that is, strictly speaking, they only apply to the ca se of complete tooth flank conta ct across the entire face width for the load under consideration ( K Hβ < 2; triangular or trapezoidal load distribution). If KHβ > 2, the deformation changes in a non-linear manner due to the contact width be < b. In this regard, the line- of-contact deviation is twice as large as the tooth deformation. Although this case ought to be excluded from practical investigations (check the construction!), the derivation with the simplifications provided from Equation (6.4/52) are specified for the sake of completeness: maxf z Hȕ z 0 0 = = mc qKq c′ƒ ′ ƒ (6.4/56) The total load of the tooth pair distributed across the partial width be (Figure 6.4/15) is given as follows: bn zk 0 = d be Fcbξ ′ − ξ ƒƒ (6.4/57) 2 e bn ezk = 2 bFc bb ′ − ƒƒ (6.4/58) From the geometric calculation from Figure 6.4/15, we obtain kz e = bbƒƒ (6.4/59) and with Equation (6.4/59) [from the transformation from Equation (6.4/58)] bn zk2 = F cbƒƒ′ (6.4/60) 1) In consequence, Equation (6.4/55) matches the corresponding equation in ISO 6336-1, DIN 3990-1 for C = 0.5. From a theoretically precise perspective, Equation (6.4/55) is only applicable when KHȕ ≤ 2; from a practical standpoint, KHȕ ≤ 2.5 is admissible. 244 6 Load Capacity and Running Performance of External and Internal Gearing we obtain k Hȕ z02 = Kƒ ƒ (6.4/61) If, in turn, additional external and internal dynamic loads are included and the effective line-of- contact deviation Fβy is set for fk, we obtain y Hȕ Av z04 = CFKKKβ ƒ for KHȕ ≥ 2 (6.4/62) If it is assumed that the two Equations (6.4/55) and (6.4/62) only deliver minor deviations of the re- sults in the transition region ( K Hβ ≈ 2) and gears with contact widths sig- nificantly less than the face width do not meet modern requirements, then an extension of the scope of Equation (6.4/55) to K Hβ ≤ 2.5 and thus the restriction for practical calculations to Equation (6.4/55) appears pos-sible. As a result of pitch deviations from a specific magnitude, both the contact stress in the tip and root meshing region and the bending stress in the tooth root region can assume greater values – despite a lower external load with double meshing – than that resulting from the total load in the single meshing region. In this process, the protruding tooth or tooth pair assumes a greater load than in the deviation-free situation, while in conjunction with a modified bending level arm or modified radii of curvature the result is therefore a greater tooth root or flank stress. For meshing with deviation, this results in the following load relationships in the double meshing region of spur gears according to Figure 6.4/16. Based on the assumptions made in Section 6.4.3, the load of tooth pairs I and II is given as kȕ I1 z = 2Fc bƒ ′ −ƒ (6.4/63) k II zk p = 2Fc bβƒ ′ −−ƒƒ (6.4/64) The total load is thus bn I I IFF F =+ (6.4/65) If a load application on the tooth tip is assumed in association with double meshing for the total load with deviation-free meshing [6.4/26] as an approximation, as in Figure 6.4/15 Tooth pair with line-of-contact deviation: a) without load, b) with load, KHβ > 2 6.4 Load Distribution on Gears 245 I bn 0 0.4F F ≈ (6.4/66) then, when an effective deviation of circular pitch occurs (the helix angle deviations primarily have no influence on the loading conditions or the transverse load distribution), the load on the tooth tip can be given as follows: () bn I kp 0.4 + ' Fc b F = ƒ (6.4/67) For meshing in the tip region, the stress should be checked with respect to the corresponding bending level arm and curvature radii compared to the single tooth meshing. For helical gears , it is possible to adopt the simple relationships relating to the face load factor in a similar manner. As the product of the transverse and face load factors is generally considered for the calculation of load capacity, an exact separation of the factors – as in Equations (6.4/36) and (6.4/46) – can be dispensed with as an approximation. The transverse load factor K Hα can then be added in each denominator alongside KA and Kv in Equations (6.4/55) and (6.4/62). If at the same time we replace the respective vul-nerable meshing position by the meshing position of the minimal length of the line of contact (L = L min), then Equation (6.4/55) or Equation (6.4.62) also applies to helical gears. Figure 6.4/16 Double meshing of a spur gear pairing The tooth deformation fz0 of the deviation-free gear can then be determined from Equation (6.4/68) as follows: ()t z0 2 = / cos εƒ ′ αF bZ c (6.4/68) where min2 = cos bLZε β (6.4/69) with Zεfrom Section 6.5.1.2, Equation (6.5/8a and b). With regard to the transverse load distribution, si mplifications of the above type for helical gears are more problematic. For larger overlap ratios, for example, a protruding tooth acquires the component forces of the other teeth simultaneously involved in the meshing for a deviation-free gear alone. The effects of deviations acting as a base pitch deviation are significantly greater for helical gears. In addition, helical gears are generally more sensitive to tooth deviations. The equations for the transverse factors, as specified in DIN or ISO [Equations (6.4/70) and (6.4/71)], are generally based on the same considerations as described for spur gears. 246 6 Load Capacity and Running Performance of External and Internal Gearing A distinction is first made: for 2γ≤ε kp F H t AvH = = 0.9 + 0.4 2 cK KFKK Kbα α β ƒ ε⋅ Ȗ Ȗ (6.4/70) 1) and > 2γε () Ȗ Ȗ kp FĮ HĮ t Ȗ AvH 1 = = 0.9 + 0.4 2 cK KFKKKbβ− ƒ ε⋅⋅ ε (6.4/71) 1) where cȖ is the mean specific stiffness of a gear pair from Equation (6.2/1), Section 6.2.4 and fkp is the effective deviation of circular pitch. A series of considerations has resulted in the reduction of the effects contained in the original relationships. These include the following reasons: • The sequence of the deviation (protruding, recessed) is not generally known unless measure- ments are available. • If the consideration is based on, say, gearing qualities in accordance with standards, then these details represent random parameters . For the known distribution type, they are characterised by the expected value and the standard deviation (half-tolerance component). The sum of the maxima of the deviations of the pinion and wheel is therefore not effective (for the given probability of an individual variation). • The ease-off amount of the pairing results from the values of the pinion and wheel. For gear ratios i ≠ 1, an unfavourable tooth pairing will generally only be repeated after a large number of revolutions. From a load perspective, this situation represents an internal load spectrum for the individual tooth pair, which would then, say, be included via a life-factor hypothesis in a calculation of load capacity. The effective deviation of circular pitch f kp can be approximated in accordance with ISO 6336-1 and DIN 3990-1 by the maximum value of the base pitch deviation f pe (pinion or wheel), reduced by the amount of running-in yα. See Table 6.5/27: for case-hardened, surface-hardened, and nitro- carburised gears, Į pe 0.075 yf= . 6.4.4.4 Determining the Line-of-Contact Deviation The line-of-contact deviation fk(i) is the gap on the tooth flanks measured in the plane of action in the normal section resulting from the change in position of the tooth flanks of the pinion and wheel from the normal position. The normal position is given by the line of contact of a deviation-free, unmodified toothing in the non-deformed state of the total gear not subjected to load. If the approach of determining the load distribution in accordance with Section 6.4.2 is continued, then it is only consistent to also include all of the other load-dependent influences as well as shaft and bearing deformations in the influence coefficients and all of the load-independent influences, such as manufacturing deviations in the gear, bearing clearance, as well as flank modifications in the line-of-contact deviation. 1) In contrast to the derivations made in Sections 6.4.3.1 to 6.4.4.3, where the ease-off amount is denoted by fk, the designation of the specific value of the effective line-of-contact deviation to be included in the load distribution factors is given by Fβy. We then have agreement with the designation in ISO 6336-1, DIN 3990-1. 6.4 Load Distribution on Gears 247 If a linear progression of the line-of-contact deviation is taken as a basis for approximation relationships (when considering flank modifications and shaft deformation for symmetrically lying sliced pinion toothings, further approximations are dispensed with) and one neglects the influence of the load distribution on the deformations of the gear components, then it is also possible and appropriate for considering the load-dependent factors to take these factors into account in the line-of-contact deviation. An effective line-of-contact deviation is created when a tooth pair is brought into contact at one point – at least – in the field of action as a result of gear tilting. Here, a reduction of the original line-of-contact deviation caused by a running-in of the gear in normal operation is taken into account in the effective line-of-contact deviation, as follows: F βy = Fβx yβ (6.4/72) (effective line-of (original line-of (amount of running-in) -contact deviation) -contact deviation) Reference values for running-in amounts y α (for pitch deviations) and yβ (for mesh misalignment) can be taken from Table 6.5/27. Therefore, Fβy1) is the line-of-con- tact deviation existing under load following the running-in. This takes into account all of the influences resulting from manufac-turing, assembly and deformation of the gear components, as well as the wear arising from running-in the gear. The following aspects should be considered in detail: the form and position deviations of the gear and the position deviations of the bearing bore ( f βZ), the bearing clear- ance or the bearing clearance differ-rences ( f βL), the elastic deformations of the gear components (shafts, bea-rings, housing) ( f βE), and the ablation due to wear yβ (Figure 6.4/17). Figure 6.4/17 Components of the line-of- contact deviation 1) In contrast to the derivations made in Sections 6.4.3.1 to 6.4.4.3, where the ease-off amount is denoted by fk, the designation of the specific value of the effective line-of-contact deviation to be included in the load distribution factors is given by Fβy. We then have agreement with the designation in ISO 6336-1, DIN 3990-1. 248 6 Load Capacity and Running Performance of External and Internal Gearing When determining the geometrically limited components ( fβZ, fβL), a distinction should be made as to whether the individual components are known as a value, or alternatively are to be considered as a random parameter . The value of the deviation is known if, say, the results of measurements are available for this special application case: y ZLE = | + + | y Fβ βββ β− ƒƒƒ (6.4/73) The deviation can be regarded as a random parameter when recourse is made to data from tolerance systems. In this case, the data should be superposed according to the rules of proba-bility theory. Since it is a question here of a large number of independent (or weakly depen- dent) random parameters, the sum can be assumed to be normally distributed. Its expected value Fβis equal to the sum of the expected values of the components () ifβ, and the spread is at least approximately equal to the sum of the spreads of the summands. The multiple of the standard deviation corresponding to half the tolerance is given by2 Zβƒ . Assuming that the individual components are statistically distributed and independent, we thus obtain the following relationship for the line-of-contact deviation for F βy as per Equation (6.4/74): yi iF y2 βββ β=ƒ + ƒ − (6.4/74) If the deviations are normally distributed and the limiting values or tolerances are selected such that they are not exceeded with the same probability, then – in compliance with this superposition law – it is possible to specify a maximum value of the sum, which is also not exceeded (or fallen short of) with this probability. An expected value Z0βƒ= is characteristic of the geometric deviations of the gear and the position deviation of the bearing bores under consideration here. The influence of the bearing clearance is composed of the mean value of the internal clearance (expected value −) and the tolerance (fluctuation ~). The deviations due to deformation (components from shaft, gear body, housing and elastic bearing deformation) are dependent on the load, whereby a minor influence of the bearing clearance on the bearing deformation also exists, which, however, can be neglected. The assumption of an initially constant external load requires that the fluctuation of the load-dependent components Eβƒ equals zero. For the majority of the cases, we thus obtain 22 yE L Z L yββ β β β β=ƒ + ƒ − + ƒ + ƒ F (6.4/75) Different methods should be used to determine the line-of-contact deviation, depending on the method chosen for calculating the load distribution. When calculating the load distribution per Equation System (6.4/7), the progression of the line-of-contact deviation can assume any (non-linear) form. For this case, the deviations of the tooth flanks (pinion and wheel) from the normal situation, measured in the plane of action in the normal section, must be added – or superposed if provided from standards in accordance with probability theory – in their full, signed value for each point i of the line of contact considered. If the intent is to estimate the influence of load-dependent deformations, such as shaft deformation and bearing deformation, on the load distribution (in this case, the progression of the load along the line of contact is meant and not the overload expressed as a coefficient), then it is possible to determine them in the form of a mean load and take account of them as a line-of-contact deviation (after an appropriate conversion). The result can be improved by means of an iteration analysis. 6.4 Load Distribution on Gears 249 When calculating load distribution factors as per Equations (6.4/45) and (6.4/48), the maximum value of the deviation is crucial (assuming a linear progression of the line-of-contact deviation in the considerations). This too can be determined eithe r from Equation (6.4/73) if measurement values are available, or from Equation (6.4/75) if tolerance ranges taken from standards are used. The line-of-contact deviation is to be determined for the meshing position to be calculated in each case. In principle, the deviations can be assigned according to Figure 6.4/17. Deviations due to deformation, which result from a tilting of the gear, such as that arising from shaft, bearing and housing defor-mation, can be regarded as mesh misalignments. If practical calculations are undertaken in accordance with the equations specified in Section 6.4.4, the line-of-contact deviations can be determined from the summary provided below. 6.4.4.5 Determining the Effective Line-of-Contact Deviation for Practical Calculations A summary intended to determine the deviations shall ensue here for the calculations taken from Section 6.4.4.3 and 6.4.4.4. The effective line-of-contact deviation F βy, resulting from manu- facturing deviations, deformations of the shafts, the bearings and the housing as well as from the bearing clearance, acts in the field of action perpendicular to the line of contact (refer to Figure 6.4/18). The determination of Fβy must ensue as per Equation (6.4/76) if measurement data is available for fβZ (index r, measurement value). The bearing clearance fβL is neglected in Equations (6.4/76) and (6.4/77): ȕy EZ = r F yββ β +− ƒƒ (6.4/76) and from Equation (6.4/77), if only tolerance values are available for fβZ: Figure 6.4/18 Effective line-of-contact deviation ȕy EZ = F yββ β +−ƒƒ (6.4/77) If no more precise analysis (calculation) is undertaken, the individual components can be approximately determined from the following equations: f βE from Equation (6.4/78) fβZ from Equation (6.4/82) fβZr from Equation (6.4/81) yβ ≤ yβmax from Table 6.5/27 a) Line-of-contact deviation due to elastic deformation fβE Under consideration of the relative size of the deformation components of the shafts, bearings and the housing of modern gear designs, only the bending deformation (B) of the gear shafts and the torsional deformation (T) of the gear bodies are taken into account in the following text in order to reduce the computational effort: f βE = f sh (6.4/78) fsh = f shB1 f shB2 f shT1 f shT2 (6.4/79) Assuming uniform load distribution, the torsional deformation component fshT1 of gear 1 and fshT2 of gear 2 is given by 250 6 Load Capacity and Running Performance of External and Internal Gearing 2 t shT 1,2 1,2 4 = ƒ π b F Gb d (6.4/80) The bending deformation component fshB1 of shaft 1 and f shB2 of shaft 2 are determined in the plane of action with suitable methods (bending line equation, Castigliano’s theorem). Approximate solutions are provided in [6.4/23], [6.4/25] for special gear layouts. When superposing the defor- mation components as per Equation (6.4/79), their sign is dependent on the relative direction of deformation, that is, from the position of the gears on the shafts as well as the position of the input and output (refer to Figure 6.4/21). The torsional deformation components have the same sign if the input and output are located on the same end face of the gear. b) Line-of-contact deviation due to manufacturing deviations fβZ If measurement values (index r) are available for the manufacturing deviations of the gears and the housing bores, f ȕZ is determined as follows: t t Zr H r 1 H r 2 r r b = cos sin cos ββ β β δ ++ α + ƒƒ ƒƒ ƒ β α bb BB (6.4/81) where B is the bearing distance ( B = LG, where LG is the reference length; see DIN 3960 and DIN 3964); r βƒ /B and r δƒ /B are the angular deviations of the gear axes in relation to the bearing distance. Remark: f Ȉį or fȈȕ possess positive signs if they increase the line-of-contact deviation caused by the positions of mesh misalignments. If only the tolerance values are known for the manufacturing deviations, fβZ is determined as follows: 2 2 22 tt ZH 1 H 2 b = + + cos + sin cos bb BBβ ββ β δ αα ⋅ ƒƒ ƒƒ ƒ β (6.4/82) 6.4.5 Methods to Improve the Face Load Behaviour The crucial parameter for the quality of the face load behaviour is the effective line-of-contact deviation achieved. Particular attention should therefore be paid to its reduction or compensation when designing and constructing gear drives. The scale and impact of the individual components making up the line-of-contact deviation (refer to Figure 6.4/19) on the face load factor are to a large degree dependent on the manufacturing quality and the gear design. The relationships depicted in, say, Figure 6.4/19 are obtained for the input and output stages of a series of two-stage cylindrical gear transmissions. The shaded region denotes the size fluctuation for the series of gears under consideration. Here the major influence can be seen of the components that are due to gear deviations and bore position deviations, which are dependent on the manufacturing quality. The components due to the bearing clearance and (particularly) the shaft deformation can also achieve values that are not negligible. Influencing the size and direction of the line-of-contact deviation (and thus the face load behaviour of the gear) will – together with ensuring a high-quality manufacturing process – in the main be possible by means of the following measures: 1. Constructive design such that the elastic (and, in borderline cases, thermal) deformations due to the bending of the shafts, torsion of the gear bodies and deformation of the bearings (and housing) either cancel each other out or superpose each other. 6.4 Load Distribution on Gears 251 2. Compensating for the mean values of the gear deviations and bore position deviations by adjustment by, say, using eccentric bushes or modified bearing clearance. 3. Applying flank modifications such that the effects of gear deviations and bore position devia- tions are thereby reduced (width crowning, topological modification). Also, as appropriate, helix angle modifications in a rotational direction to compensate for elastic deformations. Figure 6.4/19 Face load factor components; Man = (75 to 4800) Nm, a1 = (80 to 315) mm, a2 = (125 to 500) mm, iges = 12.5, β = 15 Discussion of key influences Geometric deviations Mesh position misalignments and bore position deviations are summarised under geometric deviations. It is often the case that the geometric deviations are not known with any degree of precision. Generally, only the admissible values stipulated by the tolerance standard are available for interpretation. The standardised specifications for the admissible total mesh position misalignments and bore position deviations are thereby the limiting values, which may not be exceeded with a specific probability. The values actually occurring are spread around the expected value of zero and can therefore result in a gap on the left or right side of the toothing. It is a similar situation with face wobbles. For gears designed with width crowning (C β) or with end relief (for CβI, CβII,, refer to Figure 8/39), it is possible to reduce the resulting skewed load distribution. Figure 6.4/20 depicts a load distribution for a gear designed with crowning calculated with a method described in Section 6.4.2. The reduction in the otherwise existing load peak can clearly be seen on the right face side (refer to [6.4/29]). Notwithstanding this, however, the design of such modified gears is not without problems. A too- small crowning or end relief selected results in a too-low load reduction on the tooth end face, while a too-large modification – although reducing these load peaks – unfortunately results in a strong increase in load in the tooth centre, such that lower load capacity values (compared to the unmodified gear) may eventually occur. It should also be noted that – due to the statistical distribution of the geometric deviations – it is not possible to determine the modification in accordance with the limiting value (standard value) of the line-of-contact deviation because this will result in a deterioration of the load capacity for the lower deviation occurring corresponding to a normal distribution. 252 6 Load Capacity and Running Performance of External and Internal Gearing A large number of calculations have revealed that the optimal crowning for normal gears can be roughly assumed at approximately 30% to 50% of the maximum effective line-of-contact deviation [6.4/27]. Notwithstanding this, however, it must be emphasised that a uniform flank load bearing is never possible to achieve completely and that a local load is always greater than that associated with deviation-free gears. Figure 6.4/20 Load distribution in the field of action, gear with line-of-contact deviation, depletion of part of the load increase due to width crowning Cβ (z1 = 42, z2 = 61, mn = 3 mm, β = 15 , b = 60 mm, Cβ = 10 ȝm, Fȕy = 25 ȝm) Shaft deformation, bearing displacement A significant component of the line-of-contact deviation is caused by the elastic shaft or pinion deviation as a result of the bending and torsional stress. The shaft geometry with all its shoulders and the stiffening effect (support effects) of the shaft-hub connection is critical for calculating the bending deformation. The torsional deformation in the face width region is interesting to note, particularly for pinion shafts. In general terms, it can only be neglected for face width/diameter ratios where b/d < 0.6. If the diameter of the wheel is markedly greater than that of the pinion ( i > 2.5), the torsional deformations for the wheel are generally insignificant. Whether the bending deformation of the shafts and the torsion of the gear body are added together or at least partly cancel each other out is dependent on the constructional design. Figure 6.4/21 depicts two transmission schemes. In the first, (a), the bending deformation and torsional defor-mation are added together, while in the second, (b), the bending and torsional deformation from pinion and wheel are mutually subtracted. Following a precise analysis of the torque flow and the layout of the gears, conclusions can be drawn on optimising the design. On the other hand, the individual proportions of the deformation can be minimised by applying specific restrictions, such as specifying a maximum face width/diameter ratio or large shaft diameters. The bearing displacement also has a significant effect on the resulting line-of-contact deviation of the gear. The line-of-contact deviation arises from the elastic bearing deviation and the residual bearing clearance of the roller bearings following assembly. Of these components, only the differ-rence of both the bearings of a shaft with double-sided bearings are significant. With regard to the influence considered here, roller bearings, which manage with a low bearing clearance while making effective use of the lifetime and, in particular, possess a low bearing clear-ance tolerance, are initially desirable. 6.4 Load Distribution on Gears 253 The values obtained for the displacement of the shaft due to bearing clearance or the differences in bearing clearance are generally significantly greater than the values of the elastic bearing deformation (Figure 6.4/19). The effects of the differences in bearing clearance on the load distri- bution can lie in the same range as those caused by the shaft deviation. This effect increases proportionally with the widening in shaft diameter due to the increase in bearing clearance. The layout of the roller bearings in the gears is a non-trivial factor in relation to the effect of the bearing clearance on the face load factor. In addition to the fundamental decisions, such as choice of bearing and bearing clearance, the effects resulting both from mounting the roller bearing as a fixed, loose or support bearing and from determining the bearing distance should also be included in the considerations of the design. Figure 6.4/21 Superposition of the components of line-of-contact deviation arising from bending (B) and torsional deformation (T) of the gear shafts Although increasing the bearing distance has a favourable effect on the load distribution, it should nevertheless be noted that increasing the shaft length is associated with an increase in shaft defor-mation, while an off-centre design of the gear will, in turn, have a negative effect on the load distri-bution. It is therefore necessary to balance these opposing tendencies. Bearing centring may be another cause of a relatively major influence on the bearing clearance. For ball bearings or spherical roller bearings, for example, this can result in the inner ring shifting up to one centric position towards the external ring. This means that the shaft is lifted by an amount of approximately one-half of the bearing clearance in the region of this centring position, resulting in a tilting of the shaft (Figure 6.4/22a, b). It is possible to minimise this influence by an appropriate arrangement of the roller bearings (initially disregarding functional restrictions). Minor width factors (i.e., favourable load distribu-tions) are achievable with a support bearing arrangement or fixed-floating bearing arrangement with a diagonally opposed fixed bearing (Figure 6.4/22d). Similar considerations can be applied with a so-called “flying bearing” of the gear (Figure 6.4/22c, e). Nevertheless, such bearing centring, the effect of which is deliberately used here, only occurs in association with a high axial load, due, for example, to large-scale helix angles of the gear. 254 6 Load Capacity and Running Performance of External and Internal Gearing Figure 6.4/22 Bearing arrangement in relation to the face load distribution: a, e) unfavourable; b, d) very favourable; c) very unfavourable Figure 6.4/23 Elastically designed shaft seat Figure 6.4/24 Elastically designed housing seat The load ratios 1.5 Fa/Fr > e must exist whereby e is the bearing constant dependent on the contact angle α [6.4/28]. Depending on the design parameters, the geometric relationships and the load, the bearing deformation and the opposing line-of-contact deviation thus arising can also result in an improvement of the contact pattern. This fact can be exploited by designing particularly elastic roller bearings or bearing cups or bearing brackets. It is also possible to imagine designs such as those depicted in Figures 6.4/23 and 6.4/24. In one of them the shaft seat was elastically designed, and in the other the housing seat. Influences due to the gear body design For larger transmissions, gears are no longer designed with full gear bodies for reasons of weight reduction and from a production-related perspective. Typical forms are the single- and double-web gears, consisting of hub, web(s) and wheel rim. Gear bodies so designed can significantly reduce the effect resulting from line-of-contact deviation. As far as single-web gears are concerned, the greater degree of elasticity of the face-side regions of the gear can have quite an impact on increasing the load-bearing capacity with regard to the tooth flank. For helical gears, the web stress and thus the tilting of the gear due to the axial force should be noted. Double-web gears are less sensitive in this regard, albeit nullifying the benefits arising from the elastic tooth end face. The use of non-parallel supporting plates (webs) represents a possible compromise between both cases. Figure 6.4/25 depicts the load characteristics for various line-of-contact deviations across the face width. Load peaks in the area of the tooth end face are significantly lowered, and a markedly more uniform load distribution is achieved. 6.4 Load Distribution on Gears 255 Figure 6.4/25 Face load distribution for a single-web gear, spur gearing Web gears, however, also exhibit a different behaviour than the cor-responding solid gear with regard to thermal deformation and centri-fugal stress. In addition, the gear stiffness crucial for the resonance speed of the parametric excited vibrations is also reduced. At high speeds ( /t > 100 m/s) and a high degree of elastic web-gear design, the centrifugal force acts as a cause of non-constant defor-mations across the face width. They then reach values lying in the magnitude of the static defor-mation with an operating load. Line-of-contact deviations also occur as a result of thermal deformations of the pinion and wheel due to unequal temperature distribution along the face width. In the transmission of high power levels with helical gears, temperature differences due to the transport of heated lubricant across the face width (trapped oil flow) are reached, which may result in significant deformations. For web gears, it should be noted that a significantly different thermal deformation across the face width than is the case with solid gears will occur due to differing heat dissipation in the various parts of the gear body. Design measures, such as variable oil injection and radial gaps in the face width, have to be adjusted to take this into account. Following an in-depth analysis of the force and deformation relationships of all the crucial influences, flank line modifications (width modifications) on the pinion and wheel can be under-taken to compensate for the deformations. Theoretically, it is possible for relief to be applied to the tooth flank in the face width direction such that the line-of-contact deviation and the ground modification cancel each other out under load. This would then achieve a uniform load distribution across the face width. In practice, however, the following restrictions should be mentioned (as a minimum): 1. For ideal flank line modification, special manufacturing technology requirements are required so that helix angle corrections with superposed crowning or end relief are often undertaken to achieve an approximation of the modification necessary. 2. With a non-constant load, the flank line modification required to compensate between load- independent and load-dependent components is different for each load step, and this necessitates, in turn, that statistical or lifetime investigations be included when selecting the modification to be undertaken. Taking the interaction and the complexity of the various components into consideration, the following approach would therefore be desirable: 1. The use of construction measures to influence the load-dependent deformations in such a way that they – at least partially – cancel each other out. 2. Compensating for the remaining component of line-of-contact deviation from the load- dependent, thermal and centrifugal force-dependent deformations by means of a flank line modification (helix angle modification) or by, say, adjusting eccentric bushes on the bearings or changing the bearing clearance. 256 6 Load Capacity and Running Performance of External and Internal Gearing 3. A crowning (superposed, if necessary, on the helix angle modification) to keep the line-of- contact deviation fluctuating around the expected value of zero (caused by the manufacturing- related deviations of the gear) and the deviations due to the wobbling of the wheels to a mini-mum. The theoretical minimum of the effective deviations ultimately represents the remaining part of the wobble deviation. 6.4.6 Symbols and Symbol Explanations of Section 6.4 A mm2 area A - constant B mm bearing distance C - factor C ȝm flank modification E MPa Young’s modulus E - decay function F N force Fȕy ȝm effective contact line deviation G MPa shear modulus I mm4 area moment of inertia KA - application factor Kv - dynamic factor KHĮ, KFĮ - transverse load factor KHȕ, KFȕ - face load factor L mm total length of contact lines LR mm distance R mm radius Wf - correction function for face side slant Zİ - overlap factor a ȝm/N deflection influence number b mm tooth face width bH mm half-width of flattening br mm equivalent spur gear face width bS mm web thickness c′ N/(mm ȝm) specific tooth pair stiffness (in single meshing region) c N/(mm ȝm) mean specific tooth stiffness d mm diameter dN mm hub diameter dsh mm mean shaft diameter e - basis of natural logarithm e ȝm/N deflection influence number fG - size function fHȕ ȝm helix slope deviation fL - decay function fpb m base pitch deviation fsh ȝm component of helix slope deviation due to shaft deformation fk ȝm contact line deviation, separation fz ȝm tooth deflection fz0 ȝm tooth deflection in the case of meshing without tooth deviations fȕ,F ȝm helix deviation fȈį ȝm axial tilt deviation fȈȕ ȝm axial skew deviation g ȝm residual separation of a tooth pair h mm depth of Hertzian deflection zone h mm tooth height l mm length of contact line m mm module pe mm base pitch q N/mm line load, face load distribution q ȝm/N compliance s mm tooth thickness s mm length coordinate sF mm tooth thickness at root sR mm rim thickness xK mm distance from point of load application x\* K - relative distance from point of load application xV mm distance of analyzed point of deflection from tooth border x\* V - relative distance of analyzed point of deflection from tooth border y mm level arm y ȝm amount of wear yβ - helix factor α pressure angle, angle of load application β helix angle βSt face border angle J - total overlap η ȝm deformation of bending beam 3 - shear deviation number - Poisson’s ratio ! mm length coordinate N, ȡ mm curvature radius σ N/mm2 stress σH N/mm2 Hertzian pressure 6.5 Analysis of Load Capacity 257 Indices: I pinion tooth II wheel tooth E elastic deflection H Hertzian pressure L bearing R gear body in toothed range RK gear body S shear Z gearing a constant b helical c lead crowning e end relief m mean n normal section p pitch r measured t transverse section 6.5 Analysis of Load Capacity Analyses of pitting load capacity and of tooth root capacity are explained at the beginning of this section. They are to be viewed as basic analyses of load capacity. The practical procedure for this has been summarised methodically and in table form in a specific section. The remarks which follow refer to analyses of load capacity, with the focus on tribotechnical basics. These can be just as important. However, as yet, their basic principles have been less developed. As materials, steel, cast iron and cast steel are required. For gears made of thermoplastic materials, VDI 2736 [6.5/105] contains the calculation methods and endurance limits. Typical for this are wear, heating, scuffing, as well as time and temperature-dependent strength. Principally, in more precise analyses, the strain on the entire tooth profile must be considered (three-dimensional tooth deformation, deterioration or formati on of cracks deep inside caused by Hertzian strain and bending, shear and pressure load). 6.5.1 Calculation of Surface Durability – Pitting and Deterioration at Maximum Load 6.5.1.1 Basics The subject of the following considerations is the calculation and assessment of strain in the flat- tening region and under the surface of the tooth flanks transferring the tooth force. Figure 6.5/1 Radii of curvature of special points on the line of action; gear 1 driving: a) start of meshing point (A) at root of gear 1 (start of meshing point at tip of gear 2); b) inner point of single tooth contact (B) gear 1 (outer point of single tooth contact gear 2); c) pitch point (C); d) outer point of single tooth contact (D) gear 1 (inner point of single tooth contact gear 2); e) end of meshing point (E) at tip of gear 1 (end of meshing point at root of gear 2) 258 6 Load Capacity and Running Performance of External and Internal Gearing Figure 6.5/2 Model to calculate contact stress: a) rolling model; b) radii of curvature 6.5 Analysis of load capacity 259 The tooth profiles can be approximated by their circles of curvature (refer to construction of the involute, Section 2.1 and Figure 2.1/4). Figure 6.5/1 shows this for cylindrical gears with external teeth. Along the line of action, the radii of curvature have different sizes. It is possible to calculate strain by replacing the tooth flanks by a cylinder pair at their respective contact point, on the basis of their radii of curvature. The normal stress perpendicular to the flattening produced by the normal tooth load F n represents the main quantity for assessing strain. Usually the maximum stress perpendicular to the flattening region is calculated, and this quantity is called Hertzian contact stress and labelled with pH or ıH (Figure 6.5/2). Its calculation in the case of arbitrarily curved bodies, whose main axes can be meshed against one a nother at an arbitrary angle, is given among others in Dubbel [6.5/1]. For two rolls in particular, as a result of the normal load Fn (Figure 6.5/2), the Hertzian contact stress pH results in ()n H 221FEpl=π − νN (6.5/1) Here the following applies: 12111=+NNN (6.5/1a) E EE12 112 12 122 2−= −+− ννν (6.5/1b) where Ȟ is Poisson’s ratio, E elasticity of material Note: For gearing in particular, Hertzian contact stress is labelled with the symbol ıH (refer to Section 6.5.1.2). At the centre of the cylinder at the middle of the pressure-loaded area, the stresses are ([6.5/1] and [6.5/4]) xH yH z H ;; 2 σ= − σ= − σ= − ν p pp (6.5/2) The largest shear stress is produced in the depths x = 0.78 bH in max H 0.3p τ= (6.5/2a) and according to von Mises’ maximum distortion energy theory, the equivalent stress is vGmax H 0.56p σ= at the depth of x = 0.71 bH (6.5/2b) and according to the shear stress hypothesis vGmax H 0.6p σ= at the depth of x = 0.78 bH (6.5/2c) The width of Hertzian contact deformation (2bH) is 2 n H32 ( 1 )2FbEl−ν=πN (6.5/3) The variable 2 bH can also be expressed as dependent on the Hertzian contact stress. For that the result is 2 HH128bpE−ν= N (6.5/3a) Section 7.4 contains prepared equations for the depth of the maximum of equivalent stress [Equations (7.4/15), (7.4/16)]. 260 6 Load Capacity and Running Performance of External and Internal Gearing It is also possible to calculate the components of stress along the depth coordinate x and to ascertain an equivalent stress according to von Mises’ maximum distortion energy theory. When one super- imposes the part resulting from externally applied shear stress (T ), for example resulting from sliding movement, on the stresses resulting from normal load ( N), the result is (refer to Figure 6.5/3) ()xx , Nx , T yy , Ny , T zx y xy xy,N xy,Tσ= σ + σ σ= σ + σ σ= ν σ + σ τ= τ + τ (6.5/4) Here the stress components - xz and -yz are zero. Table 6.5/1 contains a group of the equations (according to [6.5/2]) for the individual parts, in the calculation of which the integral must be solved, for example, numerically (for this refer to Figure 6.5/4). Table 6.5/1 Stress Components along the Coordinates x and y (Figure 6.5/4) ()[] () ()[] () ()[]σπ σπ τπx,N H H y,N H H xy,N H HHH HH HHdy dy dy=− − ⋅ +− =− − ⋅ +− =− − ⋅ +−−∗ ∗∗ −∗ ∗ ∗∗ −∗ ∗ ∗∗ 21 21 2123 222 2 2 222 2 2 222py bx xy y py bx xy y py bx xy ybb bb bby-y y-y () ()[] () ()[] () ()[]σπμ σπμ τπμx,T H H y,T H H3 xy,T H H2HH HH HHdy dy dy=− − ⋅ +− =− − ⋅ +− =− − ⋅ +−−∗ ∗ ∗∗ −∗ ∗ ∗∗ −∗ ∗ ∗∗ 21 21 212 2 222 2 222 2 222py bx xy y py bxy y py bx xy ybb bb bby-y y-y y-y The equivalent stress according to von Mises’ maximum distortion energy theory is () () () ( )22 2 2 vx y y z x x y132σ= σ− σ +σ− σ +σ− σ +τz (6.5/5) 6.5 Analysis of load capacity 261 Figure 6.5/3 shows the trace of the equivalent stress corresponding to Hertzian contact stress, along the depth coordinate. It can be seen that the strain maximum moves closer to the surface, due to the friction (shear stress applied to the pressure-loaded area). Figure 6.5/3 Stress sequence (equivalent stress according to von Mises’ maximum distortion energy theory, corresponding to the Hertzian contact stress) during contact of cylindrical rolls middle of the flattening region, perpendicular to the surface in the epths; x coordinate, b H half-width of Hertzian contact deformation) [6.5/2] Figure 6.5/4 Definition of quantities for calculating stresses in the half space, according to Table 6.5/1 The model on which the Hertzian theory is based does not fully correspond to reality. Between the tooth flanks of gears lubricated with liquid, an oil film form, which causes an alteration in the pressure distribution and the lubrication film curve (refer among others to [6.5/100]). However, the pressure distribution deve-loping under hydroelastic conditions does not significantly deviate from the distri-bution in cases purely of contact between solids (Hertzian contact stress distribu- tion). Figure 6.5/5 shows the typical trace of normal pressure under hydroelastic conditions. Roughness has both an influence on the effective curvature and an impact in the sense of notches. Even more strains are superimposed on the ones already mentioned. When teeth mesh, high-temperature strain occurs locally because of the sliding. This “temperature flash” causes an occurrence of thermal stress (for approaches, refer to [6.5/100]), which leads to additional strain. Owing to the different sliding paths, it differs in size for the different gearing of the pairing, and despite the equality of the Hertzian contact stress, with superimposition of all parts it works in a way that causes the pitting damage to occur mainly on one gear, for example on the root of the driving gear (in the negative slip region). Before rolling over a certain zone, there was a rolling over already at this point, likewise with damaging bending stress. However, the question of where the cracks originated cannot be clarified solely based on the calculated stresses. Often in the areas close to the surface the strength is different than in the depths. This can result from machining (internal stresses; local tempering effect) or it can result from a surface treatment, hardening, or tempering. So, in the case of large, surface-tempered gears in particular, caution is recommended concerning the assumption made earlier on, that the cracks forming pits always originate on the surface. 262 6 Load Capacity and Running Performance of External and Internal Gearing Figure 6.5/5 Principal trace of contact stress purely for contact between solids and under hydroelastic conditions From the preceding considerations it becomes evident that the Hertzian contact stress represents merely a relative and rough quantity. It must not be forgotten that it represents only one component of the stresses in effect as a whole, and so it cannot be directly compared with the tensile/com-pressive pulsating fatigue strength. Appendix 7 contains a more extensive look at the strain depth profile . However, at the present time for this specific use case, it is not possible to see in it the only path for further development. Generally there is still a need for a considerable amount of research to be done in the area of pitting load capacity. 6.5.1.2 Tooth Flank Pressure a) Introduction of the gearing quantities The relationship to Hertzian contact stress given in Equation (6.5/1) shall now be refined for application by introducing the special geometric quantities for gearing. The deduction will be done using external gearing. It also applies to internal gearing, in which case the diameter and number of teeth would be negative. Here the following results for a normal tooth load , which acts perpendicular to the tooth flank in the normal section (Figure 6.3/1): FFF nb nt t==cos cosαβb (6.5/6) Corresponding to the definition 1211 1=+NN N (6.5/7) with the quantities for the pitch point, the equivalent radius of curvature N is CC 1C 2 CC 111 1 11 1;1u =+ = + NNN NN (6.5/7a) Thereby the following is true (Figure 6.5/2b): 1, 2 C1, 2 t btancos2c o sd α=⋅ αβNwt (6.5/7b) 6.5 Analysis of Load Capacity 263 It is to be converted as follows if the equivalent radius of curvature is to be ascertained for a different point: yy yC CC,g i v e nfor =NNNNNN With NC1 + NC2 = Ny1 + Ny2 = constant (Figure 6.5/1), the result is y1 y2 yC C1 C2=NNNNNN (6.5/7c) The following is true particularly for the internal point of single tooth contact of the wheel 1 (pinion): B1 2 BC C 2 C1 C2 B1 Z== ⋅NNNN NNNB (6.5/7d) with 22 wt B1 1 b1 B2 B1 bb11 s in;2c os c osadd pα =− − = − ββ NN Nae t The variable ZB expresses the difference in contact stress at meshing point B in reference to the pitch point C due to the change in profile curvature. The following is defined for the effective contact tooth length: lb Z=cosβε b2 (6.5/8) Here Zİ2 takes into account the part not yet considered by b/cosȕb. It is common to use the following approximations (ISO 6336-2; DIN 3990-2): 1for 1 Zεβ α=ε ≥ε (6.5/8a) (Strictly speaking, Equation (6.5/8a) applies for İȕ ĺ or an integer İȕ.) () () 41for 13Zαβ β εβ α−ε −ε ε=+ ε <ε (6.5/8b) In the case of İȕ = 0 (spur gearing), according to Equation (6.5/8b) the result is a value Zİ 1, even though in the area of single tooth contact only one tooth pair is meshing. Using Zİ dependent on İĮ, this case takes into account that the danger (probability) of pitting is lower in cases of larger transverse contact ratio (i.e., smaller area of single tooth contact). Material elasticity quantities (1 – Ȟ2)/E must be determined dependent on the actual values of the pairing; Equation (6.5/1b) specifies the correlation for this. b) Summary of similar quantities and conversion for practical calculation Together with the Equations (6.5/1b) and (6.5/6) to (6.5/8b), Equation (6.5/1) results in the relationship (6.5/9) for the Hertzian contact stress. For the influence of the helix angle, which has not yet been fully considered, the factor Z ȕ is added. The calculation at the points of single tooth contact B or D is done by converting the contact stress at the pitch point on the points of single tooth contact B, D with the help of the factors ZB or ZD: pZ Z Z Z ZF bdu uHH B ,Dt=+ E εβ 11 (6.5/9) 264 6 Load Capacity and Running Performance of External and Internal Gearing Here ZE, ZH, ZB,D and Z İ express the fol- lowing: • Elasticity factor Z EEE= − +− 1 1112 122 2πνν (6.5/9a) For steel with E1 = E 2 = 2.1 105 N/mm2 and Ȟ1 = Ȟ2 = 0.3 the following is true: 2 E191.6 N / mm Z= (also refer to Table 6.5/2). • Zone factor ZH tb wt cos tan=⋅ 1 2 αβ αcos (6.5/9b) For spur gearing and a sum of the profile shift equal to zero the following is true: (for Įt = Įwt = 20 ) ZH = 2.495. Figure 6.5/6 Zone factor ZH (for Įn = 20 ) • Single mesh factor Z B or ZD C1 C2 C1 C2 BD B1 B2 D1 D 2. Zr esp Z ==NN NN NN NN (6.5/9c) In cases where the paired gears are of the same strength, the larger of the values ZB or Z D, respectively, is to be used. In normal cases, the internal meshing point of the pinion is decisive. In the case of pairings of different materials, for expediency one utilises the factor belonging to each internal point of single tooth contact and carries out the analysis of load capacity of the flank separately for both wheels with the respective ı Hlim value. After inserting the specification factors, the result is () () ()wt B 22 a1 b1 1 a2 b2 2tan /1 2 / /1 1 2 /Z dd z dd zαα= −−π ⋅ −− ε− π (6.5/9d) In cases of gear ratios in the vicinity of 1 and cases of extreme profile shift or paired gears with different strengths, the single mesh factor of the wheel ZD can also be decisive: () () ()wt D 22 a2 b2 2 a1 b 1 1tan /1 2 / /1 1 2 /Z dd z dd zαα= −−π ⋅ −− ε− π (6.5/9e) 6.5 Analysis of Load Capacity 265 • Overlap coefficient Zİ, refer to Equations (6.5/8a) and (6.5/8b) Figure 6.5/7 Overlap coefficient Zİ • Helix factor 1 cosZβ=β (6.5/9f) Note: Until now cos Zβ=β was stipulated. New results and experience prompted the proposal of the factor Z ȕ according to Equation (6.5/9f) for the further processing of ISO/DIN. This is generally an improvement. The influence of the helix angle ȕ on the load capacity is registered to some extent by the factors ZH and Zİ. A residual amount of influence has been allocated to another factor. It is labelled with the symbol Zȕ. With the introduction of this helix factor Z ȕ and the naming of the Hertzian contact stress with the stress symbol ı H or ıH0, for the resulting basic value (index 0) of contact stress, without any additional loads (vibrations, change to load distribution), the following applies: t EHB , D 11 FuZZZ ZZbd uΗ0 ε β+σ= ⋅ (6.5/10) By definition, the basic value for contact stress ıH0 does not yet take into account the rise in strain due to additional loads produced by vibrations and likewise does not take into account the rise in contact stress due to the change in the load distribution caused by deviations in the gearing, for example. If the influences from the change in load distri bution and dynamic additional loads are registered in the factor K H, for the existing contact stress the following results: HH 0H K σ= σ (6.5/11a) 266 6 Load Capacity and Running Performance of External and Internal Gearing A breakdown of the overload factor KH produces the transverse load factor ( KHĮ) for the load distribution of the tooth pairs currently meshing, the face load distribution ( KHȕ), the external additional dynamic loads ( KA), and the internal additional dynamic loads ( KHv). These factors are to some extent interdependent. The equations given for their separate calculations represent an initial approximation, which is normally accepted as sufficient. The following applies: KK K K KHA H v H H=αβ (6.5/11b) If one introduces these quantities, the following results for contact stress: t HE H B , D A H v H H 11 FuZZZ ZZ KK K Kbd uεβ α β+σ= ⋅ ⋅ ⋅ (6.5/11c) Here the following applies: ZE Moduli of elasticity, Equation (6.5/9a), Table 6.5/2 ( ZE accounts for the E moduli and Poisson’s ratios) ZH Zone factor, Equation (6.5/9b), Figure 6.5/6 ( ZH comprises the quantities accounting for the pressure angle and helix angle to calculate contact stress at the pitch point) ZB,D Single pair mesh factor, Equations (6.5/9d) and (6.5/9e) ( ZB,D converts the contact stress from pitch point C on the interior point of single tooth contact B of the pinion or D of the wheel) Zİ Overlap coefficient, Equations (6.5/8a) and (6.5/8b), Figure 6.5/7 (Z İ accounts for the difference between the effective total contact length and b/cosȕb) Z Helix factor, Equation (6.5/9f) ( Zȕ accounts for the influence of the helix angle, which is not yet fully considered in ZH and Zİ) KA Application factor, Section 6.3.3 ( KA accounts for the additional external dynamic loads not yet contained in Ft) KHv Dynamic factor, Section 6.3.4 ( KHv accounts for the effect of additional loads on the contact stress due to vibrations, which are excited within the gear unit, e.g., by tooth stiffness that changes with the rotating angle and tooth deviations) KHĮ Transverse factor for contact stress, Section 6.4 ( KHĮ accounts for the influence of the change to load distribution on multiple teeth due to tooth deviations and deformations) KHȕ Face factor for contact stress, Section 6.4 ( KHȕ accounts for the influence of the change to load dis- tribution in the axial direction of the tooth on the contact stress due to geometric deviations and de- formations) Ft Nominal circular loads (tangential loads) on the reference circle, Equation (6.3/1) b (Common) face width d1 Reference diameter, gear 1 u Gear ratio ( u = z2 /z1) Note: In the case of the internal gearing/external gearing pairing, u < 0 (negative!). In the case of load spectra it is advantageous to calculate the contact stress ıHj for each load step j with KAj, KHvj, KHĮj, KHȕj, Ftj, and to determine an equivalent contact stress, according to Equation (5.3/11). To simplify, an equivalent load Ft is often calculated [Equation (5.3/15)]. With that the factors KHv, KHĮ, KHȕ and the contact stress are then ascertained. In cases of large differences between the loads in effect, unacceptable deviations can occur when using this method. The tooth loads have a direct effect on the part of these factors which is caused by the geometric deviations, because the ratio of geometric deviation (e.g., helix slope deviation) to elastic deforma-tion is decisive. There is no corresponding constant (equivalent) load for this influence, which changes for different loads. So what can sometimes occur, for example, is that a too-large face factor K Hȕ, stipulated from the equivalent load for fatigue safety, is used for maximum load (maximum occurring tooth load, impact load) when transferred to the calculation of safety, and for that reason one has low safety levels for this. 6.5 Analysis of Load Capacity 267 Table 6.5/2 Elasticity Factor ZE ZE 191.6 1) 189 181 155 160 162 … 165 188 181 161 174 157 144 … 146 1) With E1 = E 2 = E = 2.06 105 N/mm2 the following is true: ZE = 189.8ඥ Ȁ ; . Gear 2 Poisson’ s ratio Ȟ 1 0.3 0.3 0.3 0.3 0.3 0.3 0.3 0.3 0.3 0.3 0.3 0.3 Elasticity of material E1 N/mm2 2.1 105 1) 2.0 105 1.7 105 1.0 105 1.1 105 (1.2 ... 1.3) 105 2.0 105 1.7 105 1.2 105 1.7 105 1.2 105 1.2 105 Basic material Steel Cast steel Spheroidal graphite cast iron Cast tin bronze Tin bronze Lamellar-graphite cast iron Cast steel Spheroidal graphite cast iron Lamellar-graphite cast iron Spheroidal graphite cast iron Lamellar-graphite cast iron Lamellar-graphite cast iron Gear 1 Poisson’s ratio Ȟ1 0.3 0.3 0.3 0.3 Elasticity of material E1 N/mm2 2.1 105 1) 2.0 105 1.7 105 (1.2 ... 1.3) 105 Basic material Steel Cast steel Spheroidal graphite cast iron Lamella r-graphite cast iron 268 6 Load Capacity and Running Performance of External and Internal Gearing c) Calculation of contact stress when the load alongside the lines of contact is known Through more exact analyses, for example with the help of influence numbers, if the total tooth load is known, it is possible to determine the load intensity (refer to Figure 6.4/8 for example) alongside the lines of contact. For a known line load w the contact stress results in ()H 221wEσ= π− νN (6.5/12) Here the following applies: E/(1 - Ȟ2) to be determined according to Equation (6.5/1b) and 1/ N according to Equation (6.5/1a). N1 and N2 result in 1min t et 1 b cospα +Δ ε=βNN (6.5/12a) wt 21 bsin cosa α=−βN N (6.5/12b) with 0 < ǻİĮ < İĮ as the dimensionless coordinate between meshing points A and E (refer to Figure 2.1/19a) and () ()22 1min t a 1 b 1 et /2 /2dd pα =− − εN (6.5/12c) The principal trace of Hertzian contact stress ıH alongside the line of action can be obtained from Figures 2.1/19b and 2.1/19d. Theoretically, an infinitely large contact stress exists on the base circles. Flank contact takes place in these areas only in extreme cases. 6.5.1.3 Pitting Strength and Influences The endurance limit ıHlim is based on the S-N curve obtained in the gear running tests. Since only one component of the acting stresses is used in the calculation as a measure of the material strain and other influences are neglected, such as the thermal stress due to the temperature flash during contact, the shear stress applied to the surface, and the specific sliding, until now it has been necessary to use the material characteristics obtained on the gear component during the running test. For flank pairs running in the range of finite life fatigue strength, with the help of a life factor Z N the endurance limit ıHlim is converted according to the S-N curve to the strength applicable for the existing load cycle number. As an approximation for the load cycle zone of the endurance limit, except in the cases of nitrided gears and cast iron, N = 50 10 6 is used. For cast iron, bath-nitrided and gas-nitrided steels, ISO 6336-6 and DIN 3990-2 specify NHlim = 2 106. However, for the number of load cycles N > NHlim = 50 106, some hypotheses also assume a further but smaller drop in strength. The deviation in the position of the break of the slope of the S-N curve, as well as a potential additional fall in the pitting endurance limit with N > N Hlim compared to the behaviour of test pieces in a state of vibration, may come about for several reasons. Certainly, the long-term tempering effect and impure lubricants are among them. Another reason can be sought in continuous wear, which can cause an increase in the additional dynamic load. In the case of non-integer gear ratios, only after multiple rotations do the most unfavourable flanks (pitch deviations) of the pairing mesh again, which gives the impression of a further drop in the endurance limit. The results of tests of tooth flank strength can be found among others in [6.5/6] to [6.5/16]. 6.5 Analysis of Load Capacity 269 The life factor is simplified: ZN Nq NH LH=lim (6.5/13) Equation (6.5/13) applies for 105 NL NHlim. Here the following applies: NHlim Load cycle, which identifies the break of the slope of the S-N curve for the pitting endurance limit Specified in ISO 6336 -2, DIN 3990-2 is NHlim = 2 106 for nitrided gears; cast iron NHlim = 50 106 for tempering steels; spheroidal graphite cast iron, pearlitic malleable iron; surface-hardened steels (As opposed to this, based on other experience, the former TGL 10545 (draft 1988) specified N Hlim = 100 106, with the exception of NHlim = 50 106 for structural steels which are annealed in the normal way and for tempering steels.) NL Load cycle or number of cycles of rolling of the gearing, which corresponds to the required lifetime Nn L nLp h= (6.5/13a) with np Number of meshes Lh Lifetime n Speed of the gear qH S-N curve exponent, pitting strength According to ISO 6336-2, DIN 3990-2 (refer also to Section 7.4.), the following applies for the range of finite life fatigue strength: q H = 13.2 for tempering steels, spheroidal graphite cast iron, pearlitic malleable iron, surface- hardened steels q H = 11.4 for tempering steels and nitrided steels, gas-nitrided; cast iron In other documents qH = 10 is specified (TGL 10545, draft 1988). For the area NL > N Hlim either ZN = 1 applies or, in cases of lubrication conditions which are not optimum, a further drop in strength is taken into account by a larger qH (Table 6.13, No. 11). There is an influence of the size of the gear on the strength. This is accounted for by the size factor ZX. According to Gohritz [6.5/6], it is possible to differentiate between • technological • stress-based • static and • surface-related size influences. The technological size influence includes that, in the case of case hardening, despite possibly having the same edge hardness, they could have a different core hardness and hardness characteristics, which likewise have an influence on the strength. The stress-based influence exists among others in the different stress gradients and the different support effects this causes. A static influence exists because impurities are statically distributed, and, in the case of a larger sample volume, the probability of the position of an impurity in the area of high stress is larger than for small sizes. A surface-related size influence exists among others because of the different penetration of hardening on the edge layer, which differs in relation to the stress gradient, for example because of shot peening or edge layer hardening. A proposal for ascertaining the size factor for quenched and tempered and case-hardened gears is contained in the cited work [6.5/6]. It seems necessary, however, to carry out further experi- mental tests. For the present, it is recommended in the case of surface-hardened gears to test 270 6 Load Capacity and Running Performance of External and Internal Gearing whether the maximum of the equivalent stress is still within the case, or respectively whether the distance between the depth of the maximum of the equivalent stress from the depth of case har-dening is sufficient in the case of gears under heavy strain (refer to Appendix 7). Cracks under the surface (also the outcome of flank cracks) were observed especially when brief overloading leads to plastic deformations where the case tapers off (test, Appendix 7) and/or non-metallic inclusions exist. For large wheels, it is very probable that the pitting begins under the surface. Since the potential influence of the position of the maximum of strain in the depths in the case of gears with strain-hardened surfaces is separately tested and, as needed, also accounted for (Appendix 7), it seems permissible to tentatively set Z X = 1 In the case of case-hardened and nitrided gears or gears with strain-hardened surfaces, here it is required that with respect to the stress sequence (equivalent stress), the case is adequate in both size and depth. Roughness, speed and lubrication influences must also exist. These quantities are linked because of the hydroelastic state of the contact area. For the influences currently listed in ISO 6336-2, DIN 3990-2, very low values are specified. Besides that, until now these quantities have been determined individually and without a hydrodynamic classification. So it seems justified to tentatively incorporate the quantity ( Z LZRZv) as follows: For ground or shaved gears : (ZLZRZv) = 1 (6.5/14a) For hobbed, shaped or planed by generating gears : (ZLZRZv) = 0.85 to 0.9 (6.5/14b) Where the paired gears have different strengths, the lower is used as ıHlim when this is appro- ached according to DIN 3990-2. Since pitting does not occur or is more seldom on the mating gear, or only begins when there is a larger load cycle, yet for the potential lifespan the damage of the pairing is decisive, a higher load is acceptable in some cases for a specified lifetime. This is accounted for by the hardness ratio factor Z W. In the hardness range 130 HB < HHB < 470 HB, DIN 3990 specifies the relationship HB W1301.21700HZ−=− (6.5/15a) Here HHB is the Brinell hardness of the softer gear; ZW = 1.2 is to be used for HHB < 130 and ZW = 1 is to be used for HHB > 470. On an empirical basis ISO 6336-2 additionally introduces roughness quantities, the speed and the nominal viscosity. Here it is judged, however, that this makes the load-bearing analysis neither more logical nor certain. From our own experience , it makes sense to take the following approach: Pitting safety is calculated for both wheels of the pairing. If one labels the tooth flank endurance limit of the softer gearing with Hlim′σ and that of the harder gearing with ′′σHlim, the result is the hardness ratio factor for the softer wheel, in lim H lim W Hl im0.5 ZΗ′′ ′σ+ σ=′σ (6.5/15b) Note: If ZW > 1.1 is true for spur gearing, then the following is to be set: ZW = 1.1. In the case of helical gearing, if (İȕ > 0.5) and ZW > 1.2 are true, then the following is to be set: ZW = 1.2. In the equations for the safety analysis, or respectively for determining permissible contact stress, ıHlim = ıƍ Hlim with Z W according to Equation (6.5/15b) is to be used for the softer gearing. For the harder gearing, the analysis of pitting safety is done with ıHlim = ı" Hlim and ZW = 1. 6.5 Analysis of Load Capacity 271 In the case of large differences in hardness, care must always be taken to ensure that the harder gearing has low roughness and, if possible, is ground. Otherwise, unacceptable wear can occur. In the case of helical and double helical gear pairs, pitting, which usually only occurs in certain areas, cannot have such critical consequences as in the case of spur gearing. Even then, in contrast with spur gearing, the line of contact running diagonally over the tooth ensures a certain kinematic precision (guiding precision), if in limited areas, such as in the vicinity of the pitch circle, for example, pitting exists. 6.5.1.4 Safety against Pitting; Permissible Strain Taking into consideration the symbols introduced in Section 6.5.1.3 for important influences, safety against pitting (stress safety) results in () SZZZZ ZZHHN HRLv WX =σ σlim (6.5/16) Applicable here: ıHlim Pitting endurance limit (endurance limit ascertained in running test, Section 7.4) ıH Contact stress (Hertzian contact stress), Equation (6.5/11) ZN Life factor [Compared with the endurance limit, in particular it accounts for the higher capacity to withstand stresses in the range of finite life fatigue; Equation (6.5/13).] (ZRZLZv) Hydroelastic, tribomechanical influence; with ZR roughness factor, ZL lubricant factor, Zv velocity factor (this is intended to take the tribomechanical/hydroelastic influence into account; Section 6.5.1.3) ZW Hardness ratio factor, according to Equations (6.5/15a) and (6.5/15b) (This is intended to take into account the effective increase in flank strength in the case of the “hard-soft” pairing, with regard to the endurance limit for the softer gearing.) ZX Size factor (This is intended to take into account the influences of technological, stress-based, static and surface-related quantities; Section 6.5.1.3) Tentatively: Z X = 1 as an approximation The following is necessary for avoiding damage: SH SHmin (6.5/17) Among others, the crucial quantity for safety S H is dependent on the significance of the system, potential consequential damage, how exact the knowledge of the load is, the quality of the technology, the effective probability of compliance with tolerance limits for the endurance limit, the geometric deviations and the load, the diagnostic measures, and on the agreements and commitments between the ordering party and the buying party. The value SHmin accounts for the approximated nature of the calculation method and the inaccuracies in the specification factors. When operating conditions and assumptions are observed, it is often adequately considered that SHmin = 1.1 6.5/18) (and sometimes falls below to SHmin = 1.0). If the analysis of load capacity is to be done by comparing the existing stress ıH with a permissible stress ıHP, Equation (6.5/19) can be used: ıH ıHP (6.5/19) The permissible stress (contact stress) is ()Hlim N HP R L v W X Hm i nıZZZZ Z ZSσ= (6.5/20) 272 6 Load Capacity and Running Performance of External and Internal Gearing The individual quantities are explained at Equation (6.5/16). Its calculation is summarised in Table 6.5/13. 6.5.1.5 Safety against Permanent Deformation, Cracks or Brittle Fracture on the Surface Layer under Maximum Load Because of solitary load peaks, which, because of their low frequency can hardly make any difference in the calculation of equivalent load, unacceptable permanent deformations in the vicinity of the surface and especially the transition area to the core area are nevertheless possible, which cause a reduction in fatigue strength and as a result can induce pitting damage or large- area bursting or flank breakage. In the case of very thin, hard layers, cracks are also possible. Appendix 7 offers an orientation on more precise analyses, particularly for case-hardened gearing. An initial rough, global examination , which cannot comprehensively deal with potential damage, however, is possible with Equation (6.5/21). Accordingly, a safeguard against damage in the case of maximum (intermittent) load is HSt HSt HmaxS =σ σ (6.5/21) Here the following applies: ıHSt Maximum bearable contact stress at one-off load (or low load cycle) According to ISO 6336-2, DIN 3990-2, for ı HSt the following can be selected: ıHSt = Z NmaxıHlim with Z Nmax = 1.6 for case-hardened gearing; Z Nmax = 1.3 for nitrided gearing, refer to Table 6.5/13. ıHmax Maximum occurring contact stress [ ıHmax with Ftmax according to Equation (6.5/11)] If there is to be a comparison in the analysis of load capacity of the existing maximum contact stress with a permis-sible maximum contact stress for the plastic deformation of the surface (in analogy to roller bearings), Equation (6.5/21a) applies, with the permissible values according to Equations (6.5/21b) to (6.5/21d): Hm a x H Pm a xσ≤ σ (6.5/21a) For untreated steel annealed in the normal way, or quenched and tempered steel: ıHPmax = 2.8 Rp0.2 (6.5/21b) For case-hardened, carbon-nitride-hardened, flame-hardened or induction-hardened steel: ıHPmax = 4HO (6.5/21c) For nitrided steel: ıHPmax = 3HO (6.5/21d) Here the following applies: HO surface hardness in HV; Rp0.2 yield stress (ı 0.2). Note: The load capacity limit for maximum load is usually given by Equation (6.5/21) or by the analysis according to Appendix 7. Further and different accounts are contained the following literature: [6.5/111], [6.5/112], [6.5/118], and [6.5/130]. It must be pointed out that in the case of differing distributions of stress over the face width, for example through KHβ x 1 and KHβ >> 1, there is an important difference in the probability of breakage or damage, despite having the same level of safety. The probability of failure depends on the size of the volume or area under stress. Initial attempts at this, which address the correlation mentioned, or attempt to take it into con-sideration, have long existed [6.5/129] (also refer to newer works, e.g., [6.5/111]). However, a solid foundation for a practical analysis of load capacity is yet to be established through further research work. 6.5 Analysis of Load Capacity 273 6.5.2 Load Capacity of the Tooth Root The analysis of load capacity of the tooth root is aimed both at fatigue breakage and at per- manent deformation, cracks or overload fracture . The stress in the tooth root area, the elementary stress parts and the calculation model for the commonly standardised calculation in the case of solid gears (no gear rim influence) are shown in Figure 6.5/8. Required here are load (Section 6.3), load distribution (Section 6.4), geometric quantities (Section 2.3) and the determination of stress concentration at the tooth root (notch). For this, first a few basic findings shall be presented. After that, the calculation for gearing whose tooth root strain is not affected by the gear rim thickness (solid gears) will be shown. This will be accompanied by a section on the approximate determination of the decisive stress for gears with thin rims (internal gearing in particular) and the endurance limit. Distinguishing features here are the alternating stress that occurs and the influence of mean stress. Figure 6.5/8 Model of the calculation of tooth root stress for external and internal gearing 6.5.2.1 Basics of Notch Effect The following considerations are initially some essential definitions and then devoted to the problem of stress concentration . Tooth breakage almost always originates in the tooth root area. For this, the corresponding analysis of load capacity is carried out. (Exceptions are flank breakage and gearing with a hard edge layer in cases of plastic deformation under the edge layer, abetted by inclusions.) The maximum (local) stress in the tooth root region will be used. It will refer to a stress which is calculated at a cross section at a constant position. This reference stress is here the nominal bending 274 6 Load Capacity and Running Performance of External and Internal Gearing stress ıFn, which, in the case of external gearing, is calculated at the 30 tangent ( ș = 30 , Figure 6.5/8). Since this tangent usually does not exist in the case of internal gears due to the shallower root fillet, for our purposes for gearing with internal gears the 60 tangent will be used to calculate the reference stress. The local stress, which refers to the nominal stress ı Fn, and whose position on the root fillet changes depending on the tooth data and the position of load application, is called the stress concentration factor Y S. It corresponds in a similar sense to the familiar diameter quotient, which is used in the case of rod-shaped elements, such as shafts (here significantly influenced by the shear stress, however). Because of the uncertainties felt at the start, Y S will be labelled the stress correction factor ; YS results in F S FnYσ=σ (6.5/22) Here the following applies: ıF Local stress (Maximum of the stress in the notch at a tangent angle ș. It usually lies in the range ș = 25 to 80 .) ıFn Tooth root nominal bending stress (reference stress) External gearing: Calculated at the 30 tangent, according to ISO 6336-3 and DIN 3990-3 Internal gearing: Calculated at the 60 tangent More precise analyses, such as with the help of finite element method (FEM) or boundary element method (BEM), show that the stress maximum lies in a range of the tangent angle of ș = 25 to 80 , if one factors in the manufacturing with shaper cutters and extra-depth gearing, thin gear rims and internal gearing. This alone is not sufficient reason to question the definition ș = 30 because the local stress (notch stress) can reference an arbitrarily defined quantity. A different problem arises when the stress concentration factor is determined dependent on the notch parameter s (2 NFn/sFn; hF/sFn). Here the faulty results arise when the stress maximum lies far away from the 30 tangent (e.g., at the 60 tangent); the stress concentration factor is determined dependent on the radius of curvature to tooth thickness parameter at the 30 tangent, and this ratio in the tooth root fillet changes drastically. This situation occurs among others in the case of external gearing manufactured by shaper cutters without tip rounding, especially in cases of shaper cutters with large numbers of teeth. This effect is particularly extreme when, due to profile shift, the pitch circle of the workpiece is shifted towards the tooth root. Another reason for deviations is the change in the compressive stress part, which is accounted for in neither the nominal reference stress nor in the stress concentration. However, if one assumes that hobs and rack-type cutters are always only used with a tip rounding ( Na0 ≥ 0.2 m n) and that even shaper cutters in most cases have either a tooth tip chamfer or tip rounding, then, with an eye on tradition, retaining the 30 tangent in the future for determining the notch parameter seems, as an approximation, permissible. Minor improvements would still be possible if a simple relationship for the location of maximum stress ( ș ımax) would be found for determining the notch parameter and if the change in compressive stress generally by the nominal reference stress was registered from bending (b) and pressure (d) ( ı Fnenn = ı bnenn – ı dnenn; refer also to [6.5/121]). An approximation to account for the change in the notch parameter for the point of maximum local stress in relation to the reference cross section (30 tangent) can be achieved by forming a weighted mean value of the radii of curvature at the 30 tangent and at the transition to the root diameter (minimum!). This was verified as a good improvement possibility: NFn,improved = 0.65NFn(ș = 30 ) + 0.35 NFn min (refer to [6.5/121]). With gearing that is adequately precise and sufficiently loaded, in the case of spur gearing, the largest local tooth root stress occurs at load application on the outer point of single tooth contact. In contrast, in the case of helical gearing, the highest stress exists when, originating from the front face, the line of contact runs from the tooth tip diagonally over the tooth. Because of flank modification, the roll angle, at which the largest stress occurs, can shift. Until now the local stress or the stress concentration (Figure 6.5/8) has been determined through measurements on real gears [6.5/18], photoelastically [6.5/19], analytically with the help of mapping functions [6.5/20], and numerically with the FEM [6.5/22] and the BEM/singularity method [6.5/23] to [6.5/25], [6.5/21]. The singularity method, as a special variant of the BEM, has proven itself to be extremely useful. 6.5 Analysis of Load Capacity 275 Tables 6.5/3 a), b) and c) contain some results for the stress concentration factor, which were attained with the help of the singularity method. To summarise, the following conclusions can be drawn: • With regard to standardised gearing, the influence of the pressure angle lies in the range 15%. • With a lever arm of decreasing size, the stress concentration factor YS = ı max/ıFn rises. The reason for this is the influence of shear stress on the stress gradient and the relation to the pure nominal bending stress. • The smaller the lever arm, the lower (nearer) the stress maximum lies to the tooth gap center. • The thinner the wheel rims, the lower (nearer) the stress maximum lies to the tooth gap center. • In cases of shaper cutters with large numbers of teeth without tip rounding, extremely small radii of curvature can occur in the tooth root fillet. The notch parameters can then not be determined at the 30 tangent anymore, because the stress maximum lies at larger angles and the radii of curvature in the tooth root fillet drastically change. • The stress concentration factors of two gears with different tooth root geometry behave roughly inversely proportionally as the third root of their notch parameter 2 NFn/sFn at constant hF/sFn; see Equation (6.5/23). • Ground notches are generally overload notches. An unloading only exists when the radii of curvature in the ground notch are larger than the curve generated by the pre-machining. The equation for calculating the stress concentration factor Y S or Y Sa in accordance with ISO 6336-3, DIN 3990-3 can be found in Appendix 2. The YSa for load application on the tooth tip can also be approximated for NFn ≥ 0.15m n by 1) 2) 3 Fn Sa Fn = 1.22 / 2 s Y N (6.5/23) Dependent on the lever arm , such as for a load application on the outer point of single tooth contact (then hF = hFe), under the same conditions the following applies for YS 1) 2): F Fn S F Fn Fn Fn0.0712 + 0.0657 / 0.234 = + + 0.973 2 / / shY ss h N (6.5/24) The calculation results determined for YSa with the help of the singularity method (BEM variant) are contained in Figu- res 6.5/11 for external gearing and 6.5/14 as an example for internal gearing. The increase in stress due to grinding notches can be calculated for detailed analyses according to Equation (6.5/25) for external and internal gears [6.5/109] 1) (refer to Figure 6.5/15a): 2 g0 Fn ngs Fng FnSrel n 31.041.48 1.85 1.5 1 Yth mm= −⋅ − − ⋅ + NN NN N N (6.5/25) Equation (6.5/25) according to Diss. H necke , TU Dresden 20013); YSrel for toothed rims <6.5/109>; t0; hS , Ng, NFn: Figure 6.5/15. Equation (6.5/25) is valid for Ng/NFn = 0.1 to 1.0; t 0/mn = 0.012 to 0.06, h g/mn = 0.01 to 0.1. The radius of curvature produced in the grinding notch on the 30 tangent is to be ascertained according to the relationships given in Section 2.3. From experience, the effective rounding Na0(g) of the grinding disc required for this corresponds to roughly half of the grain size. Owing to the support effect, the increase in stress concentration calculated for the case of the homogeneous, isotropic material is considerably reduced. 1) Other equations are given for YS in ISO 6336-3 and DIN 3990-3 (Appendix 2). 2) Equation (6.23) to (6.25) as well as the equations of ISO 6336 and DIN 3990 (issue 1987) represent approxi- mations. More exact values are optained from diagrams (Figs. 6.5/11 and 6.5/14). According to ISO 6336-3 or DIN 3990-3 one obtains for x 0 very low deviations from exact results (BEM), and for x 1 the stress numbers according to the standards are about 20% larger than the exact values. 3) Additionally use Equation (6.5/26) and Figure 5.5/15 (for wheel bodies, relations are given by Hantschack [6.5/109]). 276 6 Load Capacity and Running Performance of External and Internal Gearing Table 6.5/3 Tooth Root Stresses; a) Stress Gradients No. Tooth root stress No. Tooth root stress 1 toothed rack Į a0Į15 İ1.0 0.2m= = =N 2 toothed rack Į a0Į15 İ1.8 0.2m= = =N 3 toothed rack Į a0Į20 İ1.0 0.2m= = =N 4 toothed rack Į a0Į20 İ1.8 0.2m= = =N 5 toothed rack Į a0Į28 İ1.0 0.2m= = =N 6 toothed rack Į a0Į28 İ1.8 0.2m= = =N 6.5 Analysis of Load Capacity 277 Table 6.5/3 Tooth Root Stresses; a) Stress Gradients (continued) No. Tooth root stress No. Tooth root stress 7 Į a017 0 Į20 İ1.0 0.25 (hob)z x m= = = = =N 8 Į a017 0 Į20 İ1.6 0.25 (hob)z x m= = = = =N 9 Į a017 0.6 Į20 İ1.0 0.25 (hob)z x m= = = = =N 10 Į a017 0.6 Į20 İ1.6 0.25 (hob)z x m= = = = =N 11 Į a0 040 0 Į20 İ1.0 0 80 (shaper cutter)z x z= = = = = =N 12 Į a040 0 Į20 İ1.0 0 (hob)z x= = = = =N 13 Į a0 080 0 Į20 İ1.0 0 25 (shaper cutter)z x z=− = = = = =N 14 Į a0 080 0 Į20 İ1.8 0 25 (shaper cutter)z x z=− = = = = =N 278 6 Load Capacity and Running Performance of External and Internal Gearing Table 6.5/3 Tooth Root Stresses; b) Assumptions and Results No. Z Į İĮ z Na0/mn z0 (zs) )max YS 2NFn/sFn ș = 30 ș = 60 ș = 30 ș = șımax ș = 60 1 Rack 15 1.0 0 0.2 Hob 35.6 2.22 2.44 0.18 0.18 0.17 2 1.8 39.6 2.47 2.71 0.17 3 20 1.0 40.3 2.23 2.42 0.16 0.16 0.15 4 1.8 44.2 2.72 2.96 0.16 5 28 1.0 47.8 2.23 2.38 0.14 0.13 0.13 6 1.8 58.9 3.79 4.06 0.13 7 17 20 1.0 0.25 26.3 1.48 1.86 0.54 0.59 0.34 8 1.6 38.0 1.77 2.22 0.46 9 1.0 0.6 32.6 1.77 2.02 0.27 0.26 0.22 10 1.6 44.2 2.41 2.74 0.24 11 40 1.0 0 0 80 52.1 1.99 2.02 0.35 0.14 0.12 12 0.25 Hob 35.4 1.68 1.97 0.41 0.36 0.26 13 –80 0 25 62.7 2.40 2.63 0.48 0.08 0.09 14 1.8 –0.8 66.0 4.62 4.83 0.25 0.04 0.04 Table 6.5/3 Tooth Root Stresses; c) Geometric Values of the Gearing Tested No. YFa FnN sFn hFa/sFn ș = 30 ș = 60 ș = 30 ș = șımax ș = 60 ș = 30 ș = șımax ș = 60 ș = 30 ș = șımax ș = 60 1 2.49 2.27 0.20 0.20 0.20 2.19 2.22 2.34 0.91 0.91 0.88 2 0.20 2.23 0.91 3 2.05 1.89 0.20 0.20 0.20 2.41 2.45 2.56 0.82 0.82 0.80 4 0.20 2.47 0.82 5 1.55 1.45 0.20 0.20 0.20 2.79 2.87 2.93 0.72 0.71 0.71 6 0.20 2.93 0.71 7 3.08 2.46 0.50 0.54 0.37 1.85 1.82 2.16 1.04 1.04 0.97 8 0.44 1.93 1.03 9 2.17 1.91 0.30 0.29 0.27 2.21 2.23 2.42 0.94 0.94 0.90 10 0.28 2.30 0.93 11 2.48 2.24 0.38 0.16 0.13 2.15 2.27 2.31 0.92 0.91 0.90 12 2.45 2.08 0.44 0.39 0.31 2.13 2.17 2.39 0.90 0.90 0.86 13 1.87 1.71 0.61 0.11 0.12 2.54 2.73 2.72 0.77 0.76 0.76 14 1.57 1.50 0.34 0.05 0.06 2.72 2.83 2.82 0.71 0.70 0.70 6.5 Analysis of Load Capacity 279 The factor YS(smooth) , ascertained for the tooth root fillet without grinding the step “smooth”, is to be multiplied with the factor YSrel: S Srel S(smooth)YYY=⋅ (6.5/26) Figure 6.5/15 and Table 6.5/5 provide an overview of the increase in the notch effect due to grinding steps, registered by YSrel (also refer to [6.5/26], [6.5/27] and Equation (6.5/25). In some cases, the stress concentration factor YSa is given already multiplied with the form factor YFa (Figures 6.5/9 and 6.5/12). The product YFS is called the combined tooth form factor (for details, refer to Section 6.5.2.2): FS Fa Sa= YY Y ⋅ (6.5/27) The form factor Y Fa represents the nominal tooth root stress when load is applied on the tooth tip, intended for the defined root cross section (30 or 60 ), with reference to Ft /(bmn) [Y Fa from Equation (6.5/34c)]: ()Fn(a) Fa = / tnYF bmσ ⋅ (6.5/27a) (Figures 6.5/10 for external gearing and 6.5/13 for internal gearing contain examples for the form factor Y Fa). For gearing produced by hobs or rack-type cutters, if the tool tip rounding Na0 is known, it is possible to portray the combined tooth form factor YFS, dependent on the number of teeth of the equivalent gear znx z/cos3ȕ and the profile shift factor x. For this, Equations (6.5/28a) to (6.5/28c) represent (rough) approximations (TGL 10545, draft 1988). They apply for load application on the tooth tip and tooth flanks, with a basic rack according to ISO 53, DIN 867 for external gearing: • for a0 n 0.2 : m=N 2 FS nn7.63 = 4.08 + 0.18 + 15.94 xYxzz− (6.5/28a) • for a0 n 0.38: m=N 2 FS nn13.17 = 3.467 + 27.91 + 0.091 xYxzz− (6.5/28b) • with protuberance (Figure 2.3/1b) for ha0/mn = 1.4 and spr = 0.05: 2 FS nn25.28= 3.61 + 0.53 + 37.56 xYxzz− (6.5/28c) In isolated cases – especially in the case of hard edge layers – the risk of breakage can lie above the tooth root, especially when plastic deformations occur at the transition point from the hard edge layer to the core, because of Hertzian contact stress at maximum load or impact load. Because of subsequent strains, cracks can form in these zones – something which is further expedited by the presence of inclusions. In the course of further operation under Hertzian strain, the bending stress alongside the tooth height causes the crack to form and, after the crack develops, results in fatigue breakage. Table 6.5/4 shows the stress gradient above the tooth root surface and the tooth height, for the edge zone of the surface. It is evident that a relative stress maximum occurs above the tooth root fillet in some cases. 280 6 Load Capacity and Running Performance of External and Internal Gearing This is another reason why it is necessary to test the entire cross section of the tooth profile when carrying out precise analyses of strain. Table 6.5/4 Stress Curves across the Tooth Height; Profile of DIN 867, according ISO 53 No. Gearing type / stress curve No. Gearing type / stress curve 1 Įtoothed rack profile: DIN 867, ISO 53 load:İ1.0= 2 Įtoothed rack profile:DIN 867, ISO 53 load:İ1.8= 3 Įinternal gear 80 0 profile: DIN 867, ISO 53 load:İ1.0z x=− = = 4 Įinternal gear 80 0 profile: DIN 867, ISO 53 load:İ1.8z x=− = = 5 \* a \* f ĮHigh contact ratio gearing 40 Į15 1.3 1.5 load:İ1.0z hh= = == =6 \* a \* f ĮHigh contact ratio gearing 40 Į15 1.3 1.5 load: İ2.0z hh= = == = 7 \* a \* f Į28 -gearing 25 Į28 1.0 1.25 load:İ1.0z hh= = == = 8 Įstandard gearing profile: DIN 867, ISO 53 8 0.5 load:İ1.0z x= = = 6.5 Analysis of Load Capacity 281 Figure 6.5/9 Combined tooth form factor YFS for external gearing manufactured with hob or rack-type cutter using basic rack definition according to ISO 53, DIN 867 (results of calculations with the singularity method) a) n a00.25m=N b) n a00.375m=N c) Protuberance, n aP0 na00.4 1.4m hm= =N 282 6 Load Capacity and Running Performance of External and Internal Gearing Figure 6.5/10 Form factor Y Fa for external gearing manufactured with hob or rack-type cutter using basic rack definition according to ISO 53, DIN 867 for the root stress at the 30 tangent a) a0 n 0.25 m =N b) a0 n 0.375m=N c) Protuberance a0 n a0p n 0.4 ; 1.4mh m ==N 6.5 Analysis of Load Capacity 283 Figure 6.5/11 Stress concentration factor YSa for external gearing manufactured with hob or rack-type cutter using basic rack definition according to ISO 53, DIN 867 (results of calculations with the help of the singularity method). Reference stress: nominal tooth root stress at the 30 tangent (ș = 30 ) a) a0 n 0.25 m =N b) a0 n 0.375m=N c) Protuberance a0 n a0p n 0.4 ; 1.4mh m ==N 284 6 Load Capacity and Running Performance of External and Internal Gearing Figure 6.5/12 Combined tooth form factor Y FS for internal gearing manu- factured with shaper cutter using basic rack definition according to ISO 53, DIN 867 (results of calculations with the help of the singularity method). Reference stress: nominal root stress at the 30 tangent ( ș = 30 ) a) 018sz= b) 0 25sz= c) 080sz= 6.5 Analysis of Load Capacity 285 Figure 6.5/13 Form factor YFa for internal gearing for nominal root stess at 60 tangent for manufacture with shaper cutter ( z0s = 25) using basic rack definition according to ISO 53, DIN 867 Figure 6.5/14 Stress concentration factor YSa for internal gearing for nominal root stess at 60 tangent for manufacture with shaper cutter ( z0s = 25) using basic rack definition according to ISO 53, DIN 867 286 6 Load Capacity and Running Performance of External and Internal Gearing a) b) c) d) Figure 6.5/15 Stress concentration in the case of double notches (grinding step); calculation results, PCSpann calculation programme, TU Dresden [6.5/123], based on [6.5/23]: a) tooth with grinding step; b) influence of the grinding step height h S; c) influence of the grinding stock t0; d) influence of the radius of curvature NFn2; e) relative stress concentration factor (refer to Table 6.5/5 for this) 6.5 Analysis of Load Capacity 287 Table 6.5/5 Information on Grinding Steps from Figure 6.5/15e z Na02/mn hs /mn t0/mn 17; 100 0 0.05 0.07 0.10 0.035 0.04 0.05 0.07 0.10 0.035 0.08 0.05 0.07 0.10 0.035 0.15 0.05 0.07 0.10 0.035 Na02 Effective edge radius of the grinding disc 6.5.2.2 Tooth Root Stress in the Case of Solid Gears Figures 6.5/16a to c show tooth root stress curves measured with the help of strain gauges while rotating slowly. The greatest stress occurs at the outer point of single tooth contact D or B respectively. At the centre of the stress gradient shown, a slight change (step) can be seen (a and b). This small step is caused by the direction change of tooth friction, and in the curve it is dependent on the direction of rotation and the direction of loading. For technical reasons, measurements could only detect it on the tension side. As would be expected, this influence is small, and it also decreases with increasing speed because the friction declines. As such, for the sake of simplicity, in the calculation only the loads normally acting on the tooth flank are considered. a) Tension side (driven) c) Compression side (driving) b) Tension side (driving) Figure 6.5/16 Tooth root stress curves measured using strain gauges [6.3/1] 288 6 Load Capacity and Running Performance of External and Internal Gearing Basic relationships for tooth root stress To determine the strain, the model of spur gearing depicted in Figure 6.5/8 is assumed (with Figure 6.5/17), and as load, the load Fbn resulting from the (utilisable) torque to be transmitted is considered. The deductions are based on an ideal version of spur gearing. After that the results will be applied to helical gearing, using approximations to take its special features into account. Figure 6.5/17 Geometric quantities on the tooth for calcu- lating the root stress With load application on the outer point o f single tooth contact, the nominal tooth roo t bending stress is b F0n bı= M W (6.5/29) (with the bending torque b bF e F e = cosĮ MF h ⋅ with the load application angle ĮFe, the normal force Fb and the resisting moment Wb). The normal tooth load Fb and the resisting moment W b are 2 t F bb = , = cos Į 6F sbFW With the quantities for Mb, Fbn and W b, the nominal stress results in the form t F0n F = FYbmσ (6.5/30) with YF according to Equation (6.5/30a). The variable YF is called the form factor . It combines geometric quantities. () ()Fe Fe F 2 F6 / cos Į = / c o s ĮhmY sm (6.5/30a) The tooth root thickness sF can be determined according to Section 2.3, and the lever arm hFe and load application angle BFe according to the Equations (6.5/31 and 6.5/31a to e): Fe Fe = 2dhY − (6.5/31) Here the following applies: Y refer to Section 2.3 Fe b Fe = / cos Į dd (6.5/31a) with Fe B,D b Į = tanĮ ȥ− (6.5/31b) B,D b B,D cosĮ dd= (6.5/31c) 6.5 Analysis of Load Capacity 289 b = + inv Į ȥs d (6.5/31d) Outer points of single tooth contact ()2 22 2 D,B b a b Į e = + İ 1 ddd d p −− − (6.5/31e) dB driven gear dD driving gear If one includes the additional external and internal dynamic loads as additional load (application factor K A, dynamic factor KFv) and takes into account the load distribution in the circumferential direction (transverse factor KFĮ) and in the axial direction (face factor KF ), Equation (6.5/30) results in the nominal stress for spur gearing: Fn F0n FnF t AF vF Į Fȕ F = resp.ıı = ıK FKK K K Ybm (6.5/32) With an approximation, it is possible to convert the form factor YFa intended for the tooth tip to the determining load application: FF a Į / İ YY≈ (6.5/32a) With Equation (6.5/32a), the modified force application in the case of spur gearing and the larger total length of line of contact vis- -vis b/cosȕ in the case of helical gearing are to be accounted for. With the help of the stress concentration factor [ Y S = Y S (2NFn/sFn; hF/sFn)], the local stress TF = TFnYS finally results for spur gearing: t FA F v F Į Fȕ FS = FKK K K Y Ybmσ (6.5/33) The factors YF and YS are dependent on the parameters of the meshing gears ( hFe). In order to make things simpler, an overlap coefficient Yİ1 will be introduced, which converts the quantities ( YFa YSa) intended for the tooth tip to the outer point of single tooth contact: FS F a S a İ1 = YY Y Y Y (6.5/33a) The expansion to helical gearing works approximately by transferring Equation (6.5/33) to the normal section. For F t /b it is then analogous and expanded to write tn t j İ2 = 1 cos ȕcos ȕFF l b Y ⋅ (6.5/33b) with YF x YF (FB ,Fβ) to take into account the larger length of the line of contact compared to b/cosβ. With zn x z/cos3ȕ; sF = sFn and a helix factor Yȕ, which accounts for all of the effects of helical gearing which have not yet been considered, the result is (if Yİ1.Yİ2 = Yİ is set and written in abbrevi- ated form in YFa .YSa = YFS) the local stress which is decisive for fatigue stress, in t F A Fv F F FS n()FKKKK Y Y Ybmα ββ ε σ= (6.5/34) 290 6 Load Capacity and Running Performance of External and Internal Gearing or written in another way with )F0 as the local stress without additional loads and modification of the load distribution: FF F 0 t F0 FS n FA F v F Fİwith ()K FYYYbm KK K K Kβ αβσ= σ σ= = (6.5/34a) Where: KA Application factor , Section 6.3.3 (KA is intended to register additional external dynamic loads which are not yet accounted for in the load spectrum or respectively to replace the load spectrum approximately; KA refers to Ft; usually Ft is the nominal load, Ft = Ftnenn.) KFv Dynamic factor, Section 6.3.4 (K Fv is intended to account for the influence of additional loads due to vibrations on tooth root stress , which are excited by periodically changeable tooth stiffness and tooth deviations.) KFĮ Transverse factor , Sections 6.4.4.1 and 6.4.4.3 (KFĮ is intended to account for the influence of the load distribution.) KFȕ Face factor , Sections 6.4.4.2 to 6.4.4.5 (KFȕ accounts for the influence of the change to load distribution in the axial direction of the tooth on the tooth root stress due to geometric deviations and deformations.) YFS Combined tooth form factor , according to DIN 3990 with Equation (6.5/34b) and Appendix 2; newer findings in Figure 6.5/9 for external gearing and Figure 6.5/12 for internal gearing (for special cases z 0s) (for load application on the tooth tip, YFS accounts predominantly for tooth root thickness, lever arm , load application angle, and stress concentration). YY YFS Fa Sa= with (6.5/34b) YFa Form factor for load application on the tooth tip, and reference to the normal module (Figures 6.5/10 or 6.5/13) () ()Yhm smFaFa n Fan Fn n n=6 2/c o s /c o sα α (6.5/34c) YSa Stress concentration factor for load application on the tooth tip (Figure 6.5/11), approximation or Equation (6.5/34d) Sa31.22 2FnF n Ys=⋅ N (6.5/34d) sFn, NFn at 30 tangent in the case of external gearing and 60 tangent in the case of internal gearing (more detailed information for internal gearing is in Appendix 6); a different relation is given in ISO 6336-3 for YS (with denomination stress correction factor), Appendix 2. Yİ Contact ratio factor (Y İ is intended to account for the influence of contact ratios on the bending lever arm or the length of the line of contact and stress concentration. The transverse section is used as more appropriate than the normal section and Yİ is calculated with İĮ instead of İĮn.) 0.750.25 for 1 1for 1Y Yεβ α εβ α=+ ε <ε =ε ≥ε (6.5/34e) Yȕ Helix factor [Yȕ is intended to register the influence of helical gearing which is not yet fully considered in the other factors, as opposed to spur gearing (Figure 6.5/18).] 6.5 Analysis of Load Capacity 291 min 1120YYββ ββ=− ε ≥ (6.5/34f) min 1 0.25 0.75 Yββ=− ε ≥ (6.5/34g) where ȕ is the helix angle, in degrees. The value 1.0 is substituted for İȕ when İȕ > 1.0 and 30 is substituted for ȕ when ȕ > 30 . The nominal stress ıFn = ıF/YS is the reference stress for the stress concentration factor: t Fn A Fv F F F nFKK K K Y Ybmαβ β σ= (6.5/35) with YF calculated for the decisive meshing point (outer point of single tooth contact in the case of spur gearing), in analogy to Equation (6.5/30a) or as an approximation according to YY YFF a n=ε Here the following applies: YFa Form factor for load application on the tooth tip [Equation (6.5/34c) or Figure 6.5/10 (external gearing) and Figure 6.5/13 (internal gearing)] Yİn Overlap coefficient for conversion of the nominal stress of load application on the tooth tip to the outer point of single tooth contact ( ȕ = 0 ), or respectively for accounting for the length of the line of contact (helical gearing). Yεα εn≈1/ Note: For tooth root stress in the case of helical gearing, more precise analyses (e.g., FEM) sometimes result in considerably smaller values. This can apparently be traced back to the rough approximation of the factors (Y F,Y ) for ȕ 0. Newer proposals for ISO aim to improve this. In the case of load spectra, it is advantageous for each load step j to calculate the tooth root stress ıFj with KAj, KFvj, KFȕj and the tooth load Ftj and to determine an equivalent tooth root stress ıF according to Equation (5.3/12). For the sake of simplification, often an equivalent load (Equation 5.3/16) is determined, whereby, however, unacceptable deviations can arise in cases where large differences between the actin g loads exist. Figure 6.5/18 Helix factor Yȕ More precise analyses In more precise research on tooth root strain, not only the stress in the tooth root fillet, but also its distribution in the axial direction is analysed, in several mesh positions, so that 292 6 Load Capacity and Running Performance of External and Internal Gearing • specific flank modifications (e.g., lead crowning, correction of the pinion helix angle) can be realised, • the tooth root strain in helical gearing can be more precisely studied, or • the calculation of elastically designed gear bodies can be made possible. By also taking the geometric deviations into account, in this way it is possible to achieve a more uniform distribution of tooth root strain, and thus increase the load capacity of the tooth root. For this there are principally two approaches: • Based on a known load distribution, the local tooth root stress is determined for the discretised load, with the help of spatial methods (e.g., 3D FEM or 3D BEM), or • The spatial problem is partially reduced to a two-dimensional one. Here, in the case of spatial methods, first of all the distribution of the nominal stresses or the internal forces (clamping moment, lateral forces) are calculated. After that, in a second step, either slice by slice or only for the location of maximum bending moment, or the largest nominal stress, respectively, the local stress is determined, viewing it in a two-dimensional way. Particularly for those concerned with the practical side, the second approach leads to highly suitable methods (e.g., 2D FEM or 2D BEM), which involve considerably less calculation work. The lever arm and load are to be defined in the second step as ideal quantities in such a way that the bending moments (and the lateral force), determined in the tooth root using spatial methods, are available. In cases where the bending moment is known, it is also possible to approximately determine the nominal bending stresses and, with the help of a stress concentration factor YS = YS (2NFn/sFn, hF/sFn) that is already known, to infer the local stress. The lever arm hF follows yet again from the require- ment that the same ratio between lateral force and bending moment, which was ascertained in the 3D calculation, must also exist in the second step. The shear force has an influence on the stress concentration because it changes the entire stress gradient. Roughly approximated, hF can be equated with the actual lever arm of the load at this point. The nominal tooth root stress distri- butions initially required can be calcu-lated with the help of influence num- bers. Since the load on a section also leads to tooth root strain in neigh-bouring areas, the tooth root stress of a tooth section i is to be determined through a superposition of the partial stresses (nominal bending stresses) which all loaded sections k of a tooth generate (Figure 6.5/19). n Fni Fik k k1eF =σ= (6.5/35a) Figure 6.5/19 Load distribution and distribution of stress in the tooth root The calculation of the influence numbers eFik can be done in different ways, for example with the help of • the finite element method (FEM) or • approximations, attained from plate theory [6.5/33], [6.5/34], experimentally [6.5/35] or through the generalisation of results from FEM calculations [6.5/36]. 6.5 Analysis of Load Capacity 293 As follows, an approximation solution for calculating tooth root stress influence numbers will be given, which was determined by generalising the results of FEM calculations and was verified experimentally. The solution is set up in such a way that the principle path above the face width is registered by the function fıR(ȟF\*). With that, an arbitrary influence coefficient results in ()k Fik R F k 2 Fn n60.232 coshefsm∗ σ =⋅⋅ ξ α (6.5/35b) with FF n /m∗ξ= ξ where hk identifies the lever arm on which the load F k applies (Figure 6.5/20b), and ȟF identifies the distance from the load application point ( k) to the point ( i), at which the influence coef- ficient or the stress influence function fıR is to be calculated (Figure 6.5/20). Figure 6.5/20 Quantities on the tooth for using influence numbers The edge influence is accounted for as an approximation – as is the case when calculating the deformation influence numbers – by mirroring the general influence function at the tooth edge (front face). This is achieved by superimposing the edge influence function ǻ fıȕ on the influence function fı of an infinitely wide tooth (Figure 6.5/21a). Through ǻfıȕ, the edge elasticity influenced by the helix angle is accounted for. For b > 5mn, the following approach can be taken with tentative approximation1): () () ( ) ( )RF F R F R ,t a n h1 fff∗∗ ∗ ∗∗ σσ ∞ σ β ξ= ξ+ Δ ξ ξ ⋅ ξ + (6.5/35c) with FR FR nn;mm∗∗ξξξ= ξ= The following applies for the summand that registers the distribution of stress without edge influence : () ()ffσ∞ σ ξξF∗∗= () ()F with and according to Equation 6.5/35d . f∗∗ ∗ σ ξ= ξ ξ The part of the edge influence is calculated in () ( ) ()ȕ RF, 1 0.72 ff∗∗ ∗ σσΔξ ξ =ξ ⋅ β RF with 2∗∗ ∗ξ= ξ − ξ . The positive sign applies for cases where load is applied in the vicinity of the elastic (acute-angled) tooth end face; otherwise the negative sign is to be used. In the given relationship the helix angle is to be used (in radian dimension). This approximation applies in the area 0 ȕ ʌ/6. Here, ȟR is the distance from the load application point to the nearest tooth end face (corresponding to the positive direction). 1) The dissertation of Kunert [6.4/19] contains a different rendition. The results agree quite well. 294 6 Load Capacity and Running Performance of External and Internal Gearing The generalised influence function , standardised and referencing the maximum value, is calculated as follows (for ȟ\* 5), in ()2341 0.01627 0.2887 0.0908 0.0083 f∗∗ ∗ ∗ ∗ σξ= + ξ − ξ+ ξ− ξ (6.5/35d) and has tapered off at a distance of ȟ\* > 5 ( fı(ȟ\*) = 0). Figure 6.5/21a depicts the curve of the generalised influence function, appli-cable to the infinitely wide tooth, and its change for the finitely wide tooth with edge influence, and the influence of the tooth tapering off obtusely or acutely on the front face in the case of a helix angle of ȕ > 0. Because of the greater elasticity on the right face side compared to that on the left face side, the following would be true: ΔΔffσβ σ> This method is based on the conclusion that this influence function, Equation (6.5/35d), is relatively independent of the tooth profile, the load application height hk, and the location in the tooth root fillet where the stress is deter-mined, and depends only slightly on the lateral contraction of the materials used. It represents the curve of tooth root stress in relation to the maximum value. In cases of face widths of b < 5m, Equation (6.5/35c) should not be used anymore, because this would make multiple mirroring necessary. The use of general influence functions is an important basis for practical calcu-lations, which is easy to manage and should be possible with little calculation work. Figure 6.5/21b depicts a compa-rison of the general deviation influence function introduced in Section 6.4 and the stress influence function. Figure 6.5/21 Determination of the influence numbers for the tooth root stress: a) accounting for the edge influence; b) curve of the general influence function for deformation and tooth root stress a) b) 6.5 Analysis of Load Capacity 295 6.5.2.3 Root Strength and Influences For root strength, calculated endurance limits are used for the gear as a component part. This is intended to take into account influences, at least globally, which are not registered with adequate precision in calculations. Section 7.4.3 provides information on the basic values for fatigue breakage (data for the endurance limit). In the present state of the art, it is unfortunately not yet possible, for a given gear size and shape, to calculate with adequate precision the strength of the material that a heat treatment process (har-dening) will produce. For that reason some uncertainties linger, which are bypassed by the safety value which is typically stipulated. The fatigue endurance limit ascertained in the running test can be compared with optimally smooth test pieces, which corresponds to the quantity ( ı FE/YįT) (YįT in Table 6.5/8). At the very least they lie in the order of magnitude of that which is known for typical test pieces. For that reason these values can be used as a guide. Deviations – the somewhat smaller values of the gearing compared to smooth shafts – are caused among other things by the influence of the neighbouring tooth, the higher probability of fracture due to the number of teeth, and the different stress time function during the running test. Fatigue strength The endurance strength ıFE, which is either specially determined or taken from Section 7.4.4 or from ISO 6336-5, DIN 3990-5, is set as ıFE0 and adjusted for the influences of manufacturing (technology factor YT) and mode of operation (operation mode factor YA): TA FE FE0 YY σ= σ (6.5/36) Where: ıFE0 Tooth root endurance strength (basic strength) expressed by local stress in case of - pulsating load; Note: The endurance limits ı FE derived from Section 7.4 are to be set as ıFE = ıFE0 here, in order to incorporate the technology factor and operation mode factor according to Equation (6.5/36); - manufacture without special hardness increase in the tooth ground (refer to Section 7.4 for values) YT Technology factor (Table 6.5/6); use the smallest value if no tests available! YA Operation mode factor (Table 6.5/7) Table 6.5/6 Technology Factor YT Type of processing of the tooth root surface Technology factor YT Shot peening Applies for case-hardened or carbonitrided gearing; not ground in the hardened layer 1.2 to 1.4 Rolling Applies for flame-hardened or induction-hardened gearing; not ground in the hardened layer 1.3 to 1.5 Grinding Applies for case-hardened or carbonitrided gearing generally: 0.7 for CBN grinding discs: 1 Cutting processing (doesn’t apply for ground gearing!) 1 The tooth root endurance strength ı FE that applies for the specific kind of manufacturing and type of strain is implemented together with other influence factors in the equa tion for safety. Th ese influences are • Number of cycles of rolling ( YN) • Roughness ( YR) • Support effect (Y į) • Size (Y X) 296 6 Load Capacity and Running Performance of External and Internal Gearing Table 6.5/7 Operation Mode Factor YA Operation mode Operation mode factor YA Direction of loading Pulsating 1 Alternating 0.65 Reversing rev A rev A6for 10lg0.85 0.206 :0 . 6 5 NNY Y ≥=− = Note: The drop in strength already at Nrev = 1 is ascribed to microplastic deformations and changes in residual stresses. Nrev is the number of changes in load direction during operation, e.g., change in torque direction. The number of cycles of rolling (total number of meshes during the lifetime) is accounted for by the life factor Y N according to the S-N curve. YN Nq NF LF=lim (6.5/37a) Equation (6.5/37a) applies with qF for Nmin NL N Flim; here the following is true: 1 YN YNmax. Here the following applies: NFl Number of load cycles, which corresponds to the break of the slope of the S-N curve (NFlim = 3 106) NL Number of load cycles or number of cycles of rolling; refer to Equation (6.5/13a) qF S-N curve exponent (case-hardened steel qF 9; detailed information in Section 7.4.4) Nmin Lower limit of area of application: Nmin = 103; Exception: tempering steel, spheroidal graphite cast iron (pearlitic, bainitic), black heart malleable cast iron (pearlitic): Nmin = 104 YNmax Maximum increase in endurance limit to the strength limit (static strength); approximate values (ISO 6336-3, DIN3990-3): YNmax = 2.5 for steel annealed in normal way or quenched and tempered steel, cast steel; cast iron and spheroidal graphite, case-hardened, carbonitrided, flame- hardened or induction-hardened steel; YNmax = 1.6 for nitrided steel. 6.5 Analysis of Load Capacity 297 According to newer findings, in the case of N > N lim a further drop in the endurance limit is assumed. Until now this has pointed to the assumption that there is a drop of approximately 5% for each power-of-ten larger load cycle. It can be estimated according to Equation (6.5/37a), if qF = qFlim 40 is set for load cycle numbers N > Nlim . The reasons for this are not yet fully clarified, but the microinclusions are definitely worth considering. Another reason could be the long-term effect of the service temperature, which exercises a tempering effect, and the advancing wear, which, instead of an improvement, in some cases can cause an increase in the dynamic load. The notch sensitivity (structure, residual stress) is accounted for by the sensitivity factor Y į1). This Yį is dependent on the referenced stress gradient ()max ımaxȤ dı/d 1/ ı y =⋅ . Especially for gears: Fn2.3 Ȥ = N (6.5/37b) NFn in mm, refer for example to Appendix A1.2, or specially calculated (Section 2.3) Yį = Yį(Ȥ) can be taken from Table 6.5/8 or Figure 6.5/22 Figure 6.5/22 Sensitivity factor Yį for fatigue strength (endurance limit) if a smooth test piece is assumed; ıB = Rm; ı0.2 = Rp0.2 (= ıS) According to a newer way of looking at this, the notch sensitivity is dependent on the size of the area under stress. With that, in the case of pointed notches, because of the smaller probability of fracture that exists in a small region, the result is larger relative reductions of the mathematically determined stress concentrations. 1) Local root stresses endured in running tests are used as a reference in ISO6336-3 (Edition 2005) and DIN3990-3 (Edition 1987). These stresses imply the support effect at the test gear by sensitivity factor YįT. The ratio Yį/YįT = YįrelT has to be considered in the load capacity calculation, if fatigue strength values )FE obtained from running tests are used as reference (see also Appendix 2). 298 6 Load Capacity and Running Performance of External and Internal Gearing Table 6.5/8 Sensitivity Factors Yį, YįSt, YįT, YįStT Material and heat treatment of the tooth root surface Yį1) (endurance limit) YįSt (maximum load) YįT 1) (test wheel) YįStT (test wheel) Steel with distinct yield strength Annealed in the normal way ( ) 0.55 p0.20.47 875 / 1 Ȥ 10 + R+ −⋅() p0.24S2001 + 0.93 1 RY− 1.2 1.6 Quenched and tempered () 4S p0.23001 + 0.82 1 YR− 1.1 1.5 Cast steel Untreated Annealed in the normal way ( ) 0.65 p0.20.37 + / 36001+ 10R −⋅ χ () p0.24S2001 + 0.93 1 RY− 1.4 1.6 Quenched and tempered () p0.24S3001 + 0.82 1 RY− 1.3 1.5 Steel Case-hardened 0.55 0.721 + 10Ȥ−⋅ S0.77 + 0.22 Y 1.2 1.5 Carbon-nitride-hardened Induction-hardened Flame-hardened Nitrided S0.27 + 0.72 Y 1.2 Spheroidal graphite cast iron Untreated Pearlitic an-nealed Quenched and tempered () m 0.6 0.37 + / 14401 + 10ȤR −⋅ 1.0 1.4 1.0 Fn Fn2.3with = ; from Appendix 1.2 or especially calculated χ NN. For gears with grinding steps or with a ground tooth root surface, instead of NFn the product NFn (1/Y srel)3 is to be used; YSrel according to Equation (6.5/25) or Figure 6.5/15e; () SS a Į 0.8 + 0.2 İ . YY≈ For gears with grinding steps or with a ground tooth root surface, instead of YSa the product l Sa SreYY is to be used. 3Sa Fn Fn = 1.22 / ( 2 ) s Y N ISO 6336-3 and DIN 3990-3 contain numerical data for the roughness factor YR. However, the influence of the notch on this factor is ignored, so the simplifications here seem justified. In the case of gears that are quenched and tempered or annealed in the normal way, root strength, as 1) See footnote on previous page. 6.5 Analysis of Load Capacity 299 opposed to flank strength, plays a minor role anyway, which makes YR = 1 justifiable. The pro- cess of case hardening and nitriding minimises the effect of roughness. As such, YR 1 can also be approximated in the roughness range of practice Rz 40 ȝm (R a < 5 m). In the case of gears with a grinding step, whose effect is accounted for, the roughness factor is superfluous. As such, to simplify, for Rz 40 ȝ m the roughness factor YR = 1 is generally assumed. Corrosion, on the other hand, cannot be ignored or permitted. By taking characteristic measurements (Section 7.4) it is possible to partially account for the size of the gear (core strength). However, the influence of the tooth thickness still remains because in the case of s ĺ the bending pulsating fatigue strength crosses over into tensile pulsating strength. As is known, this cannot be accounted for sufficiently by adding the part of the cross section for the stress gradient of the notch radius. As such, this part is primarily registered in a size factor Y X (Figure 6.5/23 and Table 6.5/9). Figure 6.5/23 Size factor YX Table 6.5/9 Size Factor YX (For Endurance Limit, For N 3 106 Load Cycles) Material and heat treatment Normal module mn [mm] Size factor YX Steel and cast steel Normalised or quenched and tempered 5 1 5 < mn < 30 1.03–0.006 mn ≥ 30 0.85 Steel Case-hardened, carbon-nitride-hardened, induction- hardened, flame-hardened, or nitrided 5 1 5 < mn < 30 1.05–0.01 mn ≥ 30 0.75 Cast iron 5 1 5 < mn < 25 1.075–0.015m n ≥ 25 0.7 Note: In cases of approximate calculations to prevent damage due to maximum load (load cycle N 103), for all materials YX = 1 is set. Strength for maximum load For an analysis of ways to prevent damage due to maximum load, for the risk of cracks in hard surface layers (e.g., a case-hardened layer) the strength of the surface layer is decisive, and for plastic deformation of the profile, the yield stress in the core area. In general, both analyses are necessary (refer to Section 6.5.2.5 for values and analyses). 300 6 Load Capacity and Running Performance of External and Internal Gearing Tests showed that in cases of impact loading the endured strain was mostly determined by the magnitude of the stress and to a lesser extent by the speed of loading (negligible) [6.5/27], otherwise the endurance limits would also have to be dependent on the speed. Based on ISO 6336-3, DIN 3990-3, Equation (6.5/38) can be used for the edge layer (detailed in Section 6.5.2.5): FG Nmax FEı ı Y≈ (6.5/38) Here the following applies: )FG Strength limit ( Rm) YNmax Maximum life factor, values in Equation (6.5/37a) ıFE Tooth root pulsating fatigue strength; Section 7.4 (local stress) 6.5.2.4 Safety against Fatigue Breakage Safety against fatigue breakage is1) FEįT FN R Xį Fı / = ıYSY Y Y Y (6.5/39) Here the following applies: (ıFE/YįT) Fatigue strength; approximation of the optimum, smooth test piece YįT Sensitivity factor of the test wheel (Table 6.5/8) TFE Fatigue strength of the gearing according to Equation (6.5/36) or ISO 6336-5, DIN 3990-5 TF Local tooth root stress according to Equation (6.5/34) YN Life factor according to Equation (6.5/37a) YR Roughness factor according to Section 6.5.2.32) YX Size factor according to Table 6.5/9 and Figure 6.5/23 Yį Sensitivity factor according to Table 6.5/8 and Figure 6.5/22 The safety factor SF can be compared to a minimum safety factor SFmin: FF m i n SS≥ (6.5/39a) The factor SFmin is not to be viewed as generally valid. Many aspects must be considered when determining this factor [refer to notes at Equation (6.5/17)]. On the condition that the relevant load is exactly known or can be assumed, and that S F practically only expresses the uncertainty of the calculation method, the following can be considered as adequate: Fmin 1.3S ≥ (6.5/39b) If the analysis of load capacity is to be accomplished through a comparison of the existing stress ı F with the permissible stress ıFP, then Equation (6.5.39c) can be used: FF P σ≤ σ (6.5/39c) The permissible stress (which contains an element of safety!) is FEįT FP N R X į Fmin / = YYYYYSσσ (6.5/39d) 1) In this regard, refer to the note after Equation (6.5/35) 2) The relative factor YRrelT is used in ISO 6336-3 and DIN 3990-3 (Edition 2005) in similar equations; YRrelT = 1 can be set here for Rz 40 m and the assumption YR = 1. 6.5 Analysis of Load Capacity 301 The individual quantities are explained in Equation (6.5/39) (refer to Table 6.5/14 for the calcu- lation). 6.5.2.5 Safety against Permanent Deformation, Cracks or Brittle Fracture Cracks, brittle fracture: It is necessary to differentiate between cracks or brittle fracture and plastic deformation ([6.5/108]). In cases of hard (e.g., case-hardened, nitrided) surfaces, cracks are avoided when the maximum local stress ıFmax in the tooth root notch is less than the local strength, that is, the strength limit FGı. The safety factor SFt results according to the following equation: FStFG Fmaxı ıS= (6.5/40) Note that ı FG is determined by the tensile strength, along with the influence of residual stress. Usually it is simplified as ıFG = Rm,surf. On the surface of the case-hardened layer, the following is true: Rm,surf 2300 (…2450) N/mm2. This value is roughly comparable to the strength of the surface layer of nitrided CrMoV steels and nitrided steels . For the rest of the nitrided, alloyed tempering steels, the value R m,surf 1400 N/mm2 specified in VDI 2737 ([6.5/113]) can be used as a guide. Past experience has shown that cracks do not occur in case-hardened gearing where the root notches are not too sharp (roughly YS 2.5). In more precise analyses, the variable ratio R m/ıFmax starting on the surface must also be checked in the depths. The compressive residual stresses which normally exist (calculation approach: [6.5/117]) are superimposed on stresses caused by loading. They are ignored here and represent an additional element of safety. In the case of very brittle, non-ductile materials (e.g., cast iron) the full effect of the stress peak is felt. Here the element of safety S FSt must also be determined according to Equation (6.5/40) and the tensile strength used for TFG. Plastic deformation: If no risk of cracks exists, for external and internal gearing it is generally necessary to also ensure that no plastic deformation exists on any point of the profile . This is done by comparing the equiva- lent stress with the yield stress. In cases of ductile materials without a hard surface layer, plastic deformation always determines the limit of maximum load capacity. In order to roughly determine the safety (a very rough approximation) against plastic deformation, the height of the tooth profile (coordinate y) is divided into small sections. For each section the equivalent stress (e.g., with von Mises’ maximum distortion energy theory) is determined alongside the coordinates x , y. Figure 6.5/24a Accounting for maximum loads; safety against plastic deformation; example/illustration for internal gearing applies in the same way for external gearing The comparison with the yield stress shows whether and where plastic defor-mation exists (refer to Figures 5.1/13a) to c); FEM result). In the case of helical gearing with larger helix angle ()2 2 vb dsx,y x,y x,y x,y() = () ()+ 3()σσ + σ τ ()dı ,0xy < p0.2 FSt( , ) vı(x,y)xyRS = (6.5/41) It must be noted that even gearing with a hard surface layer is subject to plastic deformation. 302 6 Load Capacity and Running Performance of External and Internal Gearing In cases where the approach is extremely simplified, the notch effect is ignored and only a compari- son of the maximum nominal stress occurring at the tooth root with the yield stress of the core material is carried out (Equation 6.5/41a). However, this can only be viewed as a rough guide and not as a valid analysis because among other things the zones above the tooth root area are not included, and the risk of cracks in the case of hard surface layers is not examined. p0.2 FnmaxFSt ıR S≈ (6.5/41a) Here the nominal stress ıFnmax is to be calculated according to Equation (6.5/35) with Ft = Ftmax and R p0.2(ı0.2) used for the core material in the root cross section. Warning: In cases of small numbers of teeth and a large profile shift (small tooth tip thickness), sometimes plastic deformations go unnoticed with Equation (6.5/40/c), especially in the tooth tip region. There is another common approximation according to ISO 6336-3, DIN 3990-3. Based on Equation (6.5/39), the endurance limit is converted with the factor YNmax to the strength in N 103, and the notch sensitivity is accounted for with the so-called static sensitivity factors. For safety during load peaks the following results: FEįT St F Nmax R X Fmaxı/= ıYSY Y Y (6.5/41b) Here ıFmax is the maximum value of the local stress that occurs during the load history of the gear. The term (TFEYNmaxYX/Y T) is the strength, expressed by the maximum sustainable local tooth root stress at a one-time load (valid as an approximation in cases of load cycle numbers N 103) with the roughness factor YR = 1. The variable YįT is valid for the test wheel with which the endurance limit ıFE was determined. For YįT, information can be found in ISO 6336-3 and DIN 3990-3 (refer to Table 6.5/8), and values for YNmax in Equation (6.5/37a). Figure 6.5/24b As a guide: Sensitivity factors Y įSt, YįStT, loads with N 103 (quasi-static sensitivity factors) according to ISO 6336-3, DIN 3990-3 for GG Spheroidal graphite cast iron, tooth root surface untreated, pearlitic annealed or quenched and tempered N Steel, tooth root surface nitrided V Steel and cast steel, tooth root surface quenched and tempered EG Steel, tooth root surface case-hardened, carbonitrided, flame- hardened or induction- hardened St Steel with distinct yield strength To prevent damage, at maximum strain safety should be FSt FSt min SS≥ (6.5/42) 6.5 Analysis of Load Capacity 303 On the condition that SFSt only expresses the uncertainty of the calculation method, SFStmin = 1.3 can be viewed as adequate [refer to Equation (6.5/17) for further remarks]. More precise analyses must be based on the static distribution of stress and the strength. It is recommended to provide the valid proof of preventing the damage caused by peak loads according to Equation (6.5/40) and respectively Equation (6.5/41b). In the case of calculating the equivalent stress in the tooth profile with FEM or BEM, Equation (6.5/41) has to be used. Equation (6.5/41) can be a rough approximation by using elementary calculated stress proportions ı b, )d, Ĳs and Equation (6.5/41a). 6.5.2.6 Load Capacity of the Tooth Root in Cases of Elastically Designed Gear Rims The purpose of the following explanations is to illustrate the special issues which arise in the case of elastically designed gear rims, to further develop the notion, and to present an approach for approximate solutions that seem feasible [6.5/106], [6.5/109]. Their purpose is not to principally replace precise calculations using FEM or BEM, especially on the influence of adjacent contours. At the forefront here is the strain on gears or gear rims in the case of internal gears . Knowledge of the stress resultants (bending moment in particular) in the gear rim is assumed as a prerequisite. These can be determined either explicitly through calculations or with coarsely meshed 3D FEM models. For the breakdown of load parts in components, factors for determining local stresses are given. For external gearing these values can be used to estimate the effect of discrete dominant parts (e.g., rim bending moment, shrink stress). In the case of relatively thin elastically designed wheel rims (refer to Figure 6.5/25 for examples), the stresses in the rim have an influence on tooth root strain [6.5/28] to [6.5/31]. a) Introduction Figure 6.5/25 Elastically designed gear bodies (examples): a) external gear with rim, web, hub; b) external gear, idler gear with thin walls; c) internal gear, suspended in gearing; d) internal gear, suspended by housing with thin walls (on both sides); e) internal gear, suspended by elastic pipe shaft; d) internal gear, suspended by housing with thin walls (on one side) The method given as follows takes into account the influence of strain on the gear rim on the load capacity of the tooth root in the case of internal gearing. In its calculation, generally the spatial state of stress must be considered. Since in a first step only the nominal stresses or internal forces and moments are to be determined, the amount of effort required – with the (three-dimensional) finite 304 6 Load Capacity and Running Performance of External and Internal Gearing element method for example – is still relatively small. In special cases in a good approximation a two-dimensional case exists, and the internal forces and moments can be determined singularly. The internal forces and moments or the nominal stresses in the gear rim are used as the basis. In the two-dimensional case they are easy to calculate. For ring-shaped internal gears (internal gear rims) whose coupling to the neighbouring components is negligible, it is possible to give explicit equations for the internal and moments in the wheel rim. Here the external load – the tooth force – is broken down into a radial and a tangential component in order to be able to trace the resulting strain back to a superposition of elementary cases. The effect of the radial component on the strain on the gear rim can be approximated by a pure bending moment. The curve of the internal forces and moments (bending moment) or the curve of the nominal stress must be considered for the whole cycle, that is, for one complete gear revolution, or in the case of uniformly ordered multiple meshing of equal work, between two gear meshes. The whole cycle is necessary because, especially in the case of wheel rims with internal gearing, significant stresses can form outside the meshing area, offset by a larger angle, some even with an opposite sign (refer to Figure 6.5/26). The effect of the axial loads in the case of helical gearing is ignored here, which at roughly 15 is possible. Figure 6.5/26 Stress gradient in an elastically designed internal gear rim (inner diameter of the ring) in the case of extremely soft connecting conto urs (suspension); also refer to [6.5/106], [6.5/109] , [6.5/113], [6.5/115]: a) stress due to tangential load component FtB; b) stress due to radial load component FrB; c) stress due to normal load Fb (superposition of FtB and FrB); d) stress due to centrifugal force (and/or shrinkage), refer to [6.5/110], [6.5/113], [6.5/119]. The stresses due to tooth loads are superposed by tensional stresses, caused by centrifugal forces and shrink stresses [6.5/119], [6.5/110]. The longitudinal forces caused by the applied tooth load are to be taken into consideration in more precise analyses [6.5/109]. From the nominal stress components, for a bending load, tensile load, and tooth tangential load, the local stresses are determined and superposed with the help of stress concentration factors, which are determined in a two-dimensional approach for this single strain component on tooth segments. The methods most commonly used are based on the approximation that the local stresses which result from the individual load components or load parts are calculated and superposed on the 60 tangent (Appendix 6). The influence of mean stress is taken into account 6.5 Analysis of Load Capacity 305 in the endurance limit. A more precise approach is done by a point-wise determination of the local elements of safety in the tooth root fillet because the 60 tangent may be in the centre, but it does not always correspond to the actual position of the stress maximum [6.5/109]. The analysis of load capacity applies to both the breakage path (or crack) through the tooth root and through the gear rim. For internal gearing, determining a new reference cross section on the 60 tangent instead of on the otherwise typical 30 tangent has the effect of conforming to the inclined position of the involute in the vicinity of the tooth root, which differs compared to external gearing, and to the smaller root fillets which often exist, which both require a stress maximum that is shifted closer to the tooth gap center. The connecting contour (elasticity of the mount) has a very large influence on the deformation and stress state of internal gear rims. Based on the solution for the freely deformable ring, this can be approximately registered by clamping factors [6.5/106]. Their dependence on the dimen-sions is given for connections on both sides by cylinder shells. Ignoring the support effects of the connection contour leads to the lowest load capacity. b) Basic equations Analysis of load capacity in connection with fatigue breakage The analysis of load capacity to prevent damage caused by fatigue is done by comparing the occurring tooth root stress ı F with the tooth root stress that is permissible for fatigue load ıFP or by determining the safety factor SF in general for the sides of the tooth known as the tension side and the pressure side . The stress ı F is the double amplitude which occurs during a load cycle (e.g., revolution), expressed in the form of local stress. The term stress cycle refers to the period (given for example by the rotating angle Q of the gear) in which the stress gradient repeats at an assumed constant tooth load. For example, during a mesh, the rotating angle of a stress cycle is Q = 2ʌ or Q = 360 , and in the case of three equal meshes offset by 120 each with mating gears (such as with planetary gear transmissions) Q = 2ʌ/3 or Q = 120 . The influence of the altered mean stress component σFm on the endurance limit σFEm must be taken into consideration [6.5/114], [6.5/113]. In the case of load spectra, it is important to use suitable damage accumulation hypotheses. For the analysis of load capacity, either an equivalent load F t specific to it must be assumed, whose harmful effects are the exact equivalent of the full load spectrum, or with Ft(j) the double ampli- tudes ıF(j) associated with the stresses must be determined for the individual load steps ( j), and from that, the equivalent stress ıF calculated. Analysis of load capacity regarding damage due to maximum stress Maximum stress can cause either overload fracture, permanent deformation or cracks. Both maximum occurring tensional stress and the largest absolute amount of compressive stress (of the bending) are used. Both stresses must be considered. Generally there is also a difference between the bearable maximum tensional stress and the maximum bearable compressive stress. The analysis of load capacity for preventing damage due to maximum stress σ Fmax is accom- plished by comparing the maximum stresses which are occurring with the permissible maximum stresses σ FPmax or through the safety factors SFSt. In the analysis of load capacity for the largest tensile strain (bending and tension; FF m a xı=ı > 0) the largest of the stresses is to be considered, which are named as follows: 1. The largest local stress, which occurs on the tension side of the tooth, on which the tooth load is applied: (t) Fmax I = ıı (6.5/42a) with(t) Iσ according to Equation (6.5/49). The following is true in extreme cases: (t) I < 0σ . This analysis can then be omitted. 306 6 Load Capacity and Running Performance of External and Internal Gearing 2. The largest local stress which, compared to the field of action, can occur offset by a larger an- gle in the gear rim due to the course of the bending moment: (t) Fmax III = ıı (6.5/42b) with (t) IIIı according to Equation (6.5/50). This stress does not manifest itself in the case of rela- tively thick gear rims and external gearing with disc-shaped wheel bodies and in the case of sin-gle- and double-web gears. In the analysis of load capacity, for the largest absolute amount of compressive stress, the largest absolute amount of local stress, which occurs on the pressure side of the tooth that the tooth load is engaging, is to be used: (c) Fmax I = | | σσ , σ(c) I according to Equation (6.5/49) (6.5/42c) c) Bending moments and centrifugal loading in the gear rim of ring-shaped internal gears The bending moments and tensile forces in the gear rim are to be determined for the components of tooth loads (tangential loads and radial loads), centrifugal loads, and loads by connecting elements or connections (such as screws, bolts). Generally the spatial state of stress must be considered. The contours adjacent to the gear rim, such as cylindrical elements, must be included. To be determined at a minimum are (refer to Figure 6.5/27) the internal forces and moments in the gear rim at the root transition (1), (4) the loaded tooth, also on the neighbouring teeth (3), (5) and the bending moment maximum (2) outside the zone in which the tooth load and the tensile force acting there (e.g., due to centrifugal force) are applied. The outer point of single tooth contact B is to be used for the application of the tooth load. In the case of external gears with solid gear bodies, no strain or no further important strain will occur outside the zone where the tooth load is applied (point 2, TIII 0). Figure 6.5/27 Points where bending moment (and longitudinal forces) are to be calculated in the internal gear rim (refer to Table 6.5/12) In cases of web gears (Figure 6.5/25a), generally a negligible strain can be expected compared to ring-shaped gear bodies with internal gearing. For internal gears, in some cases it can be approximated that a two-dimensional stress state exists, and it can be assumed that the load caused in the housing by the supports is applied distributed 6.5 Analysis of Load Capacity 307 uniformly on the gear circumference. If this condition is met and if the adjacent parts (e.g., fasten- ers) do not significantly influence the stress state of the gear approximated by a simple model, it is possible to give explicit relationships for the internal forces and moments. As follows, the internal bending moments are given for the very softly mounted elastically designed internal gear. For external gears this is usually not possible. Of all the reactions produced by the tooth loads in the gear rim, as an approximation one can only consider the bending moment. Neglected here are still the rim bending moments due to axial tooth load. In the case of helix angles β 15 , they have no determinative effect on the resulting stress [6.5/109], [6.5/115]. Initially, the longitudinal forces (circumferential direction) in the gear rim will likewise be ignored. Only starting with a number of planets of p > 4 do they begin to have a significant influence on the strain in the gear rim or the tooth root. According to Equations (6.5/43) and (6.5/44), with a uniform distribution of load on the p meshing, the radial component F rB and tangential component FtB result in the bending moments in the gear rim of the internal gear. The following applies for the radial component (Figure 6.5/28b) (symmetrical course of the mo- ment): ()()() ()m br B r Bcos /,ĳ22 2s i n /p d pMF Fp π− ϕ=− − π π (6.5/43) The following applies for the tangential component (Figure 6.5/28c) (asymmetrical course of the moment): ()()() ()() m m bt B t Bsin / / 2 ,ĳ22 2sin /pp d H dMF Fp p π− ϕ − ϕ π =− ⋅ − − ϕ π ϕ π (6.5/44) The result for the normal tooth force yields () ( ) ( )br Bt B br B bt B ,,ĳ ,ĳ ,ĳ MF F MF MF =+ (6.5/45) With an uniform application of the supporting forces and supporting torques, the bending moment in the gear rim is independent of whether the internal gear is suspended by gearing or by a tubular structure, if these are very elastic. The Equations (6.5/43) to (6.5/45) and Figure 6.5/28 contain the following: dm/2 Radius of the neutral axes p Number of planetary gears q Tangential line load due to support or suspension; e.g., spline gearing or tubular structures 222m tBdqR p F Hπ= − FtB Circumferential load at the outer point of single tooth contact B FrB Radial load at the outer point of single tooth contact B H Distance of the load application to the neutral axes ĮK Pressure angle of the spline gearing Ȗ Angle for symmetry line ĳ Angle coordinate The axial load component applies a rim bending moment to the gear rim. In the case of β > 15 and b/sR < 5, for example, the effect this has on the relevant stresses is no longer insignificant. (For an orientation on the surrounding issues, refer to [6.5/127] and [6.5/128].) 308 6 Load Capacity and Running Performance of External and Internal Gearing Figure 6.5/28 Elementary calculation of the internal bending moments for ring-shaped gear rims with elastic suspen- sion without the clamping influence If the gear is a ring-shaped structure (as is often the case with internal gears) and if it rotates with the circumferential speed /, the following nominal mean stress component ım1 results from the centrif- ugal forces : 2 R m1 R0.5 sh s+σ=/N (6.5/46) or with steel 2 R m1 2 R0.5 N78.5100 m / s mmsh s+ σ≈ ⋅ ⋅ / (6.5/46a) with /= 2 2πndm, h tooth height, sR gear rim thickness (refer to Figure 6.5/37). (6.5/46b) At a constant speed, the centrifugal forces generate statically acting stresses in the gear rim, which are to be heeded as a strain on the gear body. Moreover, they have an influence on the load capacity of the tooth root. In the case of ring-shaped gear bodies which do not receive any significant support from adjacent elements (e.g., internal gears with a soft coupling), at a circumferential speed of roughly / x 100 m/s the tangential stress from centrifugal force reaches an order of magnitude of ım1 x 100 N/mm2. In the case of solid gears (e.g., pinion shafts), the influence of the stress from centrifugal force becomes noteworthy at around / x 250 m/s [6.5/119]. If there are defects in the material (dross), however, the erratic growth of cracks even at low speeds can lead to serious dam-age (gear body breakage), which, in critical systems, would have to be subjected to tests in terms of crack breakage mechanics ([6.5/120] among others). 6.5 Analysis of Load Capacity 309 d) Stress concentration factors [6.5/109], [6.5/113] (Stress) concentration factors are used to determine the local stresses from the internal loads and moments or nominal stresses. There are three concentration factors to define: Y SFt due to the tangential load component of the tooth Y SMb due to a bending moment in the gear rim (The effect of the radial component of the tooth FrB is likewise reduced to a bending moment.) Y SZ due to tensional stresses in the gear rim The definitions of these concentra tion factors can be found in Figures 6.5/29 and 6.5/30. The curva- ture of the gear bodies was assumed to be small and was ignored (for determination on rack-like structures). For their determination in terms of numbers, for internal gearing Figures 6.5/34 to 6.5/36 and Appendix 6 (Figures 6.5/31 to 6.5/33 for external gearing) or as an approximation Equa-tion (6.5/47), which apply for the respective maximum , can be used: () ( )SFt SMb SZ R Fn Fn Fn ș,ımax (, ,) / 2/bdYY Y a ss c s ≈+ N (6.5/47) The coefficients a to d are listed for internal gearing (IG) in Table 6.5/10 for the maximum stress, and in Table 6.5/11 for the stress at the 60 tangent [also included for external gearing (EG)]. Since the stress maxima of the load parts occur at different points of the root fillet, based on past experi-ence the addition of the stresses at the 60 tangent appears to be a suitable approximation (refer to Appendix 6 for equations). Figure 6.5/29 Quantities for nominal stresses and notch parameter (point B: outer point of single tooth contact) Here, the outer point of single tooth contact was generally labelled as B. It is the point of single tooth contact nearest the tooth tip (which is otherwise labelled as D on the mating gear). 310 6 Load Capacity and Running Performance of External and Internal Gearing Figure 6.5/30 Definition of the stress concentration factors YSFt, YSMb, YSZ Figure 6.5/31 Stress concentration factor YSFt for external gearing 6.5 Analysis of Load Capacity 311 Figure 6.5/32 Stress concentration factor YSMb for external gearing Figure 6.5/33 Stress concentration factor YSZ for external gearing 312 6 Load Capacity and Running Performance of External and Internal Gearing Figure 6.5/34 Stress concentration factor YSFt for internal gearing Figure 6.5/35 Stress concentration factor YSMb for internal gearing 6.5 Analysis of Load Capacity 313 Figure 6.5/36 Stress concentration factor YSZ for internal gearing Table 6.5/10 Coefficients Belonging to the Approximation Equation (6.5/47) for Stress Concentration Factors for External Gearing (EG) and Internal Gearing (IG) for Determining the Respective Maximum Local Stress Factor Coefficient from Equation (6.5/47) a b c d EG IG EG IG EG IG EG IG YSFt 0.306 0.324 -2.551 -2.561 1.332 1.418 -0.3446 -0.3388 YSMb -0.266 -0.201 -0.928 -1.102 1.877 1.595 -0.2251 -0.3045 YSZ -0.17 -0.19 -1 -1 1.7 1.66 -0.26 -0.28 Table 6.5/11 Coefficients Belonging to the Approximation Equation (6.5/47) for Stress Concentration Factors for External Gearing (EG) and Internal Gearing (IG) for Determining the Stress at the 60 Tangent (Refer to Appendix 6) Factor Coefficient from Equation (6.5/47) a b c d EG IG EG IG EG IG EG IG YSFt 0.362 0.290 -2.252 -2.727 0.997 1.092 -0.4202 -0.4188 YSMb -0.152 -0.147 -1.116 -1.094 1.279 1.220 -0.3700 -0.3908 YSZ -0.13 -0.11 -1 -1 1.25 1.24 -0.36 -0.36 e) Clamping factors for internal gears (internal gearing) In the preceding considerations on the determination of stress resultants, it was assumed that parts (fasteners) adjacent to the internal gear exercise no relevant influence on the stress state of the inter-nal gear rim. This allowed us to approximately consider the reactions in the form of bending mo-ments generated by the tooth loads, in a gear rim that is virtually freely deformable. However, if we 314 6 Load Capacity and Running Performance of External and Internal Gearing assume that a freely deformable gear rim occurs relatively rarely, then we are faced with the neces- sity of including in the determination of stresses the support effects of contours adjacent to the internal gear. Setting up an equation to calculate the bending moment for an arbitrary gear body is not possible with an adequate level of precision. For the common case of a direct connection between the internal gearing and an adjacent symme-trical housing, factors have been determined which take into account the support effects of the contours adjacent to the internal gearing (clamping factors). In this way, with arbitrary gearing data both for a very elastically mounted internal gear rim as well as for internal gearing symmetrically connected to the housing in the form of cylinder shells, it is possible to approximately estimate the tooth root strain while also taking the wheel rim into account [6.5/106]. These “clamping factors” are dependent on the geometric conditions of the internal gear rim and the symmetrically adjacent cylinder shells. They account for the influence of the support of the contours adjacent to the internal gear as opposed to the freely deformable gear rim. Three clamping factors are defined: f Fr due to radial component FrB in the gear rim f Ft due to tangential component FtB in the gear rim f res due to the normal tooth load as the result of the radial and tangential components in the gear rim Equations (6.5/48a, b) can be used to approximately determine their numerical value (for this refer to Figure 6.5/37): SR3Fr mS0.32/2lsfdh=⋅⋅ (6.5/48a) SR6Ft mS0.65/2lsfdh=⋅⋅ (6.5/48b) SR3Res mS0.37/2lsfdh=⋅⋅ (6.5/48c) Limit of validity : If the following results for the expression under the root of Equations (6.5/48a, b) SR mS0.4/2ls dh<⋅ then for fFr = fFt = fRes = 0 is to be set (transition to an “infinitely thick” gear rim). If factors f > 1 result from Equations (6.5/48a, b), then these are to be set as f = 1 (no effect from the connecting con- tours on ıF): ls Length of connecting shell h s Thickness of connecting shell s R Gear rim thickness d m/2 Radius of the neutral axes bK Width of the gear rim Figure 6.5/37 Elastic internal gear rim with connecting contour 6.5 Analysis of Load Capacity 315 f) Double amplitude, mean stress, maximum stresses Table 6.5/12 first gives an overview of the points of important stress in an elastic internal gear. Table 6.5/12 Points of Important Stress on the Point Marked as Tension Side (t) and as Compression Side (c) of the Tooth during a Stress Cycle of an Internal Gear (See Figures 6.5/26 and 6.5/27) Point (t) (“tension side”) • ıI(t) Stress of the loaded tooth (tension side) • ıII(t) . ıIII(t) Stress of the tooth not directly loaded by external loads; ıIII = ıK (rim stress), in distance of several pitches from the loaded tooth a) Neighbouring tooth b) Offset by a larger angle Point (c) (“compression side”) Note: The underlined stresses are usually the ones decisive for the double amplitude. • ıI(c) Stress of the loaded tooth (compression side); ıI(c) < 0 • ıII(c). ıIII(c) Stress of the tooth not directly loaded by the external loads, ıIII = ıK (rim stress) a) Neighbouring tooth ıII(c) b) Offset by a larger angle ıIII = ıK (rim stress), ıK > 0 The stresses on the loaded tooth (index I), that is, on the tooth on which the load to be transmitted is applied, are for internal gearing (internal gear); refer to Table 6.5/12: 316 6 Load Capacity and Running Performance: External and Internal Gearings () ()() ()() ()tF t F I IF a SFt n t d Ft FI Fa Sa n Fr b rB,D FI SMb m1 SZ 2 R6Ff KYY Y Ybm Ff f KYY Y Ybm fM F K YYbsεβ εβ+ − + − −+ −+σ= − σ (6.5/49) Here, Mb(FrB.D) is the ideal rim bending moment for an internal gear with soft suspen- sion, Equation (6.5/43), due to the radial component FrB.D of the tooth load being ap- plied on the outer point of single tooth contact B or D. Tension side (t): upper sign: ()σσIIt= ; compression side (c): lower sign (in brackets): ()σσIIc= Here the following applies: KFI Load factor for the directly loaded tooth KFI accounts for the additional loads and the influences on the change to distribution of stress: KK K K KFI A v F I F I=αβ (6.5/49a) with KFĮI and KFȕI according to, among others, Equations (6.4/41) to (6.4/43) and (6.4/47), giv- en in Section 6.4 YSFt Concentration factor YSFt accounts for the increase in stress in the tooth root due to the tangential load component Fta in the case of load application on the tooth tip, with reference to the nominal tooth root bending stress Y SFt for internal gearing: Figure 6.5/34; approximation Equation (6.5/47). Mb(FrB) Bending moment in the gear rim due to the radial load component FrB in the tooth space on the loaded tooth during load application on the outer point of single tooth contact; among others Equation (6.5/43) s R Gear rim thickness, Figures 6.5/34 to 6.5/36 YSMb Concentration factor YSMb accounts for the increase in stress in the tooth root due to the bending moment in the gear rim, with reference to the nominal bending stress of the gear rim. Y SMb for internal gearing: Figure 6.5/35, approximation Equation (6.5/47) ım1 Tangential tensional stress in the gear rim that is independent of the transmitted power (nominal stress), due to centrifugal forces and with non-positive connections (e.g., shrink-fit or circular spring clamping elements) Y SZ Concentration factor YSZ accounts for the increase in stress due to tensile strain, with reference to the nominal gear rim stress. Y SZ for internal gearing: Figure 6.5/36, approximation Equation (6.5/47) YFa Tooth form factor, Equation (6.5/34c) Yİ Overlap factor, Equation (6.5/34e) For spur gearing, Yİ registers the bending lever arm influence on the nominal stress and the stress concentration, and for helical gearing the influence of the length of the line of contact. YSa Stress concentration factor; YSa accounts for the increase in stress in the tooth root due to the tangential force component Fta in the case of load application on the tooth tip, with reference to the nominal tooth root bending stress (refer to Section 6.5.2.1). Y ȕ Helix factor, Equations (6.5/34f and g) fFr Clamping factor for internal gearing fFr accounts for the reduction in tooth root stress due to the adjacent parts in the internal gear for the radial force component, Equation (6.5/48a), ( fFr = 1 for calculation without rim influence). fFt Clamping factor for internal gearing fFt accounts for the reduction in stress in the tooth root due to the adjacent parts in the internal gear for the tangential force component; fFt = 1 for calculation without rim influence. fd Factor for approximately accounting for the compressive stress which occurs with FrB Tension side fd = 1.0; compression side fd = 1.15 6.5 Analysis of Load Capacity 317 The stress at the tooth root of unloaded teeth, that is, of teeth without directly applied load and which are not directly influenced by the root stress of neighboring loaded teeth, but where the rim bending moment has an influence, can be calculated according to Equation (6.5/50) for internal gearing : () Re b rB,D tB,D FIII SMb m1 SZ 2 R6, sfM F F KYYbsΙΙΙσ= ⋅ + σ (6.5/50) Here the following applies: M b(FrB,D; FtB,D) Bending moment in the tooth rim during load application on the outer point of single tooth contact B or D; Ideal rim bending moment [soft suspension Equation (6.5/43)] owing to radial and tangential load K FIII Load factor at the point 2 of the gear circumference; refer to Figure 6.5/27. It accounts for additional loads and influences on the change to the distribution of stress. Since the point being considered is situated several tooth pitches away from the loaded tooth, the following is set in the initial approximation for the rim stress: FIII FĮ III F III 1 KK K β === YSMb, YSZ Refer to the explanation for Equations (6.5/49 and 6.5/49a). f Res Clamping factor in the case of internal gearing for MbRes = Mb (FrB, FtB); Equation (6.5/48c) f Res Accounts for the reduction in stress due to the suspension (neighbouring contour) (Step 2 according to VDI 2737), fRes = 1 would be for calculation without rim influence For one gear revolution or one stress cycle (Figure 6.5/26) the largest difference ıF between maxi- mum and minimum stress for the tension side and the pressure side are to be calculated. This double amplitude ıF is as follows for the tension side (t) and compression side (c): Tension side (t): (t) (t) (t) FF I I I Iσ = σ= σ− σ with (t) IIIσ= Kσ 0 (rim stress), (t) Iσ 0; (6.5/51) Compression side (c): (c) (c) (c) F F I IIIσ= σ = σ − σ with (c) IIIσ= Kσ 0 (rim stress), (c) Iσ 0; (6.5/52) (Refer to Table 6.5/12.) The mean stress component ıFm1 is caused by the pre-load, due to the effects of centrifugal force and possibly the connections (e.g., shrinkage, circular spring clamping elements); refer to [6.5/119]: Fm1 SZ m1 Y σ= σ (6.5/53) Note that ıFm1 is not dependent on the operating load. The mean stress component ıFm2 results as the arithmetic mean value of the extreme values of stress during this gear revolution or during a stress cycle, and its size is different for the tension and com-pression sides. Here, ı Fm2 is proportional to the operating load: I III Fm22σ+ σσ= (6.5/54) Note that ıFm2 is proportional to ıF; ıFm2/ıF = constant. The components ( ıFm1, ıFm2) are treated differently when determining the endurance limit (refer to Figure 6.5/38) because they are not de-pendent on the operating load in the same way. The decisive stress ı Fmax which is the maximum for incipient cracks in cases of hard surface layers is to be calculated with the maximum occurring load and the associated factors KFI and KFIII. And ıFmax is the maximum value of the three stresses shown each in brackets (…+…) in Equation (6.5/55): () ()(t) (c) Fmax Fm1 I Fm1 I Fm1 III Max (ı+ı);ı+ı;ı+ ı σ= (6.5/55) 318 6 Load Capacity and Running Performance: External and Internal Gearings Notes: In some cases the tensional stress ıI(t) on the tension side of the loaded tooth can be less than the tensional stress ı III occurring offset by a larger angle. In this case the smallest safety factor will not be found on the tension side of the loaded tooth, rather either at point 2, Figure 6.5/27, or on the compression side, point 4. Table 6.5/12 gives an over-view of the important stress points and the initial quantities to be determined. g) Strength, Cyclic loading The strength at the tooth root ıFE, which is typically present as pulsating fatigue strength and expressed as the maximum local stress that can be permanently tolerated, must be converted, proportionate to the dependency on mean stress, to the actual conditions ( TFEm) that exist. Explanations for the mean stress components ıFm1 and ıFm2 are contained in sub-section f). Considering that the value ıFm1 = constant (speed assumed as constant) and in contrast the ratio ıFm2/ıF = constant is true (rise in external load, variation I), from the fatigue strength diagram, for example, the Smith diagram (Figure 6.5/38), the endurance limit ı FEm is obtained. Mathematically, the current endurance limit ıFEm, which is dependent upon the mean stresses ıFm1 and ı Fm2, can be calculated according to Equation (6.5/56). It is presumed that in addition to the local mean stresses ıFm1, ıFm2 and the local stress ıF expressed as a double amplitude, the (pure) tooth root fatigue strength under completely reversed stress ıFEW and the pure pulsating fatigue strength TFE are known. According to Figure 6.5/38, the endurance limit for the area (C 0Cu E0ƍEuƍ) is FEW Fm1 FEW FE FE FE FEm FE Fm2 FEW FF E21 2112σσ σ −⋅ − σσ σ σ= σ ⋅σσ ⋅− + σσ (6.5/56) and ı FEm = 1.6ıFE must be set for ıFEm > 1.6ıFE. The literature at [6.5/113] and [6.5/114] contains information on ıFEW/ıFE. The standard range is ıFEW/ıFE = 0.6 …(0.7). Here are the results for the following: ıFEW/ıFE = 0.6: Fm1 F FEm FE Fm 2 F1.2 0.4 10 . 4− +σ σσ= σσ σ; ıFEW/ıFE = 0.7: Fm1 F FEm FE Fm 2 F1.4 0.8 10 . 8− +σ σσ= σσ σ (6.5/56a) The Equations (6.5/56) and (6.5/57) apply if (6.5/56b) and (6.5/56c) are met: () FEm Fm2 F Fm1 e 0.5 < fR σ+ σ σ − σ (6.5/56b) () FEm Fm2 F Fm1 e 0.5 < R σ+ σ σ + σ (6.5/56c) The safety factor SF must be calculated according to Equation (6.5/56) or (6.5/56d) with the strength FEmσ which is dependent on the mean stress , and the double amplitude TF of the acting stress [Equations (6.5/51), (6.5/52)]. Note: The verification of the safety factor SF with the endurance limit according to Equation (6.5/56) represents a load reserve. If an increase in load is ruled out and if the safety factor is only aimed at the uncertainty of the calculation assumptions, then the endurance value can be calculated according to Equation (6.5/56d) (varia-tion II). ()FEW FEm FE Fm1 Fm2 FEıı 2 ıȥ ı +ıı =− with FEW FEıȥ42ı=− (6.5/56d) with ȥ0.8= for FEW FEı0.7ı= ; ȥ0.4= for FEW FEı0.6ı= If the condition (6.5/56b) is not met, the fatigue strength will be determined by the compression yield strength (area E o, Eu, 0u, Figure 6.5/38). The fatigue strength then results in eF m 1 FEm Fm2 Fııı1 2ıfR⋅+= + (6.5/57) If Equation (6.5/56c) is not satisfied, the yield stress will be overstepped. Safety against fatigue fracture is calculated according to Equation (6.5/39); the following is to be used: ıFE(ıFEm). The stress is to be determined according to Equation (6.5/51) or (6.5/52). 6.5 Analysis of Load Capacity 319 Figure 6.5/38 Determination of ıFE for ıFm1 = constant and ıFm2/ıF = constant in the Smith diagram (endurance limit) Here the following applies: ıFE Endurance limit in the case of pure pulsating load caused by the endurance limit expressed by local stress, ascertained in a test and converted to an ideal, smooth test piece (double amplitude); ı Fm = 0.5ıF ıFW Tooth root endurance limit in the case of a reverse load (amplitude); ıFm = 0 ıFm1 Local tooth root mean stress, independent of the tooth loads to be transmitted ( ıFm1 = constant in the case of a change to the transmitted torque) ıFm2 Local tooth root mean stress, dependent on the transmitted tooth loads ( ıFm2/ıF = constant) ıF Local tooth root stress; absolute amount of the difference between maximum and minimum stress during a gear revolution (stress cycle) f Factor for increase of the compression yield strength, f = 1.0 to 1.1; Strain in cases of maximum stress (strength, safety) Note that ıFG is the strength in a single occurrence of strain (or respectively in the case of a load cycle number N N0, which, compared to N = 1, doesn’t yet result in any noticeable drop in strength). N0 = 103 is generally viewed as even safer value. For tensile and compressive stress, ı FG usually has different sizes, especially in the case of cast materials . The safety of the gearing against damage due to maximum load is calculated according to Equation (6.5/40) or (6.5/41), with the maximum tensile and compressive stress that occurs [Equation (6.5/55)]. In the case of wheel rims , it is also recommended to calculate safety against an exceeding of the yield stress or in case of harded surface layer against the tensile strength, due to the maximum nominal stress of the gear rim . To do this, the bending moment in the wheel rim, Equation (6.5/45), (possibly with the longitudinal forces in the gear rim superimposed [6.5/109]) and the cross section of the gear rim, defined by the root radius and the radius of the outer contour, are to be used. 320 6 Load Capacity and Running Performance: External and Internal Gearings 6.5.3 Practical Approach for the Basic Analysis of Load Capacity For a practical analysis (without the influence of elastically designed gear bodies), this section brings together the equations to analyse fatigue damage (p its, tooth root breakage) and the damage resulting from maximum load (permanent deformation, incipient cracks or breakage). 6.5.3.1 Approach for Analysing Flank Load Capacity: Pitting and Damage at Maximum Load (Methodological Process) As follows, the equations for recalculating pitting load capacity in cases of fatigue stress, and furthermore for preventing unacceptable permanent deformation or incipient cracks at maximum load, will be brought together. I General If the materials, cases, core hardnesses or numbers of rolling cycles differ, the load capacity of the tooth flanks must be established for both gears of a pair. II Flank load capacity (pitting) In a more precise approach, the analysis of load capacity for preventing fatigue damage must include a comparison of the stress depth profile with the strength depth profile (refer to Appendix 7). The stress depth profile is registered by an equi valent stress. It is ascertained from the three stress components which result from the Hertzian contact. As a first step, the approach typical until now (in analogy to ISO 6336-2, DIN 3990-2) can be used, wherein the stress component ( ı H Hertzian contact stress) resulting right on the surface, perpen- dicular to the contact surface, is compared with an endurance limit ascertained in tests ( ıHlim). This method is briefly explained as follows: The analysis can be done by comparing • the calculated contact stress with the permissible contact stress HH 0 HH P K σ= σ ⋅ ≤ σ (6.5/58) • or the safety factor with the minimum safety factor. () SZZZZ Z Z SHHN HRLv WX H =≥σ σlim min (6.5/59) The contact stress ıH0 of the deviation-free gearing without additional load is t H0 E H B,D 11 FuZZZ ZZbd uεβ+σ= ⋅ (6.5/60) The following applies for the load factor KH: HA H v H HK KK K K α β = (6.5/61) The following relationship exists between the permissible contact stress ıHP and the tooth flank endurance limit ıHlim: ()Hl im N HP R L v W X HlimZZZZ Z ZSσσ= ⋅ ⋅ (6.5/62) 6.5 Analysis of Load Capacity 321 If a load spectrum exists, then ıH must be calculated (ı Hj) for each load step j of the equivalent spectrum ( N < NHlim). Here the factors KHvj, KHĮj, and KHȕj must be newly determined for each load step ( Ftj) because they are load-dependent. The variable ıH is determined in accordance with Equation (5.3/11) from the values ıHj belonging to the load steps j (refer to Figure 5.3/5, Section 5.3): () HH H jσ= σσ (6.5/63a) As an approximation, ı H can also be determined with the help of an equivalent tangential load FtH, Equation (5.3/15): () HH t H F σ= σ (6.5/63b) III Preventing plastic deformation or brittle fracture on the case under maximum load In a more precise approach, the analysis of load capacity for preventing plastic deformation on the tooth flanks must be accomplished by comparing the stress depth profile with the strength depth profile (refer to Appendix 7). Here the stress depth profile is determined as the equivalent stress from the three stress components which result from Hertzian contact (with von Mises’ maximum distortion energy theory). The approximation given as follows is a first step. The analysis is done by comparing the safety factor for the maximum load with the minimum safety factor: HSt HSt HSt min Hm a xSSσ=≥σ (6.5/64) For notes on ıHmax and ıHSt refer to Equation (6.5/21). An additional test, which is focussed on the deformation of the surface (similar to the static load capacity of the rolling element bearing), is done by comparing the largest contact stress with the maximum contact stress permissible: σσHH P max max≤ (6.5/65) with ıHmax in accordance with Equation (6.5/58), whereby ıH0 and KH are to be determined for the maximum load, or respectively with KA = KAS; ıHPmax according to Equation (6.5/21b, c, d). Note: Compared to Equation (6.5/65), generally according to Equation (6.5/64) the lower load capacity results. IV Determining the quantities in the calculation equations The quantities which are components of Equations (6.5/58) to (6.5/65) for calculating load capacity of the tooth flanks are found using Table 6.5/13. 322 6 Load Capacity and Running Performance: External and Internal Gearings Table 6.5/13 Determination of the Quantities for the Calculation of Load Capacity of the Tooth Flanks (Refer to Appendix 7 for Supplement) No. Name of the quantity Symbols Calculation equation and remark 1 Circumferential tooth loads on the reference cylinder in the transverse section F t Constant load over time: tt1,2 1,2/2M F d= with 1,2 1,2tPM = ω ωπ12 12 2,,= n For loads which change over time (load spectrum): F t = Ftj consecutively for each load step j [and ıHj according to Equation (5.3/9) and ıH according to Equation (5.3/11)] or approximately with Ft = FtH according to Equation (5.3/9) 2 Application factor KA Section 6.3.3 or as specified by customer; KA referring to F t 3 Dynamic (load) factor KHv Kv Section 6.3.4.4 (or 6.3.4.8) for each load step j or approximately with Ft = FtH according to Section 5.3.4, Equation (5.3/15) 4 Face load factor (contact stress) KHȕ Section 6.4; Equation (6.4/47); for each load step j or approximately with Ft = FtH according to Section 5.3.4, Equation (5.3/15) 5 Transverse load factor (contact stress) KHĮ Section 6.4; Equations (6.4/60) and (6.4/61); for each load step j or approximately with Ft = FtH according to Section 5.3.4, Equation (5.3/15) 6 Elasticity factor ZE Table 6.5/2 or according to the following equation: () ()()E 22 11 221 1/ 1/Z EE= π− ν + − ν For E 1 = E2 = E and Ȟ1 = Ȟ2 = 0.3 the following applies ( Ȟ1, Ȟ2, E1, E2 Table 6.5/2): E 0.175 Z E = ; for steel this applies: EZ=191.62Nmm 7 Zone factor ZH According to Figure 6.5/6 or the following equation: ZH tb wt=1 2 coscos tan αβ α with ()d wt t 1n d 2 21 1 n t bncos cos 12cos with tantancos sin sin cosa a zmau zuz zzα= α =+β =≥ αα=β β= β α 6.5 Analysis of Load Capacity 323 Table 6.5/13 (continued) No. Name of the quantity Symbols Calculation equation and remark 8 Single mesh factor ZB,D ()wt B 22 a1 a2 22 12 b1 b2tan 2211 1Z dd zz ddαα= ππ−− ⋅ −− ε − ()wt D 22 a2 a1 22 21 b2 b1tan 2211 1Z dd zz ddαα= ππ−− ⋅ −− ε − 9 Contact ratio factor Zİ According to Figure 6.5/7 or the following equations: 4 3Zα ε−ε= for İȕ = 0 () ()1 4 3Zαβ β αε−ε ⋅ −ε ε =+ε for İȕ < 1 1Z αε=ε for İȕ 1 9.1 Axial contact ratio (Axial contact ratio) İȕ εβ πβ=b mw nsin 9.2 Profile contact ratio (Transverse contact ratio) İĮ εε εεαα=′+′−′12 with 122 a1 b1 tt2c o sdd mεπαa 22 a2 b2 2 22 1 tt 2;2c o sdd zzzm z−′ε= ⋅ ≥πα wt Į ttsinĮ cosa mεπαa whereby the following is true: da = d + 2mn (1 + x + k) db = d cosĮ t mt = mn/cosȕ d = zmt İĮ can be determined approximately for external gearing with (z1 + z2) 250 and ȕ 25 as follows: () εε ε βαα α≈+ ′+′− 11 21c o s2 with ′′εεαα1,2 according to Figure 2.1/20 324 6 Load Capacity and Running Performance: External and Internal Gearings Table 6.5/13 (continued) No. Name of the quantity Symbols Calculation equation and remark 10 Helix factor Zȕ Zβ=1c o s β (recognized as more apt compared to the previous specification Zβ=cosβ, which was changed in DIN/ISO in 2008) 11 Life (load cycle) factor ZN ZN NqNH LH12 12,lim ,= NHlim = 2 106 for nitrided gearing and cast iron , otherwise NHlim = 50 106; for N L < NHlim; qH for it in Equation (6.5/13) and Table 7.4/27 NL according to Equation (6.5/13a) If Z N1,2 > 1.6 then ZN = ZNmax = 1.6 applies for steel annealed in the normal way or quenched and tempered steel; cast steel; spheroidal graphite cast iron; case-hardened, carbonitrided, flame-hardened or induction-hardened steel. If Z N > 1.3 then ZN = ZNmax = 1.3 applies for nitrided steel. For N L > N Hlim a further drop in the fatigue strength can be expected. It is then approximately: for N L > N Hlim: Hlim N1,2 L1,2qNZN= 1, with q = qHlim = 40 (if no other specifications or experience is available) Note: The values given for ZNmax are approximate values. Other values are permitted in cases of other experience or test results. 12 Hydrodynamic influencing factors (lubricant, roughness, speed) (Z LZRZv) The following applies for hobbed, shaped or planed gearing: (ZLZRZv) = 0.85 to 0.9. The larger value applies for gearing with good running-in characteristics. For ground or shaved gearing, the following applies: ( Z LZRZv) = 1. 13 Hardness ratio factor ZW ZW = 1 for materials of the gear pairing with the same strength ( ıHlim), otherwise according to Equation (6.5/15a); as an alternative: Equation (6.5/15b), supplements ISO 6336-5 14 Size factor ZX ZX = 1 15 Tooth flank endurance limit ıHlim Table 7.4/14a, b, c (quenched and tempered), 7.4/15 (surface-hardened), 7.4/20 (case-hardened), 7.4/23a, b (nitrided); Table 7.4/27 (approximate values: quenched and tempered, nitrided, case-hardened, nitrocarburized, surface-hardened) 16 Minimum safety factor (fatigue stress) S Hmin The specification SHmin > 1.1 can be necessary, but SHmin < 1 is also permissible, depending on requirements . To be certain, observe remarks as in Equation (6.5/17). Also observe the notes on safety in Section 6.5.1 in cases of different K Hβ values! 17 Preventing damage caused by maximum strain S HSt ; (ıHmax ıHPmax ) More precise analyses: Appendix 7 ; ıHmax = ıH according to Equation (6.5/11) with Ft = Ftmax; in cases of directly specified Ftmax for impacts KA = Kv = 1, or Ftmax = KASt Ftnenn; Equation (6.5/21) with ıHSt = ZNmaxıHlim, ZNmax No. 11 of this table. [Only in cases of deformation of the surface: ıHPmax according to Equations (6.5/21b) to (6.5/21d).] 6.5 Analysis of Load Capacity 325 6.5.3.2 Load Capacity of the Tooth Root: Fatigue Fracture and Damage at Maximum Load (Methodological Process) I General The analysis of load capacity of the tooth root must be carried out for both gears of a pairing. In borderline cases it makes sense in the case of hard edge layers to compare the stress depth profile with the strength depth profile. In the case of axial offset of the gears or of different face widths, for the protruding gear the width resulting from Equation (6.5/66) can be used as an approximation for calculations ( bw common face width): bb m=+wn (6.5/66) The following remarks refer primarily to external gearing (internal gearing: [6.5/113]). II Preventing fatigue damage To prevent fatigue damage on the teeth, the analysis of load-bearing capacity can be carried out by comparing • the calculated tooth root stress with the permissible tooth root stress FF F 0 F PK σ σ= σ ≤ (6.5/67) • or the safety factor with the minimum safety factor ()SYY YYY SFFE T N FRX F =≥σ σδ δ/ min (6.5/68) ()FE/YįT approximately represents the fatigue strength of the smooth test piece) The calculated tooth root stress of the deviation-free gearing without additional load is σβε Ft nFS 0=F bmYY Y (6.5/69)1) The following applies for the load factor KF: KK K K KFA F v F F=αβ (6.5/70) The following relationship exists between the permissible tooth root stress ıFP and the tooth root endurance strength ıFE: () σσδ δ FPFE T N FRX =/ minYY SYYY (6.5/71) In cases of load spectra, an approach similar to that in Equations (6.5/63a) and (6.5/63b) is to be used for ıF, in analogy to contact stress. III Preventing incipient cracks, permanent deformation of the tooth profile or overload fracture at maximum load To prevent incipient cracks at the tooth root (or overload fracture) at maximum load, the analysis of load-bearing capacity can be carried out by comparing • the largest tooth root stress with the permissible tooth root stress 1) Contact ratio factor Yİ=1 has to be used, when YF and Y S are determined with hFe for the highest point of single tooth contact. In that case no additional profile overlap effect has to be considered with Yİ. 326 6 Load Capacity and Running Performance: External and Internal Gearings Fmax FPmaxσ≤ σ (6.5/72) with ıFmax according to Equations (6.5/67), (6.5/69) and (6.5/70), whereby ıF0 and KF for maximum load are to be determined with KA = KAS, or with Ftmax if the largest tooth load is directly specified. FG FPmax FSt minSσσ= (6.5/73) or • the safety factor with the minimum safety factor at maximum load FG FSt FSt min FmaxS Sσ=≥σ (6.5/74a) with ıFG tooth root strength limit at N 103, ıFG = Rm set as an approximation. To prevent plastic deformation of the tooth profile at maximum load, the analysis of load capacity can be carried out • by comparing the equivalent stress in the profile sections with the yield strength; Figure 6.5/24a and Equation (6.5/41) • or as guidance with the help of a rough approximation for YS 2.5 and normalised, quenched and tempered (or case-hardened) steels: p0.2 FSt FSt min Fn maxRSS=≥σ (6.5/74b) with ıFnmax according to Equation (6.5/35) and KF with Ftmax = KAS Ftnenn and KA = KAS , Ft = Ftnenn Here the following applies: Rp0.2 Yield strength (in the tooth root cross section, core cross section) ıFnmax Maximum occurring nominal tooth root bending stress IV Determining the quantities of the calculation equations The quantities which are components of Equations (6.5/67) to (6.5/74) for calculating load capacity of the tooth root can be found using Table 6.5/14. Table 6.5/14 Determination of the Quantities for the Calculation of Load Capacity of the Tooth Root (No Influence of the Gear Rim) No. Name of the quantity Symbols Calculation equation and remark 1 Circumferential tooth load on the reference cylinder in the transverse section F t Constant load over time: Refer to Table 6.5/13, No. 1 Loads which change over time (load spectrum): Ft = Ftj consecutively for each load step j for ı Fj [and ıF according to Equation (5.3/12)] or as an approximation with Ft = FtF according to Section 5.3.4, Equation (5.3/16) 2 Application factor KA Section 6.3.3 3 Internal dynamic factor (tooth root) K Fv Kv Section 6.3.4 (or 6.3.4.8), for each load step j or as an approximation with Ft = FtF according to Section 5.3.4, Equation (5.3/16) 4 Face load factor KF Section 6.4; Equation (6.4/47) with Equations (6.4/42) and (6.4/43), for each load step or as an approximation with Ft = FtH according to Section 5.3.4, Equation (5.3/16) 5 Transverse load factor KFĮ Section 6.4; Equation (6.4/60) or (6.4/61), for each load step or as an approximation with Ft = FtF according to Section 5.3.4, Equation (5.3/16) 6.5 Analysis of Load Capacity 327 Table 6.5/14 (continued) No. Name of the quantity Symbols Calculation equation and remark 6 Combined tooth form factor for load appli-cation on the tooth tip Y FS YFS = YFaYSa; YFa Equation (6.5/34c) or Figure 6.5/10 or 6.5/13; YSa Appendix 2 (external gearing) Further possibilities: For newly calculated values refer to Figure 6.5/9a, b, c and 6.5/12a, b, c, or as an approximation (external gearing 30 tangent, internal gearing 60 tangent) according to () ()Fa Fan Fn3FS 2 Fn Fn n nn/c o s7.322ȡ /c o shm sY smα=⋅ α In special cases for external gearing for gears produced by hobs or rack cutters with a basic rack according to ISO 53, DIN 867 in rough approximation (as guide) • Na0/mn = 0.2: YFS = 4.08 + 7.63 / zn ņ 15.94 x / zn + 0.18 x2 • Na0/mn = 0.38: YFS = 3.467 + 13.17 / zn ņ 27.91 x / z n + 0.091 x2 • Protuberance and ha0/mn = 1.4 (the rest of the values as in DIN 867): with spr/mn = 0.05, Y FS as an approximation YFS = 3.61 + 25.28 / zn ņ 37.56 x / z n + 0.53 x2 For ground gears the product Y FSYSrel is to be used instead of YFS, in order to take the grinding step into account, whereby YFS is to be determined according to the equations above and Y Srel according to remark No. 6.6 in this table. 6.1 Bending moment arm for load application on the tooth tip h Fa bv n Fa Fat2cos 2 2dd dhY=− + − α with ()nn bt vn n n 23 b n Fat at t at b a ncos ;cos cos ;cos cos cos 0.5 2 tantan inv cos / 21zm zmdd zzdz mz x z dd dd m x kπ=α =ββ == ≈ββ β +αα= α − α − α= =+ ++n a Y according to Equation (2.3/4) or (2.3/21); refer to Appendix 1.1 for this. 6.2 Tooth root thickness at the 30 tangent of the virtual gearing sFn • sFn according to Equations (2.3/2) and (2.3/3) or (2.3/19) and (2.3/20), also refer to Appendices 1.1 and 1.3 • For gears produced with a hob or rack cutter with sFn/mn also according to Figure 2.3/2 or Appendix 1.3 6.3 Load application angle on the tip cylinder of the virtual gearing Į Fan n Fan an n nx0.5 2 tantan invx zπ+αα= α − α − with ()nn an ncosos21cz zx kαα=++ + 328 6 Load Capacity and Running Performance: External and Internal Gearings Table 6.5/14 (continued) No. Name of the quantity Symbols Calculation equation and remark 6.4 Radius of curvature of the tooth root fillet at the 30 tangent of the virtual gearing NFn • NFn = N according to Equation (2.3/18) and Section 2.3.3.2 or Appendix 1.1 • For gears produced by a hob or rack cutter with Na0/mn = 0.38, ha0/mn = 1.25 and ș = 30 : NFn/mn according to Figure 2.3/4 or Appendix 1.2 6.5 Radius of curvature of the ground tooth root fillet or in the grinding step at the 30 tangent of the virtual gearing Ng • According to measurement results • Ng = N according to Equation (2.3/18) and Section 2.3.3.2 with Δh mh mx mh mna ns n=− − −0 na0gN or Appendix 1.1 with ha0 Distance of tool reference line to the tool head ha0 = (1 + c\*) m n for Įw0 = Į Na0g - tip radius of the grinding tool (smallest value corresponds roughly to mesh size of the grinding tool) hs - distance from grinding disc to tooth root 6.6 Relative stress concentration factor for the load application on the tooth tip for gearing with grinding step Y Srel •For ground gears with developed grinding step according to Equation (6.5/25); precise equation! • ISO 6336-3, DIN 3990-3 specify an equation (which to some extent deviates considerably from results calculated with precise methods): Srel gg1.3 1.3 0.6 /Y t= − N with (refer to Figure 6.5/15a) tg according to measurement results or as a rough approximation tg t0 Ng according to No. 6.5 in this table • For ground gears without a developed grinding step (tooth root surface fully finish-ground) according to YSrel Fn g=NN/ 3 with NFn according to No. 6.4, Ng according to No. 6.5 in this table • YSrel refer to Figure 6.5/15 Note: For Y Srel > 2 the following YSrel = 2 and for Y S > 4 the following YS = 4 is to be used, according to experience gained until now 7 Helix angle factor Yȕ According to Figure 6.5/18 or according to YYββ β εβ=− ≥ 1120min with min 1 0.25 0.75 Yββ=− ε ≥ , İȕ according to Table 2.2/2, No. 25 8 Overlap coefficient Yİ • For İȕ < 1 according to 0.25 0.75 / Yεα=+ ε 1) • For İȕ 1 according to 1/ Yα ε=ε 1) with İĮ according to Table 2.2/2, No. 24 (Continuation of the table on the following page) 1) The value of İĮn = İĮ/cos2ȕb is used instead of İĮ for this equation in DIN 3990. 6.5 Analysis of Load Capacity 329 Table 6.5/14 (continued) No. Name of the quantity Symbols Calculation equation and remark 9 Tooth root endurance strength ıFE/YįT According to Equation (6.5/39) with ıFE Table 7.4/14a, b, c (quenched and tempered), 7.4/15 (surface-hardened), 7.4/20 (case-hardened), 7.4/22a, b (nitrided), ı FE/YįT corresponds approximately to a smooth test piece. Table 7.4/27 (roughly estimated for quenched and tempered, nitrided, case-hardened) 10 Life factor YN for N L < NFlim: FFl im N1,2 L1,2qNYN= with NFlim = 3 106; qF refer to Table 7.4/27; NL according to Equation (6.5/13a) • If Y N > 2.5 then YN = YNmax = 2.5 applies for steel annealed in the normal way or quenched and tempered; cast steel; spheroidal graphite cast iron; case-hardened, carbonitrided, flame-hardened or induction-hardened steel • If Y N > 1.6 then YN = YNmax = 1.6 applies for nitrided steel • If N L NFlim is true then a further drop in the fatigue strength can be expected. It is then for N L NFlim : Fl im N1,2 L1,2qNYN= 1, with q = qFlim = 40 (if no other specifications exist) Note: The values given for YNmax are approximate values. Higher values are permitted in cases of other experiences or test results. 11 Roughness factor YR 1)2) If no values exist derived from precise analyses of all the influences, then Y R = 1 must be used. 1) 12 Size factor YX According to Figure 6.5/23 or Table 6.5/9 13 Sensitivity factor Yį 2) YįSt, YStT According to Figure 6.5/22 ( Yį) or Table 6.5/8 2) 14 Minimum safety factor SFmin SFmin considers in total the approximate character of the method of calculation and the inaccuracies of the specification factors . For certainty’s sake observe the principle remarks for Equation (6.5/117). Note on safety for differing K Fβ values in Section 6.5.1! 15 Yield strength Rp0.2 According to Tables 7.4/14a, b, c (quenched and tempered), 7.4/15 surface-hardened), 7.4/20 (case-hardened), 7.4/23a, b (nitrided); for medium demands according to Table 7.4/27 16 Minimum safety factor at maximum load S FStmin To consider the uncertainty of the method of calculation (approaches) SFStmin = 1.3 is sufficient. Because of further aspects (such as the meaning of the system) the specification SFStmin > 1.3 may be necessary. Also observe the notes on safety in Section 6.5.1 in cases of different K Fβ values! 1) Instead of Y R ISO 6336-3 and DIN 3990-3 use the relative quantity YRrelT for which, however, here YRrelT =1 is recommended. 2) Other values are used in ISO 6336-3 and DIN 3990-3 and commonly the relative values YįrelT , YįT are used (Appendix 2). 330 6 Load Capacity and Running Performance: External and Internal Gearings 6.5.4 Lubricating Film Thickness 6.5.4.1 Calculating the Lubricating Film Thickness The main tasks of the lubricant in the transmission of load between two gears are to reduce friction and prevent damage to the tooth flank. Other tasks include the dissipation of the produced frictional heat and preventing corrosion. Compared to a plain bearing, for gear pairs the conditions for forming a hydrodynamic lubricating film are considerably less favourable. The following are essential for this: • For the most common external gearing encountered, meshing contact surfaces with opposing curvature • Constantly changing speed conditions between the contact surfaces • Incidence of high stresses in the contact area • The necessity of constantly re-establishing the supporting lubricating film at every mesh (discontinuous lubrication) For a long time many of the issues associated with the formation of a lubricating film were unre- solved, and some are still not completely settled. The unfavourable conditions that exist for gears, which have already been mentioned, to some extent led to the assumption that a hydrodynamic effect did not even exist for gears. On the other hand, the lubricant has a strong influence on friction, wear, scuffing, and micropitting, and to a lesser extent also on pitting. According to the view held here, both the hydroelastic effect as well as the adhesive strength of the lubricant exert an influence. Results from tests on the lubrication film thickness (hydroelastic effect) are provided below. Initially there was a great deal of scepticism towards theoretical solutions. Among the very re-markable phenomena associated with hydroelastic lubrication are the narrowing of the lubrication gap at the end of the parallel gap secti on and the compression peak (Figure 6.5/39). Hydrodynamics Elastohydrodynamics Hertzian theory Figure 6.5/39 Types of contact/classification of the elastohydrodynamic theory [6.5/42] Finally the measurement of the compression and temperature characteristics was successful, with the help of thin-film sensors, and the theory was confirmed. The results are depicted in Figures 6.5/40 to 6.5/42. • Cylinders set in motion under light load • Cylinder deformation insignificantly small • Hydrodynamic lubricant • Cylinders set in motion under very heavy load • Elastic deformation of the cylinders • Hydrodynamic lubricant • Inactive cylinders under very heavy load • Elastic deformation of the cylinders (flattening 2 bH) • Solid body contact (no lubricant) 6.5 Analysis of Load Capacity 331 Figure 6.5/40 Pressure distribution in the contact [6.5/51] Figure 6.5/41 Temperature and pressure distribution in the roller contact [6.5/45] Figure 6.5/42 Measured EHD pressure distribution in comparison with the calculated Hertzian pressure distribution [6.5/54] By applying the basic equations of hydrodynamics, Martin [6.5/38] initially succeeded in finding a solution for the lubrication film thickness (equation ac-cording to Martin , refer to [6.5/41] and Figure 6.5/39): 04.9h F∗ ∗=N/ (6.5/75) Here the following apply: h0 = hmin Lubrication film thickness N Equivalent radius of curvature 11 1 12NN N=+ (6.5/76) N1,2 Radii of curvature Particularly applicable for the pitch point: N1,2 = NC1,2 = 0.5 db1,2 tan Įw /\* Referenced speed (dimensionless, physical quantity equation) //∗=′η0h d EN (6.5/77) η0 Dynamic viscosity of the lubricant oil at the existing surface temperature of the teeth, before the beginning of meshing and atmospheric pressure (bulk temperature KM; KM refer to DIN 3990-4 [6.5/65], ISO TR 13989) /hd Hydrodynamically effective speed y T1 T2 h d T1, 2 1, 2 b1, 2 w ; 0.5 tan (at pitch point)22dΣ +== = ω α///// E ƍ Equivalent E-modul e 22 12 1211 1 1 2 EE E −ν −ν=+ ′ (6.5/78) E Elasticity of material; Ȟ Poisson’s ratio F\* Referenced load (dimensionless, physical quantity equation) 332 6 Load Capacity and Running Performance of External and Internal Gearing nFFbE∗=′N (6.5/79) Fn/b Specific load FF n=t cosα N According to Equation (6.5/76); Eƍ according to Equation (6.5/78) However, it soon became clear that the assumptions of Martin , that rigid surfaces and constant viscosity, which at least in the case of plain bearings result in serviceable solutions in most cases, are inadmissible here. The inclusion of these quantities as variables led to the hydroelastic theory . Here fundamental work was initially carried out by Ertel-Mohrenstein and Grubin [6.5/39], [6.5/42] to [6.5/44], [6.5/100] to [6.5/102]. Based on existing solutions of the isothermal elastohydrodynamic (EHD) problem purely for rolling (no sliding), Dowson and Higginson established an empirical equation for the smallest occurring lubrication film thickness for gearing [6.5/41]: ()0.60.7 pmin 0.131.6E h F∗ ∗′α =N/ (6.5/80)1) Here the following apply in the physical quantity Equation (6.5/80): Įp Compression viscosity coefficient Įp = (1 ... 1.6 ... 3) 10-8 m2/N for mineral oils The following correlation applies: ηηα=0p p (6.5/81) Ș Dynamic viscosity p Pressure The remaining values for the lubrication film thickness are according to Equations (6.5/77) to (6.5/79). Oster [6.5/42] specifies an equation that hearkens back to Ertel/Grubin for the isothermal lubrication film thickness in the parallel gap (refer to Figure 6.5/39): ()1/11 8/114/11 1/11 n 0p t0 1.95 'FhEb− =α η N/ (6.5/82) Oster also specifies an equation in modified form, hearkening back to Dowson/Higginson [6.5/40], for the minimal isothermal lubrication film thickness: ()0.13 0.7 0.54 0.43 0.03 n min,isoth p 0 t 2.65 'FhEb− − =α η /N (6.5/83) Notes: • Equation (6.5/83) is a physical quantity equation! • Naccording to Equation (6.5/76); vt according to the information at Equation (6.5/77) with /t = /hd; Eƍ according to Equation (6.5/78) • The calculated lubrication film thickness according to Equations (6.5/82) and (6.5/83) results in values deemed real, if the dynamic viscosity Ș0 applicable for atmospheric pressure and bulk temperature (temperature KM of the tooth flank surface) is applied right before meshing begins (DIN 3990 T4 [6.5/65]). • In the case of helical gearing, it is apparently more suitable to replace the member (F n/b) 0.13 with ( Fn/Ȉli)-0.13 in Equation (6.5/83), whereby Ȉli represents the sum of the lengths of the lines of contact in the field of action. 1) There are differences between the two Equations (6.5/80) and (6.5/83), which were published at different times, and these differences are apparently due to the required empirical approach. 6.5 Analysis of Load Capacity 333 According to [6.5/45], because of a lower viscosity of the lubricant due to warming in the running- in area of the rolling contact, the thickness of the lubricating film which actually exists is lower than under assumed isothermal conditions. The mini mum lubricating film thickness h min,th which results from the thermal theory follows with hmin,isoth , according to Equation (6.5/83): hC hmin min ,th th ,isoth= (6.5/83a) Here the following applies: Cth Thermal correction factor [6.5/60] th 0.62 th3.94 3.94CL=+ (6.5/83b) Lth Thermal load factor LkthMt=∗ηα/2 (6.5/83c) Į\* Viscosity temperature coefficient of the lubricant b M MO i l bln ∗ η η α=−KK (6.5/83d) Șb Dynamic viscosity at operating oil temperature K b and atmospheric pressure (1 bar) η νbb b=N Ȟb Service viscosity according to Figure 6.6/23 Nb Density at service temperature according to Equation (6.6/11) ȘM Dynamic viscosity at bulk temperature in the running-in zone KM and atmospheric pressure (1 bar); η νMM M=N KM Bulk temperature in the running-in zone In cases of splash lubrication: KM KOil b + (5 ... 10) K k Heat transmission coefficient Applies for lubricant: k Oil 0.133 W/mK Besides other findings, the VDI report No. 1207 [6.5/103] gives an overview of the history, publi- cations and goals of research in the area of elastohydrodynamics. 6.5.4.2 Tooth Flank Damage and the Influence of Lubricants In extensive tests it was shown that the lubricant has a notable influence on some kinds of tooth flank damage [6.5/46]. In the case of pitting the lubricant exercises an influence through the gear friction coefficient Z and the ratio of lubrication film thickness to the sum of the surface roughness values: h/ȈRa. Tests on a test rig with two discs showed an approximate connection between the middle lifetime, expressed by the load cycle number N, and the influence quantities, which are depicted in the following equation for a pairing in states 1 and 2: min1 a1 1H 1 Z 1 2H 2 Z 2 m i n 2 a 22.2 75/ /hR N Nh R−−Σ σμ =⋅ ⋅ σμ Σ (6.5/84) 334 6 Load Capacity and Running Performance of External and Internal Gearing Equation (6.5/84) applies to mixed friction with hmin/ȈRa 1. In the case of hmin/ȈRa > 1, the film thickness hmin in Equation (6.5/84) has no influence on the lifetime. The final term is then 1. According to Plewe [6.5/47], the minimum lubricating film thickness hmin is an expression for the danger of wear . In cases where mineral oils of various viscosities were used, the following approximate values were given for the occurrence of abrasive wear: • hmin < 0.5 ȝm for cases of material pairings of roughly the same surface hardness • hmin < 0.4 ȝm for cases of hard/soft material pairings, with abrasive wear limiting the lifetime on the soft wheel Flank damage from hot scuffing (for simplification’s sake and to some extent referred to as follows only as scuffing), and its precursors in the form of scratches, score marks and scuff marks, comes about, according to the current conception, when the lubrication film is penetrated. Due to high loads and sliding speeds and the resulting excessively hot temperatures in the lubrication gap, weld bridges form locally, which, due to the sliding movement, are immediately severed again. Increasing the lubricating film thickness by using higher viscosity lubricants or EP-blended lubricating oils (forming a protective coating) helps prevent hot scuffing (Section 6.5.5). According to current understanding, the formation of micropitting on case-hardened gears is particularly influenced by the relationship Ȝ of the lubrication film thickness to surface roughness, as well as by the additives (refer to Section 6.5.6 for more details): λ=h Rmin a (6.5/85) with () RR Raa a=+1 212 (6.5/86) A deeper look at the research results can be found in the literature [6.5/48] to [6.5/62] and [6.5/102] and newer dissertations [6.5/98] and [6.5/99]. 6.5.5 Scuffing Load Capacity 6.5.5.1 Introduction Whereas gear material and heat treatment are the primary influences determining flank load capacity and the strength at the root, the scuffing threshold strongly depends on the lubricant used. One differentiates between hot and cold scuffing . Particularly in the case of gearing under high thermal load, as a threshold of load capacity, the more significant hot scuffing takes on more importance. It will be dealt with below. The primary cause of (hot) scuffing is the high local temperatures between the tooth flanks (flash temperature) due to contact stress and sliding movement. The lubricating film fails, and surfaces weld together momentarily, but, due to the sliding speed, they are immediately severed again. In order for scuffing to occur, the following must happen: • The hydrodynamic lubricating film must be ruptured. • The adhesive strength of the lubricant must be overpowered. • The protective films that form at certain temperatures due to the additives in the lubricant must be destroyed. Thus, for the scuffing process, physical as well as chemical processes within the thin lubricating oil layers are decisive ([6.5/61] among others). The temperature of the tooth flanks plays an important role. It has an influence on the viscosity of the oil and thus the load-bearing capacity. To determine the distribution of temperature in involute external gearing, it is possible to use the calculation programme LVR [6.5/105], which can also be used to determine the distribution of load and stress. 6.5 Analysis of Load Capacity 335 The physical-chemical processes during meshing can only be explained with difficulty using calculation methods. Empirically and theoretically substantiated methods have been in development and subjected to testing for decades now. For example, a comparison and evaluation of earlier methods is given in [6.5/64]. At the present time two methods have emerged, which are currently recommended for use in both DIN 3990 T 4 [6.5/65] and in the draft ISO/TR 13989-1, -2 [6.5/96], [6.5/97] (integral temperature method, flash temperature method). They build on Blok’s flash temperature theory and are at the centre of further discussion here. 6.5.5.2 Damage Description and Quantities Influencing Scuffing Load Capacity Scuffing occurs already after a brief period of overload. We are not talking about fatigue damage in this case, rather processes in which, at the points of contact, weld bridges form, which are then immediately severed again. Cold scuffing usually occurs at low circumferential speeds (less than roughly 4 m/s), mainly on quenched and tempered gears of rough gear tooth quality, at negligible levels of heat development. This damage is probably caused likewise by weld bridges forming locally due to normal pressure and an exceeding of the (shear) strength of the material in regions close to the surface. Of much more significance is hot scuffing , which occurs on fast-running (more than roughly 4 m/s), highly loaded and even case-hardened spur gears, in connection with high local temperatures. Factors influencing the occurrence of scuffing damage include the material the gear is made of and its surface quality, the lubricant, the toothing geometry, and the operating conditions. Heavy foaming or impurities in the lubricant can increase the risk of scuffing. The magnitude of the influences on scuffing load capacity, with reference to torque, is listed in Table 6.5/15 [2]. Table 6.5/15 Approximate Values for Raising the Scuffing Load Capacity, with Reference to Torque; Values according to [2] Based on Niemann/ Lechner ; Values in Brackets: Experience of Our Own That Differs Influencing quantity Factor Surface, basic material roughness Flank roughness: Reduce to 1:16 Running-in, gro und tooth flanks Steel, case-hardened (drop in residual austenite content) Phosphate coating of the flanks Copper plating of the flanks (special short tests) Nitrided compared to case-hardened 2 1.5 3 (2) 1.4 (1.2) 2.8 (1.3) 2 (1.5) Lubricant EP additive (at the same viscosity of the basic oil) Nominal viscosity: Doubling at 50 C - pure mineral oils - blended oils Usual oxidation or corrosion protection agent or silicone oil as foam inhibitor Friction coefficient: Halved by a polyether oil 5 (3) 1.5 1.15 1 2.5 (1.5) Toothing geometry Modification of the toothing geometry (including module) Tip relief: - Zero or X-zero gearing - X gearing with strongly one-sided profile shift Helical gearing (ȕ = 30 ) compared to spur gearing 6 1.5 2 1.3 Operating conditions Circumferential speed Oil temperature: Lower by 20 K 2.5 (1.5) 1.18 The flash temperature hypothesis [6.5/73] (refer to [6.5/68]) published by Blok already in 1937 is based on a critical, local contact temperature, composed of bulk temperature and flash temperature. 336 6 Load Capacity and Running Performance of External and Internal Gearing Through a drop in the viscosity of the lubricant due to local heating, the metallic surfaces touch directly, and the welding which was already mentioned occurs. Most authors view this critical temperature as decisive for scuffing. 6.5.5.3 Approaches to Calculating Scuffing Load Capacity Besides an assessment according to the calculation methods contained in DIN 3990 or ISO 6336, scuffing load has also been assessed according to the following aspects, whereby, however, only part of the decisive influences are taken into consideration, and for that reason, is not considered to be generally applicable. EHD lubricating film thickness The rule states that “the existing lubricating film thickness must be thicker than the critical lubricating film thickness contingent on the roughness of the tooth flank surfaces” (refer to Section 6.5.4). The disadvantage of this method is that the rest of the decisive influencing quantities are still not registered in the methods developed for this, or not with a sufficient degree of precision. Since viscosity is the only characteristic of oil that is considered, the methods currently known apply only for unblended mineral oils. The chemical processes are not registered. Another imperfection is that the lubricating film thickness is usually calculated in the pitch circle, whereby scuffing damage is to be expected in the areas of high sliding ratios at the tooth tip or tooth root. Lubricating film strength In extensive experimental tests on gears, Lechner [6.5/63], [6.5/67] established a method of calculation which uses a maximum pressure load of a protective chemical film on the tooth flanks as a function of the permanent tooth flank temperature. For the first time, practically all of the factors affecting scuffing were taken into account. However, scuffing load tests at different circumferential speeds are necessary. This method formed the basis for further calculation methods with the mean surface temperature as the reference value, for example, according to Seitzinger [6.5/70]. Here the mean critical surface temperature is set to increase arithmetically with the circumferential speed. Flash temperature According to Blok [6.5/73], scuffing damage is to be expected when the contact temperature (sum of the bulk temperature and flash temperature) in the zone of contact of both tooth flanks reaches a critical level (flash temperature method). This critical temperature is constant for a metal-lubricant pairing, and in its original form it was applied independent of the operating conditions and only developed and tested for unblended oils. In general, the calculating approach for determining contact temperature must be applied to every meshing point with the respective local parameters, such as load, radius of curvature under load, and friction coefficient. Without flank relief, local maximums can be found in spur gearing at the four meshing points A, B, D , and E. Attempts were made by Eiselt [6.5/69] to further develop the flash temperature method. The specific scuffing load ascertained in the FZG test is converted to the data of the gear transmission to be calculated, at test speed. A conversion to the operating speed is subsequently carried out, dependent on the viscosity of the oil Ȟ 40, through a speed exponent kv. In DIN 3990-4 [6.5/65] the flash temperature method was introduced in modified or, respectively, conditioned form (integral temperature method). However, depending on the circumferential speed, the critical contact temperature is assumed as constant. The method was also made applicable for blended oils in ISO/TR 13989-1 [6.5/96], by experimental tests of the scuffing temperatures (FZG test). 6.5 Analysis of Load Capacity 337 Mean surface temperature Instead of the respective current contact temperature, it is possible to ascertain the mean surface temperature, which references mean values of influence quantities. Since the tribochemical processes depend on the time the temperature is in effect, this method s eems justified. Empirical coefficients take into account various parameters, such as tip relief, type of lubrication, roughness, and others. The mean surface temperature is compared with a critical value, which is dependent on the base oil, its type of additive and the quantity of additive. This calculation approach was further developed by Michaelis [6.5/66], [6.5/72], under the name integral temperature method . It is the foundation of DIN 3990-4 [6.5/65] and in ISO/TR 13989-2 [6.5/97]. The integral temperature method is based on the notion that, concerning scuffing, very brief local increases in strain are not to be considered as dangerous as those whose effects last longer. An averaging over the normal base pitc h should be adequate for this. The calculation of scuffing load capacity according to the flash temperature hypothesis and according to the integral temperature method derived from this will be examined more closely as follows. Based on our own experience, the latter method agrees best with test results. 6.5.5.4 Scuffing Load Capacity according to Flash Temperature This method is depicted in ISO/TR 13989-1 [6.5/96]. To the bulk temperature K M, which is the temperature of the gear teeth before meshing begins, the flash temperature Kfla (Figure 6.5/43) during meshing is added to the current contact temperature KB: BMf l a=+KKK (6.5/87) Figure 6.5/43 Curve of the contact temperature over the length of path of contact, spur gearing: a) contact temperature for deviation-free involute profile (no profile modification); b) contact temperature for profile modification or wear 338 6 Load Capacity and Running Performance of External and Internal Gearing Here the following applies: KM Mean bulk temperature The following applies approximately in cases where the material of the pinion and the wheel is the same: ()KK KMM M=+1 212 (6.5/87a) Here the influence of the time of exposure of the temperatures is not taken into account, which must necessarily lead to deviations, particularly in the case of blended oils (time for chemical reaction, among others). This and the speed-dependent scuffing temperature are reasons for the partly unsatisfactory agreement of calculation results with running results. a) Deviation-free involute profile b) (Optimally) modified involute profile Figure 6.5/44 Load distribution factor Xī, spur gearing The flash temperature is ()JT 1 T 2 fla mC HM 1 M 2 T 21.11 2n XX w bB BΓ −=μ +K// // (6.5/88) For cylindrical gear transmissions, ISO/TR 13989-1 specifies the following adapted relationship for the flash temperature: () Kfla mC J B Bt=⋅μ XX X X waM Γ3412 14// // (6.5/88a) In Equations (6.5/88) and (6.5/88a) the variables are: mC Friction coefficient (mean value): Equation (6.5/89); tests to measure local friction coefficient [6.6/43] w n Line load in direction of the normals in the meshing point being considered; Equation (6.5/91) w Bt Effective line load; Equation (6.5/89a) b H Half width of Hertzian contact deformation, refer to Figure 6.5/2; Equation (6.5/92) /T1,2 Speed of tooth 1 or 2 perpendicular to the line of action in the respective meshing point; Equation (6.1/5) / Circumferential speed on the reference cylinder B M1,2 Thermal contact coefficient of gear 1 or 2; Equation (6.5/90) X ī Load distribution factor; for cylindrical gears: Xī 1 X J Approximation factor; Equations (6.5/93) and (6.5/94) X M Flash factor X B Geometry factor; Equations (6.5/95) and (6.5/96) 6.5 Analysis of Load Capacity 339 The equations to determine these quantities are: • Friction coefficient mC 0.2 0.05 Bt mC oil L R Cr e l C0.060wXX− Σ μ= η /N (6.5/89) Here the following are true: wBt Effective line load wK K K K KF bBt A v B B mpt=αβ (6.5/89a) K A, Kv, Section 6.3 K BĮ KHĮ; KHĮ, Section 6.4 K Bȕ KHȕ; KHȕ, Section 6.4 K mp Factor of the load distribution in cases of branches in gear transmissions or additional loads; for cylindrical gear transmissions the following is true: Kmp = 1; for additional information see ISO 13989 Ft/b, Circumferential load in the reference circle per face width (is also the ideal reference line load of the load factors KA ... K BȖ) in N ηoil Dynamic oil viscosity (approximately the same as the dynamic viscosity at bulk temperature) in mPa s, refer to notes under Equation (6.5/83d) /(C Sum of the tangential speeds in the pitch point in m/s; refer to Equation (6.1/21a) X R Roughness factor 0.25 a1 a2 R2RRX+ = (6.5/89b) R a Arithmetic mean roughness in m according to Equation (6.5/86) NrelC Relative radius of curvature in the normal section in mm NNN NNrelCCC CC=⋅ +12 12 (6.5/89c) X L Lubricant factor Examples: X L = 1.0 for mineral oils X L = 0.8 for poly-alpha-olefins and esters X L = 0.7 for non-water-soluble polyglycols X L = 0.6 for water-soluble polyglycols X L = 1.3 for phosphoric esters X L = 1.5 for traction fluids • Speed components /T perpendicular to the line of action /T1 = /Ty1 and /T2 = /Ty2 can be determined according to Equation (6.1/5) with Equation (6.1/5a to i). • Thermal contact coefficient BM BcM=λN (6.5/90) Here the following are true: Ȝ Thermal conductivity coefficient; case-hardened steels: Ȝ = 40 ... 50 N/(sK) N Density; steel: N 7800 kg/m 3 c Specific heat; steel: c 490 Nm/(kgK) B M 13 103 N/(mKs1/2) for steel. • Line load wn If the line load for the mesh position in question is not known from precise analyses, the approximation according to Equation (6.5/91) can be used: 340 6 Load Capacity and Running Performance of External and Internal Gearing t n n cos cos ȕww=α (6.5/91) Here the following are true: w t Effective tooth tangential load per face width in N/mm; Equation (6.5/89a) with wt = wBt Įn Pressure angle in the normal section C Helix angle • Half width of Hertzian contact deformation bH Since no relationship has been specified in ISO to determine b H, we will resort to Blok [6.5/73]: bw EHnr e d C= ′8N π (6.5/92) Here the following are true: w n Line load in N/mm; Equation (6.5/91) NredC Reduced radius of curvature in the normal section in mm ()NredCwt b= +u ua 12sin cosα β (6.5/92a) Eƍ according to Equation (6.5/78) • Xī Load distribution factor Guide values are depicted in Figure 6.5/44. See ISO/TR 13989-1, [6.5/96] for further notes. • XJ Approximation factor Driving pinion: Jy 3 y eff a 2 Jy EA1 for 0 11 for 050X CCX=≥ − −=+ ⋅ ≥ < − Γ ΓΓΓΓ (6.5/93a) Driving gear: Jy 3 y eff a1 Jy EA1 for 0 11 for 050X CCX=≤ −=+ ⋅ ≥ > − Γ ΓΓΓΓ (6.5/93b) With the parameters on the line of action īy: ΓΓ Γyy wtAa wtEa wt=− = − ⋅ − =−tan tan;tan tan;tan tanα αα αα α1 2 11111 1z z (6.5/93c) • XM Flash factor In cases of the same materials and martensitic steels with mean values, the following can be used: X M = 50 K N–3/4 s1/2 m-1/2 mm • XB Geometry factor (in ISO/TR 13989-1: XG) External gearing: ()() ( ) () ( )1/ 2 1/2 yy1/2 B 1/4 1/4 yy11 / 0.51 1 1u XX u uαβ+− − =⋅ + ⋅ +⋅ −ΓΓ ΓΓ (6.5/94a) 6.5 Analysis of Load Capacity 341 Internal gearing: ()() ( ) () ( )1/2 1/2 yy1/ 2 B 1/4 1/4 yy11 / 0.51 1 1u XX u uαβ+− + =⋅ + ⋅ +⋅ +ΓΓ ΓΓ (6.5/94b) with XĮȕ Angle factor; refer to [6.5/96], for Įn = 20 : XĮ ȕ 1.0 Scuffing safety The scuffing safety SB is SO i l BB min Bmax OilSS−=≥−KK KK (6.5/95) Here the following are true: KBmax Maximum contact temperature alongside the length of path of contact Koil Lubricating oil temperature before meshing begins KS Scuffing temperature SBmin Smallest safety factor necessary for the contact temperature; not specified in ISO The quantities of Equation (6.5/95) are determined in KK KBMmax max =+fla (6.5/96a) Here KM is the bulk temperature, that is, the temperature of the tooth surface immediately before meshing begins. The value KM can be estimated as follows: Mn f l a n m m p S 0.47 XX =+KK K (6.5/96b) with Kfla m Mean flash temperature along the length of path of contact KK fla mfla y AE EAd =⋅ − Γ ΓΓ (6.5/96c) Xmp Lubricant contact factor Xn mpp=+1 2 (6.5/96d) n p Number of meshes XS Lubrication factor (intended to account for better heat transfer in the case of splash lubrication compared to injection lubrication) X S = 1.0 Splash lubrication and meshing with additional injection cooling X S = 1.2 Injection lubrication X S = 0.2 Gears running completely submerged in oil Based on gear test runs with the identical material-lubricant-material system, the scuffing temperature of a low-blended mineral oil for steel gears can be approximately calculated: KK KSM T W f l a m a x T=+ X (6.5/96e) 342 6 Load Capacity and Running Performance of External and Internal Gearing The index T refers to the test values. The structure factor XW can be derived from Table 6.5/16. Table 6.5/16 Structure Factor X W [6.5/65] Material/heat treatment or coating Structure factor XW Quenched and tempered steels Phosphate-coated steels Copper-coated steels Bath and gas-nitrided steels Case-hardened steels - With below-average residual austenite content - With normal residual austenite content - With above-average residual austenite content Austenitic steels (rust-resistant steels) 1.00 1.25 1.50 1.50 1.15 1.00 0.85 0.45 Supplement to ISO/TR 13989-1 (refer to DIN 3990-4 [6.5/65]) The safety factor SBmin in Equation (6.5/95) depends on whether the gear transmission is commissioned or operating only after a good running-in. After careful running-in, up to SBmin 1 no scuffing damage can occur. Without running-in, scuffing is often only ruled out at SBmin 3. Scuffing occurs when the contact temperature KB exceeds a specific value: the scuffing temperature KS. The scuffing temperature is determined in gear tests for a material-lubricant-material system and applied to the gear pair under consideration. In cases of unblended and low-blended mineral oils and hardened steels, the scuffing temperature can be determined as a characteristic of the composition of the oil, dependent on the nominal viscosity Ȟ40, as follows: KS40230 76.5 log30ν=+ (6.5/97) In cases of EP oils with a mineral base and of synthetic oils , the scuffing temperature, which is dependent on the material and operating conditions, is to be determined in tests. For the back-to-back gear test rig, standardised according to DIN ISO 14635-1 [6.5/71], the bulk temperature K MT and the maximum flash temperature Kfla maxT in Equation (6.5/96e) can be calculated as follows: MT 1 T 80 0.23 M ϑ= + (6.5/98a) 1.2 fla maxT 1T 400.4 401000.12 M−ν ϑ= ν (6.5/98b) where M1T is the pinion torque in the test, in Nm. Figure 6.5/46 depicts the bulk temperature KMT and mean flash temperature Kfla intT for the FZG test. An obvious imperfection is the fact that the scuffing factor is dependent on circumferential speed (Figure 6.5/45), which is directly accounted for by neither this calculation method nor the integral temperature method. Hydrodynamic factors play an important role. As such, because of the shorter time of exposure, the effective energy is lower in cases of high circumferential speed. Calculation of scuffing load capacity according to the contact-time method is dealt with in [6.5/104]. 6.5 Analysis of Load Capacity 343 Figure 6.5/45 Scuffing factor-speed curve for various oils [6.5/66] 6.5.5.5 Scuffing Load Capacity according to Integral Temperature This method is presented in ISO/TR 13989-2 [6.5/97]. Based on the notion that a high, but very briefly acting temperature is often not as damaging as a lower, but longer lasting one, the mean value of the temperature flash is established (Figure 6.5/43), [6.5/66] and [6.5/72]: KKfla flad AE int= 1 gαξξ ξ (6.5/99) with gĮ the length of path of contact E A; Kfla according to Equation (6.5/88). The mean value of the flash temperature is multiplied by the weighting factor C2 and added to the bulk temperature KM: KK Kint int=+Mf l aC2 (6.5/100) Here the following are true: C2 1.5 Determined in tests M Oil 1 flaint mp S CX X =+KK K (6.5/100a) with C1 = 0.7, determined in tests Xmp Lubricant contact factor, refer to Equation (6.5/96d) XS Lubrication factor, values same as in Equation (6.5/96d) The safety factor against scuffing is SSintint intmin SS S =≥K K (6.5/101) The integral scuffing temperature Kint, at which by definition scuffing begins to occur, can be approximately determined from test values of the FZG test rig. Here, there must be an identical material-lubricant system: 344 6 Load Capacity and Running Performance of External and Internal Gearing KK KintS MT W rel T fla int T≈+ XC2 (6.5/101a) Here the following are true: KMT Bulk temperature of the test gear according to Equation (6.5/96b) C2 1.5 Determined in tests XWrelT Relative structure factor according to Equation (6.5/114) and Table 6.5/16; XWrelT accounts for the properties of the material/structure on the scuffing strength Kfla intT Integral temperature for scuffing in the FZG test (refer to Figure 6.5/46) Taking the importance of the system and an understanding of the load, and so on into consideration, the minimum safety factor SSmin can be determined as follows: SintS 3 Scuffing is improbable. (1) < SintS < 3 Scuffing can be prevented if careful running-in is carried out. 6.5.5.6 Practical Calculation of Scuffing Load Capacity according to the Integral Temperature Method The information given corresponds to ISO/TR 13989-2 [6.5/97]. When added to the bulk temperature, a mean value and a weighting of the flash temperature results in the integral temperature (Figure 6.5/43): KK K Kint int int =+ ≤Mf l a PC2 (6.5/102) with KKfla fla Eint= Xε (6.5/102a) and M Oil 1 mp flaint S CX X =+KK K (6.5/102b) where KOil is the temperature of the lubricating oil before meshing. The weighting factors C1 and C2 were obtained from tests and can be set approximately: C 1 = 0.7; C2 = 1.5 Further factors are given at the end of this section. The following relationships are recommended for use in cases where more precise information is not given (method C). With reference to the tip meshing point E of the pinion, the associated flash temperature is determined from: Kfla E mC M BEBB t E QC a=⋅μαβγXX XKw aX XX()// /34 12 14/ (6.5/103) The calculation of the tooth friction coefficient, which is averaged over the length of path of contact, is done according to 0.2 0.05 Bt B mC oil R L redC C0.045wKXX− γ Σ μ= η N/ (6.5/104) Included are wBt Effective line load in N/mm wK K K KF bBt A v B Bt=αβ (6.5/104a ) KBȖ Helix angle factor for scuffing; KBȖ corrects the load determined with the rest of the factors, caused by the tendency towards scuffing in helical gearing (empirical) 6.5 Analysis of Load Capacity 345 KBγ=1 for İȖ 2 () () B10 . 2 25 Kγγ γ=+ ε − − ε for 2 < İȖ < 3.5 (6.5/104b) B1.3 Kγ= for İȖ 3.5 with İȖ = İĮ + İȕ NredC Reduced radius of curvature in the normal section in mm ()NredCwt b= +u ua 12sin cosα β (6.5/104c) /(C Sum of the tangential speeds in the pitch point in m/s Refer to Equation (6.1/21a) Șoil Dynamic viscosity for oil temperature in mPa s; refer to remarks under Equation (6.5/83d) The integral temperature for scuffing, at which, by definition, scuffing begins to occur, is deter- mined approximately from gear test runs with the identical material-lubricant-material system, from KK KintS MT W rel T fla int T=+ XC2 (6.5/105) The index T refers to the test values. In order to determine the bulk temperature KMT and the mean flash temperature throughout meshing for the KflaintT for the standard FZG test A/8.3/90, the following relationships apply: MT 1T L 80 0.23 MX =+ ⋅K (6.5/105a) flainT 1T L 401000.2MX0.02 =⋅ ν K (6.5/105b) Figure 6.5/46 shows the bulk and flash temperatures dependent on the pinion torque for the standard FZG test A/8.3/90. Figure 6.5/46 Bulk temperature KMT in C and mean flash temperature Kfla int T in K in FZG test A/8.3/90 [6.5/97] 346 6 Load Capacity and Running Performance of External and Internal Gearing In Equation (6.5/102) the allowable integral temperature defined as follows is used as a comparative value: KK intint minPS S=S (6.5/106) The smallest SSmin necessary is to be determined separately for each case. Defined as the calculated scuffing safety is SSintint intmin SS S =≥K K (6.5/107) The following guide values from practical tests are given in the ISO draft: S intS < 1.0 There is a high probability of the occurrence of scuffing marks. 1.0 SintS 2.0 If the gear transmissions are carefully run in, the contact pattern is good, and the load assumptions are true to life, then no scuffing damage is to be expected. S intS > 2.0 Scuffing is unlikely to occur. From what has been formulated, it is clear that the scuffing conditions cannot be precisely specified. In the preceding equations of the integral temperature method, the following factors are to be inserted with the X value: • Overlap coefficient X İ in Equation (6.5/102a) Without load sharing, the flash temperature at the tip of the pinion is converted by Xİ to a mean flash temperature along the flank. In the equations given, the approximation of a linear curve of the flash temperature along the line of action is assumed. Refer to [6.5/97] for further relation-ships. The following applies for 1 İ Į < 2 and İ1, İ2 < 1 (pitch point in the single meshing region): ()22 12 1 2 110.7 0.22 0.52 0.62Xεα α =ε + ε − ε + − ε ε εε (6.5/108a) If İ1 or İ 2 1 is true (pitch point in the double meshing region), then the following applies: 22 1,2 2,1 1,2 2,1 1 2 110.18 0.7 0.82 0.52 0.32X αε =ε + ε + ε − ε − ε ε εε (6.5/108b) (First index for İ1 > 1; second index for İ2 > 1) • Lubrication factor X S in Equation (6.5/102b) This factor accounts for the fact that heat transf er is better in splash lubrication, compared to injection lubrication. X S = 1.0 for splash lubrication X S = 1.2 for injection lubrication XS = 0.2 for gears running completely submerged in oil • Flash factor X M in Equation (6.5/103) Through the flash factor, the influence of the material characteristics of pinion and wheel on the flash temperature is accounted for. In cases of the same materials and martensitic steels with mean values, the following can be used: X M = 50 KN-3/4 s1/2 m-1/2 mm 6.5 Analysis of Load Capacity 347 • Geometry factor for the pinion tooth tip XBE in Equation (6.5/103) This accounts for the Her tzian contact stress and the sli ding speed on the pinion tooth tip: () ()E2 E1 2 BE 1/4 2E1 E20.51 1z uXuz− =+NN NN (6.5/109) with 22 E1 1 b1 E2 wt E1 0.5 and sinadd a =− = α −NN N • Angle factor X Įȕ in Equation (6.5/103) The influences of pressure angle Įt, operating pressure angle Įwt and helix angle ȕ on the integral temperature are registered. For Įn = 20 the following approximation can be used: X Įȕ = 1 • Mesh factor XQ in Equation (6.5/103), (refer to Figure 6.5/47) In this way the effect of the beginning of contact on the tooth tip of the drive gear in the region of high slip is incorporated in the calculation: XQ = 1.0 for İf/İa 1.5 ()Qf a41.4 /15X=− ε ε for 1.5 < İf/İa < 3 (6.5/110) XQ = 0.6 for İf/İa 3 Driving pinion: İf = İ 2, İa = İ 1 Driving gear: İf = İ 1, İa = İ 2 • Tip relief factor XCa in Equation (6.5/103) Tip relief serves to release overloading caused by elas- tic deformations in the meshing teeth, which occurs in the regions of high slip. This correction has a positive effect on the flash temperature and is registered by X Ca. For gear tooth quality 7 and finer, according to DIN 3961 (ISO 1328) the following applies: 2 aa Ca max max eff eff1 0.06 0.18 0.02 0.69CCXCC =+ + ε + + ε (6.5/111) Figure 6.5/47 Mesh factor XQ Figure 6.5/48 Tip relief factor XCa If Ca > C eff, is true, then Ca = Ceff is to be used in Equation (6.5/111). The decisive tip relief Ca (pinion or gear) is to be determined as follows: 348 6 Load Capacity and Running Performance of External and Internal Gearing - Driving pinion and İ1 > 1.5 İ 2 or Driving wheel and İ1 > (2/3) İ2: Ca = Ca1 for Ca1 Ceff Ca = Ceff for Ca1 > Ceff - Driving pinion and İ1 1.5 İ2 or Driving wheel and İ1 < (2/3) İ2: Ca = Ca2 for Ca2 Ceff Ca = Ceff for Ca2 > Ceff For a full load the effective tip relief C eff results, with which the elastic deformation is offset by flank modification: Spur gearing ()CKFb ceff At=′/ (6.5/111a) Helical gearing () CKFb ceff At=/ γ (6.5/111b) Here the following applies: c' tooth pair stiffness, cȖ mesh stiffness, according to Section 6.2.4 • Running-in factor X E in Equation (6.5/103) () XR EEa redC=+ −1130 ΦN (6.5/112) with ĭE = 1 for fully run-in gearing ĭE = 0 for newly manufactured gearing • Relative structure factor X WrelT in Equation (6.5/105) The effect of differences in materials and heat treatment on the integral temperature for scuffing is registered in the relative structure factor: XX XW rel TW WT= (6.5/113) with XWT = 1.0 for the FZG gear test. Structure factors XW which were obtained empirically can be found in Table 6.5/16. • Roughness factor X R in Equation (6.5/104) 0.25 a R redC2.2RX = N (6.5/114) • Lubricant factor XL in Equations (6.5/104), (6.5/105a) and (6.5/105b) The values correspond to those found under Equation (6.5/89). The general influence factors for determining the decisive circumferential load, with reference to the face width wBt according to Equation (6.5/100), are to be determined as follows: Application factor KA and dynamic factor Kv according to Section 6.3 Face factor KBȕ = KHȕ and transverse factor KBĮ = KHĮ according to Section 6.4 Helix angle factor KBȖ according to information given for Equation (6.5/89a) 6.5.5.7 Test Methods for Scuffing Load Capacity The calculation methods for scuffing load capacity cannot be used independently of parameters that must be established through experiment. Tradition and many years of experience lead to the construction of a back-to-back gear test rig with clearly defined technical data. It is also known as the FZG tester (FZG = Technical Institute for the Study of Gears and Drive Mechanisms of the Technical University in Munich) and is standardised in DIN ISO 14635-1 [6.5/71]. The test rig is depicted in Figure 6.5/49. The aim of the test is to determine the scuffing load capacity of a lubricant under specific operating conditions. 6.5 Analysis of Load Capacity 349 Figure 6.5/49 Back-to-back gear test rig with mechanical circulating power (a = 91.5 mm) Table 6.5/17 depicts a selection of gear data. Table 6.5/18 depicts the envisaged test conditions. DIN ISO 14635-1 refers to the standard test A/8.3/90. Table 6.5/17 Gear Data for Test Gear Pairing A; DIN ISO 14635-1 [6.5/71] Table 6.5/18 Test Conditions of the FZG Back-to-Back Gear Test Rig; DIN ISO 14635-1 [6.5/71] Duration per load stage 21700 revolutions of the motor (roughly 15 min) Motor speed 1450 min-1 3% Oil fill level 1.25 l Oil temperature at the beginning of load stage 1 Room temperature Oil temperature at the beginning of load stage 5 and in each subsequent load stage (90 3) C (must be preset on the temperature controller) Name Dimension Size a b Į m z1 z2 x1 x2 ȕ Material mm mm mm - - - - - 91.5 20 20 4.5 16 24 0.8532 -0.50 0 Alloyed case-hardened steel 350 6 Load Capacity and Running Performance of External and Internal Gearing Table 6.5/19 lists 12 power levels for the test. The test is principally carried out by running through each load stage in rising order with the envisaged starting temperature of the lubricant under the specified test conditions. The load stage at which heavier damage occurs on the tooth flanks is the damage load stage. Table 6.5/19 Load Stages in FZG Test Method A/8.3/90; DIN ISO 14635-1 [6.5/71] Load stage No. Torque on pinion M1T Nm Tooth normal load Fn N Hertzian contact stress at the pitch point ıH0 N/mm2 Total work transmitted by the test gears by the end of the load stage A kWh 1 2 3 4 5 6 7 8 9 10 11 12 3.3 13.7 35.3 60.8 94.1 135.5 183.4 239.3 302.0 372.6 450.1 534.5 99 407 1 044 1 799 2 786 4 007 5 435 7 080 8 949 11 029 13 342 15 826 146 295 474 621 773 929 1 080 1 223 1 386 1 539 1 691 1 841 0.19 0.97 2.96 6.43 11.8 19.5 29.9 43.5 60.8 82.0 107.0 138.1 Figure 6.5/50 Example for determining the damage load stage (transition to high-level wear); DIN ISO 14635-1 [6.5/71]: E = sum ǻm of pinion and wheel after load stage 4; V = sum ǻm of pinion and wheel after load stage before the damage occurs According to DIN ISO 14635-1, the load stage at which the sum of the tooth damage (width of all scratches and scuffing marks) of the tooth flanks of the pinion exceeds 20 mm is called the damage load stage. Figure 6.5/50 illustrates this point using gravimetric analysis, in which, after each load stage, the mass change ǻm for pinion and wheel is determined by weighing. The mass changes ǻm of pinion and wheel after load stage 4, as well as after the load stage just before damage occurs, are depicted by the points E and V. If no such jump occurs, then the damage load stage is above 12. The FZG test method A/8.3/90 for determining the scuffing load ca-pacity of lubricants is adequate for most appli-cations in industrial and marine transmissions. For lubricants that have to bear very high loads, such as those in vehicle gear transmissions, the more intensive test A/16.6R/120 [6.5/93] was developed, and to determine scuffing load capacity and wear characteristics of low-viscosity grease for gears, the test A/2.8/50 [6.5/94]. More information can be found in Appendix 13. 6.5 Analysis of Load Capacity 351 6.5.6 Micropitting Load Capacity Micropitting is also known as grey-staining. It is likewise a consequence of fatigue and often occurs in cases of loads below the pitting load capacity of very hard gears. It starts in the region of negative slip ([6.5/74] to [6.5/77], see also ISO/CD TR 6336-7). However, the pits are not as deep (approxi-mately 20 ȝm). Macroscopically, these regions have a grey appearance because they are grouped very close together (Figures 5.1/4 to 5.1/6). The incipient cracks are at an angle to the surface of less than 10 to 45 and form in the direction opposite to the effective friction force (Figure 6.5/51). Because of the lower depths, micropitting is not as dangerous as pitting. However, larger breaks (pits) often form later out of these regions, which is why this phenomenon is only classified as permissible in exceptional cases. Micropitting causes increa-sing additional internal dynamic load and louder noise. In favourable cases, micropitting leads to improved load distribu-tion. Based on experience, micropitting can be counteracted by the following measures: • Lower roughness • A lubricant with higher service viscosity • Running-in • Coating, among others copper-plating • Additives Figure 6.5/51 Primary direction of incipient cracks in the formation of micropitting Micropitting occurs as a result of fatigue in the boundary layer, whose strength and smoothness is influenced by the lubricant and additive ([6.5/76] among others). The path of the crack is located in the direction of the main shear stresses. Furthermore, research results hitherto allow for the conclusion that, for certain lubricating film thicknesses h c, the roughness peaks exert an influence in the course of contact stress. Compared to the values which can be calculated according to the conventional theory with the radii of curvature of the involute profile, assumed as deviation-free, this causes higher local contact stresses in the contact region. But because of the smaller radii of curvature, the maximum of strain caused by roughness moves closer to the surface, which explains the pits with smaller depths. Determination of the contact stresses due to the roughnesses cann ot be determined in calculations merely with simple models, because the curvatures of the roughness peaks change drastically already due to the elastic deformations. There is an increased risk of micropitting when the ratio of lubrication film thickness to roughness λ goes below a critical value λkrit. Thus the safety factor for micropitting formation is GF GFmin kritȜ ȜSS=≥ (6.5/115) where S GF must be calculated in the entire field of action. Newer documentation is available for this [6.5/124], [6.5/125]. A simplified method is given as follows, which, at the very least, allows an estimation [6.5/74], [6.5/76]. It limits itself to a calculation in the pitch point and roughly includes the temperature and the behaviour of the lubricant in the lubrication gap. The ratio Ȝ is C krit aȜ Ȝh = R≤ (6.5/116) Here the following applies: Ra Mean roughness value, Equation (6.5/86) hC Lubrication film thickness in the pitch point (minimal calculated EHD film thickness), Equation (6.5/116a) 352 6 Load Capacity and Running Performance of External and Internal Gearing Ȝkrit Critical film thickness, to be determined in tests; rough approximation: Figure 6.5/52 (other important influences from additives, material and its surface quality) For the calculated film thickness hC (in m) the Equation (6.5/115) is given [6.5/74]: 0.26 0.3 HȜ 0.7 0.7 C CM 0 C0.0047 856 = Ȟ hσ− Σ N/ (6.5/116a) Here the following applies: Equivalent curvature radius at the pitch point (in mm) Ct 2w bsinĮ = cos ȕ ( 1 )ua u⋅ +N M0 Kinematic viscosity at bulk temperature shortly before meshing begins (in mm2/s); [for the effect- tive temperature in the lubrication gap refer to Equation (6.5/96b)] , approximation for OM o [6.5/74] Sum of speeds CΣ/(in m/s) Cww t2 sinĮΣ=// (w/circumferential speed in the pitch circle) ıHȜ Hertzian contact stress in Z İ = Zȕ = ZB = KHĮ = 1 (in N/mm2) () HȜ HB H Į ȕİZZZ K =σσ (Refer to the FVA working paper on research project 259/I, II+III.) The critical ratio λkrit can be roughly adopted as shown in Figure 6.5/52 [6.5/76]. The newer studies and proposals are aimed at including more precisely the lubrication gap temperature, pressure viscosity, and additives actively in effect. For more information, see [6.5/107], [6.4/124] and FVA booklets 106, 152, 337, 396; FVA information sheet No. 54/7. Figure 6.5/52 Critical operating range for micropitting In order to test the influence of oils on the formation of micropitting, the application of a test in the FZG back-to-back gear test rig in analogy to DIN 51354 is usual (refer to [6.5/75]). Table 6.5/20 depicts the test conditions. Table 6.5/21 lists the torque levels. Table 6.5/23 lists the measurement data of the test gears (“C gearing” bala nced in speed), and Table 6.5/24 lists the workmanship characteristics of the test gears. Following running-in, the load is increased in stages in analogy to the FZG oil test pursuant to DIN 51354, starting with stage 5 up to damage load stage 10 at the highest. The damage load stage is deemed to have been reached when the mean value of the profile form deviation f f of three teeth (flanks) uniformly distributed in circumference amounts to ff ≥ 7.5 ȝm. If damage occurs in the 8th, 9th or 10th stage or if the flanks remain free of damage altogether, then a continuous endurance test is carried out. This runs a maximum of 80 h in the 8th load step and 5 times 80 h in the 10th load step. As usual the test is stopped in cases of scuffing and also micropitting damage, if these occur at a stage lower than 8. 6.5 Analysis of Load Capacity 353 During the continuous test, the test is stopped in the case of a mean profile form deviation of ff ≥ 20 ȝm. Likewise in the case of micropitting and a pitting area of 4% per tooth flank, micropit- ting damage is deemed as having occurred. Table 6.5/25 lists the properties of the micropitting load capacity classes. Because of the spread in the geometric deviations of the surfaces and the structure the results are spread also, so actual statistically based tests are necessary in order to achieve a reliable classification. Test results until now have shown that additives have a larger influence than does the viscosity. Even though lubricant manufacturers possess good expert knowledge, reliable assertions regarding the effectiveness of oil additives under certain operating conditions are dependent on experiments. Table 6.5/20 Test Conditions for the Micropitting Test Pinion speed n1 Roughly 2250 min-1 (high-speed gears n1 4500 min-1) Circumferential speed v at the pitch circle 0.00383 n1 = 8.6 m/s ( v 16.6 m/s at n1 = 4500 min-1) Driving test gear Pinion Lubrication Injection lubrication - Approx. 2 l/min in the mesh, total quantity of oil approx. 25 l - Paper filter 10 ȝm Injection temperature (90 2) C; (40 C or 60 C under specific condition) Time running-in 1.3 105 pinion revolutions (roughly 1 h, n1 = 2250 min-1) Time per load stage in stage test 2.1 106 pinion revolutions (roughly 16 h, n1 = 2250 min-1) Controls in continuous tests after 10.5 106 pinion revolutions (roughly 80 h, n1 = 2250 min-1) Table 6.5/21 Torque Levels for FZG Micropitting Test Load stage No. Torque on pinion M1T [Nm] Hertzian contact stress ıH [N/mm2] Running-in 5 6 7 8 9 10 28.8 70.0 98.9 132.5 171.6 215.6 265.1 510.0 795.1 945.1 1 093.9 1 244.9 1 395.4 1 547.3 Table 6.5/22 Empirical Values to Convert to the Reference Load Stage SK(0.5) = SK( Ra) – ǻ SK with Ra = 0.5 ȝm Flank Ra Correction value ǻSK ISO VG (tested lubricant) 0.4 ȝm 0.6 ȝm 32 + 1 ... 1.5 – (1 ... 1.5) 46 + 1 – 1 100 + 0.5 – 0.5 280 less than + 0.5 less than – 0.5 460 less than + 0.5 less than – 0.5 354 6 Load Capacity and Running Performance of External and Internal Gearing Table 6.5/23 Data of the Test Gears (“C gearing”) for the Micropitting Test (Various FVA Working Papers for Project 259/II+III ) Name Formula symbols Dimension Size Centre distance a mm 91.5 Face width b mm 14 Pitch diameter Pinion dw1 mm 73.2 Wheel dw2 mm 109.8 Tip diameter Pinion da1 mm 82.46 -0.087 Gear da2 mm 118.36 -0.087 Module m mm 4.5 Number of teeth Pinion z1 - 16 Gear z2 - 24 Profile shift factor Pinion x1 - 0.1817 Gear x2 - 0.1715 Pressure angle Į 20 Įwt 22.44 Helix angle ȕ 0 Tooth modification Without tip or root relief, no lead crowning Table 6.5/24 Workmanship Characteristics of the Test Gears for the Micropitting Test Material 16MnCr5 according to DIN 17210, DIN EN 10084 Heat treatment The test gears are case-hardened. Surface hardness 750 HV1, measured on the ground flank; case-hardening depth: CHD 550 HV1: 0.8 to 1.0 mm (after grinding) Core strength 1000 to 1250 N/mm 2 Under a microscope at a magnification of 500X the edge layer may have a residual austenite content of 20 % Gear tooth quality 5 according to DIN 3962; f f should be less than 5 m Base tangent length over three teeth: pinion 0.11 0.135 34.779−−; gear 0.110.13535.252−− Permissible fluctuation RW: each RW = 0.010 mm ( RW waviness) Roughness Ra, tooth flanks 0.5 0.1 ȝ m in direction of involute Grinding MAAG smooth grinding Table 6.5/25 Micropitting Load Capacity Classes GS class Damage load stage DLS in stage test Damage behaviour in continuous test Low micropitting load capacity DLS 7 Large micropitting area, micropitting more than 50% to some extent 1 x 80 h running time at stage 10 Micropitting f f clearly more than 20 ȝm Medium micropitting load capacity DLS 8 to 9 Medium-size micropitting area Micropitting approx. 30 to 50 % 1 to 2 x 80 h running time at stage 10 Micropitting f f 10 to 20 ȝm and/or pitting High micropitting load capacity DLS ≥ 10 Little to no micropitting Micropitting 20 % 1 to 5 x 80 h running time at stage 10 Possible micropitting (f f < 20 ȝm), pitting 6.5 Analysis of Load Capacity 355 The influence of roughness is much stronger for oils with lower viscosity. In tests, roughness should be in the range of Ra = 0.5 0.1 ȝm. It proved favourable to convert the results to a reference value of Ra = 0.5 ȝm (Table 6.5/22). It is known that the additives react with the component surface and change the properties of the edge layer. Since micropitting forms closer to the surface than does pitting, the influence of additives on the micropitting load capacity is also stronger. 6.5.7 The Load Capacity of Wear The following has been defined: “Wear is the progressive loss of material from the surface of a solid body, evoked by mechanical causes, i.e., contact and relative movement of an opposing solid, liquid or gaseous body.” With respect to the acting mechanisms, roughly classified, we are talking about wear due to • adhesion, • abrasion and • tribochemical reactions. The causes are also often subdivided into fatigue, plastic deformation, micro-cutting, and destructtion in the depths. Usually several mechanisms exert an influence at the same time. The literature ([6.5/83] to [6.5/90] and DIN 50320]) contains detailed classifications. Figure 6.5/53 depicts characteristic curves of wear for different mechanisms [6.5/83]. In gear engineering pitting is not counted as wear because it doesn’t act as an area itself to the same extent. Figure 6.5/53 Characteristic curve of wear over time for different mechanisms [6.5/83] For steel gears, wear is a kind of damage that limits the lifetime only in exceptional cases. These exceptions are the • adhesive wear in very slowly running gears [6.5/84], • abrasive wear in the pairing of hard and rough with soft, or in cases of lubricants containing impurities [6.5/85] and • failure of the lubrication. In cases of very slowly running gears the hydrostatic effect becomes so weak that a large increase occurs in friction and wear. The film thickness hmin i s v i e w e d a s a p a r a m e t e r . A t hmin < 0.5 ȝm, wear is often the main kind of damage. For the equidistant linear erosion due to wear W1 in mm/h, which is characteristic of cases of marked wear, the following is true as an approximation [6.5/81], [6.5/84]: 356 6 Load Capacity and Running Performance of External and Internal Gearing CW H 11 h HT CT WT1.4ȗ ı ıȗWc N =⋅ ⋅ ⋅ N N (6.5/117) with W1 Erosion due to wear (equidistant ablation) in mm/h σH Hertzian contact stress of the gearing to be calculated σHT Hertzian contact stresses of the test gear set (reference pairing, Table 6.5/26) CN Equivalent radius of curvature of the flank pairing to be calculated; 1/CN=1/C1N+1/C2N CTN Equivalent radius of curvature of the test gear set (reference pairing); CTN = 8.4 mm ȗW Assessed specific sliding ȗW =tip1 1 tip2 2ȗİ ȗİ ′′⋅+ ⋅ ; 1 Į 2 İ ;İ EC g AC gα′′== ; (1) driving ȗWT Mean specific gliding of the test gear set (reference pairing); ȗWT = 0.74 Nh Load cycle per hour The wear coefficient c1 is to be determined through tests or from information given in literature (e.g., [6.5/81]), from which the following empirical Equation (6.5/118) was also formed: () p CT C 11 T c = c hh (6.5/118) Here the following apply: c1T Wear coefficient, test gearing (reference gearing); Table 6.5/26 hCT/hC Lubricating film thickness ratio; hCT test gearing, Table 6.5/26, hC gearing to be calculated Equation (6.5/83) with hC = hmin,isoth p Exponent for influence of the lubricating film thickness; Table 6.5/26 The quality of the curve of wear of slowly running gears, ascertained experimentally by Plewe , is depicted in Figure 6.5/54 [6.5/84], [6.5/86]. The appearance of high-level wear below a rolling speed of v = 0.5 m/s is typical in cases of hardened and quenched and tempered gears. The pairing of hard and soft results in high wear values. Further works based on Plewe are aimed at ascertaining local wear and at internal gearing (Schudy, J.: “Tests on flank load-bearing capacity of external and internal gearing” (draft) Dissertation TU Munich 2010). In connection with attempts to further penetrate the wear problem theoretically, the name Dierich [6.5/87] should be mentioned, who modelled the surface as a stochastic process field. This was the basis on which Vo iek developed further approaches for the theory of wear forecasting among gear pairs [6.5/88]. Abrasive wear due to impurities in oils was studied by Stoew [6.5/85]. For the range studies (v w = 4 to 13 m/s; ıH = 150 to 650 N/mm2, C45 quenched and tempered) a dependency on sliding path could be found, but none on sliding speed. With the number of cycles of rolling and the load, the wear increased arithmetically. Permissible wear is to be determined according to multiple criteria. These include • Minimum breakage safety of tooth root (reduction of tooth thickness) • Maximum permissible quantity of abrasion in the lubricant • Pointing of the teeth or minimum tooth tip thickness • Weakening of the case hardness • Maximum **backlash** • Deterioration of the tooth form among gears which occasionally run quickly (maximum permissible helix form deviation decisive, increase in dynamic load and noise) 6.5 Analysis of Load Capacity 357 So-called running-in wear is a desirable phenomenon. Here a smoothing of the surface occurs, through wearing away of roughness and waviness. This results in a drastic increase in scuffing load capacity and an improvement in noise characteristics. Moreover, general improvements in the contact pattern occur, such as an improvement in load distribution along the face width. In cases of optimal execution of the running-in process, the final roughness is often unrelated to the roughness immediately after manufacturing. However, this is not irrelevant, because by the time it reaches the final state it will have had a damaging effect. For a quicker and more effective completion of running-in, one sometimes uses running-in oils. They react slightly with the flank surface and thus promote a wearing-away of the material without the occurrence of any scuffing or similar. After completion of the running-in process, these oils are to be replaced with the usual oils for continuous operation. Table 6.5/27 lists information for roughly accounting for the running-in amounts when calcu- lating transverse load distribution and face load distribution. Figure 6.5/54 Wear progression according to Plewe [6.5/84] Wear occurs as the lifetime delimiting damage in cases of very slowly running gears, inadequate lubrication, very large differences in hardness between the paired gears, along with a rough surface of the harder gear, and through the influence of abrasive media and plastic gears or gears made of laminates. For case-hardened gears, lubrication with standard transmission oils and rolling speeds of vw > 0.5 (to 1) m/s, wear manifesting itself as an area does not represent lifetime delimiting damage. On the contrary, it usually has the effect of increasing the load capacity because load distribution improves and the scuffing and micropitting load capacity limits increase. That is why targeted running-in is often specifically stipulated. The available performance and the potential of the system often present limits to this. Some developments in fine working, such as super finishing, act in anticipation of the effects of running-in. 358 6 Load Capacity and Running Performance of External and Internal Gearing Table 6.5/26 Guide Values for Determining Wear W1 according to Equation (6.5/118); Determination of the Wear Coefficient c1T according to Equation (6.5/119) for Wear during Slow Operation (Basis [6.5/84]), Lubrication: Mineral Oil without EP Additive Cons. No. Gear pairing Reference contact stress ıHT Wear coeffi- cient c1T Film thickness (reference value) hCT Exponent (influence of film thickness) p Validity hCmin hCmax N/mm2 mm/rev. ȝm m ȝm 1 Case-hardened/quenched and tempered (18CrNiMo7-6 / 42CrMo4) 635 1.5 10-7 10-2 0.67 5 10-3 0.5 2 Case-hardened/case-hardened (18CrNiMo7-6 / 18CrNiMo7-6) 1160 3 10-8 10-2 3.35 10-2 6 10-2 3 Quenched + tempered/quenched + tempered (42CrMo4 / 42CrMo4) 635 10 -8 10-2 1.53 5 10-3 10-1 Table 6.5/27 Running-in Wear Material and heat treatment Circumferential speed / m/s Amount of running-in ()Įȕ;0 . 5y y m Įmaxy m ȕmaxy m Steel and cast steel Normalised or quenched and tempered 5≤ Hlim160 ıf without restriction 5</10≤ Hlim12800 ı Hlim25600 ı > 10 Hlim6400 ı Hlim12800 ı Cast iron 5≤ 0.275 f without restriction 5< /10≤ 22 44 > 10 11 22 Case-hardened or nitrided steel without restriction 0.075 f 3 6 Notes: Įy Reduction of fpb (for KHĮ); with f = fpb yĮ for (single) gear ȕyReduction of Fȕx (for KHȕ) with ȕEȕZ tb2cos cosff f + =⋅ α β for yȕ Fȕx; yȕ for the pairing In cases of materials with different mechanical properties, the arithmetical mean value of the running-in amounts of both gears of a pairing is generated. These are (in m): f pb deviation of base pitch, f E line of contact deviation (through elastic deformation), f Z line of contact deviation through tooth system deviation. The following applies for case-hardened gears: max f = 40 m. 6.5 Analysis of Load Capacity 359 6.5.8 Symbols and Symbol Explanations of Section 6.5 A kWh transmitted work a mm centre distance BM N/mKs1/2 thermal contact coefficient b mm tooth face width bH mm half-width of Hertzian flattening bK mm width of toothed rim bW mm common tooth face width of a gear pair C1, C2 - weighting factors, determined in tests Ca ȝm amount of tip relief Ceff ȝm effective tip relief Cth - thermal correction factor c Nm/(kg K) specific heat cƍ N/(mm ȝm) single tooth pair stiffness per face width c1T mm/rev. wear coefficient of test gear c1 mm/rev. wear coefficient cȖ N/(mm ȝm) mesh stiffness per face width d mm diameter, general d mm reference diameter dm mm diameter of neutral fibre E N/mm2 module of elasticity Eƍ N/mm2 reduced modulus of elasticity eFik mm-2 influence number of root stress Fb N tooth load (force) at base cylinder Fn N tooth normal load (force) Ft N circumferential load (force) at reference circle in transverse section F\* - relative load fFr, fFt, - restraint factors of toothed rim ff ȝm mean profile form deviation fı - general stress influence function for infinitely wide tooth (without border influence) fıR - general stress influence function with consideration of border influence gĮ mm length of path of contact HHB - Brinell hardness HO - surface hardness in HV h ȝm lubrication film thickness, general h0 ȝm lubrication film thickness in parallel gap hC ȝm lubrication film thickness at pitch point hFa mm bending level arm for load application at tooth tip hFe mm bending level arm for load application at highest point of single tooth contact hmin ȝm minimum lubrication film thickness, general hS mm thickness of connecting shell KA - application factor KAS - application factor for maximum load KBĮ - transverse load factor for scuffing KBȕ - face load factor for scuffing KBȖ - helix factor for scuffing KF - load factor for root stress KĮ - transverse factor for root stress KFȕ - face load factor for root stress KH - load factor for flank pressure KHĮ - transverse load factor for flank pressure KHȕ - face load factor for flank pressure Kmp - factor of load distribution and addi- tional load in case of drivetrain branching Kv - dynamic factor k W/mK heat transfer coefficient kv - velocity exponent Lh h lifetime Lth - thermal load factor l mm gap length at tooth contact lS mm length of connecting shell M Nm torque N - number of load cycles NFlim - number of load cycles at inflexion of S-N curve for root stress NHlim - number of load cycles at inflexion of S-N curve for flank pressure n min-1 number of revolutions np - number of tooth contacts p bar, Pa pressure p - number of planet gears p - exponent for influence of lubrication film thickness pe mm base pitch pH N/mm2 Hertzian pressure pr mm protuberance at tool q N/m distributed load qF - exponent of S-N curve for root stress qH - exponent of S-N curve for flank pressure Ra ȝm arithmetic mean roughness Rm N/mm2 tensile strength Rp0.2 N/mm2 yield stress Rz ȝm mean roughness rb mm base circle radius SB - calculative safety factor for integral temperature SBmin - lowest necessary safety factor for contact temperature SF - calculative safety against fatigue crack SFmin - required minimum safety for avoiding fatigue cracks SFSt - calculative safety for avoiding consistent deformation, fast fracture or primary cracks in case of maximum load 360 6 Load Capacity and Running Performance of External and Internal Gearing SFStmin - minimum safety for maximum load SH - calculative safety for avoiding ma- terial fatigue due to flank pressure SHmin - required minimum safety for avoiding material fatigue due to flank pressure SHSt - safety against damage due to maximum (impulsive) load SintS - calculative scuffing safety for integral temperature SSmin - lowest required scuffing safety for integral temperature s mm tooth thickness at reference cylinder sFn mm tooth root thickness at 30 or 60 tangent of equivalent spur gear spr mm residual protuberance (gear) sR mm toothed rim thickness tg mm depth of grinding step, measured in notch perpendicular to 30 tangent t0 - stock amount u - number of tooth ratio / m/s circumferential velocity at reference cylinder /\* - relative velocity /hd m/s hydraulic effective velocity /T m/s velocity component perpendicular to line of action /( m/s sum of tangential velocities W1 mm/h equidistant linear wear amount Wb mm3 section modulus w N/mm line load wBt N/mm circumferential force per face width XE - running-in factor XJ - approximation factor XL - lubricant factor XM K N-3/4s1/2m-1/2 m flash factor Xmp - contact factor XQ - mesh factor XR - roughness factor XS - lubrication factor XW - structure factor XWrel - relative structure factor XĮȕ - angle factor Xİ - overlap factor Xī - load distribution factor x - profile shift coefficient (addendum modification coefficient) YA - operating mode factor YF - tooth form factor for load application at highest point of single tooth contact Y Fa - tooth form factor for load application at tooth tip YFS - tip factor for load application at tip YN - life load cycle factor (tooth root) YR - roughness factor (tooth root) YS - stress concentration factor for load at highest point of single tooth contact YSFt - stress concentration factor due to Ft YSMb - stress concentration factor due to Mb YSrel - relative stress concentration factor of load application at tip for considering the grinding step Y SZ - stress concentration factor due to tensile stress in the toothed rim YT - technology factor (tooth root) YX - size factor (tooth root) Yȕ - helix factor (tooth root) Yį - notch sensitivity for endurance load YįT - notch sensitivity Yį of test gear YįSt - notch sensitivity for maximum load YįStT - notch sensitivity YįSt of test gear Yİ - contact ratio factor for root stress Yİn - contact ratio factor for conversion of nominal stress from tip loading to loading at highest point of single tooth contact Z B,D - single tooth contact factor ZE mm / N2 elasticity factor ZH - zone factor (pitting) ZL - lubricant factor (pitting) ZN - life load cycle factor (pitting) ZR - roughness factor (pitting) Zv - velocity factor (pitting) ZW - material pairing factor ZX - size factor for flank pressure Zȕ - helix factor (pitting) ZF - contact ratio factor (pitting) z - number of teeth znx - equivalent number of teeth Į pressure angle ĮFe load application angle at highest point of single tooth contact Įp m2/N pressure viscosity coefficient Įw working pressure angle Į\* 1/K viscosity temperature coefficient ȕ helix angle at reference cylinder ȕb base helix angle īy - parameter at line of action ǻm mg mass change ǻK K temperature difference İ1, İ2 - partial overlap [tip/pitch point of pinion (1) respectively wheel (2)] İa - leaving overlap coefficient (pitch point/tip, driving gear) İf - arriving overlap coefficient (root/pitch point, driving gear) İmax - maximum value of İ1 and İ 2 6.5 Analysis of Load Capacity 361 İĮ - profile contact ratio (transverse contact ratio) İȕ - axial contact ratio İȖ - total contact ratio ȗW - mean specific sliding Ș mPa s dynamic viscosity Ș0 mPa s dynamic viscosity at surface temperature of the teeth before mesh and at atmospheric pressure ȘM mPa s dynamic viscosity at bulk temperature Ș mPa s dynamic viscosity at oil temp. ș tangent angle KB C actual contact temperature KBmax C maximum contact temperature versus line of action Kfla K flash temperature Kfla int K mean flash temperature versus line of action Kfla max K maximum flash temperature versus line of action KflaE K flash temperature at tip contact point E of the pinion Kint C integral temperature KintP C allowed integral temperature KintS C scuffing integral temperature KM C bulk temperature K C oil temperature KS C scuffing temperature Ȝ N/(s K) heat transfer number Ȝ - specific lubrication film thickness Ȝkrit - critical specific lubrication film thickness - number of friction mC - number of friction averaged versus line of action ȝZ - number of tooth friction Ȟ - Poisson’s number Ȟ40 mm2/s kinematic viscosity at 40 C ȟF\* mm distance from load application in relation to module ȟR\* mm distance from closest tooth end in relation to module N, ȡ kg/m3 density N, ȡ mm equivalent radius of curvature N1, N2, ȡ1, ȡ2 mm curvature radii Na0, ȡa0 mm radius of tool tip rounding NFn, ȡFn mm curvature radius of root fillet at 30 or 60 tangent of equivalent spur gear Ng, ȡg mm curvature radius of grinding notch fillet Nred, ȡred mm reduced curvature radius in normal section ı N/mm2 stress, general ı0.2 (Rp0.2) N/mm2 yielding strength of core material ıF N/mm2 local tooth root stress ıFEW N/mm2 tooth root strength for pure alternating load ı FE N/mm2 endurance strength, fatigue strength ı FEm N/mm2 fatigue strength dependent on mean stress ı FZZ N/mm2 limit strength (tooth root strength for maximum load (= ıFSt; ≈Rm) ıFm1 N/mm2 mean stress, independent of tooth load ıFm2 N/mm2 mean stress, dependent on tooth load ıFmax N/mm2 maximum local root stress during lifetime ıFnenn N/mm2 reference value of tooth root stress (nominal stress) ıFnenn max N/mm2 maximum nominal tooth root stress ıFP N/mm2 permissible tooth root stress for endurance loading ıFPmax N/mm2 permissible maximum tooth root stress for several peak loads ı H N/mm2 Hertzian pressure, existing stress ıHlim N/mm2 flank pressure endurance limit ıHmax N/mm2 maximum flank pressure during lifetime ıHP N/mm2 permissible flank pressure ıHSt N/mm2 flank strength for maximum load Ĳ N/mm2 shear stress, general ĳ rotation angle Ȥ mm-1 relative stress gradient ȥb base thickness half-angle Ȧ s-1 angular frequency Indices 0 operating temperature and atmospheric pressure 0 base value 1 gear 1 (pinion) 2 gear 2 A point A on path of action a tooth tip B point B on path of action B scuffing load b base cylinder b operating status C pitch point D Point D on path of action E Point E on path of action e highest point of single tooth contact F tooth root stress f tooth root H flank pressure hd hydrodynamic isoth isothermal theory 362 6 Load Capacity and Running Performance of External and Internal Gearing j load step j m mean value max maximum min minimum N normal loading n normal section oil lubrication oil r radial component red reduced rel relative value St static loading T test gear T shear stress t transverse section t tangential component th thermal theory w working pitch circle x coordinate x y coordinate y y arbitrary cylinder z coordinate z 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 6.6.1 Power Losses 6.6.1.1 Total Losses The total losses in a cylindrical gear are composed of various components. The total power loss is given by P PPPPVV ZV LV DV X=+++ (6.6/1) where P VZ is the gear power loss (Section 6.6.1.2) P VL the bearing power loss (Section 6.6.1.3) P VD the sealing power loss (Section 6.6.1.4) P VX the power loss of other components The power losses of the gears and bearings can be divided into load-dependent and load- independent losses. The ratio of the power loss to the input power P a of the transmission is also known as the degree of transmission loss V: VP P=Va/ (6.6/2) The gear losses form a key proportion of the transmission losses. This is illustrated in Figure 6.6/1 [6.6/1]. Here a comparison is made with the degree of transmission loss as the sum of the gear and bearing losses [6.6/2]. The sealing losses as per Equation (6.6/1) are neglected in this representation. Due to its major share, particular importance is attached to determining the gear power loss as precisely as possible. VZO Degree of gear power loss, idling VZP Degree of gear power loss, under load VLO Degree of bearing power loss, idling VLP Degree of bearing power loss, under load Figure 6.6/1 Total degree of power loss in a transmission V with its components for gears VZ and bearings VL according to Ohlendorf [6.6/1] 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 363 6.6.1.2 Gear Losses The gear losses PVZ result from the load-dependent losses PVZP and the load-independent losses (idling losses) PVZ0: P P P 0VZ VZP VZ + = (6.6/3) Friction losses occurring as heat in the meshing are due to the mutual combined rolling and sliding motion of the tooth flanks during load transmission. The size of the gear power loss primarily depends on the tooth geometry and the surface quality of the tooth flanks. The power loss in meshing is determ ined by the integration of the friction losses across the length of path of contact g Į and averaging out, as follows: PpFgg VZP eng d = 1 0μαα / (6.6/4) where P VZP is the gear power loss p e the normal base pitch F n the normal force of tooth pair ȝ the friction coefficient / g the sliding speed Remark: Fn and /g are the magnitudes existing on each meshing point. To calculate the losses, it shall be provisionally assumed that the gear friction coefficient can be taken as constant on average, except in the region of the pitch point and at the beginning of contact (Figure 6.6/2, [6.6/3]). After the introduction of a gear power loss factor H V, the following can be written for Equation (6.6/4): VZP a mZ VPP H =μ (6.6/5) where ȝmZ is the mean gear friction coefficient, Equation (6.6/7) H V the gear power loss factor, Equation (6.6/13) Pa the input power Taking into account Equation (6.6/2), the degree of gear power loss becomes VZP ZP mZ V aPVHP== μ (6.6/6) The gear friction coefficient has been experimen- tally determined, and computational relationships have been established by various authors [6.6/4]. As the chemical structure of the lubricating oil has a major influence on the friction coefficient, it should be determined experimentally. One of the ways the friction coefficients were determined was in a cylindrical gear transmission test bed for mineral oils and synthetic fluids under specific mean operating conditions [6.6/5]. Various mineral oils of the ISO viscosity classes VG 100 to 220 provided values of ȝ mZ 0.04, while synthetic and partially synthetic fluids (70 to 80% mineral oil content) of VG 68 to 220 resulted in values of ȝmZ = 0.028 to 0.037. The values of ȝ mZ only showed a strong increase with the use of low-viscosity fluids. Figure 6.6/2 Gear friction coefficient across the length of path of contact and mean gear friction coefficient after St el [6.6/3] 364 6 Load Capacity and Running Performance: External and Internal Gearings The rule-of-thumb calculation of the mean gear friction coefficient is based on experimental results obtained by Ohlendorf [2], [6.6/1]: 0.2 0.05 Ab t mZ oil R L Ȉmm/0.048KF bXX− μ= ⋅ η /N (6.6/7) where K A is the application factor F bt/b the nominal normal force in the transverse section in relation to the face width in N/mm For Fbt/b < 150 N/mm, Fbt/b = 150 N/mm should be used in Equation (6.6/7). /Ȉm the mean sum of speeds across the contact in m/s We then obtain (approximately equal to the value in the pitch point) // /ΣΣmC w w t≈= 2s i n α (6.6/8) For /Ȉm > 50 m/s, /Ȉm = 50 m/s should be used in Equation (6.6/7). Nm the mean equivalent radius of curvature in the normal section in mm At the pitch point we approximately obtain mw 1 w t C n b1=0 . 5 s i n Į1c o sȕudu≈+NN (6.6/9) Șoil the dynamic viscosity of the oil at mass temperature, approximately in the case of injection pump or oil sump temperature, in mPas: oil b b b(oil)η≈ η = ν N (6.6/10) Ȟb is the kinematic viscosity at operating temperature in mm2/s as per Figure 6.6/23 Nb(oil) the lubricant density at operating temperature in kg/dm3 From [6.6/6], the density Nb can be set as follows: oilb b2 020 1750−≈−NNN (6.6/11) N 20 is the lubricant density at 20 C (for typical values, refer to Table 6.6/1) K oilb the oil operating temperature X R is the factor for the influence of the roughness: a1 a20.25 2+= RRRX (6.6/12) R a1,2 is the arithmetical mean roughness of the tooth flanks of both gears in ȝm If no R a values are known, typical values depending on the manufacturing process can be taken from Table 6.6/2. X L is the lubricant factor; for examples, refer to Equation (6.5/89c) Table 6.6/1 Density of Mineral Oils at 20 C from [6.6/6] Table 6.6/2 Mean Arithmetical Roughness Values Ra of the Tooth Flanks of Cylindrical Gears depending on the Manufacturing Process [6.6/6] ISO viscosity class Density N 20(oil) Manufacturing process Ra in ȝm for module VG kg/dm3 1 to 8 mm > 8 mm 68 100 150 220 320 460 0.870 0.877 0.882 0.889 0.895 0.900 ground shaved milled shaped 0.4 0.8 1.6 3.2 0.8 1.6 3.2 3.2 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 365 The tooth loss coefficient HV, also known as the geometry factor, is calculated based on the assumption that the pitch point lies within the meshing region and acts in double meshing with 50% of the tooth force, as shown by () HV 12 1 1=+ ++− πβεεεzz1 cos 1 b12 22 a (6.6/13) İ1, İ2 are partial contact ratios: εε12==EC p AC p/; /et et Alongside the possibility of precisely recording the losses by means of the PC program LVR [6.4/33], a more recent work is also available on the tooth loss coefficient as a result of theoretical and experimental investigations undertaken by Wimmer [6.6/43] (refer to Appendix 14.1). The idling losses should also be added to the gear losses caused when under load. These include primarily the splashing loss (with splash lubrication) as well as the loss (with injection lubrication) caused by tooth spaces, acceleration and diversion of the injected oil as well as ventilation losses. In analogy to Equation (6.6/2), we obtain for the gear power loss PV PVZ Z a= (6.6/14) where VZ is taken from Equation (6.6/17). In analogy to Equation (6.6/3), we obtain for the degree of gear loss VV VZZ PZ=+0 (6.6/15) Figure 6.6/3 Auxiliary parameters for determining the gear idling loss; a) gear idling loss coefficient fZ0 [6.6/7] 366 6 Load Capacity and Running Performance: External and Internal Gearings Figure 6.6/3 Auxiliary parameters for determining the gear idling loss; b) splash oil coefficient CSp (splash lubrication) according to Mauz [6.6/10] The degree of gear loss when idling VZ0 can be related to the degree of gear loss VZ via the gear idling loss coefficient fZ0: Vf VZZ Z00= (6.6/16) Typical values for fZ0 with spray lubrication in Figure 6.6/3a are from Eiselt [6.6/7]. It therefore follows from Equation (6.6/15) that VV fZZP=−1Z0 (6.6/17) where VZP is taken from Equation (6.6/6). Constructive influences, such as the tooth **backlash**, the gear ratio distribution for a multi-stage gear transmission and the immersion depth on the load-independent transmission losses have been investigated by Leimann [6.6/54] and corresponding conclusions drawn. For high-speed gear drives (turbo drives), the major part of the total loss consists of idling losses. Experimental investigations to determine the idling losses of injection-lubricated cylindrical gears at circumferential speed up to 200 m/s were carried out by Maurer [6.6/44]. The subject matter of this research consisted of ventilation losses, which form the major part of the idling losses, and hydraulic losses. The ventilation losses are primarily influenced by the gearing parameters of face width and module. Empirical equations and theoretical similarity approaches have been deduced to calculate the ventilation loss torque. The ventilation losses have a significant influence for circumferential speed v > 50 m/s. These losses can be clearly reduced by gear boxes vacuumed or filled with thin air or gases with low density (e.g., helium; see patent WO 1996015392 A1). 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 367 Leimann [6.6/6] provides tips for transferring the injection lubrication idling losses to splash lubrication. Thereafter, rough approximations can be made for splash lubrication, as follows: ()Z0T Z0E 1.2 o 3t VV= (6.6/18) Here, the magnitude of the correction value is dependent on the number of gears immersed, the immersion depth, the design of the housing interior, the form of the gear body, and so on [6.6/8]. On behalf of the German Research Association for Power Transmission Engineering (FVA) in Frankfurt/Main, further investigations have been conducted at the Chair for Machine Elements at the Technical University in Munich in recent years (see [6.6/9]). In the meantime, ISO/TR 14179 is applicable, in which the issue of power losses is dealt with [6.6/53]. Splash lubrication The splashing loss of the pairing, expressed by the loss torque M H in Nm of the larger wheel is given by ()MC CC HS p ett=⋅120/// (6.6/19a) where /t0 = 10 m/s. The splash oil coefficient CSp takes into account the splash oil supply dependent on the immersion depth and the direction of rotation; see Figure 6.6/3b (ISO, however, disregards the effect due to the direction of rotation). The coefficients C 1 and C2 specify the influence of the wheel width and the immersion depth as follows: 3 12 1 000.063 0.0128ee bCeb +=+ 12 2 00.280eeCe+=+ (6.6/19b) where e0 = 10 mm and b 0 = 10 mm. For small immersion depths ( e1 pinion, e2 wheel), no influence of the viscosity was measurable, for which reason it was not taken into consideration in the calculation. The idle power loss of each gear can be calculated by multiplying the loss torque MH [from Equation (6.6/19a)] by the angular velocity of the respective shaft. The total idle power loss of the k gear pairings is given by k 3 i VZ0 Hi i11030nPM− =π=⋅ (6.6/19c) where PVZ0 is in kW, MHi is in N m, and ni is the speed of the larger gear in min-1. Injection lubrication The evaluation of the experimental results as per [6.6/10] and [6.6/42] resulted in the following equations: • Injecting into the meshing: () ()0.1 ee 06 Ho ilew 1.5 / 9 1.5 0.065 0.18 0.5 oil w oil t1.67 10 32 10 0.1ts QQMQ d dm b− −=⋅ − + ⋅ν +// /N N (6.6/19d) where the reference oil injection quantity Qe0 = 2 l/min. 368 6 Load Capacity and Running Performance: External and Internal Gearings • Injecting into the end of meshing: ()6 Ho ilew t s 8.33 10 MQ d−=⋅ +// N (6.6/19e) The equations are not dimensionally correct. The constants are selected such that the loss torque MH in Nm is obtained by setting the individual values in the units specified for both equations. The loss torque calculated in this manner also applies to the pairing of gears. The power loss of a gear pair is obtained by multiplying the loss torque M H [either from Equation (6.6/19d) or Equation (6.6/19e)] by the angular velocity Ȧ associated with the pitch diameter d w. The total power loss of all gear pairs is obtained by adding together the individual losses. The scope of application is restricted to the operating and construction parameters contained in Table 6.6/3. Example calculations have revealed that the equations can be used meaningfully far beyond this area. Table 6.6/3 Scope of the Calculation of Gear Loss when Idling in Accordance with ISO/TR 14179-2 [6.6/53] Influence quantity Symbol Unit Range of variation from to Reynold’s number Re = / t da/Ȟoil - 4125 531428 Relative immersion depth 2 e/da - 0.04 2.0 Relative wall clearance sr/da - 0.03 3.15 Tip diameter da mm 132 248 Face width b mm 10 60 Immersion depth e mm 5 135 Module m mm 3 6 Circumferential speed /t m/s 10 60 Kinematic viscosity Ȟoil mm2/s 15 240 Oil density N15(oil) kg/dm30.855 0.881 Once the gear losses have been determined, it is possible to calculate the degree of gearing loss ȘZ. The following relationship applies: ηZZVZ a=− =−11VP P (6.6/20) For multi-stage gear units, the total gear efficiency is obtained by the product of the gear efficiencies of the individual stages: ηη ηZZ I Z I I= ... (6.6/21) Experience has revealed the following gear efficiency for each cylindrical gear unit stage: ȘZ = 0.990 to 0.995 with quenched and tempered as well as milled gearing ȘZ = 0.994 to 0.997 with hardened and ground gearing 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 369 An improvement in the efficiency (reduction of the gear losses) is achievable by means of the following measures: • a small module • a large number of teeth • flank modifications • small flank roughness • low oil viscosity in association with high circumferential speeds • synthetic lubricants • with splash lubrication: immersing only one wheel, low immersion depth • with injection lubrication: the smallest possible injection volume The exchanging of input and output results in different progressions of the gear efficiency across the length of path of contact with alternating directions of the friction force in relation to the running-in side g f or the running-out side ga. Furthermore, since the friction coefficients in the running-in side and running-out side are different, the gear losses and the efficiency are – to a certain extent – dependent on the direction of rotation (current exception: AC CE 1.2 μ≈ μ). 6.6.1.3 Bearing Losses In accordance with Equation (6.6/1), in addition to gear losses, further important friction losses occur in cylindrical gear units, which may not be neglected when determining the gear efficiency and the heat balance. These initially consist of losses in the gear shaft bearings (see also Appendices 14.2 and 14.3). Rolling element bearings (see also SKF Roller Bearing Catalogue, 2008) The power loss for rolling element bearings is calculated from the equation ()VL 0 1 a in kW PM M M M−3 −3=ω ⋅ 1 0 = + + ω ⋅ 1 0 (6.6/22) where M is the friction torque of the rolling element bearing in Nm M0 the friction torque of the rolling element bearing not subject to load in Nm M1 the normal load-dependent component of M in Nm Ma the (additional) load-dependent component of M for axially loaded cylindrical roller bearings in Nm Ȧ the angular velocity; Ȧ = 2ʌ n/60 in 1/s The magnitude of the friction torque not subject to load M0 is primarily obtained as a result of the lubricant friction, the friction of the rolling elements in the cage and the friction of the cage on its guide rails. In conformity with test results, we obtain, as per Equations [6.6/9] and [6.6/53] for Ȟb(oil)n < 2000 mm2/(s min) 38 00 m1.6 10 Mf d−= (6.6/23) for Ȟb(oil)n 2000 mm2/(s min) ()2/331 0 0 0 b(oil) m 10 Mf nd−=ν (6.6/24) where for 0Min Nm: f0 is the bearing coefficient to take into account the bearing type and the lubrication type Ȟb(oil) the operating viscosity of the lubricant in mm2/s dm the mean bearing diameter in mm 370 6 Load Capacity and Running Performance: External and Internal Gearings 2 + =md Dd (6.6/25) n the roller bearing speed in min-1 The factor f0 increases with the magnitude of the rolling elements and thus with the bearing cross section. For the values f0 for oil-bath lubrication as a function of the bearing type design and series, refer to Table 6.6/4. The higher f0 values should be assigned to the wide model series. Note that 50% of these values should be used for grease lubrication and minimal-quantity oil lubrication. Table 6.6/4 Roller Bearing Coefficient f0 for Oil-Bath Lubrication as per [6.6/12]; See also Appendix 14.2 Bearing type f0 Bearing type f0 Deep-groove ball bearings 1.5 – 2 Needle bearings Self-aligning ball bearings NA48, NA49 5 – 5.5 Bearing series 12 1.5 NA69 10 Bearing series 13 2 Tapered roller bearings Bearing series 22 2.5 302, 303, 313 3 Bearing series 23 3 320, 322, 323, 329 4.5 Angular contact ball bearings (single row) 330, 331, 332 6 Bearing series 72 2 Self-aligning ball bearings Bearing series 73 3 Bearing series 213, 222 3.5 – 4 Angular contact ball bearings (paired) Bearing series 223, 230, 239 4.5 Bearing series 32 3.5 Bearing series 231, 232 5.5 – 6 Bearing series 33 6 Bearing series 240, 241 6.5 – 7 Four point bearings 4 Deep-groove axial ball bearings Cylinder rolling bearings (with cage) Bearing series 511, 512, 513, 514 1.5 Bearing series 2, 3, 4, 10 2 Bearing series 522, 523, 524 2 Bearing series 22 3 Axial cylinder rolling bearings Bearing series 23 4 Bearing series 811 3 Bearing series 30 2.5 Bearing series 812 4 Cylinder rolling bearing (full complement) Axial needle bearings 5 NCF18V 5 Axial self-aligning ball bearings NCF22V 8 Bearing series 292E 2.5 NCF29V 6 Bearing series 293E 3 NCF30V 7 Bearing series 294E 3.3 NNC48V, NNCL48V 9 NNC49V, NNCL49V 11 NJ23VH 12 NNF50V 13 The load-dependent friction torque M1 particularly results from the rolling friction between the rolling element and the bearing raceway in the loaded rolling-element bearing. The influence of the rotational speed only plays a minor role. The influencing quantities are the load, the design and the size of the bearing. From [6.6/12] we obtain 3 11 1 m 10 in Nm Mf P d−= (6.6/26) where f1 is the bearing coefficient taking into account the bearing type and bearing load as per Table 6.6/5 P1 the dynamic load (force) for roller bearings as per Table 6.6/5 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 371 Equation (6.6/26) and thus the bearing coefficients to be used differ for other bearing manufac- turers, for example, [6.6/28]. For the differing bearing coefficients f0 and f1 in ISO/TR 14179-2 [6.6/53], refer to Tables A14.2/1 and A14.2/2 in Appendix 14.2. For cylinder rolling bearings with additional axial loading, M from [6.6/12] should be increased by an axially load-dependent friction torque Ma (in Nm): 3 a2 a m0.06 10 Mf F d−= (6.6/27) where Fa is the axial force in N f2 the bearing coefficient taking into account the bearing design For cylinder rolling bearings with a cage, f2 = 0.006 for oil lubrication, and f2 = 0.003 for single- row, full-complement cylinder rolling bearings; f2 can also be determined from ISO [6.6/53] or according to Reference [6.6/12]. Special bearing designs may result in lower load-dependent friction torques M1 and Ma. In this regard, the information provided by the bearing manufacturer should be noted. Table 6.6/5 Bearing Coefficient f1 and Dimensioning Load P1 as per [6.6/12] Bearing type f1 P1 1) Deep-groove ball bearings 0.0005 to 0.0009 ( P0 / C0)0.5 2)Fr or 3.3 Fa – 0.1 Fr 3) Self-aligning ball bearings 0.0003 ( P0 / C0)0.4Fr or 1.37 Fa/e – 0.1 Fr 3) Angular contact ball bearings single row, Į = 15 0.0008 ( P0 / C0)05 Fr or 3.3 Fa – 0.1 Fr 3) single row, Į = 25 0.0009 ( P0 / C0)0.5 Fr or 1.9 Fa – 0.1 Fr 3) single row, Į = 40 0.001 ( P0 / C0)0.33 Fr or 1.0 Fa – 0.1 Fr 3) paired or paired single row 0.001 (P 0 / C0)0.33 Fr or 1.4 Fa – 0.1 Fr 3) Four point bearings 0.001 ( P0 / C0)0.33 1.5 Fa + 3.6 Fr Cylinder rolling bearings (with cage) 0.0002 to 0.0004 2) Fr 4) Cylinder rolling bearing (full complement) 0.00055 Fr 4) Needle bearings 0.0005 Fr Tapered roller bearings, single row 0.0004 2 Y F a or Fr 3) Tapered roller bearings, paired or two single rows in X or O layout 0.0004 1.21 Fa / e or Fr 3) Self-aligning ball bearings Bearing series 213, 222 0.0005 ( P0 / C0)0.33 1.6 Fa/e, if F a/Fr > e F r {1 + 0.6 [ Fa/(e Fr) ]3} if Fa / Fr e Bearing series 223 0.0008 ( P0 / C0)0.33 Bearing series 231, 240 0.0012 ( P0 / C0)0.5 Bearing series 230, 239 0.00075 ( P0 / C0)0.5 Bearing series 232 0.0016 ( P0 / C0)0.5 Bearing series 241 0.0022 ( P0 / C0)0.5 Deep-groove axial ball bearings 0.0012 ( Fa / C0)0.33 Fa Axial cylinder rolling bearings 0.0015 Fa Axial needle bearings 0.0015 Fa Axial self-aligning ball bearings 0.00023 to 0.00033 2) Fa (where F r 0.55 Fa) 1) If P1 < F r, then P1 should be set to Fr. 2) The greater value for the wider series. 3) The greater of the two values should be used in each case. 4) Subjected to radial load only. For cylinder rolling bearings subjected to additional radial load, Ma should be added to the friction torque M1: M = M0 + M1 + M a; Ma as per Equation (6.6/27). 372 6 Load Capacity and Running Performance: External and Internal Gearings As a rough estimate , the friction torque M (in Nm) can be determined via the pitch friction coefficient from Table 6.6/6: 2dMF−3= μ⋅10 (6.6/28) where F is the resulting bearing load: FF F=+ra22 This approximation can be used with sufficient precision if the bearing load F is 0.1C and good lubrication and normal operating conditions are present. Further conditions are listed in [6.6/12]: • no additional load from tilting and radial or axial tension • load angles ȕ, which are normal in the individual bearing designs • radial bearings predominantly radially loaded, cylinder rolling bearings and needle bearings only radially loaded, axial bearings only axially loaded • good lubrication condition, mean speed range (0.3 – 0.7 times the value of the kinematically permissible speed) • bearings without sliding seals Table 6.6/6 Rolling Bearing Friction Coefficient as per [6.6/12] Bearing type Bearing type Deep-groove ball bearings 0.0015 Needle bearings 0.0025 Self-aligning ball bearings 0.0013 Tapered roller bearings 0.0018 Angular contact ball bearings (single row) 0.0020 Self-aligning ball bearings 0.0020 Angular contact ball bearings (paired) 0.0024 Deep-groove axial ball bearings 0.0015 Four point bearings 0.0024 Axial cylinder rolling bearings 0.0040 Cylinder rolling bearings (with cage) 0.0013 Axial needle bearings 0.0050 Cylinder rolling bearing (full complement) 0.0020 Axial self-aligning ball bearings 0.0020 Radial plain bearings The power loss of a cylindrical radial plain bearing PVL (in kW) is calculated as 3 VL r g 10 PF−=μ ⋅/ (6.6/29) where Fr is the radial load in N the friction coefficient of the radial plain bearing / g the sliding speed in m/s It is possible to approximately determine as a function of the Sommerfeld coefficient So as follows: o1 : 3 / oSS≥μ = ψ (6.6/30) o1 : 3 / oSS< μ=ψ (6.6/31) The relative bearing clearance ȥ is the ratio of the (mean) bearing clearance S to the nominal bearing diameter d: ψ=Sd/ (6.6/32) 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 373 The Sommerfeld coefficient So is derived as follows: 2 m boȦpSψ=η (6.6/33) Whereas for the mean area pressure pm we have pF bdmr= (6.6/34) where b is the width of the bearing. For a more precise determination of the friction coefficient , refer to [6.6/13] and [6.6/14]. Axial plain bearing To determine the power loss PVL (in kW) of an axial bearing, Equation [6.6/6] can be used as follows: 3 VL a gm 10 PF−=μ ⋅/ (6.6/35) where Fa is the axial load in N /gm the mean sliding velocity, m gm1000 60dnπ=⋅/ in m/s (6.6/36) d m the mean bearing diameter in mm n the rotational speed of the shaft in min -1 the (approximate) friction coefficient: LB(oil) mgmȘ ȝ3 bp=⋅ ⋅/ (6.6/37) For a precise determination of , refer to [6.6/13], [6.6/14] and [6.6/15] (amongst others) ȘLB(oil) the dynamic viscosity of the lubricating oil in mPas at operating temperature b the segment width in mm p m the mean area pressure 2 ain N/mmFpblz=⋅⋅m (6.6/38) l the length of the segment at the diameter dm in mm z the number of segments 6.6.1.4 Sealing Losses Radial shaft seal The friction losses in the radial shaft seals (RSS) frequently used to seal the shaft outlets in the gear units depend on various factors, such as the material used for sealing, the hardness of the material used for the shaft, the surface roughness of the shaft in the region of the sealing lip, the lubricant used and the temperature at the sealing point. Various seal manufacturers provide information for estimating the power loss for mineral oils. This usually includes the shaft diameter and the circumferential speed or the rotational speed of the shaft directly and applies to specific prescribed installation and operating conditions. For many decades, it has been the practice of the Freudenberg Company to provide a diagram to determine the power loss associated with RSS [6.6/16]. This diagram is shown in Figure 6.6/4 and applies to the use of a motor oil complying with SAE 20 at 100 C. For VG 220 viscosity class gear oils, the friction losses should be increased by 50%. 374 6 Load Capacity and Running Performance: External and Internal Gearings The straight lines in Figure 6.6/4 can be represented by the following equation: 3 VD 0.1475 10 Pd−=⋅ / (6.6/39a) where PVD is the sealing power loss in kW d the shaft diameter in mm / the circumferential speed in relation to d [m/s] If these are used in the conditions specified in [6.6/16] with regard to lubricating oil and temperature, it is then possible to derive a generally valid relationship, which includes the operating viscosity as an influencing quantity on the friction power loss via the nominal viscosity (viscosity class VG from [6.6/17]) and the service temperature, as follows for qualitative orientation: Figure 6.6/4 Friction power loss (here in W) for radial shaft seals for SAE 20 motor oil at 100 C [6.6/16] () ()()21 0 VD oil b145 1.6 350log log 0.8 10 in kW PV G dn− =− + + ⋅ K (6.6/39) where K(oil)b is the oil operating temperature (e.g., oil su mp temperature with splash lubrication) in C VG the viscosity class or nominal viscosity Ȟ(oil)40 in mm2/s n the rotational speed of the shaft in min-1 For more information refer to Equation (6.6/39a). In order to present a graphical display of PVD, Equation (6.6/39) has been depicted in Figure 6.6/5. Here the scope of application has also been quantitatively marked out. The relationship proposed, Equation (6.6/39), includes data provided by the Freudenberg Company (Figure 6.6/5). Figure 6.6/5 Power loss PVD (here in W) of radial shaft seals as a function of lubricating oil and temperature 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 375 6.6.2 Transmission Efficiency The transmission efficiency Ș is determined by the power ratio at the output and input or from the degree of transmission loss, as follows: bV aa11PPVPP== − = −η (6.6/40) The degree of transmission loss V takes due account of all the power losses occurring in the trans- mission [according to Equation (6.6/1)], which were dealt with in Sections 6.6.1.2 to 6.6.1.4. Rule-of-thumb values for the transmission efficiency per gear stage with roller bearing and splash lubrication in association with nominal load and service temperature are • quenched and tempered, milled gears Ș 0.98 to 0.988 • hardened, ground gears Ș 0.985 to 0.995 The effectiveness of a gear unit is expressed by its degree of efficiency. It is influenced by various factors, listed above in Sections 6.6.1.1 and 6.6.1.2. Figure 6.6/6, for example, depicts the degree of gearing efficiency Ș as a function of the degree of gearing capacity utilisation at different speeds as per [6.6/2]. This factor must be taken into account in the practical use of a gear unit. The efficiency specified by the gear manufacturer relates to nominal / full-load operation and is only reached following the running-in of the gear. Figure 6.6/7 [6.6/18] shows how the degree of transmission efficiency is influenced by the gearing efficiency. Doleschel [6.6/46] has investigated a lubricant’s influence on the efficiency of gear drives and developed a computer software program [6.6/47]. The calculation in accordance with ISO [6.6/52], [6.6/53] can be accordingly expanded to include the results presented. Figure 6.6/6 Degree of gearing efficiency Ș as a function of the degree of gearing capacity utili- sation y [6.6/2] Figure 6.6/7 Level of gear efficiency for external gear teeth [6.6/18] 6.6.3 Thermal Balance and Thermal Conductivity 6.6.3.1 Preliminary Considerations The heat generated in cylindrical gear units as a result of friction is dissipated into the environment by convection, radiation and conduction, as well as in some cases via the cooling circuit. A thermal 376 6 Load Capacity and Running Performance: External and Internal Gearings state of equilibrium with a constant mean gear temperature is established once the gear has been in service for a certain amount of time. It is a question of determining this gear temperature or the temperature of the oil in the gearing. This is the purpose of the thermal balance, which has been incorporated into ISO/TR 14179 [6.6/53] (refer to Appendix 14.3). Measures designed to raise the power, involving a reduction in the mass-power ratio of the gear units, lead to the increase of the resulting heat loss per unit of volume. As a lower heat dissipation is associated with a relatively smaller housing surface, it is possible for the gear temperature to reach an inadmissible value. The gear temperature then becomes a decisive design criterion because the following factors are associated with its increase: • a progressive growth of the ageing of the gear oil, • a reduction in the pitting – and particularly the scuffing load capacity – of the gears, and • a reduction in the lifetime of the bearings and the shaft seals. As is partly the case with the calculation of the friction losses, the state of our knowledge in relation to determining the heat dissipation via the gear housing is not always adequate. This is primarily illustrated by the fact that the gear drives used in practice frequently either do not reach or far exceed the theoretically determined thermal power. From a thermo-technical perspective, the gear drive represents an extremely complicated structure, which does not readily allow the transfer of insights gained from simple models. Recent years have witnessed a significant increase in efforts to solve this problem by means of theoretical and experimental investigations. The influencing factors at work here are so diverse in nature that it is frequently only possible to suggest the use of approximation methods, underpinned by concrete measurements, whose applicability, say, to other gear dimensions is rarely ensured. Other influences on the thermal balance are to be expected due to external gear contamination and radiation from direct sunlight. While surface heat transfer deteriorates as a result of the accumulation of dirt on the gear, an additional absorption of thermal energy – and thus an increase in temperature – occurs with gears temporarily exposed to direct sunlight. In this regard refer to the literature [6.6/19]. In the quasi-stationary state of equilibrium , the power loss P V created is equal to the heat flow Q dissipated: Q P = V (6.6/41) In general terms, this is broken down as follows: QQ Q Q Q Q=+ + + +KSFWU (6.6/42) where QK is the heat flow due to convection QS the heat flow due to radiation QF the heat flow due to conduction via the gear baseplate QW the heat flow due to conduction via rotating shaft ends and clutches QU the heat flow due to cooling circuits (pressure-feed lubrication or cooler in the oil sump) For slow-running gears, it is sufficient to take the heat dissipation arising from convection, radiation and conduction into account. Lubrication ensues by immersing the gears in an oil bath. For fast-running gears, pressure-feed lubrication is necessary, leading simultaneously to the es-tablishment of a cooling circuit. The heat dissipated in this manner tends to predominate. If no cooling circuit exists, then th e heat flow caused by convection and radiation forms the main component. 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 377 6.6.3.2 Heat Dissipation via the Housing In this regard, the heat dissipation to the medium of air ensues by means of convection and radiation. The following equation applies (in relation to the gear housing): ()GKS K o i lRQQQk A=+ = − KK (6.6/43) where QG is the heat flow of the gear to the environment in W AK the cooling surface of the gear housing (free surface without floor space) in m2 k the heat transmission coefficient in W/(m2K) Koil the oil temperature, equated to the housing temperature in C (as a first approximation) KR the ambient temperature in C For a rough calculation, the heat conduction via the gear shafts and the baseplate is neglected. The heat transmission through a plane wall consists of the internal surface heat transfer, the heat conduction and the external surface heat transfer as per Figure 6.6/8. For plane walls, the following equation applies for the heat transfer coefficient k: ks= ++1 11 αλ αia (6.6/44) The internal surface heat transfer coefficient Įi from oil to the wall is of the order of 150 to 300 W/(m2K) and can be disregarded in comparison with the external surface heat transfer coefficient Į a 20 W/(m2K) in conditions of natural airflow. In conditions of forced airflow, Įa will reach values up to 50 W/(m2K), so Įi should be taken into account. The heat conduction through the gear wall, given by s/Ȝ 2 10 -4 m2K/W for steel housings, possesses a negligi- ble influence on k . For the heat transmission coefficient k in Equation (6.6/43), Įa is used as a first approximation, whereby the oil temperature Koil can be retained instead of using the mean housing wall KG. It therefore follows from Equation (6.6/43) that Figure 6.6/8 Heat transmission through a plain wall (gear housing) Ga K o i l , QA=α K (6.6/45) where oil,exK is the oil excess temperature in K oil,ex oil R=−KK K The external surface heat transfer coefficient is given by αα αaKS=+ (6.6/46) where ĮK is the surface heat transfer coefficient due to convection ĮS the surface heat transfer coefficient due to radiation An approximation equation for Įa has been derived empirically from experiments [6.6/8], which takes both components from Equation (6.6/46) into account and, in addition, includes the forced convection from external air movement. 378 6 Load Capacity and Running Performance: External and Internal Gearings For a temperature range GRo i l , e x 40 K 120 K ≤−≈ ≤KKK the following equation applies: ()0.15 aK o i l , e x 10 0.07fH−α= + K (6.6/47) where H is the height of the gear in m. The cooling factor fK is included in Table 6.6/7 (as a function of the air speed /L). The oil excess temperature in the gear can then be calculated from the following relationship: ()Va a a K o i l , e x 1 PP V P Q A==− η = = α K (6.6/48) Table 6.6/7 Cooling Factor fK as a Function of the Air Speed /L from [2] /L in m/s 1.25 (stationary air) 2 3 5 7 10 15 20 fK 1 1.3 1.6 2.3 2.9 3.6 5.0 6.3 With Equation (6.6/47) and resolved for oil,exK then the oil excess temperature Koil,ex for splash lubrication is derived as follows: 0.15 V oil,ex KK5100 71.40.07PH fA≈+ −K (6.6/49) where PV is the power loss here in W H the height of the gear in m AK the cooling surface (housing surface in m2) fK the cooling factor (Table 6.6/7) A value of oilK roughly equal to 120 C is considered the maximum admissible bulk temperature for mineral oils. In gear unit manufacturing, the oil temperature is given as oilK 95 C. For synthetic oils, higher limits apply, depending on the special type of lubrication used. For rough evaluations, a heat dissipation of the magnitude of PV/AK 1 kW/m2 (where AK is the housing surface) can be used in the calculation. 6.6.3.3 Heat Dissipation due to the Cooling Circuit Starting from the amount of heat caused by convection and radiation in conjunction with the maximum admissible oil temperature QGin accordance with Equation (6.6/43), it is possible to determine the amount of heat dissipated via the cooling circuit QU: QP QUV G=− (6.6/50) With pressure-feed lubrication, the amount of heat dissipated via convection and radiation is often disregarded ( QGis set to zero). The amount of heat dissipated via the cooling circuit QU is given as a function of the amount of oil Qe used in pressure-feed lubrication, the temperature difference ǻK between outlet and inlet, and the specific heat c of the cooling medium (oil), as follows: 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 379 Qc QUe=NK Δ (6.6/50a) This results in the quantity of injected oil Qe required, as follows: QQ ceU= NKΔ (6.6/50b) where QU is the amount of heat dissipated by the pressure-feed lubrication [from Equation (6.6/50)] c the specific heat of the oil in Ws/(kgK) for mineral oil and synthetic oil: c 1.9 103 Ws/(kgK) N the density of the oil in kg/dm2 for mineral oil and synthetic oil: N 875 kg/m3 ǻK the difference between the oil injection and the oil outlet temperature in K The release of the heat contained in the oil QU generally takes place in an oil cooler to a cooling liquid or the ambient air. 6.6.3.4 Supplements to the Thermal Balance A) Heat Transmission, Surface Heat Transfer Investigations carried out by Funck [6.6/20] take the position of the walls (vertical or horizontal) into account in determining the surface heat transfer coefficients and attempt to gain a better understanding of the influence of the forced convection. The theoretical derivations have been checked by experiment. The results are depicted in [6.6/21] and explained in [6.6/22] in terms of a specific gear. In the following text, several equations are specified, which can be drawn upon for more precise analyses. For ribbed gear housings, the following relation applies for the heat transmission coefficient from [6.6/22] for Equation (6.6/44): kA AsA AR ia ia ia= ⋅+ ⋅+1 11 αλ α (6.6/51) where Aa are the external wall surfaces, for gears equal to the cooling surface A K Ai the internal wall surface, for gears equal to the housing surface on the inside of the housing Aoil In Equation (6.6/51), it should be noted that for ribbed walls the surface heat transfer coefficients Įa on the air side are generally 20 to 30% less than is the case for non-ribbed housings. For the surface heat transfer coefficient in association with free convection (natural airflow), the following relationships apply – depending on the position of the walls – with KG Koil, whereby all the Į values in the text below should be understood as mean values. Vertical wall: 0.3 oil,ex 0.1 K,vert R11.06273H− α= + K K (6.6/52) 380 6 Load Capacity and Running Performance: External and Internal Gearings Horizontal wall: 0.32 0.04 oil,ex K,horiz R12.87273B− α= + K K (6.6/53) In order to approximate the calculated Į values for free convection to the measured Į values, a correction factor has been introduced and a close d-form equation established, which takes both the vertical and the horizontal walls into account, as follows: 0.3 oil,ex 0.1 vert K,free RK 0.32 oil,ex 0.04 horiz RK1.08 11.06273 12.87273AHA ABA− − α= ⋅ + + ⋅ + K K K K (6.6/54) Further simplification has taken place in [6.6/22]: 0.3 oil,ex 0.1 K,free R12.1273H− α≈ + K K (6.6/55) The surface heat transfer coefficient due to radiation is obtained from αε SSGR GR=+ −+ −CKK KK273 100273 10044 (6.6/56) where İ is the emission ratio Cs the black body radiation coefficient; 5.77 W/(m2K4) Here too an approximation has been made in [6.6/22], into which KG Koil is inserted: 3 6 oil R S5460.23 102− ++ α≈ ⋅ ε KK (6.6/57) where İ is the emission ratio, as follows: İ 0.93 for varnished surfaces İ 0.6 to 0.7 for cast iron with cast skin İ 0.80 for skin-rolled steel İ 0.15 for oxide-skinned aluminium housings (no varnish); refer to Table A14.3/1 in Appendix 14.3 [6.6/53] Surface heat transfer coefficients using different approaches are depicted in Figure 6.6/9. 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 381 Figure 6.6/9 Comparative figures for surface heat transfer coefficients: 1 from Equation (6.6/47) [6.6/8] where fK = 1 2 from Equations (6.6/54) and (6.6/56) [6.6/20] 3 from Equations (6.6/55) and (6.6/57) [6.6/20] For forced convection , the surface heat transfer from the gear into the environment is broken down into convection in the areas with tangential flow, stagnation flow and free convection in accordance with Figure 6.6/10. In doing so, a distinction is made in the two cases with primarily tangential flow (a) and primarily stagnation flow (b). For forced convection, the surface heat transfer coefficient Į K,forced is given by tang stag free K,forced K,free K,tang K,stag KKKĮ =Į +Į +ĮAA A AAA (6.6/58) where ĮK,free is calculated from (6.6/54) or (6.6/55). Reference [6.6/20] applies to the surface heat transfer coefficient of the tangential component, as follows: 0.37 0.63 K,tang L 7.6L υ∗−α= ⋅ (6.6/59) where L\* is the upstream flow length as per Figure 6.6/10 in m /L the mean air speed in m/s Reference [6.6/20] applies to the surface heat transfer coefficient of the stagnation component, as follows: 0.34 0.66 stag K,stag L stag5.6U Aυ α= ⋅ (6.6/60) where Ustag is the extent of the vertical gear wall associated with inflow with the surface Astag. The surface components Afree, Atang and Astag can be determined in accordance with Figure 6.6/10, while their sum gives the total cooling surface AK. Notwithstanding this, however, when applying the relationship for ĮK,forced as per Equation (6.6/58), it should be noted that it is only valid above a threshold value of /L,lim. Figure 6.6/11 depicts this relationship in comparison to ĮK,free in conditions of natural airflow ( /L = 0). 382 6 Load Capacity and Running Performance: External and Internal Gearings In the range 0 /L /L,lim, the progression of the curve for ĮK,erzw is replaced by a straight line, which possesses the value ĮK,free at /L = 0 and is tangential to the curve at /L = /Llim as per Equation (6.6/58). It can be seen from the diagram that /L,lim increases proportionally with the size of the gear unit and the operating temperature of the oil. With stagnation flow, /L,lim is lower than with tangential flow. The cooling factor introduced in Equation (6.6/47), fK, can now be calculated from the equation a,forced K a,freefα=α (6.6/61) Figure 6.6/10 Different surface heat transfer conditions with forced airflow The cooling factor not only depends on the air speed /L but also on the gear unit size, the operating oil temperature and the type of flow. Fundamental experimental investigations on the heat dissipation via steel baseplates and the transferring of the results into practice were also undertaken by Funck [6.6/20] and [6.6/21]. This resulted in the determining of the following key relationships: • strong influence of the size of the base- plate • low influence of the gear unit feet surface • no appreciable contribution of the heat flow via the free underside to the heat flow dissipation It may be assumed that the heat flow fed to the gear unit base in the quasi-stationary state of equilibrium is almost completely fed to the baseplate via the surface of the gear unit feet A feet and subsequently released into the environment: oil,Base Foot FQQ Q =− (6.6/62) Depending on the surface heat transfer conditions at the baseplate, the size of the baseplate, the shape of the gear unit feet and the airflow on the gear unit, a generally representative proportion (ranging from 15 to 25%) of the total heat dissipation from the gear housing was determined. For an approximate calculation of the heat flow emitted via the gear baseplate , the following calculation rule for use with foundation plates has been specified by Funck : 1 Foot Foot W FF o o t W o i l , e x Foot Wall n,Base oil1 HA fQAfA− ∗ =+ + ⋅ αλ α K (6.6/63) 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 383 Figure 6.6/11 Validity of Equation (6.6/58) for ĮK,forced The surface heat transfer coefficient on the oil-facing side of the gear unit base is approximately obtained as Įoil = 100 W/(m2K). To calculate the equivalent surface heat transfer coefficient Į\*foot on the gear unit foot, there first ensues a division of the foundation plate into individual ribs as per Figure 6.6/12 and the calculation as per () ()Fi ii wall Fi foot i qi F i foot ii Fitanh 1t a n himLmmAAmLm∗ ∗ ∗∗ ∗ ∗α + λ λ α=α + λ (6.6/64) where i is the number of the rib ĮF the mean surface heat transfer coefficient at the baseplate in W/(m2K) For foundation plates made of steel, the following applies: ĮF 7.5 W/(m2K) for foundation plates significantly bigger than the gear ĮF 10 W/(m2K) for smaller foundation plates Furthermore, for the reciprocal lengt h coefficient m\* for heat dissipation at the baseplate upwards and downwards: upwards only: q Fq F Į AU = m\* λ (6.6/65) Fq FqĮ 0.7 \*U = mAλ (6.6/66) where Uq is the circumference of a ribbed part at the imagined cut surface in m Aq the contents of the ribbed part in m2 384 6 Load Capacity and Running Performance: External and Internal Gearings The factor fW in Equation (6.6/63) is derived from model investigations: foot 0.002 W1.02ef∗−α= (6.6/67) Approximations are f W = 0.95 for small and fW = 0.81 for large baseplates in relation to the gear unit base area. The individual sizes are depicted in Figure 6.6/13 for a better appreciation. Figure 6.6/12 Division of the steel baseplate into four single ribbed parts to calculate the foundation conductivity [6.6/22] Figure 6.6/13 Thermodynamic parameters and geometric data for calculating the foundation conductivity [6.6/22] Transferring the calculations to differently designed baseplates has not yet been attempted. The foundation conductivity can be neglected when using a concrete baseplate. To estimate the heat dissipation via shafts and clutches , Funck provides relationships based on references to the literature [6.6/23]. The shaft-clutch system is divided into two replacement systems, as shown in Figure 6.6/14. 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 385 Figure 6.6/14 Equivalent shaft-flange system for calculating the heat dissipation The following applies: sh sh1 sh2QQ Q=+ (6.6/68) For an existing shaft end, we obtain () ()sh,St sh sh sh sh s h 1 s h s hq , s hs h , e x0 sh,St sh sh sh shtanh 1t a n hxmlmQm A mlm∗ ∗ ∗ ∗ ∗ = ∗ ∗α+λ=λ ϑ ⋅α+λ (6.6/69) and for a flange half () shsh2 fl fl q,fl fl,ex fl fl tanh xlQm A m l∗∗ ==λ ϑ ⋅ (6.6/70) The areas Aq are calculated from 2 sh q,sh4dA=π (6.6/71) 2 fl q,fl4dA=π (6.6/72) The equivalent surface heat transfer coefficient bet ween the shaft and the coupling is given by ()fl sh,St fl q,fl fl fl q,shtanh mA mlA∗∗ ∗ λα= (6.6/73) 386 6 Load Capacity and Running Performance: External and Internal Gearings where sh sh sh sh2 md∗ α=⋅λ (6.6/74) fl fl fl fl2 md∗ α=⋅λ (6.6/75) sh,ex oil,ex0x=≈KK with sh,ex sh R=−KK K (6.6/76) sh shfl,ex sh,exxlx l ==≈KK with fl,ex fl R=−KK K (6.6/77) () ()shoil,ex sh,ex sh,St sh sh sh sh sh shcosh sinhxl ml mlm∗ ==α+λKK (6.6/78) Generally speaking, only a small part of the heat flow is dissipated via the shaft ends and the couplings (approximately 10% and less), as has been experimentally determined (refer to [6.6/19]). It is therefore often possible to neglect this effect. The operating oil temperature or oil excess temperature can no longer be determined for these complicated interrelationships in an explicit form, as is the case with the approximate solution as per Equation (6.6/49). For this purpose, an iterative computing process is necessary, which is not a problem when using a computer. At this point, attention is drawn to ISO/TR 14179-2 [6.6/53], which deals with the performance of gear drives. The corresponding designs are included in Appendix 14.3. B) Thermal Balance in Non-steady-state Operation Mode The thermal balance hitherto conducted only applies to the quasi-stationary state of equilibrium (also known as the thermal steady-state condition), which is reached after a sufficiently long period in continuous operation under constant load (refer to Figure 6.6/19, Case 1). In practice, however, constant loads scarcely ever occur over longer periods. It is therefore necessary to expect constantly changing gear temperatures [6.6/24]. For the thermal balance associated with these non-steady-state operating conditions , one must take into account not only the heat dissipation but also the changing heat capacity C (due to the temperature gradien t) of the gear unit. The following equation then applies: ()oil,ex Vd dPQ Ct=+ K (6.6/79) Here the power loss PV and the heat flow Qլ (convection, radiation) are a function of the oil excess temperature Koil,ex present in each case. These quantities change continually as long as the tem- perature gradient satisfies ()oil,exd 0dt≠K This state of affairs is illustrated in Figure 6.6/15. By applying the relationships for the surface heat transfer and integrating Equation (6.6/79), we obtain an exponential function for the oil temperature as a function of time. The same temperature 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 387 characteristics are also known for, say, electric motors [6.6/25]. Excellent agreement has been achieved between experimental and computational results by introducing a thermal exponent x. The following relationship applies for a constant load: ()Ĳ oil,ex oil,ex A oil,ex B oil,ex A 1ext − =+ − − KK K K (6.6/80) where Koil,exA is the initial value of the oil excess temperature in K Koil,exB the oil excess temperature in a thermal steady-state condition in K - the heating-time constant in min x the thermal exponent The progression of Equation (6.6/80) is depicted in Figure 6.6/16 for heating and cooling. The application of this relationship requires know- ledge of the variables Koil,exB, - and x for the gear unit. These variables are most appropriately determined by experiment. To do this, it is neces-sary to include the progresssion of the oil excess temperatures in a steady-state condition Koil,exB as a function of the load according to Figure 6.6/17. Figure 6.6/15 Thermal balance as per Equation (6.6/79) in non-steady- state operating conditions for a single-stage cylindrical gear unit Figure 6.6/16 Oil excess temperature over time as per Equation (6.6/80) where x = 1 388 6 Load Capacity and Running Performance: External and Internal Gearings From Figure 6.6/17, it is possible to infer the following: • For a specific power of a modified thermal power rating PG, determined by the maximum allowable oil temperature, the highest oil excess temperature Koil,ex Bmax is just reached in the steady-state condition. For gear units with PG < P N, the modified thermal power rating represents the power limit, while for gears with PG > P N no thermal problems occur. • For a power P = 0 (idling), a finite temperature increase is already determined by idling losses present in the gear unit. An idling excess temperature Koil,exL occurs in the steady-state condition. Figure 6.6/17 Oil excess temperature in the steady-state condition via the gear unit power [6.6/24] The modified thermal power rating is defined as PQ VG= (6.6/81) where Q is the heat flow dissipation associated with the maximum allowable oil temperature (also known as the heat dissipation capacity) in W V the degree of transmission loss (refer to Section 6.6.1.1) The modified thermal power rating of the gear unit is specified in the manufacturers’ product documentation. It represents a crucial selection parameter. The heat dissipation in this information is generally taken into account by the gear housing and by secondary measures such as air filters and further additional cooling. For a known modified thermal power rating P G for a maximum oil excess temperature and a known or estimated idling excess temperature Koil,exI, the oil excess temperature associated with a given power P can be estimated using the following relationship: ()22 2 2 oil,ex B oil,ex A oil,ex L oil,ex L y =− +KK K K (6.6/82) 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 389 where y is the relative load: yP P= G The heating-time constant - can be read from the heating or cooling curves as per Figure 6.6/16. From Equation (6.6/80) with t = - and oil,ex A1K = 0, there then follows for Curve 1: ()1 oil,ex1 oil,ex B1 oil,ex B1 1 e 0.632−=− =KK K (6.6/83) Curve 2: ()11 oil,ex 2 oil,ex B2 oil,ex A2 oil,ex B2 oil,ex A2 1 e e 0.632 0.368−−=− + = +KK K K K (6.6/84) Under identical conditions (constant speed and ambient conditions), the same value for - is determined from both curves. The heating-time constant only possesses a constant value for a permanently assigned operating condition. It varies depending both on the movement of the oil in the gear unit and the movement of the air around the gear unit. In both cases, the surface heat transfer is affected. For gear unit cooling when at a standstill, a value of - is obtained that is about twice as great as when the gear is in use. Fan-cooled gear units possess a lower - value than do gear units with no filter. The thermal exponent x causes the theoretical progression of Equation (6.6/80) to approach the experimental progression for x = 1. This influence is illustrated in Figure 6.6/18. A calculation from experimental measurements ensues after transposing Equation (6.6/80): oil,ex oil,ex B oil,ex A oil,ex Bln ln lnĲxt −− − =KK KK (6.6/85) Previous experiments have resulted in the approximate value range 0.7 x 1 with a concentration at x = 0.8. One approach for the calculation of - can be followed from Equation (6.6/79) by using Equation (6.6/80). If Equation (6.6/80) is differentiated by time and t is set to -, we obtain () ()oil,ex A oil,ex B nn Ĳ0.368 d d Ĳtx t t=− =−KK (6.6/86) Inserted into Equation (6.6/79) and resolved for -, we then obtain () ( ) ()oil,ex A oil,ex B oil,ex V oil,exĲĲ0.368 Ĳ ttxC QP ==− = − KK KK (6.6/86a) For the case of cooling when the gear unit is at a standstill with PV = 0, Koil,ex B = 0, Koil,ex = 0.368 Koil,ex A and assuming heat dissipation only from convection and radiation, then, by means of Equations (6.6/45) and (6.6/47) with the associated thermal exponent xS, we obtain for the cooling-time constant -S in min. 390 6 Load Capacity and Running Performance: External and Internal Gearings Figure 6.6/18 Influence of the thermal exponent x on the progression of the curve Koil,ex = f(t ) ()0.15 S S Ko il,ex A KĲ 60 10 0.026xC H fA= +K (6.6/87) The heat capacity of the gear unit is determined by the products mc for the various materials present, for example: St St GG GG oil oil Cm c mc m c=+ + (6.6/88) where m is the mass in kg with the indices St Steel GG Cast iron oil Oil For the specific heat c the following values can be used: c St = 460 Ws/(kgK) cGG = 545 Ws/(kgK) coil = 1675 Ws/(kgK) The values calculated in this way accord with those determined experimentally to within an order of magnitude. Several experimental values, determined in industrial gear units, are listed in Table 6.6/8 as indicative quantities [6.6/26]. Table 6.6/8 Experimentally Determined Heating-Time Constants - for Cylindrical Gear Units [6.6/26] Centre distance of output stage a Gear ratio i Number of stages Heating-time constant - Cooling-time constant when gear unit at a standstill -S mm - - without additional cooling min with fan min min 160 2 1 45 - 125 160 16 2 50 - - 200 6.3 54 33 130 250 8 130 90 260 200 100 4 92 - - 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 391 To determine the resulting gear temperatures, statements on the operating mode of the gears are required. A distinction is made between the following operating modes (including long-term operation): • Long-term operation Operation with constant load and a duration sufficient to reach the steady-state condition. • Short-term operation Operation with constant or changing load, the duration of which is not sufficient to reach the thermal steady-state condition, followed by an operating standstill (intermission) of such duration that the gear unit can to all intents and purposes cool down to the ambient temperature. • Intermittent operation Operations with a series of cycles of the same type, consisting of constant or changing loads and intermissions, the duration of which is not sufficient to reach the thermal steady-state condition within one cycle. • Uninterrupted operatio n with intermittent load Operations with a series of cycles of the same type, consisting of constant or changing loads and idling periods without intermission, the duration of which is not sufficient to reach the thermal steady-state condition within one cycle. • Uninterrupted operation with non-periodic load changes Operation with an irregular series of changing loads. These operating conditions are graphically depicted in Figure 6.6/19, whereby a constant load was assumed during the period of operation t B for each of the cases 2 to 4. Changing loads within one load cycle can generally be described by means of a load spectrum. This load spectrum is composed of different periodically recurring load components, each of which affects a specific time slice of the total duration of the cycle. The example of such a load spectrum is depicted in Figure 6.6/20 for long-term operation in relation to the duration of the cycle t S, whereby an ascending sequence has proven its worth from a thermal perspective. Figure 6.6/19 Operating conditions of a gear drive 392 6 Load Capacity and Running Performance: External and Internal Gearings Figure 6.6/20 Long-term operation with periodically changing load (load spectrum) and associated oil excess temperatures for two cycles in equilibrium Determining a representative load spectrum can generally only ensue from measurements on a gear unit over a sufficiently long period. The results of the measurements are classified. The highest value of the oil excess temperature occurring in the gear unit in non-steady-state operating conditions following a sufficiently long period is defined as the oil excess temperature in equilibrium: oil,ex G oil,ex max=KK (6.6/89) In practical terms, Koil,ex is calculated using Equation (6.6/80) for each load component with its associated time slice. This value forms the initial value Koil,ex A for the following load component. This process ensues until the maximum is reached. The thermal load factor oil,ex G G oil,ex Bmax1 f=<K K (6.6/89a) characterises the thermal load capacity of a gear unit compared to long-term operation at maximum load. For both load spectra and practical calculations, a thermal (correction) exponent of x = 1 can be used for the calculation (refer to Figure 6.6/21) with mostly satisfactory precision, whereby the calculation is particularly simplified at conditions of non-constant load. 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 393 Figure 6.6/21 Temperature characteristics for a load spectrum with x = 0.8 compared with x = 1 [for x thermal (correction) exponent]; refer to Equation (6.6/85) and Figure 6.6/18 6.6.4 Lubrication and Cooling 6.6.4.1 Selection of Lubrication In Section 6.5.4.1, the tasks of the lubricant in the meshing were described and the problems faced in fulfilling them listed. It is necessary to select the lubricant suited to best realising these demands from the range of products available on the market in the most practicable manner. Based on its coefficients, chemical structure and properties under the influences of load, relative rotation and temperature, the lubricant selected must be able to form a load-carrying lubricating film. The mathematical relationships to calculate th e lubricating film thickness required in cylindrical gear pair meshing are specified in Section 6.5.4.1. When using mineral oils without additives, a large number of methods only refer to the pitch point. For this reason, they can only provide a rough indication. When selecting a lubrication for a cylindrical gear unit, the following aspects, which lead to corresponding compromises, should be taken into account: • High ratios of transmission power to gear transmission volumes require lubricants with high additive treatment that have a sufficient scuffing load capacity and reliable protection against micro-pitting. • Multi-stage gear units possess different geometry meshing and operating conditions. • The tooth flanks are most vulnerable on the tooth tip and the tooth root due to the highest sliding speed occurring at those points. The lubricating film thicknesses (or the oil additives) must be selected accordingly to be sufficient. 394 6 Load Capacity and Running Performance: External and Internal Gearings • Not only the meshing but other lubrication points, such as the roller or plain bearings, must also be provided for. • Other functional elements, such as the hydrodynamic converter elements, can have a critical influence on the choice of oil. • The heat created by friction must be discharged to the outside to avoid the permissible gear unit (or oil) temperature being exceeded. • For reasons of economy, the performance limits of the lubricant should be exploited as far as possible. On the other hand, it is in the interest of the user to keep the range of lubricants as low as possible (by the use of so-called multi-grade oils). As a matter of principle, not only the choice of lubrication, price and product range, but also the following influencing variables should be taken into account [6.6/27]: Design features Operational features - type of toothing - dimensions of gear unit - design of housing - surface roughness - material of gear - lubricant feed - tooth flank load - circumferential speed - service temperature - ambient temperature The first decision required is the lubricant type to be used. Table 6.6/9 contains the possible lubricant types, their application limits and the primary lubrication type after Niemann/Winter [2]. Appendix 14.4 contains new insights on alternative lubricants in powder form. Up to now, lubricating oils have mainly been used in industrial gear units. The main advantages over lubricating grease consist of their easy application to the friction point, good heat dissipation properties, and the large area of application in relation to circumferential speed and load. For this reason, the further designs refer to lubricating oils. Table 6.6/9 Lubricant Types and Global Operating Conditions [2] Circumferential speed m/s Lubricant type Method of lubrication Gear design up to 2.5 Adhesive lubricant Application lubrication Open possible up to 4 (possibly 6) Viscous grease Spray lubrication up to 8 (possibly 10) Splash lubrication Sealed up to 25 (possibly 30) 1) Lubricating oil Splash lubrication or injection lubrication over 25 (possibly 30) Injection lubrication 1) Experiments were undertaken up to 60 m/s [6.6/40]. The kinematic viscosity required is to be determined and the lubricant type selected depending on the circumferential speed and the mechanical or thermal load, whereby it may be necessary to recal-culate the scuffing load capacity. The viscosity required increases with higher load and less speed. The required lubricating oil viscosity can be determined based on ISO TR 18792 (lubrication of industrial gear drives”). From operational experience, the required lubricating oil viscosity Ȟ 40req for mineral oils can be determined from the following equation (refer to [2] and [6.6/29]): 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 395 ()oil,ex 2.22 lg 20 0.52lg 1.42 40 10req f ν +− − ν=⋅K / (6.6/90) where / is the circumferential speed in m/s For multi-stage gear units, / should be replaced by the arithmetic mean value /m of the circumferential speeds of the individual gear stages. oil,exK the oil excess temperature in K; oil,ex GK = Koilb - KR fȞ the viscosity factor, a function of the ambient temperature KR from Table 6.6/10 Table 6.6/10 Viscosity Factor fȞ KR C -30 -20 -10 0 10 20 30 40 50 60 70 80 fȞ - 0.115 0.17 0.26 0.40 0.63 1.0 1.6 2.5 4.0 6.5 10 16 Equation (6.6/90) is depicted in Figure 6.6/22 in diagram form. For gear units with rolling element bearings, the operating viscosity Ȟb required for the bearings or the viscosity class from Figure 6.6/23 is to be deter mined according to the information provided by the manufacturer [6.6/30]. The two values obtained for the toothing and the rolling element bearings (or plain bearings) are to be compared with each other and the higher value used for the selection of the lubricating oil. A rough orientation to the lubricating oil viscosity required (albeit not fulfilling all of the requirements) is provided by DIN 51509 T 1 [6.6/31]. For suitable lubricants, refer to the relevant standards: Lubricating oils C, CL and CLP [6.6/32], HL and HLP [6.6/33] and [6.6/34]. Special experimental investigations on car gearboxes performed by Wienecke [6.6/45] have resulted in further insights, which should be taken into account when choosing a suitable lubricant to improve the efficiency. Figure 6.6/22 Approximate calculation of the required lubricating oil viscosity and viscosity class for cylindrical gear units for qualitative orientation according to TGL 38404 [6.6/29] Note: ISO TR 18792 (lubrication of industrial drives) is recommended for actual designs 396 6 Load Capacity and Running Performance: External and Internal Gearings Figure 6.6/23 Calculation of the operating viscosity Ȟb (minimum value) and the viscosity class for rolling element bearings [6.6/30] Alongside the mineral oils common in the trade, synthetic lubricants (e.g., poly-alpha-olefins and polyglycols) are coming to the fore, particularly in special individual cases. The selection of a synthetic lubricant is only warranted if the properties required when using specific additives in the mineral oil are not (or are not economically) viable. The following characteristics may be specified as advantages provided by synthetic lubricants : • thermal stability • oxidative stability • viscosity-temperature behaviour • flow behaviour at low temperatures • volatility at high temperatures • high temperature-range applicability • favourable friction behaviour • radiation-resistant • low inflammability Attention should also be paid to specific disadvantages, which offset the above benefits. These include • hydrolytic behaviour • corrosion behaviour • toxic behaviour • incompatibility with other materials • poor solubility for additives • availability (general or in specific viscosity indexes) • costs These positive and negative properties as per Equation [6.6/35] do not, of course, occur with every synthetic liquid. One exception to this, however, is provided by the costs, which are principally higher than for mineral lubricants. 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 397 Lubricant manufacturers often provide their own guidelines for the purpose of their products being selected. Mobil Oil AG uses EHD directly [6.6/36]. The theoretical background is presented in Section 6.5.4.1. From [6.6/36], the specific lubricating film thickness Ȝ\* to be expected can be determined from Figure 6.6/24 and compared with the minimum value depending on the circumferential speed /w. If the specific lubricating film thickness determined by using the EHD theory lies below a required size (Mobil specifies the condition Ȝ\* 1), then the use of lubricants with EP and/or effective anti- wear additives is recommended. Most gear oils contain these additives. Another approach is to determine the oil viscosity as a function of the speed and several secondary conditions on an empirical basis [2]. The scuffing safety is then recalculated and the conclusion drawn on the (EP) additives. In a few cases, the type of oil and the viscosity are determined by other functional elements (e.g., for fluid transmissions), and then a selection based on the EHD theory is not possible. (For an alternative lubrication refer to Appendix 14.4.) Ȝı h∗= , ()1 22 2 12ııı=+ , 1,2 a1,2ı 1.3R ≈ Figure 6.6/24 Specific minimum lubricating film thickness at 5% wear probability [6.6/36] 6.6.4.2 Lubricating and Cooling Systems Oil lubrication is almost exclusively used in sealed gear units because this is the best way to realise both the lubricant supply to the meshing and the dissipation of the resulting frictional heat. Grease lubrication is only envisaged in special cases, sa y, when the gear unit has to be absolutely sealed. For further information refer to [2]. The choice is between two lubrication methods: splash lubrication and injection lubrication (also known as pressure-feed lubrication). A combination of the two is also possible. The characteristic indicators, their advantages and disadvantages, as well as their special features, are given below. Splash lubrication is a simple, reliable and economical method of lubrication. Gears immersed in the oil reservoir deliver the oil into the mesh (Figure 6.6/25) [6.6/37]. The bearings of the gear shafts are supplied with splashed oil, thus ensuring a continual lubrication and cooling. A disadvantage here is the limited quantity of oil, which makes high demands on the lubricant due to a lack of filtering and additional cooling. The cooling can be improved by using fans [6.6/38] or a cooling coil [6.6/39]. The circumferential speed of the gears should be taken as the limit for using splash lubrication. Splash lubrication is applicable up to a speed of 15 (20) m/s with no special constructive measures. It is still possible to use splash lubrication over this speed up to 30 and even 60 m/s [6.6/40] if the oil guide is secured by baffles, oil pockets, and the like. The maximum permissible centrifugal acceleration when using splash lubrication is specified as approximately 550 m/s 2 [6.6/34]. 398 6 Load Capacity and Running Performance: External and Internal Gearings Figure 6.6/25 Splash lubrication of a sealed cylindrical gear unit [6.6/37] The key parameter for a secure lubrication procedure using splash lubrication is the immersion depth of the gear. A too-low oil level will result in inadequate lubrication with consequences including heat dissipation and increased wear, while a too-high oil level causes a high splashing loss and thus greater heating of the oil, accompanied by a reduction in the oil viscosity, resulting in poorer adhesion and pressure absorption characteristics. Moreover, foaming and a higher noise level must also be expected. Oil-level measurements should ensue during operation, as the level is lower than when the gear unit is at a standstill. Figure 6.6/26 depicts an example for measuring the oil level when the gears are running. The following immersion depths are recommended depending on the circumferential speed: / < 5 m/s / = 5 to 20 m/s / > 20 m/s e = (3 to 5) m e = (1 to 3) m e 2.5 m, according to the mounting position of the gears where m is the toothing module. The losses from splashing should not exceed 0.75% of the total losses. An oil-filling quantity ranging from 3 to 10 l per kW power loss is recommended in association with pure heat dissipation via the gear unit housing. Pressure-feed lubrication is a more complex procedure. The higher investment and maintenance costs are compensated for by the benefits provided. These include dosable lubrication with less lubricant loss, better cooling and oil maintenance and the higher service lives of the gears and oil guide resulting. Pressure-feed lubrication has been successfully used in operations from / = 10 to 250 m/s. A distinction is made between a circul ation lubrication system with an external oil reservoir provided to supply the gears and a circulation lubrication system with an oil sump, where the lubricant is 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 399 directly fed into circulation from the oil sump of the gear unit using a pump. The oil is fed directly into the meshing using injection nozzles or injection sprinklers. Jet forms and outflow elements are depicted in Figure 6.6/27. The determination of the quantity of injected oil re-quired is particularly important. As a rule of thumb, the following quantity applies: Q e = 0.5 to 1.0 l/min per cm of face width. A too-small quantity of oil leads to inadequate lubri-cation with the same consequences as in the case of a too-low oil level with splash lubrication. A too-large quantity of oil, on the other hand, results in excessive oil squeeze losses. Particular care should be taken with the direction of injection. At low circumferential speeds, the priority task lies in realising the formation of the lubricating film. The cooling only plays a seconddary role in this regard. At high circumferential speeds, the major part of the oil is required for cooling. Although a lubricating film forms if the lubricating oil is injected before meshing, higher oil squeeze losses are to be expected in association with heating. Figure 6.6/28 illustrates the process during the meshing. The reduction in the oil volume between the tooth flanks is clearly visible. If the oil is injected after the meshing, then a good cooling of the tooth flanks is obtained. For this reason, it is necessary to make compromises. Figure 6.6/26 Measuring the oil level when the gear unit is operational [6.6/2] Figure 6.6/27 Jet forms and outflow elements for injection lubrication [6.6/2] For safety reasons (pressure fluctuations), for example, injection from above is preferred to injection from below with a horizontal layout of the gears (Figure 6.6/29a). Recent investigations intended to assess the effects of lubrication and cooling with injection lubrication are depicted in [6.6/41]. Even the smallest quantities of oil suffice to ensure lubrication at a safe level, as they improve the efficiency due to lower non-load-dependent hydraulic losses. Notwithstanding this, however, far greater quantities of oil are required for the cooling of cylindrical gears. This oil can also be fed to 400 6 Load Capacity and Running Performance: External and Internal Gearings the gears outside the toothing area. The results of the experiments conducted have shown the combination of these methods of oil feed ing to be superior to the methods hitherto commonly used in practice. This enables a reduction of the mean temperatures of the teeth in high-performance gears. Figure 6.6/28 Oil injection before meshing The positions of the jets and directions of injection in relation to the meshing are depicted in Figure 6.6/29. From [2], the oil pressure up-stream of the gears should lie in the range from (0.5)1.0 to 3.5(10) bar. A jet diameter of (1)1.5 to 4(5) mm is recommended. Figure 6.6/30 depicts the relationship between the jet diameter and rate of flow for various oil pressures [2]. The distance of the jet from the surface of the tooth-ing should not exceed 150 to 200 mm. In order to avoid non-uniform heating in helical gear units via the face width due to axial dis-placement of the lubricating oil and the resulting increase in oil squeeze losses, non-uniform oil feeding using suitable sprinklers is recommended (as shown in Figure 6.6/31). The separate oil reservoir should be designed with a capacity corresponding to four or five times the delivery volume per minute. Rule-of-thumb values for differences in tempera-ture between the lubricating oil and the cooling water are listed in Table 6.6/11. Figure 6.6/32 depicts the diagram of a complete pressure-feed lubrication system as a circulation lubrication system with an external oil reservoir for a cylin-drical gear unit. The components primarily used in this regard are listed. The combination of splash lubrication and pressure-feed lubrication is sensible when it is intended to take advantage of the benefits of both lubrication systems together. Critical lubrication points, such as the high-speed gear stage of a multistage gear unit and specific bearings, are supplied by pressure-feed lubrication and the slow-speed gear stages by splash lubrication. The cleaning and any necessary cooling of the lubricating oil take place outside the gear unit. Complicated inputs require suitable constructive lubrication and cooling solutions to ensure secure operation. Such an example is depicted in Figure 6.6/33. The lubricating oil is guided into the inside via boreholes and radial grooves. A lateral exit is provided in the borehole B 3, through which a part of the oil is branched off to the bearing of the hollow shaft W1. At the same time, this oil is also available to lubricate the spline gearing KZ2. The major part of the oil penetrates into the interior of the hollow shaft W1 via the jet D. Here it is uniformly distributed onto the internal wall of the hollow shaft for the purposes of cooling. The oil escapes via boreholes B4, collects in the interior of the hollow shaft W 2, finds its way via the boreholes B 5 to the spline gearing KZ1 and then – in the form of oil mist – into the region of the cylindrical gearing ZS. With such constructions, it is particularly important to accurately estimate the actual path taken by the oil. 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 401 Figure 6.6/29 Positions of the jets and directions of injection for injection lubrication [2]: a) direction of injection tangential to the pitch circles before and after meshing b) direction of injection on the circumference of the gears before and after meshing c) direction of injection parallel or inclined to the gear axes in the meshing depth or directly onto the gear body for cooling only d) centrifugal injection lubrication Figure 6.6/30 Relationship between the jet diameter and the rate of flow [2] 402 6 Load Capacity and Running Performance: External and Internal Gearings Figure 6.6/31 Temperature characteristics across the face width for helical gear units [6.6/2]: a) uniform oil supply, b) non-uniform oil supply Figure 6.6/32 Diagram of a pressure-feed lubrication system [6.6/37] 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 403 Table 6.6/11 Common Differences in Temperature between the Lubricating Oil and the Cooling Water [2] Lubricating oil - with no cooling (only lines and pump outside the housing) - with cooler for large gear units; long-term operation usually at nominal power - with cooler for small gear units; intermittent operation, key time splices below 70% nominal power 3 to 5 K 10 to 15 K 15 to 20 K Cooling water 10 to 20 K Figure 6.6/33 Design of a lubricating and cooling system for a planetary gear unit (marine gears with gas turbine drive; Entwicklungsbau Pirna) 6.6.5 Symbols and Symbol Explanations of Section 6.6 A mm2 area AG mm2 internal cross section area of the housing in the plane of gear section AK m2 cooling area of the housing (air- facing side) Aoil m2 internal area of the housing (oil- facing side) Aq m2 area of the section of a fundamental dividing rib a mm centre distance B mm face width of the gear unit b mm tooth face width b mm segment width, bearing face width C N basic dynamic load rating C Ws/K heat capacity of the gear box C0 N static load rating of a roller bearing C1, C2 - factors influencing gear face width and immersion depth CSp - splash oil factor CS W/(m2K4) emissivity coefficient of black body c Ws/(kgK) specific heat D mm outer diameter of a roller bearing D mm shaft diameter d mm diameter, general d mm nominal bearing diameter da mm tip diameter dm mm mean bearing diameter dw mm working pitch diameter ED % duty cycle e mm immersion depth e - bearing factor according to dimensioning table F N bearing force Fa N axial force Fbt N nominal tooth force in transverse section Fn N normal force Fr N radial force f0, f1, fa - roller bearing coefficients fG - thermal loading factor fK - cooling factor fW - factor of heat transfer to basement fZ0 - gearing loss factor of idling cycle fv - viscosity factor ga - leaving path of contact ga - entrance path of contact 404 6 Load Capacity and Running Performance: External and Internal Gearings gĮ mm path of contact H mm gearbox height HV - gearing loss factor h mm lubrication film thickness h mm tooth height hC mm height of pitch point above deepest point in tooth gap of the immersing gear i - gear ratio KA mm application factor k W/(m2K) heat transfer coefficient, general kR W/(m2K) heat transfer coefficient at ribbed wall L mm length of the gearbox L\* mm upstream length l mm length, general l mm segment length lh mm hydraulic length of gearbox in oil stream direction M Nm friction moment of roller bearing M0 Nm friction moment of unloaded roller bearing Ml Nm load-dependent proportion of friction moment M Ma Nm friction moment due to axial loading of cylindrical roller bearings MH Nm hydraulic loss moment m mm module m kg weight m\* m-1 reciprocal length coefficient n min-1 driving speed nG min-1 driving speed limit P0 N static equivalent load P1 N decisive bearing load for Ml P kW power, general Pa kW input power Pb kW output power PG kW heat limit power PN kW nominal power PV kW power loss of the gear unit PVD kW power loss of the sealing PVL kW power loss of the bearings PVX kW power loss of other parts PVZ kW power loss of the gearing PVZ0 kW gearing power loss of idling cycle PVZP kW gearing power loss of duty cycle p bar pressure pe mm base pitch pm N/mm2 mean contact pressure Qլ W heat flow Qe l/min oil injection quantity QլF W heat flow due to transfer to basement QլG W heat flow of gearbox to the surroundings QլK W heat flow due to convection QլF W heat flow due to transfer to basement QլS W heat flow due to thermal radiation QլU W heat flow due cooling circuit QլW W heat flow due to transfer across rotating of shafts and bearings Ra m arithmetic mean roughness index Re - Reynold’s number S mm mean bearing clearance So - Sommerfeld’s number s mm wall thickness sr mm wall distance t min, h time tB min operating time tL min idle time tP min pause time tS min cycle time U mm circumference of the wetted area of housing cross section AG Uq m circumference of the section of a basement rib u - gear ratio V - degree of gear unit losses VG - viscosity grade VL - degree of bearing losses VL0 - degree of bearing losses in idling cycle VLP - degree of bearing losses in duty cycle VZ - degree of gearing losses VZ0 - degree of gearing losses in idling cycle VZP - degree of gearing losses in duty cycle / m/s circumferential velocity /g m/s sliding velocity /L m/s air velocity /s m/s jet velocity /t m/s tangential velocity /Ȉm m/s mean sum velocity across path of action XL - lubricant factor XR - surface roughness factor x - thermal exponent x - coordinate Y - axial factor y - coordinate z - number of teeth z - number of segments α pressure angle α W/(m2K) heat transfer coefficient αw working pressure angle α∗ W/(m2K) equivalent heat transfer coefficient β helix angle ǻK K difference between oil injection and outlet temperature J - emission ratio İĮ - transverse overlap 6.6 Power Losses, Efficiency, Gear Heating, Lubrication 405 İ1, İ2 - partial overlap [tip/pitch point of pinion (1) and wheel (2)] Ș - efficiency Ș(oil) mPa s dynamic viscosity ȘZ - gearing efficiency KG C housing wall temperature Koil C oil temperature Koilb C working oil temperature Koil,ex C oil excess temperature KR C surrounding temperature Ȝ - heat transfer coefficient Ȝ\* - specific lubrication film thickness - number of friction Z - number of gearing friction mZ - mean number of gearing friction (oil) mm2/s dynamic viscosity Ȟ40 mm2/s nominal viscosity at 40 C '(oil), N(oil) kg/dm3 density '20, N20 kg/dm3 density at 20 C 'C, NC mm equivalent radius of curvature at pitch point 'm, Nm mm mean radius of curvature in normal section ı - sum roughness depth ıH N/mm2 Hertzian pressure at pitch point - min heating time constant -S min cooling time constant ȥ - relative bearing clearance Ȧ s-1 angular frequency Indices: 0 reference value 1 gear 1 (pinion) 1 1st cycle 2 gear 2 2 2nd cycle I, II gear stage A initial state a external B steady state b operating state b base circle base gear unit base C pitch point E injection lubrication ex excess temperature F fundamental fl coupling (flange) foot gear unit base (stand) forced forced convection free free convection G equilibrium state GG cast iron horiz horizontal areas i internal i rib number K convection L idling lim limit value M mass m mean max maximum n normal section oil lubrication oil R surrounding req required value S radiation S standstill St end face St steel sh shaft stag stagnation flow T dip lubrication t transverse section tang tangential flow vert vertical wall housing wall w working pitch circle 6.7 Noise Behaviour 6.7.1 Basic Principles 6.7.1.1 Introduction and Important Acoustic Terminology The following sections on noise behaviour will compile several new or repeated basic concepts, whereupon sound generation for transmissions and measures and experiences relating to noise reduction will be discussed. Further explanations of terminology relating to acoustics and specific acoustic relationships can be found in standard textbooks, such as [6.7/38], [6.7/39], [6.7/47], [6.7/50], [6.7/74], [6.7/107], and [6.7/158]. Depending on whether vibrations in the audible range occur in air, structures or liquids, we talk about airborne sound (audible sound) , structure-borne sound or fluid-borne sound . The healthy human ear senses pressure variations p˾ 0 in the air above a certain size within a frequency range f = 16 to 20000 1) Hz. At f = 2000 Hz, the perception threshold is about p˾0 = 2 ⋅ 10-5 Pa. For larger and also for smaller frequencies, the perception thresholds for p˾0 are larger. 1 ) The upper limit is often indicated as f = 16000 Hz. 406 6 Load Capacity and Running Performance: External and Internal Gearings Below the perception threshold ( f 16 Hz), the sound is known as infrasound; above the threshold it is known as ultrasound . A tone is understood as pressure variations in air perceived as sound, which proceed according to the function p(t) = p˾ cos(Ȧt+ĳ0). Sounds are created from the mixture of pure tones. When sound disturbs or annoys us and when it is harmful to our health, we refer to this as noise. For a given location where people are present, for example the workplace or home, the sound pressure is decisive. The noise generator, for example the machine or the transmission, must be characterised by a measurement that includes the whole subject, and not just the effect on a specific point. Therefore, the emitted sound power of the sound generator was chosen. To make it easier to detect the major areas of the occurring sound pressures p˾ = (0) ... > 20 Pa and sound powers ( P = (0) ... 10 8 W) and to better adapt these quantities to the human sensitivity, a logarithmic measure can be used. The measurements formed from the logarithm of the ratio between the existing sound pressure, the existing sound power and the reference values are known as the sound pressure level and sound power level (dimension is decibels, dB). The preferred reference values are specified in DIN EN ISO 1683 [6.7/153], such as p 0 = 2 ⋅ 10-5 Pa for the sound pressure and P 0 = 10-12 W for the sound power. To take into account the difference between the objective sound pressure level and the subjective perception of loudness, the sound pressure level is frequency weighted . A weighting has been developed from the different possibilities, which is used almost universally in technical acoustics and which is referred to as A-weighting. The corresponding weighted sound pressure level and sound power level is indicated by adding the letter (A) to the dimension in brackets [dB(A)]. After it is generated, the chain of effects of the sound consists of conduction and radiation. Even in the project planning and construction phase, it is particularly important to specifically influence a low noise emission of the driving equipment, which includes the transmission. The measures to reduce noise generation represent the primary method of noise reduction. In the case of secondary noise reduction, a work zone enclosure can, for example, minimise the impact on the work environment. Active measures should be introduced first. Noise protection measures for humans (ear plugs, helmets) are the last resort. Due to the increasing demands for the environmental safety of technical products, the noise problem plays an increasingly important role. Emphasis is also given to this situation through the revision and amendment of European Union directives and laws at the national level with partially lower immission limiting values, such as the EC Directive “Noise” 2003/10/EC [6.7/116] (and/or the EC Machinery Directive 2006/42/EC [6.7/117]). The following summary listed in alphabetical order should simplify navigation. • Admittance h (f) The mechanical admittance describes the sensitivity of a structure that has been excited by force and is formed by the ratio of the ĺ velocity to the exciting (alternating) force. It is a complex quantity. The point or input admittance is present at the point of excitation: e ee e(,)ˆ(,)ˆ(,)ȣxfhx fFxf= (6.7/1) In the practical case, the simplified ĺ effective values for the force and the velocity are used. The transfer admittance hT(f) results in evidence as to what energy is present at a given excitation force at a certain point in the centre (temporal and spatial) of the structure. The quadratic transfer admittance is used for the description of the sound power: i22 2()() ()Tȣfhf Ff= (6.7/2) The reciprocal value of the admittance is called impedance. The acoustic admittance combines sound particle velocity and sound pressure and is a real value in a plane wave. It is common practice to use acoustic impedance or characteristic impedance: 6.7 Noise Behaviour 407 i1 pcȣ h== ⋅ N (6.7/3) Ɣ Airborne sound Sound that propagates through air. Ɣ Bandwidth Frequency range of a number of adjacent spectral lines. Ɣ Coupling frequency f k Frequency at which the dynamic bending stiffness merges into the static bending stiffness [6.7/82]: ) 4 /(g2 k2 k f l c f = (6.7/4) Ɣ Critical frequency f g General: lower and upper frequency of a certain range. Specific: significant frequency (coincidence frequency) at which the bending wave length of the structure ȜB, which is determined by the material and dimensions, is equal to the airborne sound wavelengths Ȝ0. Below fg, the vibration excitation of the air goes back through bending vibrations, for example, from plates, Section 6.7.2.4, Equation (6.7/40); critical frequencies for plates made of steel, cast iron with lamellar graphite, and aluminium are given for plate thickness s in mm: Steel f g = 11830/ s Hz Cast iron EN-GJL-200 fg = 14620/ s Hz Aluminium fg = 11870/ s Hz Ɣ Effective value indicates the power transported by the sound waves and is formed from the root mean square of the time function of the respective physical variable. Example: Sound pressure i2 1() d Tp p t tT= (6.7/5) Ɣ Far field Range in which the flow processes (reactive energy), which change in shape, present in the immediate vicinity of the noise source have subsided; the area followed by the near-field region in which sound pressure p and sound velocity v are in phase (Figure 6.7/10). Ɣ Field quantity Quantity, the square of which is proportional to the power. Ɣ Force excitation Excitation of a structure by a time-dependent force; present, for example, if the operating forces are introduced via rigid structure areas. Force excitation occurs primarily in transmissions. Ɣ Free field (also direct sound field or free sound field) Present in the case of undisturbed sound propagation (free flat terrain): i~~211p ; J r r where p is the sound pressure, J the sound intensity, r the distance from the centre of the sound source. This principle is referred to as the distance law. Ɣ Frequency response The sound has a ĺfrequency spectrum that contains different frequency components with different amplitudes. The knowledge of the vibration amplitudes as a function of the frequencies is required in the studies of the noise generation mechanism, the sound radiation, the sound propagation in space, as well as the noise protection measures to be introduced and the assessment of the effects on humans. ż Frequency weighting The sensitivity of human hearing is frequency dependent. Therefore, the noise is frequency weighted as part of the metrological determination: the sound level is corrected by the quantity ǻL = ǻi Terz (Figure 6.7/1), which can be performed automatically by filters from the measuring instrument. Of the standardised weighting curves, the 408 6 Load Capacity and Running Performance: External and Internal Gearings A-weighting is of greatest significance for sound that is both annoying and harmful to hearing (Figure 6.7/1). At low frequencies and at very high frequencies there is a deduction; at medium frequencies there is a small addition. Thus, the following are commonly used: - The unweighted (linear) sound pressure level L p - The A-weighted sound pressure level LpA - The A-weighted sound power level LWA ż Frequency spectrum Decomposition of the noise into different frequencies or frequency spectra ( ĺ bandwidths). ż Octave band spectrum In theory, the frequencies that delimit the octave band are 21 / = 2ff (6.7/6) For m1 2 = f ff (6.7/6a) the following centre frequencies fm for the normal range in transmission noises are technically defined: (63) 125 250 500 1000 2000 4000 (8000) Hz ż Third-octave band spectrum The ratio of the frequencies that delimit the third-octave band is 213 / = 2ff (6.7/7) Three third-octave bands form one octave band. The technically defined third-octave centre frequencies are (50) (63) (80) 100 125 160 200 250 315 400 500 630 800 1000, and so on. The same steps are repeated in subsequent decades. ż Narrow band spectrum The chosen bandwidth can be as small as desired, depending on the measuring instrument. It can be relatively constant, for example, ( f 2 - f1)/fm = 0.03 (i.e., 3% bandwidth), or absolutely constant, ( f2 - f1) = 1 Hz. Table 6.7/1 Sensitivity Level of the Frequency Weighting, Curve A [6.7/155] Centre frequency of 3rd octave band Hz ǻiTerz dB Centre frequency of 3rd octave band Hz ǻiTerz dB 50 63 80 -30.2 -26.2 -22.5 800 1000 1250 -0.8 0.0 +0.6 100 125 160 -19.1 -16.1 -13.4 1600 2000 2500 +1.0 +1.2 +1.3 200 250 315 -10.9 -8.6 -6.6 3150 4000 5000 +1.2 +1.0 +0.5 400 500 630 -4.8 -3.2 -1.9 6300 8000 10000 -0.1 -1.1 -2.5 Ɣ Impedance Complex vibration resistance; reciprocal value of the ĺ admittance. Ɣ Insertion loss Obstruction of sound propagation through reflection (transfer of the mechanical ĺ admittance). Ɣ Mesh frequency f z Product of rotational frequency and number of teeth. 6.7 Noise Behaviour 409 Ɣ Near field Near field is the sound field in the immediate vicinity of the sound source; in this area, the sound power is not proportional to the square of the sound pressure, and sound pressure and sound speed do not have the same phasing. (Figure 6.7/10). Ɣ Noise Disruptive (annoying or harmful) sound. Ɣ Noise emission ( or sound emission) describes the emitted noise (from the transmission). Significant quant ities are the sound power level (ĺ sound power ĺ level quantities) and the workplace-related emission value according to DIN 45635 T1 [6.7/130]: pAeq pAeq 2A 2A L LK k K′=− − ⋅ (6.7/8a) with pAeqL′the time-averaged A-sound pressure level (energy equivalent) and k, K2A correction factors. It is used in cases where, due to the size of the transmission or the entire drive system, the determination of sound power is not possible or useful. In the absence of a defined workplace, the 1 m measuring surface sound pressure level LpA,1m is indicated. It is the averaged A-weighted sound pressure level of the gear on the measuring surface at a distance of 1 m from the gear. Ɣ Noise immission ( also sound immission) includes the noises that affect people and is dependent on the environmental conditions and the ratios of the sound propagation in the vicinity of the transmission and people. The sound pressure level ( ĺ sound pressure ĺ level variables) is used to identify the noise immission. ż Equivalent continuous sound pressure level L eq describes time-varying sound effects: p 0.3 ( ) eq T 1 = lg d dB 100.3 TtL qqt L (6.7/8b) The equivalence parameter q is defined in standard textbooks ([6.7/122], q = 3 corresponds to an energy-related averaging of the sound pressure level Lp(t)). Ɣ Radiation factor ı(f) Ratio of radiated sound power, for example, of a structure with bending vibrations, to the radiated sound power of a notional plate of the same surface vibrating in phase, the velocity square of which corresponds to the mean velocity square of the structure; see Section 6.7.2.4, Equation (6.7/39). Ɣ Radiation index L ı Results from the radiation factor ı; see Section 6.7.2.1, Equation (6.7/25). Ɣ Reverberant field (also diffuse field or reverberation chamber) Chamber (sound field) with a constant sound energy density that is independent of location; caused by the properties of a room (reflection). Ɣ Sensitivity level describes the characteristics of the frequency filter. Ɣ Sound Sound with many tones of an arbitrary frequency (broadband and tonal spectral components). Ɣ Sound absorption Obstruction of sound propagation through absorption of sound energy. Ɣ Sound intensity J( t) results from the product of the two sound-field quantities, the sound pressure and the sound particle velocity: 1() () d TJp t ȣttT=⋅ JGG (6.7/9) It is the sound energy passing per unit time through a surface element dS. The directions of the intensity vector and the sound particle velocity are the same. 410 6 Load Capacity and Running Performance: External and Internal Gearings Ɣ Sound level sizes (sound pressure level, speed level, sound power level, and so on) For better handling and possibility of comparison, a logarithmic measure with the dimension dB (decibels) is introduced. The sound pressures, for example, from different sound sources often have values that differ by up to several orders of magnitude. Such level quantities based on reference values are ż Sound pressure level dB ~ lg 20 = 0pppL Sound pressure reference value p0 = 2 ⋅ 10-5 Pa (6.7/10) ż Speed level 0 = 20 lg dBȣȣLȣ Reference speed /0 = 5 ⋅ 10-8 ms-1 or 1) (6.7/11) ż Sound power level W 0 = 10 lg dBPLP Reference sound power P0 = 10-12 W (6.7/12) ż Sound intensity level J 0= 10 lg dBJL J Reference sound intensity J0 = 10-12 Wm-2 (6.7/13) Ɣ Sound particle velocity (or velocity) / (t) describes the local sound particle velocity at the rest position in the case of progressive waves; is almost set equal to the vibration velocity. For sound radiation, the quadratic effective speed is decisive. Ɣ Sound power P(t) is the sound energy flowing in a time unit through a surface S (e.g., the surface encasing the gear). With the vertical component of the intensity belonging to the surface element dS, the following applies: S P = JdS (6.7/14) The sound power is used for the product-specific marking of the noise emission; through it, comparisons with any sound sources are possible. Ɣ Sound pressure p(t) An alternating pressure superimposed on the direct pressure of the air; it is a field quantity. Its effective value is used for the determination; it indicates the power carried by a progressive sound wave. Ɣ Structure-borne sound Vibrations and wave processes in solid structures. Bending vibrations are particularly important in the case of transmissions. Due to the sound generation process, the structure-borne sound processes are responsible for the airborne sound power to a decisive degree. Ɣ Structure vibration transmission level L Sh(f) describes the behaviour of a structure with a specific surface under force or velocity (speed) excitation and results from the transfer admittance ( ĺ admittance) and the radiating surface; see Section 6.7.2.4, Equation (6.7/33). Ɣ Time weighting For the time of averaging the effective value, three quantities are defined: S (slow) 1000 ms F (fast) 125 ms I (impulse) 35 ms Ɣ Tonality Occurrence of single tones in noise, the amplitudes of which are significantly higher than the adjacent noise components. The conditions are currently defined in pollution regulations. According to [6.7/160], [6.7/161], [6.7/162], the annoyance is corrected using a tonality surcharge. Ɣ Velocity excitation A time-dependent vibration velocity is imposed on the structure. A change in the admittance of the structure has virtually no effect on the exciting vibration velocity. Velocity excitation is present, such as in thin-walled panels. 1) In DIN EN ISO 1683, v 0 = 10-9 m/s is specified for other media (liquids and solids). 6.7 Noise Behaviour 411 6.7.1.2 Arithmetic Operations with Level Values Incoherent (independent) sources of sound are superimposed in a room, and the powers and power quantities are added together. This is then the sound power level: Wnn i Wges i=1 i = 1 00.1dB = 10 lg dB = 10 lg dB 10iL PLP (6.7/15a) This also applies analogously to the sound pressure level L p, since the power is proportional to the square of the effective value of the pressure. pi2 nn i 0.1 dB pges 2 i=1 i = 1 0 = 10 lg dB = 10 lg dB 10L pL p (6.7/15b) It therefore applies to the superposition (addition) of level values in general: n 0.1 dBges i = 1i = 10 lg dB 10 LL (6.7/15) where Li and L ges can be a sound power level but also a sound pressure level. When an L2 interference signal occurs, the overall sound level Lges according to Equation (6.7/16) can be freed from this and the desired signal (useful signal) L1 eliminated. ges 20.1 0.1 dB1dB 10 lg dB 10 10L LL= − (6.7/16) Examples: - A second source with the same component sound power increases the total sound power level by 3 dB. - If the sound pressure p increases to twice the value, according to Equation (6.7/10), the result is an increase in the sound pressure level by 6 dB. - If two incoherent sources with the same size sound pressure are superimposed, the overall sound pressure level increases by 3 dB. If an averaging of noise levels should be performed, such as from several measurement results of sound pressure levels on an enveloping surface around the measured object, the averaging of the squares of the sound pressure must be carried out (average of the energies): i2 nn i 0.1 dB m 2 i = 1 i = 1 011 = 10 lg dB = 10 lg dB 10nnL pL p (6.7/17) If the maximum level difference between the individual levels is not greater than 5 dB, it is possible to use the arithmetic mean by approximation : n mi i = 11 nL L ≈ (6.7/18) 6.7.2 Mechanical Noise Generation 6.7.2.1 Basic Equation, Sound Propagation The knowledge of the mechanical and acoustic relationships in the case of mechanical noise generation is essential for 412 6 Load Capacity and Running Performance: External and Internal Gearings • assessing the frequency response and • influencing the noise behaviour. From the basic equation for the sound power of a plane wave (6.7/10) and the relationship of sound intensity to the sound pressure and the sound particle velocity (6.7/9), with the introduction of the acoustic quantities • transfer admittance, • radiation factor )(f) and • acoustic impedance (characteristic impedance) p˾/ /˾ = N c the basic equation of the mechanical noise generation and the machine-acoustic basic equation for the sound power results: i22 T () () () ı() Pf cF fhfS f =⋅ ⋅ ⋅ ⋅⋅N (6.7/19) F˾(f) Excitation force, frequency-dependent hT(f) Mean transfer admittance (spatially and temporally averaged) ı(f) Radiation factor, frequency-dependent S Radiating surface N, c Density and sound velocity in the medium surrounding the transmission In the level notation, the sound power level LW (f) is WF h T S ı () () () ()LfL fL fL L f=+ + + (6.7/20) The shares of the sound power level are determined by the following main influencing factors : LF Ɣ tooth deformation Ɣ tooth system deviations Ɣ mounting deviations Ɣ dynamic properties of the gear-shaft-bearing system LhT, LS Design and dynamic properties of the • gear bodies • shafts • bearings • housings LhT, LS, Lı Ɣ housing form Ɣ stiffness Ɣ wall thickness Ɣ material Ɣ material and groove damping The definitions of the shares are as follows: Power level F 0( ) = 20 lg dB FfLF N 1 =0F (6.7/21) Transfer admittance level T hT T0( ) = 20 lg dB hLfh 81 1 T0 51 0 m s N h−− −=⋅1) (6.7/22/23) Envelope surface level S 0= 10 lg dB SLS 2 0= 1 mS (6.7/24) Radiation index , dB ıı lg10 ) ( 0ı = f L 1ı0= (6.7/25) The sound propagation in a transmission is clearly illustrated in Figure 6.7/2 with an example of a transmission cross section. 1) Based on 8- 1 051 0 m s υ−=⋅ ; F0= 1 N; according to DIN EN 21683 [6.7/153], other values apply that, however, do not appear useful [6.7/39]. 6.7 Noise Behaviour 413 Figure 6.7/2 Sound propagation in a transmission The (occurring) airborne sound measured in the vicinity of the transmission results primarily from the structure-borne noise that propagates through the gear bodies, shafts, bearings, bearing brackets and housing walls. The dynamic bearing loads (6.7.2.2) are responsible for the ĺ structure-borne noise excitation of the housing [6.7/32], [6.7/78]. As a rule, “force-excited” structure-borne sound in transmissions is decisive, wherein the housing structures are located in the force flow and are particularly excited to bending vibrations through alternating forces or torques. It is, however, also possible that structure-borne sound vibrations occur in elements outside the operating force flow, as vibration velocities are imposed by the coupling points, such as adjacent housing walls. This involves velocity excitation, as opposed to force excitation. Airborne sound occurring inside a closed transmission, which results from the radiation of the vibrating parts located inside, also excites the housing walls, but its involvement in the airborne sound power of the transmission is generally negligible in size (<10%). In the case of special constructions, (e.g., large-scale housings with a pure envelope function), the influence can increase significantly. For certain characteristic sounds of gear units, terms such as howling, whistling, grinding or singing are used, depending on the subjective impression [6.3/5], [6.7/49]. Flank separation and abutment against the non-working flank can occur with strong external excitation. With gear pairings w ith low or no load, this effect is also caused by tooth system deviations. These usually clearly audible phenomena are called rattling and clattering [6.7/36], [6.7/58], [6.7/77]. Flank separation is also possible due to tooth stiffness excitation in loaded transmissions (particularly in the case of spur gearing in the main resonance range [6.3/1]). 414 6 Load Capacity and Running Performance: External and Internal Gearings 6.7.2.2 Excitation of Structure-Borne Sound A. Internal excitation in the mesh of loaded transmissions The key main sound source is the mesh. The dynamic tooth force Fdyn(t) represents the key para- meter of the excitation mechanism in the vibration system gear-shaft-bearing, which is discussed in detail in Section 6.3.4. The main causes are as follows: I The time-varying tooth mesh stiffness acts as a parametric excitation (e.g., [6.7/21], [6.7/48], [6.7/96], [6.7/107]). Characteristics for this excitation of vibrations are - The average tooth stiffness c m (determines the resonant frequencies) - The tooth stiffness variations ǻc (the frequency composition of the excitation results from the form) - The amplitude (a measure of the excitation intensity [6.7/124]; the fundamental vibration often has the largest amplitude with mesh frequency) II The manufacturing-related tooth system deviations and flank modifications represent external excitations (path excitations) (Section 6.7.6.3). They emerge as exciting variables especially with low tooth loads. Tooth flank modifications that are designed for much higher loads also act in the same way. The deviations that affect the mesh result not only directly from the tooth system deviations, but also from the deformations of shafts, bearings and the housing, as well as from mounting and assembly deviations and bearing clearances. Under load, the deviation of rotation can serve as a characteristic [6.7/2], [6.7/12], [6.7/48]. The exciting frequency is usually not an integer ratio of the mesh frequency. III Meshing interference through premature meshing caused by load-induced deformations and tooth system deviations. Thus, a force impulse is generated with the mesh impact [6.7/83], [6.7/105], which, however, is negligible in comparison to the excitations through I and II [6.7/48]. IV Materials, geometry and damping, load and speed The effectiveness of the different influences I to III varies and is largely dependent on the influences of IV. Except in exceptional cases, the friction noise (due to the friction force reversal) and the pitch circle impulse noise components fade noticeably into the background compared to the major influences [6.7/48]; this happens even more noticeably the greater the circumferential speed and the transferred forces become [6.7/33]. In addition to the internal excitation, resulting from the ratios inside the gear from the mesh to the bearing, an external excitation may also exist due to external influences (Section 6.3.3 and [6.7/60]). B. Dynamic bearing loads The dynamic bearing loads act on all bearings of the transmission. They are determined by the excitation in the mesh, the properties of the vibration system gear-shaft-bearing, including the momentum [6.7/60], as well as by the periodic change of the non-linear roller bearing stiffness. The dynamic of the flexible shafts also plays a role in this case. While the actual quantities of bearing stiffness and damping generally do not have a significant influence on the dynamic tooth loads, they are essential parameters for the dynamic bearing loads. Thus, the time profile and the amplitude-frequency correlations of the tooth and bearing loads also differ in principle. The difference is exemplified in Figure 6.7/3, as the mathematically determined force levels with a variable input rotational speed of a straight-tooth cylindrical gear transmission are contrasted with a lower mesh damping. Theoretical calculations on the effect of bearing stiffness on the structure-borne sound of a transmission are made in [6.7/66]. Both the axial and radial dynamic bearing loads are responsible for the excitation of the gear housing walls (refer to Section 6.7 Noise Behaviour 415 6.7.6.4). Even greater structure-borne sound amplitudes may result from the axial force excitation, due to higher input admittance. Examples of frequency responses for radial and axial bearing loads with the cylindrical gear transmissions studied in [6.7/61] are shown in Figure 6.7/4. Figure 6.7/3 Comparison of the curves of the dynamic tooth and bearing forces (example) [6.7/78] Figure 6.7/4 Example of the frequency response of the radial and axial bearing forces [6.7/61] In addition to experimental research results [6.7/60], [6.7/61], [6.7/78], mathematical simulations can also be found, for example, in [6.7/34], [6.7/60], [6.7/71]. In [6.7/71], a bearing load level for multi-stage cylindrical gear transmissions is defined as a parameter 1). Reference [6.7/60] describes a modularly constructed spatial vibration system, in which the dynamic bearing loads can be deter-mined from the inner and outer excitation under the consideration of the flexible shafts. The simu-lation program of the Research Association Antriebstechnik e.V. described in [6.7/22] can be used for this purpose. Calculations of bearing load have led to successful studies on gear motors [6.7/23]. In contrast to the size of the static bearing loads, the size of the dynamic bearing loads with the fast-speed stages are generally significantly higher compared to the slow-speed stages, so these bearing positions have primary importance and ar e considered priorities for noise-reduction measures. On the basis of a good concordance with measurement results, information about the frequency-dependent values of both the radially and axially acting dynamic bearing loads can be obtained. They act on the transmission housing while superimposed, but in different coordinate directions. 1) Compensation effects are discussed, among other things, in order to reduce the excitation of, for example, two- stage gear transmissions, the stages of which have the same meshing frequencies. 416 6 Load Capacity and Running Performance: External and Internal Gearings Attention must also be paid to the transfer conditions from the bearings to the housing. Thus, for example, depending on the pre-load of rolling element bearings according to the pressure distribution, the force-transferring area also changes. A high elasticity of the housing may also be of importance. However, the excitation by the bearing itself, due to the rotating angle-dependent stiffness variation caused by the finite number of roller elements, is generally of minor importance. For worm gear transmissions, which are used in elevators, for example, it can be superficial, since in this case the excitation is very low due to the gearing. The bending vibrations of the shafts can be dominant for the noise behaviour, bot h in transmissions with a low sound power level and in transmission stages operated beyond the critical speed. 6.7.2.3 Excitation Frequencies A. Shares Transmission noises are composed of broadband stochastic and tonal components. The noise component consists of friction, air turbulence and oil injection noises [6.7/17]. The tonal com-ponents are determined, among other things, by function-related excitation frequencies. When they are in congruence with component natural frequencies, they generate resonance vibrations. Function-related excitation frequencies, which are relevant to the sound radiation, are as follows: • Rotational frequency n f=n (6.7/26) • Mesh frequency n z f z f⋅ = (6.7/27) • Machine frequency n M Mf z f⋅ = (6.7/28) • Frequency of the same tooth position 2n1 1n2 kzfkzfk f ⋅ = ⋅ = (6.7/29) • Rolling bearing frequencies (stiffness variations) 1) o Rotating inner ring, fixed outer ring (outer ring damage) ()() cos / - 1 2 = r wkIδ d d nwf (6.7/30) o Rotating outer ring, fixed inner ring (inner ring damage) ()() wk rA 1 + / c o s į 2wfn dd = (6.7/31) where n speed of the shaft and roller element bearing ring in question z number of teeth of the gear in question zM number of teeth of the worm gear portion of the gearing machine on which the gear in question was toothed. k greatest common divisor of the number of teeth. It is calculated by dividing the number of teeth into prime numbers and multiplying the common prime numbers. w number of rolling elements dwk rolling element diameter dr rolling circle diameter; average of the raceway diameter of the inner and outer ring į contact angle between the rolling elements and outer/inner ring; contact angle 1) Rolling element bearing frequencies do not need to appear. They can also be recognised through an amplitude modulation via amplitude vari ations of the modulation frequency. 6.7 Noise Behaviour 417 B. Mesh frequency The mesh frequency has priority in force excitation; with it, the largest energy feed takes place in the housing. If the excitation force curve is not known as a function of frequency, the value for the amplitude and frequency range of the excitation force can be roughly correlated to the mesh fre-quencies of the gear stage with the highest up to the gear stage with the lowest circumferential speed. Stage I 1 1 |zI60|nf z =⋅ Stage II 11|zII 360Inf zi=⋅ ⋅ | (6.7/32) Stage III 111|zIII 560II I|nfzii=⋅ ⋅ with z1, z3, z5, and so on, the number of teeth of the drive pinion of the respective stage. Multiples of the excitation frequencies of each gear stage can be significant for effective gearing deviations, depending on the force spectrum and correlations. In practical cases, the multiples of the mesh frequencies appear up to four times (also up to ten times in the case of spur gearing): 2 f Z, 3 fZ, 4 fZ, and so on [6.7/30]. The probability that a machine frequency occurs at such a high energy that it determines the noise significantly is relatively small. The levels of mesh frequencies are most often dominant. The frequency range, in which the mesh frequency of the gear stage with the largest pitch circle velocity is located, generally exhibits the highest level. Depending on the position of the mesh frequencies and their harmonics relative to the position of the first housing resonant bending fre-quency and other component natural frequencies, different relative frequency spectra may result. If, for example, the mesh frequency is below the first housing resonant bending frequency, the amplitude determined by the sound power level may well occur in the case of a harmonic of the gear mesh frequency. The sound spectrum is dependent on both the toothing geometry and the toothing accuracy, as well as the transmission housing geometry. C. Rotational frequency f n and frequency of the same tooth position f k For the absolute sound power of a transmission, the occurrence of these frequencies and their multiples is less important than for the subjective impression of a smooth run. They are below the audible range or in the region of low auditory sensitivity and generally have low values. At a high intensity they lead to complaints; when they are between approximately 0.1 and 15 Hz, they occur as a modulation of a carrier frequency (e.g., mesh frequency) and are audible as a modulating level. The level variations are noticeable even at a low absolute level. Measures to prevent the occurrence of such frequencies by limiting their causal origin according to [6.7/17] are listed in Table 6.7/2. Table 6.7/2 Excitation Frequencies fn, fk: Causes and Remedies [6.7/17] Description Cause Remedy Rotational frequency fn Individual deviation per revolution, e.g., imbalance, concentricity variation, large individual tooth-to-tooth pitch deviation - Balancing - Precise alignment of the workpiece during gear cutting - Step by step over several teeth with flank grinding 418 6 Load Capacity and Running Performance: External and Internal Gearings Table 6.7/2 Excitation Frequencies fn, fk: Causes and Remedies [6.7/17] (continued) Description Cause Remedy Wobble frequency 2 fn Wobbling of the gear - Mounting of the gear on the shaft before the gear cutting - Alignment on both bearing positions - Long hub length in g ears that are pulled onto the shaft after gear cutting Frequency of the same tooth position fk Cumulative pitch variation; concentricity variation or individual variation on both gears of a pairing - Avoid or reduce the variations referred to as causes - In the case of integer ratios, a relative rotation between the pinion and gear of 90 to 270 helps if necessary 6.7.2.4 Transfer and Radiation Behaviour of Transmission Housings A. Transfer behaviour The transmission housing may be generally considered as a box structure in which vibrations are induced in the individual housing walls through the exciting forces F input at the bearing posi- tions. As a parameter for the vibration characteristics, the surface-weighted transfer admittance is used. In level spelling, the structure vibration transmission level (ĺ structure-borne sound) is defined as follows: Sh S hT () ()Lf LL f =+ (6.7/33a) with the Equations (6.7/22) and (6.7/24). Taking into account the radiation index according to Equation (6.7/25), the parameter of the struc-ture transfer measurement, which is also commonly used, results: S h () () ()Lf LfLfσ =+ (6.7/33b) Analytical and numerical simulation methods can be employed to describe the dynamic structural properties, and theref ore the resulting speed ȣ~(ĺ sound particle velocity) of the housing surface. It makes sense to use numerical calculation methods, such as the finite element method (FEM) for structure-borne sound, the infinite element method for airborne sound (explained in [6.7/11]) and the multibody simulation (MBS) method (described in [6.7/60], for example), but these can provide only approximate solutions [6.7/39], [6.7/77]. It is important to always be in correlation with the real structures. In this regard, the experimental modal analysis [6.7/75] is an effective method in which the experimentally determined structure vibration transmission level is represented by a mathematical model. As a numerical calculation method, the boundary element method (BEM) can be used for the particular circumstances of the sound radiated through the transmission [6.7/39]. For ease of understanding, the correlations of structure-borne noise behaviour at the “plate” structure element are explained below. In most cases, the individual housing walls can also be correlated to it, depending on the boundary conditions. For this purpose, there is extensive knowledge of the structure vibration transmission level. Relevant documentation has been developed by F ller [6.7/19], Welp [6.7/99], Storm [6.7/81] and Bock [6.7/6] for the elaboration of estimation methods (refer to Section 6.7.6.5). Since the housing shape influences the sound radiation, the use of the “cylinder” element is appropriate for planetary gear transmissions. 6.7 Noise Behaviour 419 The principal schematic of the structure vibration transmission level of a plate as a function of the exciting frequency is shown in connection with the influence of the bearing in Figure 6.7/5. A distinction is made between different frequency ranges subject to different principles, which should be considered especially in terms of noise-reduction measures: • Quasi-static range • Eigentone range • Transition range To estimate the structure vibration transmission level of a rectangular plate, for a central force excitation according to [6.7/81] (really only from five resonant frequencies in the frequency band), the following applies: Quasi-static range ( ) 2 (11f f≤ () ()2 Shq32 2 11 Str T010lg dB 8( 1 ) fL fm m B S hπη = +⋅ (6.7/34) Eigentone range ( f f≤11 ) () ()2 She2 Str T0ʌ10lg dB 512Ș L fm m BS h = ⋅ (6.7/35) In the Equations (6.7/34) and (6.7/35) where ()2 T0Sh Reference structure vibration transmission level, ()215 4 2 2 T 0= 2.5 10 m s N Sh−−−⋅ ȘStr Structural damping, data for transmission housing in [6.7/81], ȘStr = 0.001 to 0.1 f11 First resonant frequency of the plate with the basic mode of vibration respectively in the two coordinate directions x and y of the plane [6.7/38] m Compound coating of the plate with the thickness sm m m = s′ ⋅N ; N density (6.7/36) B Bending stiffness of the plate with the thickness sB and the elasticity of material E ()3 B 212 1 - ȞEB = s⋅′ with Poisson’s ratio (steel: = 0.3) (6.7/37) For inhomogeneous plates: s m Thickness of a homogeneous plate that has the same mass as the inhomogeneous plate in question s B Thickness of a homogeneous plate with the same bending stiffness as the inhomo- geneous plate in question s m, sB According to documents on the basis of [6.7/81] or program MASAK [6.7/63] The calculation of the first resonant frequencies of transmission housing walls (rectangular plates, circular plates and beams) is, for example, given in [6.7/81]. It refers to simple rough calcula-tions primarily on homogeneous structures. The resonant frequencies of inhomogeneous housing walls can thereby be calculated using a reference thickness s E, which can be determined with reference to [6.7/81] according to Equation (6.7/38): E3Bm= / s ss sm, sB Equations (6.7/36) and (6.7/37) (6.7/38) 420 6 Load Capacity and Running Performance: External and Internal Gearings Figure 6.7/5 Schematic of the structure-born sound index of a plate Of particular importance are the mounting and fixing conditions of the structural elements, the resonant frequencies of which are to be determined. The experimentally determined boundary conditions in housing dimensions (length/width > 2 and height > width according to [6.7/79]) can be used for the range of the first bending resonant frequencies of a box-shaped transmission housing. In the estimations, plates that are simply supported on all sides are often calculated. Studies on coupled plate and housing structures using the finite element method are described, for example, in [6.7/43]. The individual housing walls of a transmission generally have different structure vibration transmission levels; they exhibit different resonant frequencies and different amplitudes at the same excitation frequencies. As an example, Figure 6.7/6 illustrates these relationships in a modern industrial gear transmission (measured according to [6.7/79]). The measurement methods used for the experimental determination of the transfer admittance of a transmission housing, that is, the vibration behaviour of the passive structure, are essentially characterised by • mechanical excitation via an electro-dynamic vibration exciter or impulse hammer, thereby measuring the exci- tation force by means of a force sensor and the response in the form of the radiated sound power or accelerations or velocities of the structure surface, • acoustic excitation through a sound field, thereby measuring the sound power and the accelerations generated on the structure surface (reciprocity method [6.7/17]). While mainly static behaviou r and no dynamic effects can be seen in the quasi-static range, all fnx,ny resonant frequencies are i n the eigentone range of the plate. In principle, the amplitude dropin the eigentone range of a single plate is on average 10 dB per fre- quency decade (10 dB/decade)(at constant excitation), while themaximum amplitudes usually de-crease much faster (20 to 30 dB per frequency decade) [6.7/39]. Figure 6.7/6 Various structure vibration transmission levels of individual walls of a modern industrial gear transmission [6.7/79] 6.7 Noise Behaviour 421 B. Radiation behaviour To describe the radiation behaviour, the radiation factor ()fσ is used. ()2 00() ()Pff cS fȣσ = ⋅ N (6.7/39) N0c0 = 408 Ns/m3 (acoustic impedance of the air) It links the radiated sound power P (f) with the mean velocity square ()fȣ2~, which is formed over the entire radiating surface S. For the radiation index, Equation (6.7/25) applies. The sound radiation is dependent on the ratio of the wavelengths of the airborne sound ( Ȝ0) and the bending wave ( ȜB) of the structure. Here, Ȝ0 = ȜB occurs at the coincidence frequency or critical frequency fg. 2 22 00 g L12(1Ȟ) ' 2ʌ '2ʌcc mfBc s−== ⋅ (6.7/40) with s Plate thickness (in m) Ȟ Poisson’s ratio (steel: Ȟ = 0.3) c0 Sound velocity in air (in ms-1) cL Longitudinal wave velocity in plate material (in ms-1) The frequency dependence essentially exhibits three ranges (Figure 6.7/7): Figure 6.7/7 Frequency dependence of the radiation index Lı The physical backgrounds for the course of the radiation index, particularly in the range of the critical frequency, are clearly explained by the so-called track adjustment (coincidence) [6.7/41]. Statements on the accuracy of the calculation can be found in [6.7/39]. Taking into account [6.7/20] and [6.7/27], the calculation and measurement results for the radiation index of plates are discussed in [6.7/74]. According to data, for example, in [6.7/39] and [6.7/74], the radiation index can also be greater than zero. Approximation formulas for calculating the radiation factor for the individual frequency ranges are summarised in [6.7/17], [6.7/81], and [6.7/86]. In [6.7/39], comparisons of analytical and numerical methods and the use of FEM and BEM are discussed. • Piston radiator range : The housing behaves like a piston dia- phragm or a zero-ord er monopole sound source, in which all points of the struc- ture surface vibrate in phase with the same amplitude. • Short range : For Ȝ0 > ȜB, a partial hydrodynamic pressure balance of adjacent vibrating shaft regions in phase opposition, and thus a reduced radiation, results. • Radiation range: There is a maximum radiation of energy. 422 6 Load Capacity and Running Performance: External and Internal Gearings In the range f = (0.75 to 2) fg, insertion losses and, therefore, high sound intensities occur in excited structures. It is therefore to be avoided. For optimum sound insulation, certain corre- lations of exciting frequency, material and material thickness (wall thickness) must be observed (Figure 6.7/8). The choice of the wall thickness for the edge length can be made independent of the diagram in Figure 6.7/8: mm 10 3403 errf < l ; with ferr in 1/s. (6.7/41) 6.7.2.5 Determination of the Total Sound Power With the machine-acoustic treatment of a transmission, the task is to find a model structure that reflects the real structure and the same acoustically important properties in terms of structure-borne noise, structure-borne sound transmission and sound radiation. The housing is divided into individual structural elements (such as plates and cylinders), the mass density (m ), bending stiffness (B ), position of the first resonant frequencies ( f xy), and coupling and radiation factor ( ı) of which must be recordable. The sound power level of the entire transmission LW then consists of the component sound power levels LWi of the individual structural elements. In most cases of cylindrical gear transmissions, this can emanate from box structures, where the housing walls constitute plate elements. The component sound power level of a wall is determined by the excitation force, structure vibration transmission level and radiation index (Figure 6.7/9): Wi0.1 W 110lg 10 dBn L iL == (6.7/42) From the formal logarithmic summation, it can be deduced that individual gear housing walls can determine the overall level, which is also often the case in practice. Caution: Do not overestimate the effect of ribs as a boundary of reference surfaces within larger surfaces! Figure 6.7/8 Selection of the wall thickness according to the occurring exciting frequency to achieve optimum sound insulation 6.7 Noise Behaviour 423 Figure 6.7/9 Schematic of the formation of the component sound power level of a transmission wall 6.7.3 Metrological Determination of Sound Power 6.7.3.1 Airborne Sound Measurement Method The metrological determination of sound power, for which different standardised methods are available depending on the sound field properties (Section 6.7.3.3), can be carried out by deter-mining the sound pressure level or even the sound intensity. The criteria for the selection of a measurement method are primarily the required accuracy, the size of the transmission, the measurement environment, the background noise level and the available measuring equipment [6.7/138]. When applying the sound pressure measurement to determine the sound power through the integration of the sound pressures over a plurality of measurement points, it is as-sumed that the sound pressure and sound particle velocity are in phase. For measurements in the ĺ near field , where this condition is not present, the direct measurement of the ĺ sound intensity is advisable. To explain the correlations of the near field, far field, direct sound field, reflection sound field and the measured distance to the sound source, the sound field structure is shown in a semi-reverberant room in Figure 6.7/10 [6.7/73]. The most important measurement methods based on the sound pressure measurements are as follows: 424 6 Load Capacity and Running Performance: External and Internal Gearings Figure 6.7/10 Sound field structure in a semi-reverberant room [6.7/73] Free-field method [6.7/130], [6.7/142], [6.7/143], [6.7/144] The temporally and spatially averaged sound pressure level L ժP is determined on a trans- mission enveloping surface (measuring surface S). The measuring points must • be located in the far field of the trans- mission • be located in the direct sound field of the ambient space • exhibit the same direction for the sound field quantities / and d S. In the enveloping surface method, usually a microphone distance of 1 m from the transmission is used. 0pW lgSSL L+ = (6.7/43) S0 = 1 m2 Reverberation chamber method [6.7/130 T2 and T3], [6.7/140], [6.7/141] A diffuse sound field and a steady state of equilibrium between radiated and absorbed sound power are assumed: Wp 010lg dBALLA=+ (6.7/44a) A0 = 4 m2 (reference sound absorption surface of the chamber) The methods mentioned have significant disadvantages: • The interference components of the external noise sources must be low. • A very high measurement effort with high precision of the measurement result is required. • Measurements must not take place in the near field. For most practical applications, the sound intensity measurement method is particularly suitable for measurements in the near field, as it can largely compensate for the background noise and the influences of the reflected sound. It is an enveloping surface method that is described in [6.7/147] and [6.7/148] and explained in detail, for example, in [6.7/74]: 0J W lg10SSL L+ = (6.7/.44b) A sound intensity probe is used to record the noise. With one-dimensional probes, two micro-phones are placed in the probe at a certain distance from each other. The sound pressure is detected through the average of the two microphones. The sound pressure gradient necessary for the determination of the sound particle velocity is approximately 6.7 Noise Behaviour 425 calculated by the phase difference of the sound waves at the two microphones. In order to cal- culate the phase difference accurately, both microphones have to have phase characteristics that are as identical as possible. The sound intensity is then determined by forming the product of the sound pressure and sound particle velocity. There are two standard methods of measurement, which relate, on the one hand, to the measurement at discrete points and, on the other hand, to the scanning method [6.7/147], [6.7/148]. The principal chain of measurement instruments for measuring airborne sound consists of • Microphone o as recorder of the measurement signal (transducer of the sound pressure into an analogue electrical signal); condenser microphones for precision measurements, ceramic micro-phones for lower accuracy. • Sound level meter o with amplifier o with third octave, octave or narrow band filter o with evaluation options (A, S, F, I) o with analogue or digital display for measuring the sound pressure or sound intensity • Recording instrument as o recorder o coupled computer system (especially for further processing of the measurement signals up to the sound power) The technical characteristics of the measuring devices and the accuracy requirements are laid down in standards, such as in general terms in DIN EN 60651 [6.7/155], and in particular terms, for example, for intensity measurement devices in DIN EN 61043 [6.7/156]. There is a wide range of industrially produced measuring devices with standardised characteristics. Suppliers include, for example, the companies Bruel & Kjaer, Norsonic, and Onosokki. For the primary noise control, a variable deployable laboratory equipment system is necessary, while most porta-ble devices with integrated filters are sufficient for practical applications. A list of candidate standards for the determination of sound power and other acoustic quantities is contained in Section 6.7.3.3. 6.7.3.2 Structure-Borne Sound Measurement Method If airborne sound measurements do not make sense because the external noise is too loud or the influences of reactive sound fields are too great, the application of structure vibration transmission levels is possible [6.7/135]. One such method is standardised in DIN 45635 T 8 [6.7/131], according to which the sound power can be determined. The vibration velocity is used as a measurement variable, which is determined via a structure-borne sound sensor. Due to the high number of measurement points, among other things, (e.g., with an acoustic emission area < 1 m 2 of about 10), this method is relatively time consuming. Through a conversion of the measurement results for the sound radiation of a piston radiator ( ı = 1), an upper estimate of the airborne sound power level is obtained. For structure vibration transmission levels, measuring instruments are often used that can handle both airborne and structure-borne sound signals (Section 6.7.3.1). Piezoelectric transducers, which produce a voltage proportional to the acceleration (accelerometer), have proven to be a structure-borne sound sensor. The vibration velocities or displacements can also be determined with the help of integrating amplifiers. For low frequencies and high accelerations, quasi-charge amplifiers are required. 426 6 Load Capacity and Running Performance: External and Internal Gearings 6.7.3.3 Standards In the international and national framework, standards are created that, independent of • the environmental conditions, • the size of the object, • the character of the noise generated, or • the accuracy of the sound power level reading, establish uniform procedures for the determination of the sound power level of technical sound sources, such as equipment, machinery, machinery parts, assemblies, and facilities, so that results are comparable. In Table 6.7/3, information on the standard deviation σR for different measurement methods of the accuracy grades 1 and 2 is partially reproduced in summarised form; σR includes the uncorrectable error. Table 6.7/3 Standard Deviation of Reproducibility σR in dB for Measurement Methods Accuracy grade Standard Publi- cation Environment Measurement methods σR for 3rd octave band 50 to 10000 Hz σR for 3rd octave band 63 to 8000 Hz σR for overall sound pressure level A-weighting 1 Precision method DIN EN ISO 3745 [6.7/143] 2004 Anechoic chambers and half-spaces Free field method ≤2 - - DIN EN ISO 3741 [6.7/139] 1999 Reverberation chamber Reverberation chamber method≤3 ≤2.5 0.5 DIN EN ISO 9614 T 1 [6.7/147] 1995 Reverberant sound field; outdoors over reflecting plane Intensity method ≤21) 2) ≤2 - 2 Technical method DIN EN ISO 3743-1 [6.7/140] 1995 Test room with reverberant walls Reverberation chamber method- ≤3 1.5 DIN EN ISO 3744 [6.7/142] 1995 Outdoors, indoors over reflective plane; anechoic half-space Free field method ≤5 1) 2) ≤5 1)2) 1.5 DIN EN ISO 3747 [6.7/145] 2000 Reverberant sound field Comparison method - - 1.5 DIN EN ISO 9614 T 1 [6.7/147] 1995 Reverberant sound field; outdoors over reflecting plane Intensity method ≤3 1) 2) ≤3 - DIN EN ISO 9614 T 2 [6.7/148] 1996 Reverberant sound field; outdoors over reflecting plane Intensity method ≤3 ≤3 1.5 DIN 45635 T 23 [6.7/159] 2003 Free field; free field over reflecting plane Enveloping surface method (free field method) ≤3 ≤3 2 1) Refer to [6.7/138]. 2) Limited frequency range. 6.7 Noise Behaviour 427 Guidelines for the application of basic standards for the determination of the sound power of noise sources are contained in DIN EN ISO 3740 [6.7/138]. An overview is provided on the correlation of measurement methods to • accuracy grades • test environment • character of the noise • volume of the sound source • limitation of external noise • measuring devices • measurement uncertainties Information on accuracy grades is contained in [6.7/130 T1], [6.7/134], and [6.7/138]. Measurement methods of the accuracy grade 3 (survey grade) are explained in [6.7/144], [6.7/145], [6.7/147], and [6.7/148]. The precision measurement methods are only rarely used for trans-missions. DIN EN ISO 4871 [6.7/146] makes statements on the “Declaration and Verification of Noise Emission Values”. The following are used specifically for transmissions: • DIN 45635-23 [6.7/159] • DIN 45635 Part 8 [6.7/131] • ISO 8579-1 [6.7/136] • ISO 8579-2 [6.7/137] Since the standards are subject to revision, it is necessary for users to always find out about the latest versions. 6.7.4 Sound Power Level of Gear Drives 6.7.4.1 Expected Values According to VDI 2159 [6.7/121] The expected value of the acoustic emission is the arithmetic average of a large number of meas- urements of the sound power level of gear drives that are structurally identical to the last detail, manufactured with normal industry-standard but randomly varying degrees of accuracy and operated under the same operating conditions. It is not exceeded by 50% of the gear drives. At the present time, there is no standardised documentation for predicting the expected noise emissions based on the noise generation mechanism. Empirical relationships for individual noise-relevant parameters can be found in the literature. Based on experience, but also due to the physical noise generation mechanisms, the various gear types also exhibit different sound emissions. However, with most types, the processes can be reduced to the same or similar qualitative noise-related parameters. A document for the prediction and classification of sound power levels with a relatively high degree of generalisation is the Guideline VDI 2159, ed. 7/85 [6.7/121] (refer to [6.7/102]). However, it originated from the production period from 1977 to 1981 on the basis of measurements on approximately 150 transmissions for the various gear types, taking into account the power transmitted from the transmission under the following operating conditions: nenn 0.3 MM≥ ; nenn0.9nn≥ The acoustic evaluation of a transmission noise is carried out with the statistical data of a 50% and an 80% line. 428 6 Load Capacity and Running Performance: External and Internal Gearings The regression relationships for the sound power level are (VDI 2159): WA 1 2lg dB(A) LK K P=+⋅ (6.7/45) with LWA in dB(A), P in kW, K1 in dB(A); K1, K2 according to Table 6.7/4a. Table 6.7/4a Regression Coefficients for the Sound Power Level Regression coefficient K1 K2 50% line (expected value) 80% line (80% of the values according to Equation (6.5/45) are below) Cylindrical gear transmission (industrial gear transmission) 71.1 77.1 12.3 Cylindrical gear transmission (high-speed gear transmission) 83.9 85.6 6.4 Planetary gear transmission 85.8 87.7 4.4 Table 6.7/4b Boundary Conditions for the Regression Relationships in VDI 2159, ed. 7.85 [6.7/121] Cylindrical gear transmission Planetary gear transmission Industrial gear transmission High-speed gear transmission Power in kW 0.7 to 2400 380 to 42000 6 to 12500 Input rotational speed in min-1 1000 to 5000 1000 to 12700 350 to 16500 Pitch circle velocity in ms-1 1 to 20 > 35 Stage number 1 to 3 1 to 2 1 to 2 Gear tooth quality, DIN 3962 5 to 8 3 to 5 Number of test transmissions, on which the statistical evaluation is based 29 21 18 The confidence coefficient, based on the studied quantities and other boundary conditions as shown in Table 6.7/4b, is given as 90%. 6.7.4.2 Comparisons with Other Literature Sources While [6.7/121] refers primarily to the volume of production of the 1980s, [6.7/102] is based on over 2000 measurements on transmissions in the period 1967–1974. In Figure 6.7/11, the expected values of the sound power level of cylindrical gear transmissions of universal application from [6.7/121] and [6.7/102] are compared according to [6.7/79]. The values for bevel cylindrical gear transmissions (refer to [6.7/1], [6.7/42], [6.7/65]) are listed for comparison. 6.7 Noise Behaviour 429 While in [6.7/121] and [6.7/102] a comparable quantity is described by the sound power level, typical maximum sound pressure levels for di fferent gear types have been summarised in [6.7/128], in which more statements were made on the influence of variable input rotational speeds and loads. The particular value of prediction equations in [6.7/102], which were used until the end of the 1980s for establishing limits and guaranteed values, lies in the fact that, in addition to power, even more noise-related parameters that are easy to determine have been taken into account: • nominal power • effective power • working pitch velocity • stage number In [6.7/79] it is indicated that, through the application of new knowledge and technological opportunities, the sound emission of modern power transmissions can also be well below VDI 2159, ed. 7/85 [6.7/121]. As a rough estimate, the following can be determined for the ranking of the sound emission depending on the gear types with the same ratio, ordered from the quietest to the loudest: Up to approx. 50 kW: Worm and belt transmission Bevel and spur gear stages Planetary gear stages without load balancing Disk and roller chain transmission Above approx. 50 kW: Planetary gear stages, high speed Spur gear stages Planetary gear stages, standard speed Bevel gear stages For predicting the sound power of lightweight planetary gear transmissions, refer to [6.7/59]. 6.7.4.3 Influence of Noise-Related Parameters With the development and utilisation of machine acoustics and the study of the systematic corre-lations of the noise generation mechanism, it is envisaged that mathematical estimates of the sound power level are increasingly possible in advance. Thereby, noise-related influencing factors, such as the structural design of the housing and the location of the transmission resonant frequencies, must be taken into account. Figure 6.7/11 Expected values (50% line) of the sound power level L WA [6.7/102] and VDI 2159, ed. 7/85 [6.7/121] 430 6 Load Capacity and Running Performance: External and Internal Gearings It will be useful here, despite the use of computer facilities, to refer to real reference transmissions. Because of this, the relative changes of the noise-relevant parameters are used in the first instance, because, for the practical use of the projection methods, it will always be problematic to calculate the absolute effects of the parameters for each studied transmission with high accuracy. Approximations are always under consideration and take into account the experiences of the influence of individual variables, such as for modern industrial gears, in general use particularly for cylindrical gear transmissions. Influence of the rotational speed 1): For single transmissions with constant torque: dB lg ) 33 ... 14 ( = BWnnLΔ with a reference rotational speed (rotational speed on the basis of which the change takes place) in the range nB = 1000 to 1500 min-1 From statistical considerations for a number of transmissions: dB lg 11 = BWnnL ⋅ Δ (see above for reference rotational speed nB) In the calculated influence of the rotational speed, the noise-relevant parameters power and velocity are included in the working pitch, to which it should be usefully traced back. Although two wheel pairings from different centre distances or a different ratio can be operated at the same rotational speed, they do not have the same circumferential speed. Conversely, pairings with different rotational speeds can have the same circumferential speeds. Influence of the circumferential speed: w W wB w W wB = 5.5 lg dB for = const. = ( 14 ... 33 ) lg dB for = const.LP LMυ υ υ υΔ Δ with a reference speed (speed on the basis of which the change takes place) of wBυ= 4 m/s. Influence of the torque2): For single transmissions at a constant rotational speed (1500 min-1), the following is derived from [6.7/121] for cylindrical gear transmissions: nom W B = ( 12 ... 21 ) lg dBMLMΔ with a reference torque (speed on the basis of which the change takes place) in the range of MB = 1 kNm. For a variety of gears from serial measurements: nom W B = 6.6 lg dBMLMΔ (see above for reference torque MB) 1) According to data in [6.7/53], a doubling of the rotational speed results in a level increase of 6 dB. Refer to [6.7/5] or [6.7/154] with other relations. 2) According to data in [6.7/53], a doubling of the torque results in a level increase of 3 dB. 6.7 Noise Behaviour 431 According to other sources in the literature, the partial load influence, which is of particular interest and which is related to the nominal torque, varies even more [6.7/37]. In some cases, the sound power level remains approximately constant between the nominal load and 0.5 the nominal load and then decreases with approximately W nom = 14 lg dBMLMΔ From [6.7/117], the following can be derived for a cylindrical gear transmission: nom w W B = 8.3 lg dB for , = constant ,MLn ȣMΔ (see above for reference torque MB) Compared to a nominal load, under partial load an increase in the level values can also occur instead of a reduction. This occurs when the flank modifications designed for a nominal load lead to a reduction of the loaded flank portions (decreasing transverse contact ratio and overlap ratio) in the case of a partial load. For example, it can be determined for the flank modifications of bevel gears (elliptical contact pattern at a nominal load) in models according to the previous design rules. A significant increase in noise can also be caused by flank separations, which already result due to small tooth system deviations under partial load. Influence of the gear ratio : This is a derived influence and should be attributed to the influences of the circumferential speed and the transmitted torque and possibly even to the reference diameter of the pinion. 6.7.4.4 Limiting Values of the Sound Emission From the expected values, the confidence intervals and corresponding standard deviations of which are known, maximum noise emission values or limiting values can be determined, depending on the informational value of the result. They can be used in the sense of guaranteed values, for example, as a noise emission value agreed upon between gear manufacturer s and operators, or as an infor- mation value. A guaranteed sound power level must be formed from the measured value according to DIN EN ISO 4871 [6.7/146] with respect to the production variation and the accuracy of the measurement method. Determination and verification procedures must therefore exist. The verify-cation procedure must be based on both a clearly defined measuring method and a concept for handling the variations of the measurement results, including the production-related variations of the sound emission values. Verification procedures for the given sound emission level referred to as a noise marking value are defined by the series of standards DIN EN 2 7574 T 1 to 4 [6.7/133] and relate to machines in a general sense. In the case of gear drives, a transfer to the ratios is possible. The noise marking value L c [in dB(A)] is defined by ()cW A t M t = + 1.5 ı + ı - ı LL k (6.7/46) with LWA Sound power level 22 tP Rin dB (A ı = ıı )+ (6.7/47) Pσ Production standard deviation in dB(A) Rσ Standard deviation of reproducibility of the measurement method used in dB(A) 432 6 Load Capacity and Running Performance: External and Internal Gearings Mσ Reference standard deviation in dB(A) k Acceptance constant; 95% for probability of acceptance: 1.514 1.645 /kn=− n Sample size In modern transmissions and technologies, the production standard deviations, which must always be redefined when a noise marking value has to be verified, lie in the range P (1.5 to 2.5) dB(A) σ= The values given in Table 6.7/3 apply to the comparative standard deviations. The reference standard deviations can be determined on the basis of previous measurements for spur, bevel and combined spur and bevel gears: M (3 to 4) dB(A)= σ For the transmissions described according to [6.7/102], the following can be defined as a limiting value based on a probability of 95%: (limiting value) = (expected value) + 4.8 dB(A) (6.7/48) In their catalogues (e.g., [6.7/67]), various gear manufacturers specify sound power levels and sound pressure levels at a distance of 1 m from the transmission. 6.7.5 Examples of Sound Power Levels of Various Machines To provide an orientation for the classification of the sound emission of transmissions compared to other driving equipment or special types of machines, sound power levels of gear motors and motors are exemplified ( internal combustion engines [6.7/41]). Gear motors Figure 6.7/12 gives an example of four-pole three-phase cylindrical gear motors, with benchmarks according to VDE 0530 [6.7/111] and measured values according to [6.7/24]. Figure 6.7/12 Scatter band ( a) of the A-weighted sound pressure level in the case of four-pole hree- phase cylindrical gear motors and benchmarks ( b) [6.7/24] [6.7/111] Ventilators For radial and axial ventilators, measurement values of the sound power level are compiled in [6.7/126] in order to describe the state of the art [6.7/74]. Electrical machines The first internationally agreed limit values are contained in the Recommendation IEC34-9 [6.7/120]. They are given as a function of rotational speed and load ranges and apply only to idling. 6.7 Noise Behaviour 433 In [6.7/17], measurement values from different manufacturers (dispersion range) for two-, four- and six-pole low-voltage induction motors with the degree of protection IP44 are compared with the limiting values according to IEC 34-9 [6.7/120] (stage lines) (Figure 6.7/13). DIN EN 60034-9 [6.7/154] describes the current state. In addition to the highest A-weighted sound power levels when idling, which have lower values compared to [6.7/120] and the previous standards (according to the trend of the measurement values), data can be provided on the increase in the sound power level from id-ling to the rated power. Figure 6.7/13 A-weighted sound power level of two-, four- and six-pole low-voltage in duction motors with a degree of protection of IP 44 (from [6.7/17]) 6.7.6 Structural Measures for Noise Reduction 6.7.6.1 Regulations From the perspective of a low noise level required for people, laws, regulations and guidelines have been developed that also extend to the sound emitted by machines and equipment, in-cluding transmissions. Directive 98/37/EC [6.7/114] refers to a primary noise reduction at the source in particular. With Directive 2003/10/EC [6.7/116], lower immission limit values are determined compared to the superseded Directive 86/188/EEC [6.7/112] (for example, instead of a noise exposure value of 85 dB(A), the value now stands at 80 dB(A)). Further basic regulations: • Ordinance on workplaces (Workplaces Ordinance) of 12 August 2004 • BG Regulation B3 (UVV) “Noise” of 1 January 1990, as amended in 1/2005 • 32nd Ordinance on the Implementation of the Federal Immission Control Act of 1 September 2002, as amended on 6.3.2007 • Sixth General Administrative Regulation to the Federal Immission Control Act (Technical instructions on protection against noise – TA Noise) of 26 August 1998 • Act on the Reorganisation of the Safety of Technical Work Equipment and Consumer Products of 6 January 2004, as amended on 14 July 2004 • Directive 98/37/EC of 22 June 1998 [6.7/114] • Directive 2000/14/EC of 08 May 2000 [6.7/115] • Directive 2003/10/EC of 06 February 2003 [6.7/116] • Directive 2006/42/EC of 17 May 2006 [6.7/117] The state of the art must be observed in the activities for noise reduction, which is recorded, for example, in VDI Guidelines (e.g., VDI 3720 [6.7/123], [6.7/124]) or in the Report of the Federal Institute for Occupational Safety and Health (e.g., [6.7/25], [6.7/16]). Predetermined limit values or guidelines of sound emission must be observed. 434 6 Load Capacity and Running Performance: External and Internal Gearings According to [6.7/110]: “The noise-reduction measures must also take into account rules in safety technology, ergonomics and environmental protection and must be adequate and technically and economically feasible.” A targeted low noise emission can be achieved through consistent evaluation, implementation of laws and knowledge of the machine acoustics. 6.7.6.2 Basic Principles for Noise Reduction The first basic requirement of noise reduction is, of course, to develop a quiet transmission through primary measures. In this sense, the basic principles listed below and the possibilities to influence described in Sections 6.7.6.3 to 6.7.6.5 must be considered [6.7/80]: Principle 1 Systematic approach based on the mechanical noise generation mechanism Principle 2 Primary noise reduction measures generally befor e secondary measures Principle 3 Noise reduction as close as possible to the source Principle 4 Separation of driving elements from those with a radiating function Principle 5 Acoustic optimisation of the time profile (or frequency curve) of the dynamic opera- ting forces Principle 6 Conduct the dynamic operating forces to the baseplate along the shortest path Principle 7 Machine-acoustic design of the transmission walls and the individual components in the order of their share of the sound power level of the transmission Principle 8 Useful combination of the influencing variables of mechanical noise generation Principle 9 Use of statistically significant experiences and the latest scientific knowledge For all noise reduction measures it is crucial to achieve the greatest effect with the least reasonable effort. The main task resulting from Principle 7 is also related to this; for example, in connection with the influence of the noise excitation, the transmission and emission properties are to be designed so that the component sound power level of the individual gear walls are approximately the same size. If one share is significantly higher than the others, the total sound power level will essentially be determined by this transmission wall. In the practical case, this often applies, for example, to the cover surface of the transmission or also to the large bearing cover, which in modern transmission designs have a significant share of the total surface of the gear [6.7/79]. These basic conditions are illustrated in the following numerical examples. Case 1 L1 = 90 dB L 2 = L3 = L4 = L5 = L6 = 80 dB L ges = 91.8 dB Case 2 Reduction of all partial levels by 4 dB compared to Case 1 L 1 = 86 dB L 2 = L3 = L4 = L5 = L6 = 76 dB L ges = 87.8 dB Case 3 Reduction of only the partial level L1 by 10 dB compared to Case 1 L 1 = L2 = L3 = L4 = L5 = L6 = 80 dB L ges = 87.8 dB The amount of effort required for all six surfaces in Case 2 is too high in relation to the effect. There are hardly any universally valid “recipes” [6.7/82] that are suitable for any particular case. There will be increasing dependence on numerical calculations and comparison with reality. 6.7 Noise Behaviour 435 6.7.6.3 Influence of the Excitation of the Mesh with Cylindrical Gearing In the case of different excitation mechanisms, such as in conical tooth gearing, cylindrical gearing or worm gear gearing, various special noise reduction measures apply in addition to general measures (refer to [6.7/95]). In principle, in the influencing of the main causes of excitation, three measures are of particular importance (Section 6.7.2.2): A Targeted influencing of the toothing geometry through the following key variables: 1 Helix angle 2 Module, gearing width, number of teeth 3 Contact ratio 4 Tooth flank modification 5 Special gearing B Influencing the tooth system deviations through the improvement of the gear tooth quality with the same or modified manufacturing method: 1 Tooth system deviations 2 Manufacturing method C Design strategy of the entire drive system The internal excitation mechanisms are discussed extensively below (Complexes A and B), and the importance of the outer mechanisms (Complex C) is noted only briefly. The mentioned information on the possibilities to influence is provided on the basis of the state of knowledge of the art. The consideration of the individual sizes is shown in a schema, according to which information is given on principle and purpose, and examples with information on effects are shown. A1 Helix angle Principle: - Transition from spur to helical gearing - Increase of the helix angle up to max. 30 is useful [6.7/54], [6.7/70], [6.7/109] - Influencing the overlap ratio İȕ [6.7/48] Purpose: - With helical gearing, there are much lower tooth stiffness variations and a more uniform load take-over than with spur gearing [6.7/106]. - Reduction of the mesh impact - Due to the form of the tooth stiffness deviation, higher-frequency components of the excitation force, among others, can be reduced. - Even with a small helix angle, as well as with İȕ < 1, a significant reduction in excitation can be seen [6.7/54]. Results: Examples: - Figures 6.7/14, 6.7/15 and 6.7/16 - Reduction of sound emission in the transition from straight to helical gearing according to [6.7/78]: 0 to 10 dB according to [6.7/53]: 2 to 6 dB according to [6.7/55]: 3 to 10 dB Notes: Through large gearing deviations, poor gear tooth quality, partly as a result of a total helix deviation, or through misalignment of the shafts, the advantages of helical gears are quickly negated. This applies primarily to low loads (example: Figure 6.7/16). 436 6 Load Capacity and Running Performance: External and Internal Gearings Figure 6.7/14 Transition from spur to helical gearing [6.7/78] Speed range n = (600 to 1000) min -1 Ground toothing b = 40 mm; 0Hȕ≈f Figure 6.7/15 Influence of the helix angle [6.7/62] Figure 6.7/16 Effect of helix slope de- viation in helical gearing [6.7/78] Speed range n = (600 to 1000) min -1 Ground toothing B = 40 mm; ȕ = 24.301 6.7 Noise Behaviour 437 A2 Module, gearing width, number of teeth Principle: Establishing or changing the gearing geometry taking into account the load carrying capacity Purpose: Influencing of the size and form of the tooth stiffness variation Changes are primarily effected through - specific tooth normal force - overlap ratio - working pitch velocity - exciting frequency Results: Examples: - Figure 6.7/17 - Effects result primarily from the joint influence (of m, b, z ) on excitation and resonant frequency (refer to Section A3). - The module has a small influence on the tooth stiffness curve. Notes: With a decreasing specific tooth load, the excitation can move into the foreground through tooth system deviations. Literature: [6.7/56], [6.7/62], [6.7/78]. ȕ = 0 b = 40 mm m = 4 mm ȕ = 15.841 b = 40 mm m = 4 mm ȕ = 0 b = 9.2 mm m = 4 mm Figure 6.7/17 Influence of tooth width on the excitation [6.7/78] A3 Contact ratio Principle: - According to [6.7/2], a constant total length of the line of contact of all meshed teeth must be ensured. - In addition to an integer overlap ratio, an integer total contact ratio is of significant importance. - Decisive influence of the transverse contact ratio among others through special gearing, for example, extra-depth gearing (refer to Section A5). Purpose: Minimising the gearing stiffness fluctuations and thus the tooth deformation fluctuations and weakening the mesh impact through - low tooth stiffness c min at the beginning of the meshing - minimising the load-induced pitch deviations above a high total contact ratio - primarily due to integer overlap ratio If only the change in stiffness needs to be minimised, then the total contact ratio İ Ȗ should be an integer [6.7/88] (if possible also İĮ and İȕ). Results: Examples: - In the literature, different significance is attached to an integer overlap ratio and an integer total contact ratio (e.g., [6.7/48] and [6.7/70] or [6.7/30]). - Modify the overlap ratio by changing the helix angle. - Compared to İ ȕ = 1, no major improvements can be achieved with İȕ = 2, 3, 4 [6.7/32] because, for example, tooth system deviations come to the fore as causes of excitation. 1 2 3 438 6 Load Capacity and Running Performance: External and Internal Gearings - High sensitivity with integer overl ap ratio through flank variations. - Extra-depth gearing enables significant improvements (Figure 6.7/22 and 6.7/23). - Increasing İĮ values counteract the disadvantage of a profile slope variation [6.7/12]. - A relatively low excitation level can be expected even with helical gearing with 0 < İ ȕ < 1 and a high transverse contact ratio İĮ > 1.7 for high tooth loads [6.7/48]. - Select the highest possible transverse contact ratio. For an integer transverse contact ratio, a constant total length of the line of contact exists independently of İȕ [6.7/21]. - According to [6.7/2], [6.7/48], and [6.7/71], the dependence of the excitation of vibrations on the overlap ratio is generally independent of the size of the transverse contact ratio. - For an integer İȕ value, the first harmonic of the excitation force level is comprised mainly of low amplitudes [6.7/21]. - Figure 6.7/18 uses examples to show a comparison between sound power level values for gearing with an integer overlap ratio and a non-integer total contact ratio, as well as with a non-integer overlap ratio and an integer total contact ratio according to [6.7/70]. Figure 6.7/18 Sound power level at integerβεand Ȗε[6.7/70] A4 Tooth flank modification Figure 6.7/19 Principle diagrams of flank modifications Profile modification or profile crowning in the form of tip relief Ca at pinion and wheel or tip relief Ca and root relief Cf at the pinion only. Mostly applied are short and long involute-type modifications with tangential transition to the true involute Flank line modification in the form of end relief or lead crowning in different geometrical shapes; parabolic (or circular arc) traces with tangential transitions are mostly used General 3D modification (topography) combined profile and flank line correction with tangential transitions; manufactured with numerically controlled grinding machines Principle: 6.7 Noise Behaviour 439 Purpose: - Reduction of the parameter excitation caused by the tooth stiffness curve - Weakening of the load-related mesh impact - Reduction of the excitations caused by effective helix slope deviations through compensation of the deformations of the gear body and the shaft-bearing system Results: Examples: - Profile correction in spur gearing o According to [6.7/78], the sound level reduction is between 0 and 8 dB. o When operating near the reference conditions, a long involute modification is preferable from an acoustic point of view. o Figure 6.7/20 - Profile correction in helical gearing: In the case of high gear tooth quality, an increase in the sound power level can also occur for certain load and speed ranges through a profile modification that is constant across the flank line [6.7/78]. - Flank line crowning for spur gearing: Depending on the size and shape of the total helix deviation, both an increase and a decrease in the sound emission may be possible [6.7/56], [6.7/78]. - Flank line crownings and long end reliefs are beneficial in flank line angle deviations, but, according to [6.7/29], can have different effects. - The combination of profile and flank line crownings has a positive effect when the modification values are determined exactly [6.7/21]; this has not yet been sufficiently examined. - Correction in the direction of meshing In [6.7/70], relatively high noise reductions ( ǻLp = 7 to 10 dB) are shown (Figure 6.7/21). Figure 6.7/20 Effect of short and long tip relief according to [6.7/78] Speed range n = (600 to 1000) min -1 Ground toothing, b = 40 mm, ȕ = 0 Figure 6.7/21 Sound power level in case of unmodified gearings and gearings modified in mesh direction [6.7/70] 440 6 Load Capacity and Running Performance: External and Internal Gearings Remarks: - A gear quality better than seven according DIN 3962 is required for full effect of tooth modifications - The largest effect of the modification is achieved in noise reduction for the load level, for which the modification is designed. Often a lower amount of noise reduction is the result at loads which deviate clearly from the design load. Even an increase of noise can appear compared to unmodified gears. - The noise level is very sensitive in case of helical gearings with integer values of axial overlap ratio İȕ and lead crowning. If İȕ is non-integer, a noise reduction is achieved in the case of large helix slope deviations. - The largest noise reduction is achieved in low multiples of the mesh frequency (low order numbers). - Special modifications with periodic traces for compensating mesh excitation were investigated in projects of the research association “Forschungsvereinigung Antriebstechnik e.V.” [6.7/31, 6.7/98]. Those modifications are used in addition to other common modifications (for balancing the load distribution) without affecting the load distribution. The effect of those modifications is based on the removal of material on the peaks of the transmission error curve and the addition of material at the valleys of the transmission error curve. So the variation of the transmission error curve is compensated for by a given design load in a large scale. The effect of the periodic modification is not optimal for other loads. The amount of this modification is very low for optimizing gears with low transmission error variation. The necessary high level of accuracy in phase and amount is then a problem for reliable application. Additional literature: - Profile modifications [6.7/2], [6.7/48], [6.7/54], [6.7/71] - Flank line modifications [6.7/70], [6.7/48] - 3D modifications [6.7/21] A5 Special gearing Extra-depth gearing Principle: Geometry so that İ Į 2.0, often using a different pressure angle of 20 , for example 15 . Purpose: - Mesh impact is reduced through a more uniform distribution of load. - Tooth load surges in the case of mesh changeover can be substantially reduced. - Increase in the meshing time [6.7/87]. - Increase in the mesh damping [6.7/70]. Results: - Figure 6.7/22 according to [6.7/62] for extra-depth gearing with pressure angle 15 . (Examples) - Figure 6.7/23 according to [6.7/87] for extra-depth gearing with pressure angle 20 . - According to [6.7/62], [6.7/70] and [6.7/87], reduction of the sound emission over a wider speed range 2 to 6 dB. Remarks: - Low load dependence - Increase in sliding speed compared with standard gearing - Advantages especially at high rotational speeds - Special tools may be required Figure 6.7/22 Comparison of extra-depth gearing with standard gearing [6.7/62] 6.7 Noise Behaviour 441 Figure 6.7/23 Influence of load on standard and extra-depth gearing [6.7/87] B1 Tooth deviations The excitation from the tooth deviations represents a position excitement as a separate excitation [6.7/71]. Particularly extensive experimental studies were carried out in the 1960s (e.g., [6.7/55], [6.7/83]). In [6.7/54], an attempt is made to establish relations between the change in the sound emission and the gear tooth quality according to DIN 3962. Thereafter, in the range between the quality grades 4 and 8 in accordance with DIN 3962, the sound level would change by about 2 dB per quality grade. The main tooth system deviations of the noise excitation a) helix form deviations, such as described in [6.7/54], b) pitch deviations, including tooth-to-tooth pitch deviation, such as described in [6.7/62], [6.7/48], c) base circle variations, pressure angle deviations, such as described in [6.7/48], [6.7/54], d) total helix deviations, such as described in [6.7/78], [6.7/54], except position deviations, are attributable to the first- and second-order structural deviations according to DIN 4760 [6.7/129]. It is the macro-geometric deviations, and not the surface structures through different manufacturing methods or roughness, which are primarily decisive for the excitation [6.7/30]. The effect is largely dependent on the excitation of vibrations of the deviation-free gearing (e.g., whether an integer overlap ratio is present) and on the operating conditions, in what ratio the existing load is to the rated load (static load). At a low excitation force, a partial load has a disadvantageous effect on the influence of tooth system deviations at a low trace of stiffness; at a high excitation force from the trace of stiffness, tooth system deviations under partial load do not have a significant effect [6.7/48], [6.7/54]. Therefore - on the basis of a rated load - with a decreasing moment of load the sound power level of a transmission can remain approximately constant, fall in the presented sense or even increase. Toothing with corrected deviation exhibits a greater excitation force in the lower load region than toothing that is uncorrected and deviation-free [6.7/48]. Wittke [6.7/105] proposes a design concept for the design of the gear contact geometry, in which the deviation influences for flank line and profile modifications are taken into account in addition to the elastic deformations; at the same time, a shift of the centre position of the tolerance field is performed. The excitation behaviour of helical cylindrical gear pairings with flank modifications is described in [6.7/98] with respect to process-induced deviations. In the case of tooth flank modifications, the requirement for a high toothing accuracy inevitably results from the tooth modifications. It should not be coarser than a quality grade of 6 according to DIN 3962. In the case of multi-stage transmissions, Sattelberger [6.7/71] recommends providing the first (high-speed) stage with a higher gear tooth quality. 442 6 Load Capacity and Running Performance: External and Internal Gearings Tooth trace variations elicit greater changes in stiffness and manifest themselves as the mesh frequencies. Profile form variations can also affect the excitation amplitudes of the harmonics of the mesh frequency with a corresponding waviness. Pitch and base pitch deviations or concentricity variations randomly distributed on the circumference can be identified by side bands around the mesh frequencies [6.7/54]. In principle, the accuracy of toothing plays an important role in noise reduction. It is related, of course, to the manufacturing methods used. With the current high level of manufacturing, individual deviations will be significant only in the rarest of cases. B2 Manufacturing methods The finishing method most widely used is the grinding of the tooth flanks by various methods, which can achieve gear tooth qualities of 8 to 3 depending on effort. Evidence of this is set out, for example, in [6.7/61] and [6.7/128]. According to [6.7/54], surface treatments such as nitriding and phosphating have no influence on the excitation behaviour. However, shaving, honing and lapping are also used for finishing. In the case of hard-skiving or honing, so-called “ghost frequencies” [6.7/30] can occur, for example, but which have no or very little influence on the strength of the overall level. If the method is applied properly, significant reductions in noise level can be achieved through lapping, which extend not only to the mesh frequencies, but also to harmonics. An improvement in the surface structure can also be achieved through super finishing (or “isotropic finishing”), without which the profile structure is adversely affected. Statements identifying trends with details of examples to improve the noise behaviour are given, for example, in [6.7/85]. C Design strategy of the entire drive unit The search for such noise-reduction measures has been the focus of noise research for many years (e.g., [6.7/76]). A reduction in the sound power level results from an • advantageous assignment of the driving machine and transmission. A low rotational speed of the motor often enables the reduction of the number of stages of the transmission and possibly allows for the reduction of the gear unit size or, by changing to a different type series • for an appropriate in stallation [6.7/44]. Optimised drive systems can be smaller, lighter and cheaper at the same time! 6.7.6.4 Influencing the Structure-Borne Sound Excitation at the Bearing Position Stiffness of the bearing environment: The immediate environment of the bearing position in the transmission housing is of great impor-tance. Its elasticity is a decisive factor in the vibration system, which is decisive for the radial and axial dynamic bearing forces. Thereby the input admittance of the gear housing transmission behaviour can be influenced and an effective noise reduction ca n be achieved, which extends over a large frequency range. Effective basic possibilities are shown schematically in Figure 6.7/24: Figure 6.7/24 Effective basic possibilities for the bearing bracket design 6.7 Noise Behaviour 443 • Separation between the force-carrying elements and the radiating surfaces (housing): o A reduction of 10 to 20 dB in the sound power is achieved through an elastically suspended transmission housing [6.7/103]. o A further advantage is the low material usage, since the housing is made of thin sheet metal. o Problems: - shaft seals - use in mining o Absorption of axial forces through a wheel flange. Reduction of the sound power level, as an example, by approximately 2 to 3 dB. • Elastic mounted transmission shafts [6.7/104]: o Special metal-elastic spring elements are used, which support the shafts and bearings relative to the housing. They produce a bearing stiffness that is significantly lower compared to a conventional bearing (e.g., 1/10). o Reduction of the sound power of 6 up to 12 dB(A) has been demonstrated experimentally. o Another advantage: reduction of structure-borne sound input into the baseplate. o Problems: - Special requirements for the production of the elastic rings - Limiting the static load capacity • Reduction of the bearing bracket admittance through material accumulation and advantageous conduction of the operating forces up to the housing walls and their excitation points, as well as into the baseplate along the shortest path [6.7/8], [6.7/81], [6.7/163]: o Reductions in sound levels of up to 10 dB(A) are possible. o Figure 6.7/25 shows the material accumulation at the bearing positions corresponding to the pressure distribution ratios between the bearing and housing, which are highly dependent on the stiffness of the bearing and the bearing clearance or pre-load. o Advantage: Increase in the static and dynami c stiffness of the transmission housing. o Design in such a way that excitations through radial and axial forces will be taken into account. Figure 6.7/25 Advantageous bearing bracket design through material accumulation and conduction of operating forces Damping measures : A large number of studies on the influence of the damping of the gear, shaft, bearing and housing vibrations and of the dynamic bearing forces have already been carried out, among others, in [6.7/4], [6.7/81], [6.7/84], [6.7/123], and [6.7/125]. The measures do not deliver the expected effect in general. They are also difficult to control because of the dependence of the damping on the frequency and amplitude. An advantage is that their effect has a broad bandwidth. Plain bearings 444 6 Load Capacity and Running Performance: External and Internal Gearings generally have better damping properties than do rolling element bearings. A change in the external damping is possible via the friction damping of coupling points. Surface coating is not carried out in transmissions with high power density at the bearing positions. Figure 6.7/26 Influence of the bearing type on the Figure 6.7/27 Influence of the rolling element bearing type on the transmission noise [6.7/56] transmission noise [6.7/109] Selection of the bearing type : The proportion of the actual bearing noise is low in the entire gear noise in transmissions with a high power-to-weight ratio. Worm gear transmissions and defective bearings may present exceptions. In quiet transmissions, high-quality rolling element bearings must be used. Information on the stiffness and damping values of different types of rolling element bearings is contained in [6.7/64]. The stiffness of rolling element and plain bearings is of the same magnitude. The structure-borne sound transmission behaviour of rolling element bearings is dependent on frequency and direction [6.7/64]; only at low frequencies are the vibrations transmitted to the housing in the feed direction; in the other frequency ranges, radial or axial components dominate. As a result of the greater damping of the oil gap, a plain bearing has a level-reducing effect, espe-cially at higher frequencies, for example, above 2 kHz according to [6.7/109]. The oil film provides a structure-borne sound bridge [6.7/15]. For a transmission with spur gearing, Figure 6.7/26 demonstrates the sequential application of plain bearings before cylinder roller bearings and deep groove ball bearings. Exciting frequencies can be located both in the exciting frequency range of gears and far above it. The information contained in the literature on the selection of the bearing type in rolling element bearings varies and depends on the special operating conditions. As a practical example, the appli-cation in a transmission is shown in [6.7/109] (Figure 6.7/27). 6.7.6.5 Design of the Transmission Housing A Reference transmission Since it is difficult and uncertain to determine the absolute structure vibration transmission level or structure transfer measurement in new constructions, even when using modern methods of calcu- 6.7 Noise Behaviour 445 lation, it is often advantageous to use a comparison or reference gear, which can be the starting point for much simpler and more accurate relative observations. For this reference transmission, the most important machine-acoustic parameters must be known (refer to Section 6.7.2). The housing that must be constructed or changed should not significantly differ from the outer design of the reference transmission. It should be noted that there are differences between a housing that is empty and a housing with inner parts assembled with respect to the structure vibration transmission level, resonant frequencies and the normal mode of vibration [6.7/39], [6.7/75]. B Basic change in the design parameters Through their change, the parameters used for the description of the machine-acoustic properties of the transmission housing wall thickness s compound coating m’ surface S damping (material, joint) bending stiffness Bƍ influence the change in the structure transfer measurement [6.7/33b]. Using the laws of similarity [6.7/17], [6.7/86], [6.7/57], the basic ratios for plate elements of housing structures with force exci-tation are compiled in Table 6.7/5. The choice of measures should be based on the pre-existing values (stiffness, mass loading, damping) [6.7/75]. The operating conditions must be taken into account in the case of structure specifications or changes. In the case of a high resonant frequency density of the structure and/or widely varying operating conditions, it is useful to implement statisti-cally broadband effective design measures [6.7/39]; in the case of narrow-band excitation, this can be disadvantageous. As an example, to illustrate the level changes, the values when the size of the respective parameters is doubled are shown in Table 6.7/6. It is important for the notion that, for example, a doubling of the bending stiffness Bƍ can be achie- ved with a quantity of material that corresponds to a fraction of the increase in the wall thickness. A weight optimisation often forms part of the task. The shifts in the first plate resonant frequencies can be identified when the values of the design parameters from Table 6.7/7 are doubled. Table 6.7/5 Changes in the Structure Transfer Measurement through the Influence of Individual Design Parameters Parameter Quasi-static range 1) Natural frequency range f f11 < fM f < fM < f11 fM f < f11 f11 < f < fM f fM > f11 fM < f11 < f s \* 60 lgs- s \* 30 lgs- s S \* 40 lgS+ S \* 30 lgS+ S \* 10 lgS+ S 0 Bƍ \* 20 lg B- B′ ′ \* 5l g B- B′ ′ mƍ 0 \* 15 lg m- m′ ′ 1) Plate elements of transmission housings behave like springs in this range. The parameter of the sequence variant is marked with \*. Level reductions are negative data. 446 6 Load Capacity and Running Performance: External and Internal Gearings Table 6.7/6 Level Changes When the Size of Construction Parameters Is Doubled Parameter Quasi-static range Natural frequency range f f11 < fM f < fM < f11 fM f < f11 f11 < f < fM f fM > f11 fM < f11 < f s -18 dB -9 dB S +12 dB +9 dB +3 dB 0 dB Bƍ -6 dB -1.5 dB mƍ 0 dB -4.5 dB Table 6.7/7 Shift in Frequency When the Size of Construction Parameters Is Doubled Parameter f11\* fg\* fM\* s 2 f11 0.5f g constant S 1.4 f11 0.7 fg constant Bƍ 0.7 f11 1.4 fg constant mƍ 0.5 f11 constant 0.7 fM C Design influence of substructures of the housing With the aim of reducing the bending vibrations excited through transverse forces or bending moments, ribs, beading, bends, buckles and double-walled structures can be used to increase the bending stiffness. Thus, the low-frequency range (<1 kHz) is primarily influenced, while the medium- and high-frequency ranges are mainly modified through mass loading [6.7/82]. With a favourable combination of stiffness and mass influence, an effective binding of the kinetic and potential energy of the mode shape can be achieved [6.7/75]. For the superficial implementation of the basic principles 3 and 7 (refer to Section 6.7.6.2), the principles of the relationships are represented in the practical examples set forth below. Ɣ Use of ribs for housing walls : - In general, ribs are mounted to support the bearing positions. - The direction of the dynamic bearing forces must be considered, which is dependent on the bearing clearance and the frequency range [6.7/64]. Figures 6.7/25 and 6.7/28 make it clear that the forces are conducted as quickly as possible into the baseplate. - For a rib to work effectively to increase the bending stiffness, it is important that the rib runs from plate edge to plate edge. The choice of rib height is more important than the rib width. Ribs that are heavily stepped have impaired effectiveness. - Ribs should closely conduct the forces in the sense of a skeleton construction. - The number of resonant frequencies (normal modes of vibration) can also be reduced through the arrangement of the ribs (e.g., cross ribs). - Structures with high flexural elasticity could also be used to influence the medium- and high- frequency range, for example, through so-called “negative ribs” (grooved surface) according to [6.7/82]. - According to [6.7/81] and [6.7/74], the static and dynamic bending stiffness must be distinguished in the ribbing of the housing walls. The static bending stiffness is observed up to the coupling frequency f k. According to [6.7/81], the dynamic frequency-dependent bending stiffness describes the stiffness of the area between the ribs, which has an advantageous effect 6.7 Noise Behaviour 447 with high flexural elasticity (decoupled) and acoustically, if the distance between the ribs (coupling length l k) is greater than half the bending wavelength (critical frequency) of the observed plate (wall of the housing) [6.7/81], [6.7/74]. - A reduction of the structure vibration transmission level through ribs or an increase in the wall thickness can be partially offset by the increase in the radiation factor [6.7/27]. - The effect of ribs can be identified most reliably by using the finite element method. Further reading: [6.7/75], [6.7/100] Ɣ Material accumulation at the bearing positions : - The material accumulation at the bearing position illustrated in Figure 6.7/25 is used to reduce the input admittance. - The concentration of mass at the position of excitation has the disadvantage that the first resonant frequency is shifted to even lower frequencies. Ɣ Design of the cover surface: In most conventionally designed transmissions, the cover surface represents the housing wall with the highest component sound power level [6.7/79]. Figure 6.7/28 shows a possible change in the macro form of the upper housing portion of the transmission. Ɣ Design of the bearing covers In modern constructions, the bearing covers have a relatively high proportion of the total surface area of the radiating surface of the transmission and, thus, their contribution to the total sound power level should not be underestimated. Furthermore, there can be an excessively high workplace sound level with tonal components due to a disadvantageous radiation ratio as a result of resonances. The structure vibration transmission level of a bearing cover can be significantly influenced, for example, - by increasing the wall thickness. Using an example of an industrial gear transmission, Figure 6.7/29 demonstrates the clearly measured reduction of the structure-borne sound level through the replacement of a “con-ventional” bearing cover with a small wall thickness with a cover with an increased wall thickness. - by positioning ribs in the centre area of the bearing cover. • Annular ribs are an interesting variant that save mass when two or three annular ribs are mounted at a certain distance from the centre point [6.7/79], [6.7/164]. • As an example, Figure 6.7/30 shows the change in the structure-borne sound level at the first and second normal mode of vibration of the bearing cover using measurements and calculations. - by mounting thin plates in order to exploit the principle of structure-borne sound absorption through the use of thin air layers (e.g., [6.7/4], [6.7/84], [6.7/165]). This possibility is particularly suitable for subsequent noise-reduction measures. The distance between the reference surface and the plate should not exceed a few tenths of a millimetre. It is advantageous to use sheet steel as material for the thin plate. Figure 6.7/31 shows a construction example for the advantageous influencing of the radiation behaviour. Figure 6.7/28 Example of a stepped upper housing portion of a transmission 448 6 Load Capacity and Running Performance: External and Internal Gearings Figure 6.7/29 Impact of a reinforced bearing cover on the structure-borne sound level of the cover Figure 6.7/30 Influence of annular ribs on covers [6.7/79] Figure 6.7/31 Construction example for the mounting of a plate to the outer wall of the cover 6.7 Noise Behaviour 449 D Influence of the material The replacement of the housing material is quite often the subject of publications, for example, in [6.7/28], [6.7/38], [6.7/108], [6.7/109]. The descriptions of the impacts on noise behaviour, con-sidered theoretically or experimentally determined, are at times contradictory. The causes lie in the often not comparable conditions. Changes in the stiffness, mass and damping are also linked with the variation of the material, due to the influence of the elasticity of the material, the density and Poisson’s ratio [6.7/75]. In addition, different materials require different structural design rules. Essentially, two complexes can be distinguished for the replacement: 1) The mutual replacement of conventional materials, such as aluminium with steel or cast iron, and vice versa (e.g., [6.7/38]). 2) The replacement of conventional materials such as steel, cast iron and aluminium with polymer concrete or plastics. To achieve low vibration amplitudes of the housing walls, a high stiffness and a high material damping are decisive for the influence of the housing material. It is described by the longitudinal wave velocity and Poisson’s ratio. When making comparisons, however, the construction conditions that differ the most must be observed. If steel and cast iron housings are compared, different wall thicknesses or bending stiffnesses usually form the basis of the executed constructions. This will be explained in the following examples of force-excited structures: • When changing the material at a constant bending stiffness ( Bƍ = constant), the change in the structure vibration transmission level (+ increase, - decrease) is calculated as shown in Table 6.7/8. • On the other hand, the geometric dimensions of wall thickness and radiating surface remain constant, so the change in the structure transfer measurement is calculated according to Table 6.7/9. In [6.7/56], the cited results for the difference between cast iron and steel gears show differences that are within the measuring accuracy range (Figure 6.7/32). As a housing material that until now has been unusual as a replacement for steel and cast iron housings, possible applications of polymer concrete have been studied in various ways [6.7/26], [6.7/90]. It meets the requirements for a low sound emission due to high stiffness and high material damping. In [6.7/89], results indicated that the same weight of a housing made of cast iron with lamellar graphite can be achieved with a transmission housing made of polymer concrete with the same dynamic stiffness. For a transmission with helical gearing, a lower sound power of 5 to 10 dB(A) was observed (Figure 6.7/33) in a concrete housing compared to a cast iron housing. In [6.7/108], a further use of concrete is described, in which the cavities of the housing, which has been constructed using the honeycomb and skeleton construction method and welded, have been filled with polymer concrete. Sound pressure level reductions of up to 10 dB have occurred as a result. The use of plastics, such as polyamide [6.7/109], tends to produce a level increase compared to steel and cast iron. Advantages will result for housing parts outside of the force flow [6.7/28]. The use of concrete in power transmission construction will only be possible if strength properties comparable to cast iron or steel are achieved. The acoustic basic principles are covered in [6.7/28]. E Influence of damping Structure-borne sound absorption measures should then be used when only a small construction dam- ping is present. In addition, the manifestation of a damping measure must be examined based on the structural properties compared to the exciting frequencies [6.7/74]. Due to the existing groove damping, it does not make sense to increase the damping by varying the material in transmissions [6.7/75]. At high tightening torques, the separating joint does not exert an influence on the structural damping [6.7/75]. Different welded joints have different damping ratios depending on the surfaces of 450 6 Load Capacity and Running Performance: External and Internal Gearings the groove [6.7/75]. Damping of the joint exhibits much larger values than does material damping [6.7/38], [6.7/75]: cast iron 0.0126, steel 0.007 compared with 0.001 and 0.0001. A transmission equipped with inner parts has a 0.7% to 0.9% higher damping value compared with an empty housing [6.7/75]. Additional ways of increasing the damping of bending vibrations are, for example: • Provision of additional joining and friction surfaces applied over a large area [6.7/82]. • Damping through an air layer (Figure 6.7/31). • Use of simple damping coverings (especially for thin sheet steel [6.7/39], [6.7/74] or of the better solution in terms of material use with constrained coverings and a top layer of sheet steel or metal foil [6.7/74], [6.7/10]); this is not recommended for power transmission housings due to the poorer heat output capability. F Influence of the radiation factor This possibility is dependent on the position of the dominant frequencies in relation to the coinci-dence frequency (Section 6.7.2.4, Figure 6.7/8). The benefit is determined by the complexity of the noise generation; thus a smaller wall thickness has a disadvantageous effect on the structure vibra- tion transmission level. Table 6.7/8 Change in Structure Vibration Transmission Level with Material Substitution and Constant Bending Stiffness ǻLSh Noise behaviour trend Quasi-static range Natural frequency range EN-GJL-200 → Al 0 +5 dB louder EN-GJL-200 → St 0 +1 dB no change Table 6.7/9 Change in Structure Vibration Transmission Level with Material Substitution and Constant Geometric Dimensions ǻLSh Noise behaviour trend Quasi-static range Natural frequency range EN-GJL-200 → Al +5 dB +8 dB louder EN-GJL-200 → St -5 dB -2 dB quieter Figure 6.7/32 Influence of different materials in gears [6.7/56] 6.7 Noise Behaviour 451 Figure 6.7/33 Sound emission from trans- mission housings made of polymer concrete [6.7/89] 6.7.6.6 Sound Protection Enclosures A Task If the primary noise-reduction measures relating to the influencing of the gear-shaft-bearing system and the transmission housing are not sufficient to achieve a certain required sound emission, it is possible to provide sound-protection enclosures. The so-called integrated enclosures [6.7/17] can still be counted among the range of primary noise-reduction measures, while the separately installed partial or full enclosures are counted among the secondary noise-protection measures, which are mounted subsequently. B Physical principles Sound-protection enclosures envelop sound sources completely or partially and reduce the pro-pagation of sound between the sound source and the insulated areas. The basic construction of a sound-protection enclosure is shown in Figure 6.7/34. This figure also shows the principal possible sound-propagation paths for both airborne and structure-borne sound with reference to [6.7/74]. The noise-absorbing effect of an enclosure is described by the insertion loss D eK as the difference of the sound pressure or sound power level with and without a sound-protection enclosure [6.7/73]. Test codes are formulated in [6.7/149] and [6.7/150]. In an approximate observation, an insertion loss results [6.7/73]: W eK 10 dBĮ = + 10 lg dB ı10DRDD− + (6.7/50) D Damping for the opening or of the sound absorber S O Open area of the enclosure R Sound-reduction index of the enclosure wall S W Surface of the enclosure wall ı Opening ratio of the enclosure ı = S O/SW Įw Absorption coefficient of the enclosure wall 452 6 Load Capacity and Running Performance: External and Internal Gearings The sound level reductions through the enclosure walls, openings and rigid connections are discussed in detail in [6.7/74]. The following can be used for the estimation of a level reduction in sound propagation through the enclosure walls: dB lg10Wα Δ + R = L (6.7/51) The openings in an enclosure can have a considerable influence on the insertion loss effect. A sound absorber must be provided for larger openings. Figure 6.7/34 Schematic diagram of a sound protection enclosure C Basis for dimensioning For acoustic reasons, a minimum distance between the surface of the transmission and the en-closure wall must be observed, which is dependent on the frequency of the airborne sound and the mass (per unit area) of the enclosure wall. The calculation of the amount of heat accu-mulating in the enclosure and the decision for self- or forced ventilation are very important. The possibility of repairing or inspecting the transmission must be ensured. The sound reduction index R, the damping D and the sound absorption coefficient Į W of the ab- sorbing material are frequency dependent and at low frequencies are much smaller than 1000 Hz (refer to Tables 6.7/10 and 6.7/11 according to [6.7/18] and [6.7/73]). The lower the frequency to be damped is in relation to the absorber, the greater the thickness of the absorbing material must be. Usual thicknesses are from 30 to 80 mm. The sound-reduction index of a wall is determined by its surface weight. Sheet steel with a thick-ness of 1 to 3 mm is often used for the enclosure wall. In order to prevent significant bending vibrations, large and smooth wall surfaces should be avoided. Openings for rigid or moving parts must be well sealed. An essential requirement for the use of enclosures relates to a structure-borne sound-proofed installation. Examples It can be typically deduced that it is difficult to achieve high insertion loss in the low-frequency range. For frequency-dependent sound absorption coefficients Į W: Table 6.7/10 For frequency-dependent sound reduction indices R: Table 6.7/11 For the frequency-dependent effect of an executed sound protection enclosure: Figure 6.7/35 6.7 Noise Behaviour 453 Figure 6.7/35 Example of the effect of a transmission sound protection enclosure Table 6.7/10 Frequency-Dependent Sound Absorption Coefficients and Įw Mean Values Įࡄw for Selected Materials Material Thick- ness s mm Density N kg/m3 Centre frequency in Hz Įࡄw 63 125 250 500 1000 2000 4000 8000 Sound absorption coefficient ĮW Polyester-based PU foam 30 30 0.1 0.1 0.2 0.6 1 0.9 0.85 0.8 0.41 Resin-bonded mineral fibre board 15 30 50 140 120 100 0.1 0.1 0.1 0.1 0.1 0.2 0.12 0.45 0.60 0.39 0.82 0.83 0.68 0.83 0.95 0.92 0.92 0.98 1 0.9 0.98 1 0.9 0.96 0.36 0.47 0.55 Wall clearance dW = 0 for all materials. Table 6.7/11 Frequency-Dependent Sound Reduction Index R and Mean Values Rժ Material Thickness mm Centre frequency in Hz Rժ dB 63 125 250 500 1000 2000 4000 8000 Sound reduction index R in dB Aluminium sheet 1 2 3 8 10 21 10 13 24 12 15 25 14 22 26 19 26 28 25 30 32 28 33 27 31 35 25 18.0 23.2 27.0 Sheet steel 1 1.5 3.5 15 18 26 17 22 29 23 27 33 30 33 36 32 36 39 35 38 41 38 36 31 40 35 27 29.2 32.0 34.8 6.7.6.7 Anti-sound One possibility for active noise reduction is to consciously manipulate sound wave fields by ex- ploiting the principle of interference. With the method “noise reduction through anti-sound”, in addition to the primary sound source, a second electroacoustic sound source exists with a sound field in which the individual frequency components must have the same amplitude and a phase shift of 180 . High technical requirements are placed on this approach. Applications that have become known relate to low-frequency noise components, for example, in air ventilation ducts [6.7/74]. 454 6 Load Capacity and Running Performance: External and Internal Gearings 6.7.7 Structure-Borne Sound and Diagnostics In addition to the noise behaviour, the features of functional capability, such as availability, safety and accuracy, represent the most important quality assurance criteria. For preventive and condition-based maintenance, the application of diagnostic systems is necessary. The elements at risk particularly through malfunction should be monitored in their function, and the damage occurring in their progressive change should be observed. Therefore, it is important to register and interpret the features using measurable physical quantities. The focus of the studied and applied methods, on which the technical diagnostic is based, is the vibration and structure-borne sound method. Extensive tests have been carried out internationally and described in numerous publications concerning which parameters and parameter curves within different diagnostic systems are suitable for a reliable assessment of damage development at an economically acceptable cost. With the rapid development of microchips, microsystem-based diagnostic methods are central. Because of the different functions that are to be assessed, it is difficult to find a universal method. Application-ready and tested diagnostic systems already exist for special machines, such as for turbines and DC motors. Meltzer [6.7/45] includes reports on the state of vibration monitoring and diagnostics at the turn of the millennium. Basic possibilities for early damage detection are already contained in [6.7/9] and [6.7/40]. For transmissions, it is much more difficult than originally believed to determine the appropriate features and physical quantities and to separate and assign recorded signals to the individual damage mechanisms. The following types of damage in toothing have significant importance: • tooth root breakage, tooth edge disruption • pitting • grey staining • scuffing • wear (profile deviation) A sole analysis of the main excitation frequencies of a transmission and the generally unknown transmission behaviour of the transmission elements, including the housing, is not sufficient for a diagnosis. In the 1980s, the following diagnostic features were the focus of discussion: • overall sound pressure level of the (structure-borne sound) acceleration signal • frequency level of the main exciters with their harmonics and their difference from the background noise level • gamnitudes and quefrency from the cepstrum analysis • level sums of side bands • kurtosis and skewness from the statistical analysis of the amplitude distribution of the structure-borne sound spectrum [6.7/69] In contrast, in connection with the microsystem-based diagnostic methods in the literature (e.g., [6.7/46]), new features in correlation with the damage development are proposed, which relate to the time domain (e.g., effective value of the vibration acceleration) and the frequency range (e.g., noise band effective value). Hitherto conventional features are also explained in [6.7/46]; thus, no satisfactory correlations with damage development have been determined for the kurtosis and crest factor. Primarily, correlations between the different features and the course of the damage are to be obtained reliably through measurements. 6.7 Noise Behaviour 455 Other literature that also provide an overview of the development of diagnostic methods for transmissions in recent years are • gearing: [6.7/13, 14, 35, 40, 51, 52, 69, 93, 94, 127, 157] • bearings: [6.7/3, 7, 32, 92] 6.7.8 Symbols and Symbol Explanations of Section 6.7 A m2 sound absorption area A0 m2 reference value of sound absorption area (4 m2) a mm centre distance Bƍ Nm specific bending stiffness B, b mm face width bR, bRa mm rip width C m relief, general Ca m tip relief Cc m lead crowning Cf m root relief CL ms-1 longitudinal wave velocity in the material c ms-1 sound velocity cm Nmm-1 m-1 mean specific tooth stiffness cmin Nmm-1 m-1 minimum specific tooth stiffness c0 ms-1 sound propagation velocity in the air cȖ Nmm-1 m-1 total specific tooth stiffness D dB damping of a sound absorber DeK dB insertion insulation level dec - decade dr mm roll circle diameter dS - differential of measuring area d W mm wall distance dwk mm diameter of rolling element E Nmm-2 Young’s modulus F N exciting force FL N dynamic bearing force FN N static tooth normal force Fdyn N dynamic tooth force F0 N reference force (1 N) ˆF N peak exitation force F N effective exitation force f Hz frequency fA Hz outer ring frequency ferr Hz frequency of excitation ff m profile form deviation fg Hz limit frequency, frequency of coincidence fHĮ m profile angle deviation fHȕ m helix slope deviation fHȕw m effective helix slope deviation fI Hz inner ring frequency fK Hz frequency of identical tooth position; coupling frequency fM Hz machine frequency fm Hz mean frequency (third octave) fn Hz rotation frequency fp m single pitch deviation fp Hz breaking frequency of radiation factor fTerz Hz mean third frequency fwk Hz rolling element frequency fz Hz mesh frequency f0 Hz corner frequency of piston radiator f11 Hz first eigenfrequency G dB standard gauge of deviation HR mm rip height h ms-1N-1 admittance h - tooth height factor hT ms-1N-1 transmitting admittance hT0 ms-1N-1 reference transmitting admittance (5 10-8 ms-1N-1) i - gear ratio, variable J Wm-2 sound intensity J0 Wm-2 reference sound intensity (10-12 Wm-2) K1 dB(A) regression coefficient K2 - regression coefficient K2A - factor for surrounding correction k - assumption constant kL kNsm-1 bearing damping L dB sound level Lc dB(A) noise characterisation value Leq dB equivalent noise characterisation value LF dB force level LFerr dB excitation force level Lges dB total sound level LhT dB transfer admittance level Li dB sound level proportion LJ dB sound intensity level Lm dB mean sound pressure level Lp dB sound pressure level (linearly weighted) LpA dB(A) A-weighted sound pressure level LpAeq dB(A) workplace-related emission value (A- weighted) LƍpAeq dB(A) time-averaged workplace-related emission value (A-weighted) LS dB area index LSh dB structure-borne sound index LShe dB structure-borne sound index in eigentone range 456 6 Load Capacity and Running Performance: External and Internal Gearings LShq dB structure-borne sound index in quasi-static range L dB structure transfer index Lv dB vibration velocity level, structure-borne sound level LW dB sound power level (linearly weighted) LWA dB(A) A-weighted sound power level Lı dB radiation index l mm edge length, rip length lk mm coupling length l1, l2 mm partial rib length M Nm torque MB Nm torque of reference gear unit m kgm-2 compound coating of a plate mn mm normal module n - sample size, count number n min-1 actual driving speed n, n N min-1 nominal driving speed n0, nB min-1 reference driving speed n1 min-1 input speed P W sound power P kW transmission power PN kW nominal power P0 W reference sound power (10-12 W) p Pa sound pressure p - number of poles p(t) Pa instantaneous sound pressure at a given location p֥ Pa effective sound pressure p֥0 Pa effective sound pressure reference (2ͼ10-5 Pa) q - equivalence parameter R dB sound reduction index r mm distance from sound source rg mm border radius in sound field S m2 surface, measuring surface, radiating surface S0 m2 reference surface (1 m2) (Sh2 T)0 m4s-2N-2 reference structure-borne sound index (2.5 ͼ10-15 m4s-2N-2) SO m2 area of casing opening SW mm surface of casing wall s mm plate thickness, wall thickness sB mm thickness of a plate with equivalent bending stiffness sE mm substitute plate thickness sm mm thickness of a plate with equivalent compound coating t s time / ms-1 vibration velocity, sound particle velocity ΔiTerz dB sensitivity level (in case of A- weighting) / ms-1 effective vibration velocity averaged over a surface / ms-1 effective vibration velocity, effective sound particle velocity / ms-1 peak sound particle velocity 0 / ms-1 sound particle velocity reference /w ms-1 velocity at worki ng pitch circle w - number of rolling elements x - profile shift coefficient z - number of teeth zM - number of teeth of pitch worm gear Į pressure angle ĮW sound absorption coefficient of a wall ȕ helix angle į contact angle (roller bearing) ΔL dB difference level ΔLW dB sound power difference level ΔLp dB sound pressure difference level Δc Nmm-1 tooth stiffness variation İĮ - transverse overlap İȕ - axial overlap İȖ - total overlap ȘStr - structure damping ȜB m free bending wave length Ȝ0 m airborne sound wave length Ȟ - Poisson’s number ĳ0 - phase angle at t = 0 ', N kgm-3 density, general '0, N0 kgm-3 density of air ı - radiation factor ı - ratio of casing opening SO/SW ıM dB reference standard deviation ıP dB production standard deviation ıR dB comparison standard deviation ıt dB total standard deviation Ȧ s-1 angular frequency Indices B reference value ges total i count number m mean value max maximum N nominal w working circle 0 reference 1 gear 1, level 1, frequency 1 2 gear 2, level 2, frequency 2 I, II, III caption of gear stage 11 base vibration 12 first harmonic EFFICIENT GEARBOXES FOR AGRICULTURAL MACHINERY AND OFF-HIGHWAY APPLICATIONS – THIS IS OUR BUSINESS. As an engineering partner and global supplier of gearboxes GKN Walt erscheid has extensive knowledge of complete systems, intelligent constructions and modern manufacturing processes. Our components and technology significantly influence the performance of our customers’ products.INNOVATION POWEREFFICIENCY Gearbox technology made by GKN Walterscheid www.gkn-walterscheid.de/en 459 7 Design of Gearing and Gear Transmissions 7.1 Number of Stages and Splitting the Total Transmission Ratio 7.1.1 Preliminary Considerations Depending on the demands made on a gear transmission, certain criteria in terms of quality can be used to evaluate the various draft designs, such as minimal individual dimensions, small moments of inertia, or low transmission losses. In addition to high operational reliability, the kinds of demands made on optimised industrial gear transmissions today include especially the lowering of manufacturing costs [7.1/1], [7.1/14] and the mass of the gear transmission. However, frequently value is placed on satisfying not only one but several of the criteria. After the task has been clearly outlined, a more precise analysis of how to establish a model to solve the optimisation problem can begin. The optimisation model is complete once the target functionality and the secondary conditions have been formulated. Here not all of the factors and relations to the actual system can be taken into consideration. The goal is to simplify the model as much as possible in order to ensure that the problem remains manageable and clear, and to make sure that the amount of work required for calculations is kept to a minimum. During this process it is impossible to avoid the loss of some information. A model can depict merely an approximate picture of reality. For that reason, the results of the optimisation process can only consist in a statement about the optimal conditions of the model. The extent to which these results are actually relevant to the real system depends on the quality of the model itself. Whether or not the model is meaningful should therefore be tested before any optimisation calculations are done. From the numerous possibilities available, in the following section will be presented the results of the optimisation of the design of industrial gear transmissions for minimum mass, with case-hardened gearing, based on the permissible contact stress [7.1/2], [7.1/3]. 7.1.2 Design for Minimum Mass The total mass of a gear transmission is essentially composed of the masses of the gears, the shafts, the bearings and the housing. It is affected by the choice of the number of transmission stages, the splitting of the total transmission ratio into the individual stages ([7.1/4] to [7.1/9]), and the relative width b/d. It is to be expected that the kind of housing and the arrangement of shafts that are chosen won’t play any insignificant role. In order to determine the main parameters of the gear transmission, which are the number of stages, partial transmission ratios and width ratios, it is sufficient to incorporate the gear masses and the housing mass in the target function (Figure 7.1/1). Based on the assumption that instead of gears, solid cylinders can be used, the following equation can be formulated for the mass of the set of gears: 460 7 Design of gearing and gear transmissions ()23 (gear set) i 1 iʌ14 = ⋅ =⋅ + ⋅ ikbmu ddN (7.1/1) d Pinion diameter of the ith stage k Load stage Width ratio of the ith stage (width/pinion diameter) In order to come as close as possible to the real gear trans- mission, the bearings, fastening rails, bracing and other details which differ from the calculation model are accounted for in the housing mass in the factor F 1 [refer to Equation 7.1/2)]. This resulted in F1 = 1.5 from tests on gear transmissions by notable manufacturers. The dimensions of the respective housing can be deduced dependent on the dimensions of the set of gears. The difference concerning the density of the materials of the gears and of the housing can be registered by a factor F 2. With that the volume proportional to the mass (quality criterion) is given: m transmission ĺ V = K ǜ (VZ + F 1 ǜF2 ǜVG) (7.1/2) V Gear transmission volume K , F1, F2 Correction factors V Z Volume of the gear sets V G Housing volume In order to correspond to real gear transmissions, the factor K is introduced, which accounts for the proportion of the shafts, bearings and other surrounding components of the total mass. This is also the result of investigations on gear unit series by some manufacturers of industrial gear transmissions ( K = 1.5). As a proportionality factor, however, K has no influence on the deter- mination of the main parameters for the optimisation process [7.1/10]. Despite its limited validity, the Hertzian theory has been proven to be a reasonable criteria for the evaluation of contact pres-sure in gearing. With that, gear transmission optimisations considering the permissible contact stress according to DIN 3990 [7.1/11] are justified. Otherwise, the safety factor S H, calculated and evaluated with this stress, would also be useless. The manufacturing deviations which have an effect on the face load distribution are considered according to the tolerance equations in compliance with DIN 3962 [7.1/12], or respectively through approximations, as functions dependent on the face width. Even the misalignment during meshing caused by elastic deformations were included in the iterative calculation. The minimal volume index for a one- to three-stage gear transmission, depicted in dependence on the total transmission ratio [Figure 7.1/2], leads, through the determination of the points of intersection between the respective curves and while taking into account the given load, to the conclusion to make gear transmissions of a medium level of precision with one-stage at i < 5, two-stage at 5 i 15 and three-stage at i > 15. The shape of the housing plays a minor role here. Other aspects, such as manufacturing costs, can lead to a shift in these limitations. Optimal splitting of the total transmission ratio into the individual stages for medium gear tooth qualities (Q5 and 6 according to DIN 3962 and DIN 3964 [7.1/13]) is depicted in Figures 7.1/3 and 7.1/4. Figure 7.1/1 Model of a two-stage gear transmission (axial section) [7.1/14]; assumptions for optimisation i db 7.1 Number of Stages and Splitting the Total Transmission Ratio 461 Evident are minor differences in the results when the output torque varies. Tables 7.1/1 and 7.1/2 list, among other things, information about approximation equations for selecting the main para-meters of one- to three-stage gear transmissions, determined for a design with minimum mass [7.1/3]. Whereas in cases of different shapes of housing the results for the number of stages and the splitting of the total transmission ratio differ only very little from one another, the ideal face width ratios ( b/d) i are dependent on the chosen shape of the housing (Figure 7.1/5) and are only meaningful if the face load distribution factor KHȕ has been included in the calculations. The following recommendations were elaborated for the housing shapes most frequently found in modern industrial gear units, with the requirement of minimum mass . For the chosen designs and assumptions, the ratio of face width b to pinion diameter d of the respective stage i resulting from the optimisation process is ig e s iiȞ bidχ =⋅ (7.1/3) i Number of the stage ( I, II, III ), I = input stage The parameters Ȥi and Ȟi can be found in Table 7.1/3 de pendent on the number of stages k, the shape of the housing (Figure 7.1/ 5) and the stage Number I, II, III [7.1/3]. Figure 7.1/2 Volume-moment ratio for one- to three-stage gear transmissions The values which result from Equation (7.1/3) are to be treated as guiding values . For the final design other aspects can be important, such as existing gear sets, other components, cost issues or values taken from experience. In cases where the optimum runs flat (e.g., for optimisation according to the minimum mass of the gear transmission), small deviations do not have a very negative effect on the individual criterion [7.1/3]. 462 7 Design of gearing and gear transmissions Figure 7.1/3 Splitting of the total transmission ratio of two-stage gear transmissions for gear tooth quality 6 Figure 7.1/4 Splitting of the total transmission ratio of three-stage gear transmissions for gear tooth quality 6 In summary, one can assert: When the gear housing is considered in the mathematical model, in addition to determining the number of stages and splitting the total transmission ratio into the individual stages, it is also possible to make specific statements concerning the selection of face width ratios. These depend on the chosen shape of the housing (Figure 7.1/5) and are only reasonable if the face load distribution factor K Hȕ has been included in the calculation. The recommendations available generally agree with the findings of the gear transmission industry. Tests have also shown that limited deviations of the main parameters from the optimum don’t necessarily mean a drastic increase in mass. Because of this, it can be expected that the optimisa-tions can be well expanded to other criteria, such as minimum costs. No considerable reduction in weight is to be expected with a change in the shape of the housing. Further information on including the costs and observing a series expansion can be found in the literature ([7.1/1], [7.1/14] among others). 7.1 Number of Stages and Splitting the Total Transmission Ratio 463 Figure 7.1/5 Housing shapes (Shape 2: The inclination of the cover surface is identical to the tangent on the tip circles of the large gears of the two last stages) Table 7.1/1 Selection of Number of Stages for Industrial Gear Transmissions (Shafts on One Plane) One-stage Two-stage Three-stage 1 | iges | 5 5 < | iges | 15 15 < | iges | 60 Table 7.1/2 Approximations for the Most Favourable Splitting of the Total Transmission Ratio for Industrial Gear Transmissions and Medium Manufacturing Quality (Quality 5 to 7 (8) DIN 3962, ISO 1328, Shafts on One Plane, Transmission Range Table 7.1/1) Number of stages Gear ratio k uI = z2 / z1 uII = z4 / z3 uIII = z6 / z5 1 iges - - 2 0.7332 iges0.6438 iges/uI - 3 0.4643 | iges|0.609 1.205 | iges |0.262 | iges | / (uI ǜ uII) Table 7.1/3 Approximations for Favourable Width Ratios for Mean Gear Tooth Qualities (Quality 5 to 7 (8) DIN 3962, Shafts on One Plane, Transmission Range Table 7.1/1) k Shape ȤI ȤII ȤIII 1 1 - 0.0028ǜܶ + 0.6087 2 - 0.0024ǜܶ + 0.5723 2 1 - 0.0043ǜܶ + 0.3495 0.0816ǜܶ + 0.4167 2 0.349ǜ ܶ 0.0826ǜ ܶ + 0.2658 3 1 0.0581ǜܶ +0.2744 0.2513 -0.025ǜܶ + 0.325ǜܶ +0.05 2 0.047ǜܶ - 0.459ǜTժ + 1.4394 - 0.005 ǜܶ + 0.031ǜܶ + 0.268 0.143 ǜܶ +0.1245 k Shape ȞI ȞII ȞIII 1 1 0.0142ǜܶ + 0.1326 2 0.0141ǜܶ + 0.1389 2 1 0.0195ǜܶ +0.0403 0.009ǜܶ +0.0344 2 0 -0.0218ǜܶ + 0.031ǜ ܶ + 0.268 3 1 -0.0127ǜܶ +0.128ǜܶ -0.0984 0.1548 0 2 -0.0483ǜܶ +0.481ǜܶ - 0.935 0.0407 ǜܶ - 0.0617 - 0.0292ǜܶ + 0.158 (k number of stages; I, II, III, number of the stage ( I = input stage); ܶ = log ( Mtab/Mto); M tab in Nm; Mto = 1 Nm) Area of validity: 102 Nm Mtab 106 Nm; Mtab output torque of the gear 464 7 Design of gearing and gear transmissions 7.2 Approximate Determination of the Dimensions After choosing the material, the heat treatment, the type of gearing (spur or helical), and the splitting of the total transmission ratio, the problem now comes down to the design of an individual stage. For this, simplifications are used, which won’t be valid anymore for the final calculation. 7.2.1 Draft Calculation for the Tooth Flanks (Pitting) In most cases, a design according to the load capacity of the tooth flanks provides a suitable initial approximation for the main dimensions. Particularly for non-hardened gearing, that is the decisive criterion. The smaller the number of teeth, the more decisive flank pressure becomes. In cases of typical numbers of teeth for industrial gear transmissions and similar, a design according to flank load capacity has its advantages. From Equation (6.5/11) for contact stress and Equation (6.5/16) for the safety factor, if the single pair mesh factor ( Z B, ZD), the helix factor ( Zȕ), the roughness, speed and lubricant factors ( ZRZvZL), the size factor ( ZX) and the hardness ratio factor ( Zw) are all disregarded, with Ft = M1/(d1/2), the following results for the pinion diameter: ()222 H1 E H İ31 2 NH l i m H 12 1 /KM ZZZ udu ZS b dσ⋅⋅⋅ ⋅ ⋅ +≥⋅ ⋅ (7.2/1) Here the following apply: KH Load factor, tooth flank KH = KAKvKHĮKHȕ, with KA = KAB application factor, a reasonable factor for overloading operational loads that act as fatigue stress. KA = KAB is to be chosen according to experience (Section 6.3.3 or ISO 6336-6 and DIN 399 0 T1). If the torque M 1 already contains the load spectrum including the additional external loads, then use KA = KAB = 1 (Sections 5.3 and 6.3.3); Kv dynamic factor (Section 6.3.4), KHĮ transverse load factor (Section 6.4.4.1). Here K v ǜ KHĮ = 1.2 is set as an approxi- mation. KHȕ face load factor (Section 6.4.4.2). If KHȕ is not determined in the optimisation process (Section 7.1.2) or known from experience, it is set as KHȕ = 1.5. M1 Pinion torque of the respective stage; M1 = Pnenn/Ȧ1 , (P nenn nominal power) ZE Elasticity factor , according to Equation (6.5/9a) ZE = 190 N/mm for a steel/steel gear pair ZH Zone factor , according to Equation (6.5/9b) With spur gearing and a sum of the addendum modification Ȉx = 0, the following is used: ZH = 2.5, with helical gearing according to Equation (6.5/9b) or Figure 6.5/6 for Ȉx = 0 and the chosen helix angle ȕ. The helix angle is chosen taking into consideration the overlap ratio, the axial force supportable by the bearing, and the operational specifications (refer to DIN 3978). With industrial and vehicle gear transmissions, the helix angle is in the range of ȕ = 8 to 25 . With double helical gearing, values up to ȕ = 45 are common. Zİ Overlap coefficient , according to Equation (6.5/8a,b) Spur gearing: Zİ = 1; helical gearing: Zİ = 0.85 u Gear ratio , u = z 2/z1; z2 z1 Note: • u transmission ratio of the gear stage • With internal gear pairs the number of teeth of the internal gearing is negative. ZN Life factor according to Equation (6.5/13) In the endurance limit range, the following is true: ZN = 1. ıHlim Flank strength (pitting), Section 7.4 For alloyed case-hardened steels with a surface hardness in the range of σHlim = 1400 … 1500 N/mm2 may be selected. HRC H3 360+ − = 7.2 Approximate Determination of the Dimensions 465 SH Safety factor The safety factor is to be determined taking into consideration any possible agreements or existing requirements, the importance of the system, the accuracy of knowledge of the operating loads, the impact of possible damage and the manufacturing, testing and operating conditions anticipated. If no special requirements or agreements exist, then S H = 1.2 can be assumed here. Note: From experience, the value ( ZNǜıHlim/SH) is often known as a specific, permissible strain value. b/d1 Face width / diameter ratio If b/d1 is not pre-determined or has not been calculated in the optimisation process, or if the Equation (7.1/3) derived from this is not to be used or doesn’t apply to this type of gear transmission, then b/d1 is to be specified. Decisive here is the entire design (deformations, quality, material / heat treatment, adjustment measures). Values such as b/d1 = 0.6 to 1.2 are common. After determining the pinion diameter, the number of teeth z1 can be chosen and, with the help of the previously determined helix angle ȕ, the modulus mn determined. The value calculated from the diameter is generally to be rounded up or down to a standardised value in accordance with ISO 54 / DIN 780 (refer to Table 2.1/3). The recalculation must then confirm that the required bending safety factor is achieved. After determining d 1, mn, z1, d2, z2 and ȕ , the profile shift must be determined (Section 7.3.2), and with that the final recalculation can be carried out (Section 6.5.3). The other way is the calculation of the minimum value of the modulus. Based on the determined diameter d 1 in accordance with Equation (7.2/1), after converting Equation (6.5/34) for the tooth root stress and Equation (6.5/39) for the safety factor, the required smallest module can be roughly determined. After disregarding the size factor Y X, the roughness factor YR, the sensitivity factor Yį and the helix factor Y ȕ, the result is Ft F S İ n NF E F() (/ )KFYYmbY S σ⋅⋅ ⋅≥⋅⋅ (7.2/2) Here the following apply: KF Load factor, tooth root KF = KH; for KH refer to information given at Equation (7.2/1) Ft Tangential force on the reference circle Ft = 2Mt1/d1 (YFSYİ) Stress factor For gearing with a protuberance: hf/mn = 1.4; Na0/mn = 0.4; other values ISO 53, DIN 867: ( YFSYİ) 2.9 For gearing without a protuberance: hf/mn = 1.25; Na0/mn = 0.25; other values according to ISO 53, DIN 867: ( YFSYİ) 3.1 b Face width b = (b/d1)d1; b/d1 from Equation (7.2/1) ıFE Tooth root endurance strength Expressed by the maximum local stress (Section 7.4). For case-hardened gearing (alloyed case- hardened steel), the calculations can use ıFE = 800 N/mm2. YN Life factor , according to Equation (6.5/37a) In the endurance limit range N 3ǜ106 is YN = 1 SF Safety factor against bending fatigue fracture The safety factor is to be determined taking into consideration any possible agreements or existing requirements, the importance of the system, the accuracy of knowledge of operating loads, the impact of possible damage and the manufacturing, testing and operating conditions anticipated. If no special requirements or agreements exist, then S F = 1.3 can be assumed here. Note: From experience, the value ( YNǜıFE/SF) is often known as a specific, permissible strain value. 466 7 Design of gearing and gear transmissions With the diameter d1 and the modulus (after choosing ȕ), the number of teeth z1 is known. In some cases a change of the determined quantities d1 or m n may be necessary, if the resulting number of teeth turns out to be unfavourable or other demands are made on this quantity. 7.2.2 Draft Calculation for the Tooth Root In cases of larger numbers of teeth (high-speed gear unit) and surface-hardened or nitrided gearing, the load capacity of the tooth root comes close to the load capacity limit. In particular, in the case of high thermal load and scuffing risk, the aim is to design the module as small as possible. This also means that the load capacity of the tooth root is always a decisive factor. From Equation (6.5/34) for tooth root stress and Equation (6.5/39) for the safety factor, with F t = 2M 1/d1, d1 = z1mn / cos ȕ and b = (b/d1) d1 and disregarding the size factor YX, the roughness factor YR, the sensitivity factor Yį and the helix factor Yȕ, for the normal module of the associated stage results in the following equation: or 32 1 1 F N FE2 İ FS 1 F n) / ( ) / ( cos) ( 2 z d b S YY Y M Km⋅ ⋅ ⋅⋅ ⋅ ⋅ ⋅ ⋅≥σ 3 1 n F N FEİ FS 1 F n) / ( ) / ( cos) ( 2 z m b S YY Y M Km⋅ ⋅ ⋅⋅ ⋅ ⋅ ⋅ ⋅≥σ (7.2/3) (7.2/4) Here the following apply: KF Load factor, tooth root KF = KH; (KH = KAKvKHĮKHȕ); for K H refer to information given at Equation (7.2/1) M1 Torque of the pinion of the respective stage; M1 = Pnenn /Ȧ1 (Pnenn nominal power) (YFSYİ) Stress factor Refer to remarks under Equation (7.2/2) ȕ Helix angle Refer to remarks under Equation (7.2/1) at ZH ıFE Fatigue strength Expressed by the maximum local stress (Section 7.4.) For case-hardened gearing (alloyed case-hardened steel) calculations can be done here with ı FE = 800 N/mm2. SF Safety factor against fatigue fracture Refer to remarks under Equation (7.2/2) YN Life factor Refer to remarks under Equation (7.2/2) b/d1 Face width / diameter ratio Refer to remarks under Equation (7.2/1) z1 Number of teeth of the smaller gear of the pairing For industrial gear transmissions one chooses z1 = 14 to 25; for high-speed gear units z1 = 25 to 45 is typical, and in vehicle gear transmission manufacturing z1 = 8 to 25. b/mn Face width / module ratio If b/d1 is not pre-determined or has not been calculated in the optimisation process, or if the Equation (7.1/3) derived from this is not to be used or doesn’t apply to this type of gear transmission, then b/m n is to be specified. Decisive here is the entire design (deformations, quality, basic material/heat treatment, adjustment measures). Frequent values are b/mn = 10 to 20, if space considerations do not require other relations. 7.3 Selection and Partitioning of Profile Shift 467 Usually a standardised value is selected (ISO 54, DIN 780 and Table 2.1/3) for the module mn. After the main dimensions are settled ( z1, z2, mn, ȕ, b) it is time to determine the profile shift. The information given in Section 7.3.2 can be used to determine the sum ( x1 + x 2) and to split it. With the normal module mn in accordance with Equation (7.2/3) or (7.2/4), the chosen number of teeth z1 and the helix angle ȕ, the diameter d1 is preliminary determined. Next, using either the general recalculation or with the help of Equation (7.2/1) (and possibly b/d1 = (b/mn) (1/z 1) cos ȕ ), the minimal diameter of the pinion due to contact stress must be checked. A rough determination of the dimensions is never a substitute for recalculation. This must be done for all types of strain and is ultimately decisive. 7.3 Selection and Partitioning of Profile Shift The profile shift has an influence on the properties of involute gearing. The following sections will examine tendencies towards an increase in tooth root, flank, and scuffing load capacity (sliding speed and specific sliding) through the selection of a suitable profile shift sum and its partitioning. Practical recommendations for design will follow. 7.3.1 Criteria and Tendencies Load capacity of the tooth flanks The radii of curvature of the tooth flanks in contact are of decisive importance for the tooth flank pressure. The farther the corresponding flank sections are from the base circle, the larger the radii of curvature. Thus a positive profile shift with favourable partitioning between pinion and wheel has the effect of increasing the load capacity on the tooth flank, especially in cases of small numbers of teeth [7.3/1]. a) Radii of curvature b) Hertzian contact stress along the line of action Figure 7.3/1 Contact conditions for pairs of gears (external gearing) 468 7 Design of gearing and gear transmissions Contact stress can be decreased when the profile shift is used to move the path of contact along the line of action towards the area of the contact stress minimum (Figure 7.3/1). Theoretically the minimum of contact stress has been reached with y1 y2 y1 y2 y1 y2! Max achieved at⋅==+NNNNNN It can be found in the middle of the distance 2 1T T, i.e., ̄ y1 = ̄y2 = (a/2) sin Į wt. The possibilities, however, are limited. In cases of typical numbers of teeth, before a significant reduction in contact stress can be realised, pointed tooth tips occur or the minimum contact ratio is not reached. In various sources in the literature (e.g., [7.3/1] to [7.3/5]), it has been pointed out repeatedly that the influence of the profile shift on contact stress drops with an increasing num-ber of teeth and/or speed. In cases of higher numbers of teeth, it is almost impossible to influence the strain due to profile shift. Determining the location of the stress maximum along the path of contact at high speeds is not an easy matter because of the influence of dynamic forces. Frequently more effective is a larger pressure angle that deviates from the standardised profile. Mathematically, the most favourable result in terms of a minimum in Hertzian contact stress comes about with a pressure angle of Į = 45 . Load capacity of the tooth root Formerly, when for gearing a constant design strength was compared to the nominal stress, the profile shift was used to mathematically verify a generally significant rise in the load capacity of the tooth root. According to today’s principles of calculation, there is only an insignificant or no increase in load capacity for gearing under load in the endurance limit range. The reasons are the opposing tendencies of the tooth root thickness, the bending lever arm (transverse contact ratio) and the tooth root curvature radius dependent on the profile shift. In extreme cases, instead of an increase in load capacity, a reduction even occurs. Since in cases of very low load cycle numbers (area of quasi-static strength) the notch effect is practically cancelled out by the sensitivity effect in conventional tempering and case-hardened steels, nevertheless, according to newer guidelines, there is an increase in load-bearing capacity. • Influence of profile shift on load capacity of the tooth root in the endurance limit range: To assess the tooth root stress the following formula is used: įS F ntX RrelT įTFE F YY Y m bFY YYS⋅⋅⋅⋅ ⋅σ = (7.3/1) With respect to the root load capacity, the profile shift is favourable if Y = (YFǜYS/Yδ) for gear 1 and gear 2 is the same and strives for a minimum. The profile shift factors are considered favour- able if the following is true: ( YF1 ǜ YS1/Yδ1) = (Y F2 ǜ YS2/Yδ2) (7.3/1a) The reduction of the effect of the increase in stress by the notch is considered globally by the sensitivity factor Y į. This occurs because the material, due to its structure and residual stress, practically does not react with full sensitivity to stress peaks. Whereas YS and Y F are used in the following in accordance with ISO 6336-3, DIN 3990-3 (method B) [7.3/8], the sensitivity factor 7.3 Selection and Partitioning of Profile Shift 469 Yį is calculated according to Equation (7.3/2), which, according to our own experience, corresponds better to reality. Due to the sensitivity, the influence of the notch effect is reduced, especially in the case of small moduli. This means that in cases of very large moduli the notch is fully in effect. For case-hardened gearing, according to our experience [7.3/6] the equation 0.55 0.72 į Fn2.3 mm11 0− =+ ⋅ YN (7.3/2) can be used ( NFn in mm). The larger the gear ratio and number of teeth, the harder it is to influence the load capacity with the profile shift. Figure 7.3/2b shows the most favourable partitioning of various sums of the profile shift at a number of teeth of the pinion of z 1 = 20 and u = 2. The larger the sum of the profile shift is, the less influence smaller changes have in its splitting on the tooth root stress, particularly in the area x 1 > 0. The decrease of the form factor, particularly due to the increasing tooth root thickness and the size of the sensitivity factor, leads to a reduction in the tooth root stress, which however is counterbalanced to some extent by an increasing notch effect. The larger the module, the smaller the sensitivity factor and its influence on the stress becomes. This means that the reduction of the tooth root stress through the profile shift is small and only effective for small numbers of teeth. Even an ideal profile shift sum with favourable partitioning cannot exert an influence on the load capacity to the desired degree (Figure 7.3/2a). A clear improvement is to be expected only in the case of small numbers of pinion teeth (14 to 17), transmission ratios (or respectively gear ratios of u = 1 to 2) and small moduli. The results depicted in Figure 7.3/2a are valid when the tooth root stress is the same for pinion and gear ( Y 1 = Y 2). The size of the calculated benefit in load capacity in cases when the tooth root stress of the pinion and gear is balanced, or respectively when an optimal profile shift is selected, is within the variance of commonly used approximations for Y S. This also applies to the results when changes are being made to the tool geometry. According to [7.3/7], when the profile shift is increased and grinding notches appear, only a worsening in tooth root stress can be expected. Figure 7.3/2 Influence of profile shift on tooth root stress (external gearing, h a0/m = 1.25; Na0/m = 0.25; ȕ = 0 ; Įn = 20 ) a) Referenced local stress for Y1 = Y2 with variation of the number of pinion teeth ( u = 1) 470 7 Design of gearing and gear transmissions Figure 7.3/2 b) Stress factor of the local stress, dependent on the sum of the profile shift and its partitioning (u = 2) Figure 7.3/2 c) Referenced nominal stress for Y F1 = Y F2 with variation of the number of teeth ( u = 1) Figure 7.3/2 d) Form factor, depen-dent on the sum of the profile shift and its partitioning ( u = 2) 7.3 Selection and Partitioning of Profile Shift 471 In summary, with respect to the optimal profile shift for minimum tooth root stress in cases of operation in the endurance limit range, the following can be asserted: Due to the opposing tendency of Y F (sFnĹ) and YS (NFnĻ) at rising profile shift, the calculated improvements using optimum profile shift factors are insignificant. • Influence of profile shift on tooth root stress under quasi-static load Gearing is also subject to single impact loading during operation. These maximum loads usually occur with low load cycle numbers. For this reason, among other possibilities, a comparison with strength limits that apply for low load cycle numbers is used for the assessment of the load capacity of the tooth root. Tests have shown that, in the area of static strength, the structural, tempering and case-hardened steels typically used for gearing are virtually insensitive to notches. For the calculation of the relationship between the permissible and existing tooth root stress, as an approximation it can be assumed that the notch effect and the sensitivity effect approximately cancel one another out, as in Equation (7.3/3): Y S = Yį (7.3/3) A possibility to increase the load capacity of the tooth root through the profile shift consists in influencing the form factor YF: YF = σFn / (Ft /b m n) (7.3/4) with σFn = f(z, x, Na0, εα) and YF1 = YF2 != Min In this particular load case, the favourable partitioning of a given sum of the profile shift can clearly lead to a reduction in tooth root stress. Here too, the largest gain in load capacity can be found in small numbers of teeth and small gear ratios (Figure 7.3/2c, d). The selection of the number of pinion teeth has a smaller influence on the partitioning of the profile shift than does a variation of the gear ratio. This conclusion is independent of the given basic rack of the tool. The intersection between the curves for Y F1 and Y F2 (Figure 7.3/2d) marks the most favourable place to partition the respective sum of the profile shift (same nominal tooth root stresses). Recommendations according to DIN 3992 [7.3/13], which are oriented towards an increase in tooth root load capacity through large sums of the profile shift and an adjustment of the factors Y F1 and Y F2 for balanced tooth root stresses, are justified for peak load conditions. The influence, which the sum of the profile shift at optimal partitioning has on the tooth root stress (peak load) or respectively on the form factor Y F, is depicted in Figure 7.3/2c for u = 1. Figure 7.3/2c suggests that the curve progression has a minimum. For small numbers of teeth, this stress optimum is not being reached because it is located outside the permissible range (s a samin, İĮ 1). Generally, a tendency towards preferably high positive sums of the profile shift can be detected here. We already know that, in terms of tooth root stress, helical gearing can be approximated by virtual spur gearing. As such, the results for spur gearing are at least partially representative for helical gearing, if the equivalent number of teeth is being used. More precise examinations of this matter incorporate the load distribution in its entirety and are not appropriate in this context because the uncertainty of the assumptions is in the range of the effects realised. Scuffing load capacity The problems associated with the scuffing load capacity have not yet been sufficiently clarified. Of the methods currently known, the best congruency compared to running results is realised through the application of the integral temperature criterion [7.3/8], [7.3/9]. This is why preference is often given to the integral temperature method. This method is based on the fact that a locally confined peak value is not as much a measure of critical stress as is the mean value of the contact 472 7 Design of gearing and gear transmissions temperature, Equation (7.3/5). Here the effect of single influence factors, which alter the course of the flash temperature (profile modification, wear, dynamic forces, etc.), are weakened: Įflam flaĮ Į1=⋅ gdggϑϑ (7.3/5) As opposed to the exact mathematical flash temperature curve according to Blok [7.3/8], [7.3/10], partially linear function curves along the length of path of contact can be used as a basis in order to calculate a mean temperature value. The deviation resulting from this is considered to be so small that it can be ignored.1) In order to calculate the most favourable partitioning of the sum of the profile shift, the minimum of the geometric influences, consolidated in a geometry factor, is being aimed for [7.3/11]. According to present-day experience, it is recommended to differentiate between the friction coefficient in the area of pushing and pulling friction. The Figures 7.3/3a to d depict the calculated profile shift factors at favourable partitioning with respect to maximum scuffing load capacity, where this differentiation is taken into consideration with 2 . 1 / CE AC= . This results in different recommenddations for the partitioning of the sum of the profile shift for speed-decreasing ratios and for speed-increasing ratios. Tests showed that for the partitioning of a given sum of the profile shift it is generally irrelevant whether the flash temperature or the integral temperature criterion is chosen. Modifying the helix angle ȕ itself has been shown to be unimportant for the partitioning of the profile shift. In summary, regarding the most favourable flash temperature strain for a speed-decreasing ratio, for the partitioning of the profile shift the following is initially to be recommended in terms of quality: For small sums of profile shift, the pinion should receive the larger share of profile shift. For sums of profile shift around Ȉx = (x 1 + x 2) = 1 it is beneficial to select the factors xi roughly the same size ( xi = Ȉx/2). In cases of larger values for Ȉx, the gear receives the larger profile shift. Depending on the gear ratio, beginning at Ȉx > 2 the ratio x2/x1 moves towards z2/z1. For speed- increasing ratios, however, there is generally a predominating tendency for the gear to have the larger share of the sum of the profile shift. Equations (7.3/7) and (7.3/8) respectively correspond to these recommendations and make a quantitative determination possible. Figure 7.3/3 Examples of the partitioning of the sum of the profile shift for a minimum integral temperature (ϑ flam = Min, external gearing) a) Speed-decreasing ratio, z1 constant, u = 2 1) As opposed to the integral temperature method in accordance with ISO/TR 13989-2 and DIN 3990-4, here the inclinations of the temperature curves were calculated separately based on the meshing points A, B, D, E . This should prevent unnecessary deviations. 7.3 Selection and Partitioning of Profile Shift 473 Figure 7.3/3 b) Speed-increasing ratio, z 1 constant, u = 2 Figure 7.3/3 c) Speed-decreasing ratio, z 1 = 20; u constant Figure 7.3/3 d) Speed-increasing ratio, z 1 = 20; u constant 474 7 Design of gearing and gear transmissions The partitioning of the profile shift recommended here concurs well with the recommendations in DIN 3992 [7.3/13] in the case of small gear ratios (or gear teeth ratios u = 1 to 2). With increa- sing gear ratio, however, larger deviations occur (Figure 7.3/4). The reason for this is the ratio of partial length of path of contact opt/CE AC , for which an optimum exists (integral temp. ⎯→⎯! Min), dependent on the gear ratio; see Figure 7.3/4 b. Here, opt/CE AC is jointly responsible for the fact that neither the inclusion of the helix angle (distinctly helical gearing), the sliding friction coefficient, nor the differentiation between the flash temperature and integral temperature criterion has a decisive influence on these recommendations [7.3/12] regarding the partitioning of the sum of the profile shift. a) Profile shift partitioning for speed-decreasing ratio in comparison with DIN 3992, example b) Ratio of partial length of path of contact at optimum partitioning of profile shift for minimum integral temperature, examples Figure 7.3/4 Gearing parameters at optimum ( x 1 + x2) partitioning for minimum integral temperature (examples, external gearing) Due to the influence of helix angle, wear, profile modification, dynamic forces, and so on, the temperature maximum shifts on the length of the path of contact. As a consequence of the con-sideration of load distribution in operation in the supercritical speed range, the zone of high forces (contact stress) shifts to the area of high sliding speeds. 7.3 Selection and Partitioning of Profile Shift 475 The solutions to an optimal sum of profile shift are oriented towards the smallest possible contact ratio (particularly in the case of distinct helical gearing). An optimum within the design limits only exists in the case of small numbers of (pinion) teeth. This is due to a change in the tem-perature maxima in the points on the length of the path of contact. Taking into account various influences, such as wear and dynamic tooth forces, this leads to larger deviations in the results for Ȉx. It is extremely difficult, however, to specify with sufficient precision the development of the influencing quantities, namely dynamic tooth force, friction coefficient, curvature under load, and so on, at each position throughout contact. That is why the sum of the profile shift can only approximately come close to the actual optimum. The depicted results [procedure: Equation (7.3/6)], however, can be used as a rough guide. In general, Ȉx 3 should not be exceeded. To keep this limit the number of teeth is decisive. Differentiated guidelines can be found at [7.3/12]. 7.3.2 Recommendations EXTERNAL GEAR PAIRS As a guide, for the design of external gearing with a sum of the profile shift freely chosen, the range of values as per Figure 7.3/5a can be used. For the partitioning , for speed-decreasing ratios Figure 7.3/5b and for speed-increasing ratios Figure 7.3/5c can be used. Generally the influence of the sum of the profile shift on the load capacity is larger in cases of small numbers of teeth and gear ratios. In each case the partitioning of the profile shift is significant. Figure 7.3/5 Determination of profile shift for external gearing (in analogy to DIN 3992) a) Typical range for choosing the sum of the profile shift factors Load capacity of the tooth root If there is a special demand for higher load capacity of the tooth root, a sum Ȉx that has already been calculated or is chosen according to Figure 7.3/5a or which arises from optimisations can be effected according to Figure 7.3/6b for an optimal static load capacity of the tooth root (single peak loads) for external gearing. Figure 7.3/6a can be used for the partitioning according to the ideal tooth root endurance strength. The corresponding relative optimum Y F1 = YF2 (YFa1 = Y Fa2) or respectively YFS1 = Y FS2 of a given sum Ȉx is found in the same way, as depicted in Figure 7.3/5c, with the partially curved lines at YF or YFS linearised. 476 7 Design of gearing and gear transmissions Figure 7.3/5 b) Partitioning of the sum of the profile shift factors on pinion and gear for speed- decreasing ratios Approach to determine x 2, x1 using the example: z 1 = 23; z2 = 52; Ȉx = 1.25: 1. Determine the point P (Ȉz/2, Ȉx/2) in the diagram 2. Place the interpolation line through the found point 3. Read the profile shift factor x 1 on the ordinate with the help of z1 and the interpolation line 4. x2 = Ȉ(x–x1) Figure 7.3/5 c) Partitioning of the sum of the profile shift factors on pinion and gear for a speed- increasing ratio For the partitioning of Ȉx for YF1 = YF2 or Y FS1 = Y FS2 it must be noted that, with respect to scuffing, the sliding conditions are not favourable in cases of speed-increasing ratios. Deviations from the relative optimum (same tooth root stress) can sometimes make sense in the range of limited life strength because of the higher number of load cycles of the pinion or also in the endurance limit range due to the consideration that the gear is the larger of the components, and for that reason it often has a lower strength (lower tempered strength or core hardness). Scuffing load capacity Based on mathematical tests specifically designed for high scuffing load capacity (integral temperature method), Equation (7.3/6) resulted as a guide for the selection of the sum Ȉx: ( 0.0123u 0.538) 1 2 opt. n1 ( ) (0.0718 0.2448)−++= ⋅ + ⋅xx u z (7.3/6) valid for 14 zn1 50; 1 u 6; 0 ȕ 30 7.3 Selection and Partitioning of Profile Shift 477 Figure 7.3/6 Partitioning of the sum of the profile shift for external gearing in cases where the maximum load capacity of the tooth root is required (h f /mn = 1.4; spr /mn = 0.02; Na0/mn = 0.4; basic rack acc. to ISO 53, DIN 867) a) Strain in the endurance limit range Figure 7.3/6 b) Strain in the low- cycle range (N 10 4); basic rack according to ISO 53, DIN 867) Equation (7.3/7) or respectively (7.3/8) can be used for the partitioning . For the speed- decreasing ratio (gear 1 driving; 14 zn1 50; 1 (u = | iStage |) 6; 0 ȕ 25 ): 2 11 12 3 ()exe u xx e=⋅ ⋅ + + (7.3/7) 0.0045 1n 1 2 2n 1 n 1 1.1408 0,017 3n 1 n1with e 0.4916 z e 0.0003 z 0.0222 z 1.0322 e 0.1075 0.0007 z 4.386 z ln (u) u−=⋅ = − ⋅+ ⋅− =− ⋅ + ⋅ ⋅ ⋅ For the speed- increasing ratio (gear 2 driving; 14 zn1 50; 1 (u = 1/| iStage|) 6; 0 ȕ 25 ): []u 3 2 1 1 1 ) (2X e x x u e xe⋅ + + ⋅ ⋅ = (7.3/8) 478 7 Design of gearing and gear transmissions stage 2 1 0.0261 1n 1 2 2n 1 n 1 0.7231 3n 1 n 1 2 uwith 1/ | i | u z / z e 0.4705 z e 0.0001 z 0.0058 z 0.605e 0.0007 z 0.0257 1.301 z ln (u) 8.134X 0.0023 u 1.022 u−== =⋅ =⋅ +⋅ − =− ⋅ − + ⋅ ⋅ =− ⋅ − + After determining x 1 from Equation (7.3/7) or Equation (7.3/8), and with Ȉx, x2 is also fixed: x2 = Ȉx – x 1 (7.3/9) Helical gearing can be treated approximately as spur gearing with znx. The approximations [Equations (7.3/6) to (7.3/8)] are based on experience, according to which differences in the friction coefficient exist with respect to the influence of pushing and pulling friction. That is why CE AC/μ μ = 1.2 was used. The selection and partitioning of the profile shift in accordance with Equations (7.3/6) to (7.3/8) lead also to favourable values for tooth root and pitting load capacity. INTERNAL GEAR PAIRS Besides checks to avoid interference (Section 3.3), for internal gear pairs the following rules should be noted as a guide. Speed-decreasing ratio The sliding speed at the tip of the external gearing should be somewhat higher than at the tip of the internal gearing: /gE (1.2 to 1.5) /gA (7.3/10) The profile shift of the external gearing should be x1 > 0. Speed-increasing ratio The sliding speed /gE at the tip of the internal gearing should be the same or higher than at the tip of the external gearing: /gE (1.0 to 1.2) /gA (7.3/11) The profile shift of the external gearing should be x 1 > 0. With regard to the optimisation of the load capacity of the tooth root, more comprehensive analyses are necessary for internal gearing due to the influence of the gear rim material, which is usually different, and its different kind of heat treatment. Principally the following must be examined in every design of gearing for external and internal gears: • Interference (primarily internal gear pairs) • Tooth tip thickness (external gearing) • Tooth root space width (internal gearing) • Undercut (external gearing) • Transverse contact ratio 7.4 Selection of Materials: Strength Values 479 For external gearing (external gear pairs) when tip easing is applied to a constant tip clearance at gear ratios of 1 z2/z1 6, the tooth tip thickness comes not below the minimum. This risk exists solely in cases of very large numbers of teeth of the mating gear (rack), because then the tip shortening coefficient is k 0. Furthermore, this applies to cases of internal gear pairings, be- cause here k results as positive at Ȉx > 0 for a constant tip clearance, and usually k = 0 is set. 7.4 Selection of Materials : Strength Values 7.4.1 Basics of Selecting Materials and Basic Values for Material Strength In order to develop and manufacture high-capacity gear drives, it is necessary to possess a com- prehensive understanding of the strain affecting gears and of the materials which can be used for gears and how to enhance the properties of these materials through bulk heat treatment and surface-layer hardening (heat treatment, coating, and mechanical hardening). The strength values strived for in the final state are intrinsically connected to the chosen material and its treatment and hardening. Since the maximum of the complex strain on the teeth is to be found near the surface, the case and the core (Table 7.4/1) must be subject to special requirements. Table 7.4/1 Strain and Material Properties, Based on [7.4/1] Effect of the strain Requirements Fatigue Fatigue (fatigue fracture), contact fatigue/surface layer fatigue (pitting, micropitting) High endurance limit with the highest possible ductility in the case region in particular, high structural homogeneity (ferrite-free), continuous drop in state of hardness/strength from the sur- face to the core, residual compression stresses in the surface layer Plastic deformation (up to overload fracture) High static strength with high ductility, structural homogeneity Wear Abrasion (wearing away) High hardness and sufficient ductility; structure with hard phases (e.g., carbide, nitride) in a basic solidifying material Adhesion (scuffing) Dissimilar case with low adhesive binding strength (low predisposition to scuffing); i.e., avoidance of “purely” metal-metal pairing, good running-in characteristics (contact pattern) Tribo-oxidation (wearing away) Reaction-resistant protective layer; e.g., oxidation layer Note: Large-area wear can also be caused by fatigue, micro-cutting, crenation and chemical reactions. For ferrous materials the various surface-layer hardening methods listed in Table 7.4/2 are suitable. The methods relevant for gears are explained in Section 10.3.2. Steels which have not been subjected to bulk heat treatment or surface-layer hardening are only used in cases of low strain. To achieve an optimum ratio of performance to mass at high load conditions, steels and their hardening treatments are of importance. General design rules for proper heat treatment have long existed [7.4/2]. To guarantee the desired mechanical properties, the rules ensure that distortion through heat treatment is minimised and crack formation is ruled out. From this it is also possible to derive some recommendations for designing gears. Information regarding the proper design of gears for heat treatment can often be found in connection with older measurements of distortions (e.g., [7.4/3] to [7.4/6]). On this matter, results pertaining to gears are also to be expected from research in the DFG special research field 570 “Distortion Engineering” [7.4/7]. 480 7 Design of gearing and gear transmissions For a given design and a selected material, an optimal heat treatment suitable for the design (e.g., [7.4/2]) can also contribute to minimise distortion and prevent cracks. When ordering materials, it is the task of the designers and engineers to decide whether the material should be delivered in a condition where the microstructure is ready for manufacturing or, in cases where high requirements exist in the company itself or on its behalf, heat-treatment processes suit-able or necessary for manufacturing should be integrated into the manufacturing process. The goals when selecting the material and applying bulk heat treatment and surface-layer hardening are • to select an economical material, which fulfils all the requirements in terms of design and manufacturing, and • to select one or more treatment processes, through which the manufacturing costs are kept to a minimum and through which hardening is achieved that is appropriate for the strain through suitable structural characteristics, to rule out potential damage. Therewith the load capacity necessary for the tooth flank, the tooth root and the entire cross section is to be ensured. The basic principles of this can be found in standards which have existed since 1996 (1987), particularly ISO 6336-1 to 3; 5; 6 (DIN 3990-1 to 3; 5; 6). There are only slight differences between ISO 6336 and DIN 3990. The many individual considerations from ISO 6335-5 (DIN 3990-5) are stated in a very simplified form in the heat-treatment information on the part drawing (refer to Section 9.3). It is mandatory that the technical content of this information entered by the designer is coordinated with the technical engineers and manufacturing specialists. 7.4.1.1 General Basic Principles According to the general principles of selecting the material [7.4/8, 7.4/9] and the basic principle for calculating the strength, for gears, specifically, a level of strength must be guaranteed by the material (with or without treatment appropriate for the strain) that exceeds the external strain with a sufficient degree of probability (flank: σ HP > σH; root: σFP > σ F; cross section: Rp0.2/SFStmin > σFmax) at every position on the tooth and the gear body. The selection of materials associated with the development of gear transmission engineering and the suitable treatment methods have changed over the years. New combinations of materials and treatments have been developed for which there is still much to do in the research of their properties. For gears made of ferrous materials 1) case-hardened steels in various conditions of thermo- chemical hardening and tempering steels dominate, with the increasing trend towards thermal and thermochemical surface-layer hardening. Coatings produced by chemical vapour deposition (CVD) and physical vapour deposition (PVD) are used. The higher quality of material production is used, for example, to make very pure steels and micro-alloyed steels available. Newly developed steels or steels with special properties are used in combination with enhanced and new hardening methods. The selection of materials for gears made of ferrous materials was standardised for the first time in the manufacture of industrial gear transmissions in 1971 [7.4/10]. In analyses of gear tests in running and pulsator tests for various groups of materials, ISO 6336-5 (DIN 3990-5) contains ranges of pitting endurance limit σ Hlim and tooth root endurance limit σFlim or respectively tooth root strength σFE. 1) Ferrous materials: Iron-carbon alloys with a carbon content of up to 2% are called steels; materials with more than 2% carbon content are called cast iron. 7.4 Selection of Materials: Strength Values 481 482 7 Design of gearing and gear transmissions The groups of materials can be divided into • steels, iron casting materials and cast iron materials, which are used primarily as delivered or in a normalised or quenched and tempered condition, without surface-layer hardening, and • steels and iron casting materials, which are used in a surface-hardened condition as depicted in Tables 7.4/3 a, b. Appendix 7.3 contains a comparison of international designa-tions for materials. Characteristic for each group are heat-treatment conditions that are manufacturing- specific/appropriate or appropriate for the strain. The final conditions appropriate for the strain are roughly specified by the “Requirements on the quality of materials and heat treatment” in the three levels of ML (“Low requirements on material and heat treatment”), MQ (“Requirements which can be satisfied by experienced manufacturers with adequate cost expenditures”) and ME (“Requirements which can be satisfied by experienced manufacturers only with very high cost expenditures”) in accordance with ISO 6336-5 (DIN 3990-5). In diagrams, the lines of the endurance limit are allocated to the three levels, depending on the surface hardness. In all cases as follows the level MQ is referenced. The selection of materials for steels and iron casting materials is based on German standards. The conversion of most national material standards (e.g., DIN) into EURO standards (DIN EN) has led to new material designations and codes (refer to DIN EN 10027-1, DIN EN 10027-2, DIN EN 1560). For gears without surface-layer hardening, in terms of the material itself, the hardness of the tooth is decisive for the endurance limit. Excerpts of the strength/hardness of structural steels in accordance with DIN EN 10025, unalloyed tempering steels in accordance with DIN EN 10083-2, open-die forgings made of unalloyed stainless steels in accordance with DIN EN 10250-2, and unalloyed cast steel in accordance with DIN EN 10293 (all in normalised condition), as well as the unalloyed types of cast iron in cast condition in accordance with DIN EN 1561, DIN EN 1562 and DIN EN 1563, are specified in Tables 7.4.4a to f, where it should be noted that the values depend on the different characteristic dimensions (e.g., as nominal thickness, diameter d, thickness t , thickness of the decisive cross section t R, diameter of the sample d, and decisive wall thickness t). With their given chemical composition, the strength/hardness after the quenching and tempering of unalloyed and alloyed tempering steels in accordance with DIN EN 10083, open-die forgings made of unalloyed and alloyed stainless steels in accordance with DIN EN 10250, and alloyed cast steel in accordance with DIN EN 10293 is especially influenced by the representative dimension s (refer to Figure 7.4/4; see “decisive heat treatment cross section” in accordance with DIN EN 10083-1), the quenching during hardening, and by the tempering temperature. Specific statements about this require knowledge of the hardenability. This includes the heat-treatment terms for ferrous materials in accordance with DIN EN 10052 (standard undergoing revision, replacement by DIN ISO 4885 planned), the determination of hardenability in the end-face quenching test in accordance with DIN EN ISO 642, knowledge of hardenability scatter bands, for example, in accordance with DIN EN 10083-2 and -3, the calculation of hardenability from chemical analysis in accordance with SEP 1664, and knowledge of conversion characteristics [7.4/11] and of heat treatability [7.4/12]. Information on the selection of steel based on the hardenability can be found in DIN 17021-1. Because of the lower material costs for gears made of structural steels in accordance with DIN EN 10025 and unalloyed cast iron in the cast state in accordance with DIN EN 1561, DIN EN 1562 and DIN EN 1563 (refer to Table 7.4/2), it is common in actual practice to thermally or thermochemically harden the surface layer. 7.4 Selection of Materials: Strength Values 483 Table 7.4/3a Material Groups and Heat Treatment Conditions for Gears without Surface-Layer Hardening Possible stress-oriented heat treatment (final state) normalised, normalising rolled, thermo-mechanical rolled precipitation hardened normalised (unalloyed steels), quenched and tempered (unalloyed and alloyed steels) normalised (unalloyed steels), quenched and tempered (unalloyed and alloyed steels) normalised (unalloyed cast steel), quenched and tempered (unalloyed and alloyed cast steels) in cast condition ferritic-perlitic, perlitic, quenched and tempered possible in cast condition undecarburising annealed, pear- litisised, quenched and tempered possible in cast condition ferritic to pearlitic, pearlitisised, quenched and tempered, austempered in cast condition bainitic-ferritic 1) From hot forming temperature precipitation hardening ferritic-perlitic steels Possible heat treatment suitable or necessary for production (prior or intermediate state) untreated, normalised, normalising rolled, thermo-mechanical rolled, stress relief annealed untreated, precipitation hardened, stress relief annealed normalised, soft annealed, stress relief annealed soft annealed, quenched and tempered, stress relief annealed normalised, stress relief annealed, quenched and tempered in cast condition ferritic-pearlitic or pearlitic, stress relief annealed in cast condition undecarburising annealed, stress relief annealed in cast condition ferritic to pearlitic, stress relief annealed in cast condition bainitic-ferritic, stress relief annealed Standards DIN EN 10025-2 to DIN EN 10025-4 DIN EN 10267 DIN EN 10083-2 DIN EN 10083-3 DIN EN 10250-2 to DIN EN 10250-4 DIN EN 10293 DIN EN 1561 DIN EN 1562 DIN EN 1563 DIN EN 1564 Group of material with non-hardened surface structural steels AFP steels 1) quenched and tempering steels open-die forgings out of steel for common use cast steel for common application cast iron with lamellar graphite malleable iron, black cast iron with spheroidal graphite bainitic cast iron 484 7 Design of gearing and gear transmissions Table 7.4/3b Groups of Material and Heat Treatment Conditions for Thermal and Thermo-chemical Surface-Hardened Gears Possible stress-oriented heat treatment (final conditions ) precipitation hardened, surface hardened, nitrided quenched and tempered, austempered, surface hardened, nitrided, carbonitrided, boron-treated quenched and tempered, surface hardened, nitrided, carbonitrided, boron treated nitrided, nitrided + hard coated (PVD, CVD) case hardened, carbonitrided, nitrided, hard layer coated (PVD, CVD) quenched and tempered, surface hardened, nitrided, carbonitrided, boron-treated pearlitic hardened, quenched and tempered, surface hardened, nitrided pearlitic hardened, quenched and tempered, surface hardened, nitrided pearlitic hardened, quenched and tempered, austempered, surface hardened, nitrided austempered, surface hardened, nitrided 1) from hot forming temperature precipitational hardening ferritic-pearlitic steels 2) alternative: soft annealed to maximum hardness requirements or treated for hardness range 3) alternative: treated for ferritic-pearlitic structure and hardness range Possible heat treatment suitable or necessary for production (pre or interned conditions ) untreated, precipitation hardened, stress relief annealed untreated, normalised, soft annealed, quenched and tempered, stress relief annealed normalised, soft annealed, quenched and tempered, stress relief annealed soft annealed, quenched and tempered untreated, soft annealed 2), normalised 3), isothermal annealed, stress relief annealed normalised, quenched and tempered, stress relief annealed in cast condition ferritic-pearlitic or pearlitic, stress relief annealed in cast condition undecarburising annealed, stress relief annealed in cast condition ferritic to pearlitic, stress relief annealed in cast condition bainitic-ferritic, stress relief annealed Standards DIN EN 10267 DIN EN 10083-2 DIN EN 10083-3 DIN EN 10250-2 DIN EN 10250-3 DIN EN 10085 DIN EN 10084 DIN EN 10293 DIN EN 1561 DIN EN 1562 DIN EN 1563 DIN EN 1564 Group of material with hardened surface AFP steels 1) quenched and tempered steels open-die forgings out of steel for common use nitriding steels case hardening steels cast steel for common use cast iron with lamellar graphite malleable iron cast iron with spheroidal graphite bainitic cast iron 7.4 Selection of Materials: Strength Values 485 7.4.1.2 Selection of Steel according to Hardenability Hardenability means the steel’s capability to increase and deepen its hardness. Hardening capac- ity is the highest hardness achievable through optimal quenching. Hardness deepening is the change in hardness through the cross section, which is confirmed with a hardness depth if applicable. The hardenability is determined in the end-face quenching test (Jominy test) (DIN EN ISO 642). The sampling, dimensions, heat treatment, normalising and processing are specified in detail. The cylindrical sample of 25 mm in diameter and 100 mm in length is heated to the hardening temperature, standardised according to the type of steel, and quenched by a water jet in a fixture, hanging from the end face. After cooling down, the sample is ground in the axial direction along two lines offset by 180 to a depth of 0.4 to 0.5 mm. On these areas the Rockwell hardness is measured at defined distances from the end face. Figure 7.4/1 depicts the test principle. The hardness curve along the distance from the end face leads to the so-called end-face quench curve (Jominy curve). Figure 7.4/1 Schema of the end-face quenching test in accordance with DIN EN ISO 642: 1 Fixture to centre the sample 2 Sample in the holder 3 Cover 4 Opening of the water feed pipe 5 Quick shut-off tap 6 Water feed pipe The standards list the hardenability scatter bands for all of the case-hardened steels and tempering steels (refer for example to Figure 7.4/2b) according to the spread range of the chemical analysis. They reflect a material property that is independent of the technological conditions. For a “comparative selection of materials” according to hardenability, the correlations between diameter and distance from the end face in accordance with [7.4/13] (refer to DIN EN 10083-1) and their extension to gears and pinion shafts can be used. Figure 7.4/2 shows how the distribution of hardness on the end-face quenching sample can be estimated using the diagram at [7.4/13]. Figure 7.4/2a represents the relationship between the diameter of the cylindrical bar and the distance from the end face for the quenching intensity H = 0.4, which corresponds to quenching in oil. The quenching intensity (thermal flow equivalent) [7.4/14] is a measure (dimension 1/inch) for the quenching effect of the respective medium. The H values range from 0.02 (inactive air) through 0.3 to 0.8 (oil), and 1 to 6 (water), up to 7.5 (brine) [7.4/13]. With this it is possible to make comparative assessments of the quenching effect of liquid media. The processes during heating and cooling down are explained in Section 10.3. The rising sets of curves depict the distance from the surface of a sh aft to its centre (m easured in half shaft length). Points of intersection result with the dropping sets of curves, from which, with reference to the distance from the quenched end face, hardness values can be read. Figure 7.4/2b shows the projection on to the hardenability scatter band of the steel 42CrMo4 for a cylindrical rod 100 mm in diameter. According to Figure 7.4/2b, at minimum hardenability of the steel 42CrMo4 (lower limiting curve), the maximum hardness possible is 53 HRC at a distance 486 7 Design of gearing and gear transmissions from the end face of 1.5 mm. The ratio of the measured hardness to the maximum hardness possible, which is determined only by the carbon content, is the degree of hardening R; here R = 1 (see also Section 7.4.2.4). With 37 HRC, at half the length l (l ≥ 3d), the surface of the shaft has the same hardness as at a distance from the end face of 15.2 mm, R = 0.70. At a depth of 12.5 mm below the surface, which corresponds to the sampling location according to DIN EN 10083-1 of a shaft for tensile and notch impact samples, at 32 HRC the same hardness exists as at a distance from the end face of 25.5 mm, R = 0.60. With 30 HRC, at the centre of the shaft the hardness is equal to the hardness at a distance from the end face of 40 mm, R = 0.57. The distribution of hardness determined this way after quenching with oil is rendered in Figure 7.4/3 along the shaft cross section at half the length of the shaft. Figure 7.4/2 Relationship between shaft diameter, quenching, distance from the end face of the end-face quenching sample and th e hardenability scatter band; readings: example with D = 100 mm; a) diameter D = f(distance from the end face), quenching intensity H = 0.4 (moderately moving component in motionless oil); b) hardenability scatter band of the steel 42CrMo4+H, DIN EN 10083-3 ( R degree of hardening); dashed curve: end-face quench curve for the chemical composition as per Figure 7.4/5 The very wide range of variation in hardness specific to the material can occur in actual practice if parts made of different batches of melted mass are heat-treated together. Figure 7.4/3 Distribution of hardness after quenching in oil along the cross section of a shaft of 100 mm in diameter made of the steel 42CrMo4, determined from the hardenability scatter band in accordance with DIN EN 10083-3; dashed curve: distribution of hardness according to the hardenability for the chemical composition as per Figure 7.4/5 In actual practice, for a certain batch of melted mass the hardenability is determined either in a test or is calculated mathematically. For this there is a simple regression [7.4/15], or, for example, the regression equations available for unalloyed tempering steels (carbon steels) and for chrome- alloyed and chrome-molybdenum-alloyed tempering steels are used to calculate hardenability in accordance with SEP 1664 [7.4/16]. 7.4 Selection of Materials: Strength Values 487 Since it is rare for toothed parts to take the form of a smooth shaft, the concept of the characteristic dimension s was defined [7.4/10, Sheet 1] as an approximation and illustrated by examples, which Figure 7.4/4 shows for shaft and wheel-shaped parts. For the distinguishing dimension, ISO 6336-5 introduced the term “controlling section” and also illustrated it with examples. Figure 7.4/4 Examples of the distinguishing dimension for shaft and wheel-shaped parts (according to [7.4/10], [Bl.1]) As such, the distinguishing dimension represents the link to the selection of materials on the basis of the hardenability. The diagrams at [7.4/13] provide information for estimating the hardness in the teeth of shafts and wheels. DIN EN 10052 defines cooling as “lowering the temperature of a workpiece” and quenching as a “heat-treatment step in which a workpiece is cooled at a higher speed than in inactive air”. The ideal goal is to be able to determine the hardness, residual stresses and the size and form changes (distortion) at arbitrary points for a specific gear form out of the time-temperature curve during quenching and the resulting structures. Research by the Computer Aided Simulation of Heat treatment (C.A.S.H.) project, particularly in the area of case hardening, identifies ways to achieve this specified goal through experimental and mathematical means [7.4/17]. The general requirement of having a homogeneous structure of the highest possible strength and ductility means a ferrite-free structure in the area of the teeth (ISO 6336-5: a maximum of 10% ferrite for tempering steels or respectively no ferrite for surface-hardened steels). In continuous time-temperature-transformation diagrams [7.4/11], information is available about the structure being produced during quenching and the associated hardness. 488 7 Design of gearing and gear transmissions Figure 7.4/5 shows the continuous time-temperature-transformation dia- gram for the steel 42CrMo4. As with any other time-temperature-transfor- mation diagram , it only applies for the given chemical composition. The properties of the structure produced by the transformation during cooling down to room temperature are deter-mined along the cooling curves indi-cated in the diagram and defined using the cooling time t 8/5 from 800 to 500 C. Figure 7.4/5 Continuous time-temperature- transformation diagram for the steel 42CrMo4 (melt 5 from [7.4/11, Vol. 1]) The fourth curve from the left will be chosen as an example. The cooling time t8/5 can be determined at approximately 40 s. After the steel cools to above 600 C, the structure field F (ferrite) is touched. When leaving the field the number “2” indicates that 2% ferrite has formed up to that point (slightly below 600 C). At slightly above 300 C, 75% bainite (f ield B) has formed. The rest of the share of the austenitic structure (field A) turns into martensite beginning at the martensite starting point M s. With the appropriate chemical composition, at room temperature the steel 42CrMo4 now has a structural mixture composed of 2% ferrite, 75% bainite and 23% martensite, and a hardness of 34 HRC (circle on the x axis of the diagram). For the terms austenite, bainite, ferrite and martensite, refer to DIN EN 10052. An analysis of this cooling curve shows that the requirement for <10% ferrite has been satisfied, but to achieve a ferrite-free structure it would be necessary to cool faster. A good approximation of the cooling time from 800 to 500 C and the associated hardness values after continuous cooling can be found for 100% martensite and ferrite-free structure using a regression analysis of more than 50 time-temperature-transformation diagrams of primarily low-alloyed structural steels [7.4/18]. For the “comparative selection of materials” the lower limit of the hardenability curve (refer to Figure 7.4/2b) should be used as a guide to reliably achieve the specifications for structure and hardness. Here the same principle is applied as for the endurance limit values according to ISO 6336-5 (failure probability 1%). There test gears were deemed to have failed by pitting when one of the following conditions is met: when 2% of the total working flank area of through-hardened gears, or when 0.5% of the total working flank area of surface-hardened gears, or 4% of the working flank area of a single tooth, is damaged by pitting. The percentages refer to test evaluations; they are not intended as limits for product gears (ISO 6336-5). 7.4.2 Materials for Gears without Surface-Layer Hardening: The Basics 7.4.2.1 Gears Made of Unalloyed Steels and Cast Steel and Unalloyed Cast Iron in Cast Condition Potential candidates in a normalised condition include structural steels in accordance with DIN EN 10025, unalloyed tempering steels in accordance with DIN EN 10083-2, open-die forgings made of unalloyed stainless steels in accordance with DIN EN 10250-2, unalloyed cast 7.4 Selection of Materials: Strength Values 489 steel in accordance with DIN EN 10293, and unalloyed types of cast iron in cast condition in accordance with DIN EN 1561, DIN EN 1562 and DIN EN 1563. Structural steels in accordance with DIN EN 10025-2 The steels St50-2, St52-3N, St60-2 and St70-2, which have been common until now, were named according to their minimal tensile strength. The new designations are based on the minimal yield stress and are called E295 (E for steels for machine construction; 295 is the minimum yield stress in MPa), S355J0+N (S for steels for steel construction; J0 means 27J notch impact energy at 0 C, +N normalised), E335 and E360. The mechanical properties for the minimum yield stress and the tensile strength (according to DIN EN 10025-2, tables 7 and 8) are shown in Table 7.4/4a. Values for minimal fracture elongation for the steels E295, E335 and E360 are also listed in DIN EN 10025-2, Table 8, but values for notch impact energy in contrast are not. The conversion from tensile strength R m to the Brinell hardness (HBW) can be done according to DIN EN ISO 18265, Table A1, using the equation hardness H [HBW] = 0.295 Rm. The calculation of guide values for the pitting endurance limit σHlim and the tooth root strength σFE for the material quality MQ can be done according to Appendix 12. Refer to DIN EN 10027-1 for steel designations. Unalloyed tempering steels, normalised according to DIN EN 10083-2 Unalloyed tempering steels are delivered as quality steels or stainless steels. There is no differ- ence between them in terms of the mechanical properties of minimum yield stress R e, minimum tensile strength Rm and minimal fracture elongation A. All quality steels contain up to 0.045% sulphur. Stainless steels are called, for example, C35E (maximum sulphur content 0.035%) and C35R (sulphur content ranging from 0.020 to 0.040%). Table 7.4/4b contains the mentioned mechanical properties according to diameter d or respectively thickness t. Values for notch impact energy are not listed. The conversion from tensile strength R m to the Brinell hardness (HBW) can be done according to DIN EN ISO 18265, Table A1, according to the equation hardness H [HBW] = 0.295 R m. The calculation of guide values for the pitting endurance limit σHlim and the tooth root strength σFE for the material quality MQ can be done according to Appendix 12. Refer to DIN EN 10027-1 for steel designations. Open-die forgings made of unalloyed steel, normalised as well as normalised and tempered according to DIN EN 10250-2 Just as in the case of unalloyed tempering steels, open-die forgings are delivered as quality steels or stainless steels (E qualities with lower S content), and do not differ from one another in terms of the mechanical properties of minimal yield stress R e, minimal tensile strength Rm and minimal fracture elongation A . In contrast with unalloyed tempering steels, however, values are given for notch impact energy. Table 7.4/4c shows the mechanical qualities according to the thickness of the decisive cross section t R. According to DIN EN 10020, the steels 28Mn6 and 20Mn5 are considered unalloyed steels. The conversion from tensile strength R m to the Brinell hardness (HBW) can be done according to DIN EN ISO 18265, Table A1, according to the equation hardness H [HBW] = 0.295 R m. The calculation of guide values for the pitting endurance limit σHlim and the tooth root strength σFE for the material quality MQ can be done according to Appendix 12. Refer to DIN EN 10027-1 for steel designations. 490 7 Design of gearing and gear transmissions Table 7.4/4b Mechanical Properties 1) at Room Temperature of Unalloyed Quenched and Tempered Steels, Normalized (+N) according to DIN EN 10083-2,Table 10 Mechanical properties for products with the diameter d or for flat products with thickness t of 100 < d 250 mm; 100 < t 250 mm Amin [%] - 19 17 16 14 12 11 18 1) Re: upper yield strength or if no distinctive yield strength exists, then the 0.2% yield strength Rp0.2 2) The steels C35 and C60 without additional symbols are quality steels; the steels with additional symbols E (mandatory max. S c ontent) and R (mandatory range of S content) are special steels 3) For 100 < d 250 mm; 100 < t 250 mm no values are given Rm min [MPa] - 500 530 560 590 620 650 590 Re min [MPa] - 245 260 275 290 300 310 290 16 < d 100 mm; 16 < t 100 mm Amin [%] 25 19 17 16 14 12 11 18 Rm min [MPa] 410 520 550 580 610 640 670 600 Re min [MPa] 210 270 290 305 320 330 340 310 d 16 mm; t 16 mm Amin [%] 24 18 16 14 13 11 10 17 Rm min [MPa] 430 550 580 620 650 680 710 630 Re min [MPa] 240 300 320 340 355 370 380 345 Steel type 2) C22E, C22R 3) C35E, C35R C40E, C40R C45E, C45R C50E, C50R C55EW, C55R C60E, C60E 28Mn6 Table 7.4/4a Mechanical Properties of Structural Steel according to DIN EN 1 0025-2, Table 7 and 8 Tensile strength Rm [MPa] Distin guishin g thickness [mm] > 150 250 440 to 610 450 to 600 540 to 710 640 to 830 1) E: steels for mechanical engineering without values for the notch impact energy; S: steels for steel work with values for the notch impact energy 2) Former denomination according to DIN 17100 > 100 150 450 to 610 450 to 600 550 to 710 650 to 830 > 3 100 470 to 610 470 to 630 570 to 710 670 to 830 < 3 490 to 660 510 to 680 590 to 770 690 to 900 Minimum yield strength Re min [MPa] > 200 250 225 275 255 285 > 150 200 235 285 265 295 > 100 150 245 295 275 305 > 80 100 255 315 295 325 > 63 80 265 325 305 335 > 40 63 275 335 315 345 > 16 40 285 345 325 355 16 295 355 335 360 Steel type according to DIN EN 10027-1 1) E295 (St50) 2) S355J0+N (St 52-3 N) 2) E335 (St 60) 2) E360 (St 70) 2) 7.4 Selection of Materials: Strength Values 491 Table 7.4/4c Mechanical Properties of Open-die Forgings of Unalloyed Steels in Normalized Status (+N) and Normalized and Tempe red Status (+NT), Status according to DIN EN 10250-2 Thickness of distinguishing cross section tR 250 < tR 1000 mm 2) 3) KV [J] min. tR - - - 15 - 12 - 10 - - - 15 27 1) The notch impact bending tests have to be carried out at -20 C 2) l, tR in longitudinal and lateral direction 3) For 20Mn5 a maximum of tR = 750 mm is valid l - - - 20 - 17 - 12 - - - 20 40 A [%] min. tR - - - 16 - 14 - 11 - 8 7 11 20 l - - - 22 - 18 - 15 - 11 10 17 22 Rm [N/mm2] min. - - - 390 - 470 - 530 - 590 620 540 490 Re [N/mm2] min. - - - 180 - 210 - 230 - 250 260 260 250 250 < tR 500 mm 2) KV [J] min. tR 15 15 15 15 - 12 - 10 - - - 15 27 l 27 27 27 25 - 20 - 15 - - - 25 40 A [%] min. tR 17 17 12 17 - 15 - 12 - 9 8 12 20 l 23 23 18 23 - 19 - 16 - 12 11 18 22 Rm [N/mm2] min. 340 340 450 400 - 480 - 540 - 600 630 540 500 Re [N/mm2] min. 165 165 265 190 - 220 - 240 - 260 275 270 260 100 < tR 250 mm 2) KV [J] min. tR 20 20 20 20 - 15 - 10 - - - 20 35 l 30 30 30 30 - 25 - 18 - - - 30 50 A [%] min. tR 17 17 12 17 - 15 - 12 - 9 8 12 20 l 23 23 18 23 21 19 17 16 14 12 11 18 22 Rm [N/mm2] min. 340 340 450 420 460 500 530 560 590 620 650 570 520 Re [N/mm2] min. 175 175 275 210 230 245 260 275 290 300 310 290 280 Steel type S235JRG2 1) S235J2G3 1) S355J2G3 1) C25, C25E C30 C35, C35E C40 C45, C45E C50 C55, C55E C60, C60E 28Mn6 20Mn5 492 7 Design of gearing and gear transmissions Cast steel for general use, unalloyed, normalised according to DIN EN 10293 There are five types available. The GS qualities differ from the GE qualities because of lower P and S contents and because of higher values for notch impact energy (refer to Table 7.4/4d). The material G28Mn6 is considered as unalloyed. Table 7.4/4d Mechanical Properties of Unalloyed Cast Steel for General Use, Normalised (+N) according to DIN EN 10293 Name Thickness Mechanical properties Tension test at room temperature Notch impact test 1) t mm Rp0.2 min. MPa Rm MPa A min. % KV min. J Temperature 2) C GE200+N (GS-38) 3) 300 200 380 to 530 25 27 RT GS200+N 100 200 380 to 530 25 35 RT GE240+N (GS-45) 3) 300 240 450 to 600 22 27 RT GS240+N 100 240 450 to 600 22 31 RT GE300+N (GS-60) 3) 30 300 600 to 750 15 27 RT 30 < t 100 300 520 to 670 18 31 RT G20Mn5 (GS-20Mn5) 4) 30 300 480 to 620 20 27 -30 50 RT G28Mn6 250 260 520 to 670 18 27 RT 1) If two values are given for the notch impact energy, the buyer must specify which value is required 2) RT means room temperature 3) Former name according to DIN 1681 4) Former name according to DIN 17182 The conversion from tensile strength Rm to the Brinell hardness (HBW) can be done according to DIN EN ISO 18265, Table A1, according to the equation hardness H [HBW] = 0.295 R m. The calculation of the guide values for the pitting endurance limit σHlim and the tooth root strength σFE for the material quality MQ can be done according to Appendix 12. Refer to DIN EN 10027-1 for steel designations. Lamellar-graphite cast iron in accordance with DIN EN 1561 DIN EN 1561 contains six sorts with values for tensile strength in large ranges with the designations such as EN-GJL-200 for separately cast samples, for integrally cast samples, and for anticipated values for the casting (DIN EN 1561, Table 1). Aside from that, there are six sorts with designations, for example, EN-GJL-215HB with values for Brinell hardness in large ranges. Moreover, for both sorts, areas of decisive wall thickness exist (DIN EN 1561, table 2), whereby tensile strength or Brinell hardness increases as wall thickness decreases. For a structure with a maximum of 5% ferrite, ISO 6336-5 (DIN 3990-5) requires in Table 3 for the quality MQ only information on the Brinell hardness, without prescribed graphite development. Information about the structure for sorts with the name based on the tensile strength, for example, EN-GJL-200, is provided in DIN EN 1561 Table A1, “Mechanical properties in separately cast samples with a cast diameter of 30 mm”. According to that, all sorts from EN-GJL-200 to EN-GJL-350 have a pearlitic structure. With respect to the specific gear blank, the complexity of these interrelated issues always requires coordination between the manufacturer of the cast part and the user. 7.4 Selection of Materials: Strength Values 493 In order to achieve high strength, lamellar-graphite cast iron can be quenched and tempered. Through surface-layer strengthening in combination with quenching and tempering, higher load capacity values can be achieved. Potential procedures are nitriding/nitrocarburising and surface-layer hardening. No parameters are available for this. Refer to DIN EN 1560 for material designations. Malleable iron in accordance with DIN EN 1562 For gears, only “black heart” malleable cast iron (non-decarburising annealed), which is heat treated by the manufacturer, with designations such as EN-GJMB-500-5, is considered. The two numbers describe tensile strength and strain. DIN EN 1562 specifies nine sorts, of which, because of the high ferrite content, only the sorts starting at EN-GJMB-550-4 with hardnesses above 180 HBW are appropriate. With increasing tensile strength, yield point (0.2%) or hardness (refer to Table 7.4/4e), the strain decreases. Verification of the yield point (0.2%), hardness, ductility and impact resistance must be arranged with the manufacturer. The use of black heart malleable cast iron is recommended only for small gears with low load and with wall thicknesses of 3 to 10 mm (max. 60 mm). Through quenching and tempering combined with the subsequent nitriding or surface-layer hardening, the gear load capacity can be significantly raised compared to the cast state. The calculation of guide values for the pitting endurance limit σ Hlim and the tooth root strength σFE for the material quality MQ in cast state can be done according to Appendix 12. Refer to DIN EN 1560 for material designations. Table 7.4/4e Mechanical Properties 1) of Non-carburising Annealed Malleable Iron, in Accordance with DIN EN 1562 Material name Sample diameter (nominal size) d mm 2) Tensile strength Rm N/mm2 min. Strain A % min. Yield point Rp0.2 N/mm2 min. Brinell hardness (only informative) HBW EN-GJMB-300-6 3) 12 or 15 300 6 - 150 max. EN-GJMB-350-10 12 or 15 350 10 200 150 max. EN-GJMB-450-6 12 or 15 450 6 270 150 to 200 EN-GJMB-500-5 12 or 15 500 5 300 165 to 215 EN-GJMB-550-4 12 or 15 550 4 340 180 to 230 EN-GJMB-600-3 12 or 15 600 3 390 195 to 245 EN-GJMB-650-2 12 or 15 650 2 430 210 to 260 EN-GJMB-700-2 12 or 15 700 2 530 240 to 290 EN-GJMB-800-1 12 or 15 800 1 600 270 to 320 1) On separately cast samples, representative of the properties of the casting, heat treatment together with casting 2) If agreed d = 6 mm is possible 3) For requirements on pressure tightness Spheroidal graphite cast iron in accordance with DIN EN 1563 Table 7.4/4f shows the selection available for sorts with a tensile strength of more than 600 N/mm2. For pearlitic sorts no values for notch impact energy are given. The calculation of the guide values for the pitting endurance limit σHlim and the tooth root strength σFE for the material quality MQ in the cast state can be done according to Appendix 12. Spheroidal graphite cast iron is hardenable, so it can be quenched and tempered (refer to Section 10.3.2.1). A condition for this is knowledge of the hardenability from the end-face quenching test. End-face quenching curves are available (refer to [7.4/21]). 494 7 Design of gearing and gear transmissions It is possible to considerably increase the load capacity of the gear through subsequent nitriding or surface-layer hardening. No parameters are available for this. Refer to DIN EN 1560 for material designations. Table 7.4/4f Mechanical Properties of Spheroidal Graphite Cast Iron in Accordance with DIN EN 1563, Taken from Samples Machined from Integrally Cast Samples 1) Material name Decisive wall thickness t Tensile strength Rm N/mm2 min. Yield point (0.2%) Rp0.2 N/mm2 min. Strain A % min. Brinell hardness range HBW EN-GJS-600-3U 2) (EN-GJS-HB230) 3) t 30 30 < t 60 60 < t 200 600 600 550 370 360 340 3 2 1 190 to 270 - - EN-GJS-700-2U (EN-GJS-HB265) 3) t 30 30 < t 60 60 < t 200 700 700 660 420 400 380 2 2 1 225 to 305 - - EN-GJS-800-2U (EN-GJS-HB300) 3) 4) t 30 30 < t 60 60 < t 200 800 - - 480 - 2 - - 245 to 335 - - EN-GJS-900-2U (EN-GJS-HB330) 3) 4) t 30 30 < t 60 60 < t 200 900 - - 5) 600 - - 5) 2 - - 5) 270 to 360 - - 5) 1) With that the properties of the actual casting cannot be precisely indicated. 2) The U appended to all of the sorts means: properties mechanically produced from integrally cast samples. 3) DIN EN 1563, Appendix A (informative): material name according to hardness classes 4) Not recommendable for cast pieces with large wall thicknesses 5) The dashes “–” mean that the values are to be agreed between the manufacturer and the buyer. 7.4.2.2 Gears Made of Bainitic Cast Iron In analogy to the heat treatment of steel, in the past bainitic cast iron was called “ austempered ” cast iron. Since the mid-1970s it has been offered as bainitic-austenitic spheroidal graphite cast iron for use as gear material [7.4/19]. This sort of cast iron was first entered into the German standards in 1997. According to DIN EN 1564:2006, bainitic cast iron is “sometimes called ADI (austempered ductile iron)”. DIN 3995, T. 5 contains no information about load capacity. Compared to spheroidal graphite cast iron in accordance with DIN EN 1563, bainitic spheroidal graphite cast iron according to DIN EN 1564 is characterised by “higher strength and ductility properties because of the heat treatment” (compare the properties in Table 7.4/5 with those in Table 7.4/4f). For the internationally used term “ADI”, “ austempered ductile iron ”, the name “austenitic-ferritic ” spheroidal graphite cast iron is widely used in German-speaking regions [7.4/20]. It can be traced back to the structure which is composed of austenite with ferrite content saturated with carbon. Depending on the strength requirements and the wall thickness of the casting, unalloyed and low-alloyed sorts are used. A condition for the manufacture of castings made of austenitic-ferritic cast iron is a high-quality structure in the cast condition with well-formed spheroidal graphite and the lowest possible solid solution segregation [7.4.20]. The main alloying elements are Si (always contained in unalloyed spheroidal graphite cast iron with approx. 2.5%) as well as Mn, Ni, Cu, Mo and Cr [7.4/21]. 7.4 Selection of Materials: Strength Values 495 The advantage of bainitic cast iron compared to steel is the same strength at lower density (approx. 10% less) and the good damping ability. The machinability of bainitic cast iron ranges from simple to complicated [7.4/22]. The base material for the four standardised sorts according to Table 7.4/5 is unalloyed, ferritic spheroidal graphite cast iron EN-GJS-400-18. The four sorts listed are suitable for gears with wall thicknesses between 25 and 45 mm [7.4/23]. By alloying with Mn up to 0.3%, Ni up to 2%, Cu up to 1% and Mo up to 0.2% it is possible to improve the hardenability [7.4/22] so that gears with large wall thicknesses (up to 150 mm) can be manufactured. For gears made of austenitic-ferritic cast iron, varying views exist about the limits of application with respect to optimum alloy combinations and wall thickness, such as [7.4/22], [7.4/23], and [7.4/24]. For gears with cast or mechanically produced teeth, it is recommendable to carry out the heat treatment at the manufacturer of the cast part, who has the necessary know-how. The change in dimensions associated with austempering must be taken into consideration [7.4/22]. Table 7.4/5 shows the mechanical properties, measured in samples mechanically produced from separately cast samples. The hardness values which were inserted into the table must be agreed upon. The hardness range refers to the influence of the wall thickness. Table 7.4/5 Mechanical Properties of Bainitic Cast Iron in Accordance with DIN EN 1564, Measured in Samples Machined from Separately Cast Samples Material name Tensile strength Rm N/mm2 min. Yield point (0.2%) Rp0.2 N/mm2 min. Strain A % min. Brinell hardness range 1) HBW EN-GJS-800-8 800 500 8 260 to 320 EN-GJS-1000-5 1000 700 5 300 to 360 EN-GJS-1200-2 1200 850 2 340 to 440 EN-GJS-1400-1 1400 1100 1 380 to 480 1) DIN EN 1564, Appendix A (normative) information about hardness: “For each sort the hardness ranges show the influence of wall thickness” Through mechanical cold work hardening, the surface hardness, and with that also the gear load capacity, can be increased. Thermal surface-layer hardening, for example through induction hardening, is possible. Refer to DIN EN 1560 for material designations. Refer to Section 10.3.2.2 for austempering. 7.4.2.3 Gears Made of AFP Steels AFP steels (in accordance with DIN 10267) are “ferritic-pearlitic steels produced by pre- cipitation-hardening from the hot working temperature”, which obtain their properties through fine-particle precipitation while cooling down from the hot working temperature. The properties are different for rods (which are mostly machined consecutively) than they are for forged blanks. Tables 7.4/6a, b list values for five AFP steels. The steels achieve characteristics which are not possible through quenching and tempering for a comparable chemical composition. Surface-layer hardening or nitriding can improve the load capacity. Refer to DIN EN 10027-1 for steel designations. 496 7 Design of gearing and gear transmissions Table 7.4/6 Mechanical Properties of AFP Steels in Accordance with DIN 10267 a) Mechanical Properties in Precipitation-Hardened state (+P) for Rods for Machining Steel name Mechanical properties 1) Re 2) min. N/mm2 Rm N/mm2 A min. % Z min. % 19MnVS6+P 390 600 to 750 16 32 30MnVS6+P 450 700 to 900 14 30 38MnVS6+P 520 800 to 950 12 25 46MnVS6+P 580 900 to 1050 10 20 46MnVS3+P 450 700 to 900 14 30 1) The values apply for sizes from 30 to 120 mm. The mechanical properties of other sizes must be agreed. 2) Re: Upper yield stress, or in cases where the yield stress is not distinct, the 0.2% yield point Rp0.2 b) Guide Values for the Mechanical Properties of Forged blanks after Precipitation-Hardening Steel name Mechanical properties Re 1) min. N/mm2 Rm N/mm2 A min. % Z min. % 19MnVS6 420 650 to 850 16 32 30MnVS6 470 750 to 950 14 30 38MnVS6 520 800 to 1000 12 25 46MnVS6 570 900 to 1100 8 20 46MnVS3 470 750 to 950 10 20 1) Re: Upper yield stress, or in cases where the yield stress is not distinct, the 0.2% yield point Rp0.2 7.4.2.4 Gears Made of Quenched and Tempered Steels and of Quenched and Tempered Iron Casting Materials Tempering steels For only quenched and tempered gears, the steels and cast steel in accordance with the standards listed in Table 7.4/3a and combined with the “quenched and tempered” (short name +QT) final condition – suitable for the load – can be chosen. DIN EN 10083-3 contains 22 low-alloyed steels, to which six steels alloyed with boron belong. Boron improves the hardenability and in some cases the fineness of the grain. To calculate approximate values for the pitting endurance limit σ Hlim and the tooth root strength σFE, the steels C45E, 38Cr2, 34Cr4, 34CrMo4, 42CrMo4, 34CrNiMo6, 30CrNiMo8 and 36NiCrMo16 were chosen depending on their dimensions and with increasing hardenability (refer to Section 7.4.4.4, Table 7.4/14a). Steels for open-die forgings and low-alloyed cast steel can be found in Tables 7.4/14b and 7.4/14c. An excerpt from DIN EN 10083-2, -3 for the tempering steels in Tables 7.4./7a, b documents the influence of the carbon content and the alloying elements on the mechanical properties of yield point R p0.2, tensile strength R m, fracture elongation A, reduction of area at fracture Z and notch impact energy KV. With increasing carbon content and only a slight increase in manganese content at a given size, the strength increases. Elongation, reduction of area and notch impact energy decrease (Table 7.4/7a). 7.4 Selection of Materials: Strength Values 497 Table 7.4/7 Influence of Carbon Content and Alloying Elements on the Mechanical Properties of Tempering Steels (from DIN EN 10083-2 and DIN EN 10083-3) a) Mechanical Properties of Unalloyed Quenched and Tempered Steels Dependent on Carbon Content (16 s 40 mm) Type of steel Mean C content % R e (Rp0.2) min N/mm2 Rm N/mm2 A min % Z min % KV min J C22E, C22R C35E, C35R C40E, C40R C45E, C45R C50E, C50R C55E, C55R C60E, C60R 0.205 0.355 0.405 0.460 0.510 0.560 0.610 290 380 400 430 460 490 520 470 to 620 600 to 750 630 to 780 650 to 800 700 to 850 750 to 900 800 to 950 22 19 18 16 15 14 13 50 45 40 40 35 35 30 50 35 30 25 - - - b) Mechanical Properties of Quenched and Tempered Steels with Approximately the Same Carbon Content, Dependent on Content of Alloying Elements (16 s 40 mm) Type of steel Mean alloy content in % Re (Rp0.2) min N/mm2 Rm min N/mm2 A min % Z min % KV min J C Mn Cr Mo Ni C35E, C35R 34Cr4 35NiCr6 34CrMo4 34CrNiMo6 36NiCrMo16 0.355 0.335 0.335 0.335 0.34 0.355 0.65 0.75 0.75 0.75 0.65 0.65 0.20 1.05 0.95 1.05 1.50 1.80 0.05 - - 0.225 0.225 0.35 0.20 - 1.40 - 1.50 3.83 380 590 740 650 900 1050 600 800 880 900 1100 1250 19 14 14 12 10 9 45 40 40 50 45 40 35 40 35 40 45 30 At approximately the same carbon content and at the same size, the strength increases with the share of the hardenability-increasing alloying elements manganese, chrome, molybdenum and nickel. With rising strength the elongation drops, and the reduction of area and notch impact energy remain almost constant (Table 7.4/7b). When selecting tempering steels for gears, the hardenability scatter bands provided in the standard DIN EN 10083, including subdivision in +H (full scatter bandwidth) and +HH or respectively +HL with limited scatter bandwidth, are to be observed. It is useful to select using the bottom-most limiting curve +H or respectively +HH or the specific end-face quench curve for the melted material in question. The requirement for the most ferrite-free martensitic-bainitic transformation possible for the position of the component under scrutiny (surface, sampling location) can be assessed in a chemical analysis of the steel in accordance with [7.4/18]. The t 8/5-times for 0% ferrite and 100% martensite lead to the hardness for 0% ferrite (a ferrite- free condition) and 100% martensite [refer to Equation (7.4/14)]. From this the degree of hardening R – a measure for the quality of the hardening – can be determined as the quotient from the achieved hardness and the maximum hardness possible [7.4/25, 7.4/26]. The following applies: R = H measured /Hmaximum (7.4/1) with Hmeasured hardness measured on the component [HRC] Hmaximum maximum hardness possible = 60 ǜ C0.5 + 20 [HRC] The Equation (7.4/1) applies for 0.2 C [%] 0.6 and R 1. The maximum hardness is determined only by the carbon content. 498 7 Design of gearing and gear transmissions The link between the quenching medium, the hardenability, the degree of hardening and the distribution of hardness for a shaft of 100 mm in is illustrated in Figures 7.4/2a, b and 7.4/3. The final hardness that is to be adjusted through tempering is dependent on the tempering tem-perature (duration of usually 2 h), the degree of hardening and the chemical composition of the respective steel [7.4/12] (refer to Section 10.3.2.1). With knowledge of the tempering hardness H it is possible with the relation tensile strength m3.22≈⋅R H[HV] [N/mm2] (7.4/2) according to DIN EN 18265, Table A1, valid for H 430 HV with R2 = 1 (refer to Appendix 11) and the ratio of yield point to tensile strength R p0.2/Rm = 0.37 R + 0.54 (7.4/3) in accordance with [7.4/25] to estimate the 0.2 yield point with the degree of hardening R. Equation (7.4/3) applies for 0.4 R 1, martensite content MG = 200 R – 100 [%], [7.4/26]. With the help of time-temperature-transformation diagrams from [7.4/18], degrees of hardening were determined for a ferrite-free transformation with R between 0.60 and 0.75. For a reliable selection, it is recommended to calculate with R = 0.75. Then, the following is true, according to Equation (7.4/3): Rp0.2≈ 0.82 R m (7.4/3a) for a ferrite-free, martensitic-bainitic (quenched) structure. Refer to Section 7.4.4.4, Table 7.4/14a and b for a selection of steels and endurance limit values. Quenched and tempered iron casting materials In addition to the unalloyed sorts in accordance with Table 7.4/4 d with C contents above 0.3%, DIN EN 10293 also contains six low-alloyed sorts which can be used for quenched and tempered and likewise for surface-hardened gears: G34CrMo4, G42CrMo4, G30CrMoV6-4, G35CrNiMo6-6, G32NiCrMo8-5-4 and G30NiCrMo4. Compared to the tempering steels the groups of dimensions (wall thickness t) are graded differently, and two different tempering conditions are given for all of the sorts named (QT1: higher tempering temperature range; QT2: lower tempering temperature range). The mechanical properties differ correspondingly. The standard doesn’t contain hardenability scatter bands. Principally, selection according to hardenability and heat-treatability at a given chemical composition is possible. This approach is depicted in detail in [7.4/27] for tempering cast steel. The selected sorts for calculating approximate values for pitting endurance limit σ Hlim and tooth root strength σFE are specified in Section 7.4.4.4, Table 7.4/14c. At the same static strength, low-alloyed tempering cast steel has lower endurance limit values than do the tempering steels (see Table 7.4/14a and 7.4/14c). Refer to Section 10.3.2.1 for quenching and tempering. 7.4.3 Materials for Gears with Surface-Layer Hardening: The Basics For gears with surface-layer hardening , depending on the material and the treatment the selection according to hardenability must be expanded to include selection according to surface-layer hardenability , case-hardenability, carbonitriding hardenability, and nitriding hardenability. The achievable properties are based on the formation of martensite (transformation hardening). Selection according to nitridability (refer to Section 7.4.3.5) utilises the mechanism of precipitation hardening through nitrides in the surface layer. 7.4 Selection of Materials: Strength Values 499 For gears with surface-layer hardening, all influencing factors on the tooth endurance limit depending on material and heat treatment are expressed in terms of surface hardness, hardness depth, and core hardness (characterisation of the hardening curve). The hardening curve incorporates the effect of residual stresses induced by the heat-treatment method, which, as residual compression stresses, have a large positive influence, particularly on the tooth root endurance strength (refer for example to Section 7.4.4.6, strength values of case-hardened gears). The groups of steels to which these criteria can be applied are tempering steels, including the steels for flame and induction hardening (DIN EN 10083), open-die forgings made of steel for general use (DIN EN 10250), cast steel for general use (DIN EN 10293), cast steel for flame and induction hardening (SEW 835), case-hardened steels (DIN EN 10084), nitrided steels (DIN EN 10085), and in special cases stainless steels in accordance with DIN EN 10088. 7.4.3.1 Surface-Hardened Gears Surface-layer hardenability is the hardness increase and deepening achieved in a workpiece through surface-layer hardening (primarily flame and induction hardening, and electron-beam or laser-beam hardening). Only the carbon content of the material determines the hardenability , with the surface hardness as the measured quantity. The hardness deepening with the case depth (DIN EN 10328) as measured quantity is dependent on the chemical composition of the steel, the depth of the method-dependent austenitised surface layer, and on the quenching effect of the medium used. The achievable hardness profile is depicted by the hardness-depth curve (refer to Section 7.4.4.5). Primarily unalloyed and alloyed tempering steels in accordance with DIN EN 10083 are used for surface-hardened gears. Unalloyed and alloyed open-die forgings in accordance with DIN EN 10250, cast steel with similar composition in accordance with DIN EN 10293 and SEW 835, and AFP steels in accordance with DIN EN 10267 can also be surface-hardened. DIN EN 10083-2, -3 recommend steels for flame and induction hardening, which are listed with minimal hardness values of 48 to 58 HRC in Table 7.4/8a. Table 7.4/8 Surface Hardness Values after Flame or Induction Hardening a) Steels Recommended by DIN EN 10083-2 and DIN EN 10083-3 for Flame and Induction Hardening and Other Steels Suitable for Th is from DIN EN 10083-3 Steels recommended for flame and induction hardening Surface hardness 1) HRC min. Steels suitable for flame and induction hardening Surface hardness 1) HRC min. C35E, C35R 48 34Cr4, 34CrS4 49 C45E, C45R 55 34CrMo4, 34CrMoS4 49 C50E, C50R 56 34CrNiMo6 50 C55E, C55R 58 35NiCr6 49 46Cr2 54 36NiCrMo16 50 37Cr4, 37CrS4 51 38Cr2, 38MnB5 52 41Cr4, 41CrS4 53 39NiCrMo3 52 42CrMo4, 42CrMoS4 53 39MnCrB6-2 51 50CrMo4 58 51CrV4 57 1) After surface-layer hardening and subsequent tempering 180 C, 1 h holding period The table is supplemented by other tempering steels from DIN EN 10083-3. Moreover, Table 7.4/8b lists further steels and casting materials that can be surface hardened. 500 7 Design of gearing and gear transmissions In their initial pearlitic condition, the surface-layer hardening of spheroidal graphite cast iron (e.g., EN-GJS-600-3), lamellar-graphite cast iron (e.g., EN-GJL-350) and black-heart malleable cast iron (e.g., EN-GJMB-350-10) can be used for setting a minimum of 0.4% bound carbon. Minimal surface hardness values of 55 HRC can be achieved. Table 7.4/8 b) Open-Die Forgings Made of Steel in Accordance with DIN EN 10250-2 and -3 Suitable for Flame and Induction Hardening, Cast Steel in Accordance with DIN EN 10293, Cast Steel in Accord ance with SEW 835 and Ferritic-Pearlitic Steels Produced by Precipitation Hardening from the Hot Working Temperature in Accordance with DIN EN 10267 (AFP Steels) Open-die forgings made of steel Surface hardness 1) HRC min. Cast steel Surface hardness 1) HRC min. Ferritic-pearlitic steels produced by precipitation hardening from the hot working temperature (AFP steels) Surface hardness 1) HRC min. DIN EN 10250-2, -3 DIN EN 10293 DIN EN 10267 C35E 48 G34CrMo4 49 38MnVS6 51 C45E 55 G42CrMo4 53 46MnVS6 53 C55E 58 G35CrNiMo6-6 51 46MnVS3 54 34CrMo4 49 G32NiCrMo8-5-4 50 42CrMo4 53 SEW 835 36CrNiMo4 51 GC45E 53 34CrNiMo6 50 G36Mn5 50 36NiCrMo16 50 G46Mn4 53 40CrMoV13-9 52 G42CrMo4 54 51CrV4 57 G50CrMo4 57 1) After surface-layer hardening and subsequent tempering 180 C, 1 h holding period For using electron-beam or laser-beam hardening for gears, the same tempering steels, cast steels and iron casting materials (on condition of a pearlitic or quenched and tempered basic structure) can be used as for flame and induction hardening. For electron-beam hardening, the unalloyed tempering steels with more than 0.25% C to 0.6% C and the low-alloyed tempering steels with more than 0.3% C to 0.5% C are specified. What’s interesting is that carburised case-hardened steels, such as C15, 16MnCr5 or 14NiCr18, and the nitrided steels 31CrMoV9 and 34CrAl6 can also be surface hardened using the electron beam [7.4/28]. Refer to Section 10.3.2.3 for surface-layer hardening. 7.4.3.2 Case-Hardened Gears In gear manufacturing, case-hardened gears predominate. They are manufactured with dimen- sions of up to 1000 mm tip diameter for pinion shafts and up to 4000 mm tip diameter for disc-shaped gears. The reason for this is that, compared to other surface-layer hardening methods, it has the highest load capacity values for the tooth root and the tooth flank. The basis for selecting steels for case-hardened gears is the case hardenability . The case hardenability comprises the terms surface hardenability , hardness deepening , and core har- denability . When selecting steels for case-hardened gears based on case hardenability, it is necessary to use knowledge derived from the section on case hardening, Section 10.3.2.4. 7.4 Selection of Materials: Strength Values 501 As a consequence of the carbon profile produced during carburisation and the hardness profile which results from this during hardening, according to the three hardenability criteria the maximum surface hardness achievable for the given carbon content (surface hardenability), the possible case-hardening depth (hardness deepening) and the core hardness (core hardenability) must be analysed for each case-hardened steel. The hardness profile as a measure of the “strength” of the case-hardened surface layer is dependent on the carbon content, and essential on the case hardenability of the respective melted steel material, the geometry (characteristic dimension), the mass of the component and on the quenching effect of the hardening medium, and is influenced by tempering. The core hardenability is specified by DIN EN 10084 in scatter bands from the end-face quenching test (DIN EN ISO 642) on samples which were already normalised. For this, standardised austenitising and quenching conditions apply (consistent technology for each melted steel material). DIN EN 10084 contains the hardenability scatter bands for the low-alloyed case-hardened steels. The hardenability of the MnCr, MoCr(Ni), CrNi, CrNiMo and MnCrB case-hardened steels can be calculated according to SEP 164 [7.4/16] from chemical analysis. This possibility is often used instead of the costly end-face quenching test, for example to predict the hardenability of a melted material in the steel plant. Figure 7.4/6 shows the hardenability scatter band for the non-carburised steel 16MnCr5+H in accordance with DIN EN 10084. Depending on the application, steels can be ordered with a scatter band narrowed by one-third. The designation 16MnCr5+HH applies for the narrowed tolerance from the upper limiting curve and 16MnCr5+HL in analogy from the lower limiting curve. An additional restricting tolerance, for example to 2HRC, can be agreed where required with the steel manufacturers. For gears made of case-hardened steel of the levels MQ and ME, ISO 6336-3 (DIN 3990-5) recommends the scatter band with restricted tolerance HH. From Figure 7.4/6 it can be seen that a rod of 112 mm diameter made of the steel 16MnCr5+HH min can achieve a hardness of 20 HRC on the surface only when quenched in oil in motion (H = 0.35). Figure 7.4/6 Hardenability scatter band of non-carburised steel 16MnCr5 in accordance with DIN EN 10084 502 7 Design of gearing and gear transmissions The standardised case-hardened steels important for gears are compared with one another in Figure 7.4/7 using the lower limiting curve of their hardenability scatter band HH min. To differentiate, the hardness at 1.5 mm distance from the end face was chosen as the criterion for the hardenability (100% martensite), and the hardness at 10 and 25 mm distance from the end face as the criterion for the hardness deepening. For example, J 42/44 - 1.5 means that 42 to 44 HRC exists at a 1.5 mm distance from the end face. a) Steels of “low” core hardenability : J 42/44 - 1.5; J 25.5/30.5 - 10; J 20/22 - 25: Steels: 16MnCr5 (identical with 16MnCrB5 and 20MoCr4); 20NiCrMo2, 18CrMo4, 16NiCr4 and 28Cr4. This steel has, because of its higher carbon content, with 48 HRC the best hardenability of all case-hardening steels according to DIN EN 10084. b) Steels of “medium” core hardenability (medium-strength values achievable) ) : J 42/45 - 1.5; J 32/39.5 - 10; J 24/27 - 25: Steels: 15NiCr13, 18NiCr5-4; 20MnCr5 (almost identical with 17CrNi6-6); 20NiCrMo6-4 c) Steels of “high” core hardenability (high-strength values achievable) : J 42/44 -1.5; J 39/43 - 10; J 34/42 - 25: Steels: 14NiCrMo13-4, 18CrNiMo7-6, 20NiCrMo13-4 Because of the very similar C contents (with the exception of 28Cr4), the J1.5 values are between 42 and 45 HRC. The hardenability-increasing alloying elements have a decisive effect on the J10 values and J25 values. In addition to carbon, hardenability-increasing alloying elements in case- hardened steels include, with increasing influence, nickel, chrome, manganese, and molybdenum (the strongest) and the trace element boron (hardenability-increasing and fine-grain-forming). The Cu content (on average 0.11 to 0.16 %) can have a positive influence on the hardenability. Compared to the former standard DIN 17210, the number of case-hardening steels has risen by 19 sorts in DIN EN 10084. The steel 15CrNi6 was dropped (in its place is 17CrNi6-6). The steel 20NiCrMo13-4 was adopted. Despite the large number of standardised steels, a large number of other case-hardening steels are used in actual practice, some of which were listed in past editions of DIN 17210. Examples of some steels used include 27MnCr5; 18MnCrB5, 20MnCrB5, 21MnCrB5-4, 27MnCrB5-5; 28CrMnB7, 20CrMo5, 24CrMo5, 25CrMo4, 27CrMo4, 25MoCr4; 18NiCr13, 18CrNi8, 18NiCrMoS4; 18NiCrMo5, 18MnCrMoB5, 21MnCrMo5, 23MnCrMo5 (JOMASCO) and 25MnSiCrVB6 (METASCO). The steels 21CrMnCu8-6 (0.2% C; 2% Cr, 1.6% Mn, 0.5% Cu, as well as Nb and Ti) and Ovatec277 (0.14–0.17% C; 1.20–1.40% Mn; 2.10–2.30% Cr; 0.45–0.55% Ni; 0.45–0.55% Mo; 0.15–0.25% V) are characterised by a uniformly high core hardenability. The hardenability of the steel 21CrMnCu8-6 only drops from J 50 - 1.5 to J 45 - 50 [7.4/29]. The hardenability of the steel Ovatec 277 is similarly good [7.4/30]. With these steels, large gears with core hardnesses of ≥ 40 HRC can be manufactured, thus with the highest pitting and tooth root endurance limit on the material side. The aerospace steel M50NiL is in a category of its own (AMS 6278, remelted in a vacuum) [7.4/31]. It is listed in DIN EN ISO 683-17 with the designation 13MoCrNi42-16-14 (also X12CrMoNiV4-4-3) as a “heat-resistant” roller bearing steel for case hardening. Due to its composition, after carburisation and low-distortion hardening in compressed gas, the tempering temperature can be raised to 400 C, combined with high core strength at high ductility. The combination with nitriding or respectively case hardening and coating with hard material as well as superfinishing methods led to values in flank endurance limit and tooth root endurance strength which are above the areas of the ME level for case-hardened gears which were known until now. 7.4 Selection of Materials: Strength Values 503 Figure 7.4/7 End-face quench curve HH min of case-hardening steels acc. to DIN EN 10084 504 7 Design of gearing and gear transmissions A very important development for almost all case-hardened steels is the improvement of the fine-grain stability for carburisation temperatures of up to 1100 C (radically shortened carburisa-tion time) through microalloying with aluminium (150 to 400 ppm), niobium (up to 500 ppm), titanium (up to 500 ppm) and nitrogen (up to 150 ppm) 1). Research in German steel plants contributed a great deal to this [7.4/32], [7.4/33], [7.4/34], [7.4/35]. Interesting are the results concerning the increase of hardenability in the end-face quenching test up to 5 HRC through the use of quenched and tempered samples as opposed to normalised samples. Aside from the potential increase in core hardness achieved through “pre-quenching and tempering”, there is also a positive influence on distortion characteristics [7.4/36]. Occasionally case-hardened gears are manufactured from low-alloyed cast steel. Examples of the sorts which are suitable for this include those according to DIN EN 10293, such as G15CrMoV5-9, G17CrMo5-5, G17CrMo9-10, G26CrMo4, G20NiCrMo4, and G17NiCrMo13-6. In order to homogenise the structure and to minimise distortion, the condition prior to heat treatment should be +QT. Principally it is possible to differentiate between the sorts named using their case-hardenability. When selecting case-hardening steels, in many cases the machinability has an important role. This is why nearly all case-hardened steels of low and medium hardenability in accordance with DIN EN 10084 are available with a defined sulphur content (0.020 to 0.040 %). On the other hand, the finest desulphurised case-hardened steels are used too, with sulphur contents of <0.010%. The surface hardenability and the hardness deepening of carburised steels are also to be determined in the end-face quenching test. Because of the diversity of the technological pos-sibilities, there are no standardised scat-ter bands. Figure 7.4/8 [7.4/11, Volume 2] depicts the case hardenability of the steel 16MnCr5. A carburised and tempered end-face quenching sample was used to deter- mine it. To measure the hardness this is polished in layers, and for each polished layer the hardness is measured and the C content determined (refer to diagram in lower sub-picture). Figure 7.4/8 Case hardenability of the steel 16MnCr5, determined in the end-face quenching sample ( AC3 = 845 C) with 0.16% C; 0.22% Si; 1.12% Mn; 0.03% P; 0.008% S, 0.015% Al; 0.99% Cr; 0.12% Cu; 0.02% Mo; 0.12% Ni; from [7.4/11, Volume 2] Definitions: 825 C, 930 C: Austenitisation temperature for the carburised end-face quenching sample; 870 C, 1050 C: austenitisation temperature for the non-carburised end-face quenching sample 1) 1 ppm (parts per million) = 0.0001% 7.4 Selection of Materials: Strength Values 505 In a given steel composition, in addition to the carbon content (here 0.68% is optimal) in the surface layer, the austenitisation temperature (930 C is considerably more favourable than 825 C) has a considerable influence on the case hardenability to be determined in the end-face quenching test. For a comparative selection of materials based on case hardenability, the regression formulas based on SEP 1664 can be used to calculate the core hardenability (non-carburised). For example, Table 7.4/9 specifies the regression formulas for the MnCr case-hardened steels [7.4/16]. Table 7.4/9 Calculation of the End-Face Quenching Hardness of Case-Hardened Steels Alloyed with Manganese and Chrome according to SEP 1664 [7.4/16] Limits of the Chemical Composition of the Considered Meltings of the Case-Hardened Steels Alloyed with Manganese and Chrome (Mass Content of Manganese ~1.2% and Chrome ~1%) C Si Mn P S Cr Min. 0.130 0.020 1.020 0.006 0.002 0.820 Max. 0.231 0.380 1.480 0.033 0.044 1.290 Mean 0.179 0.200 1.220 0.013 0.026 1.057 Mo Ni Al Cu N Min. 0.010 0.020 0.012 0.040 0.0060 Max. 0.090 0.300 0.063 0.350 0.0180 Mean 0.045 0.125 0.035 0.158 0.0118 Coefficients for the Equation for Calculating the End-Face Quenching Hardness of Case-Hardened Steels Alloyed with Manganese and Chrome (Mass Content of Manganese ~1.2% and Chrome ~1%) Coefficients of the elements J m Constant C Si Mn P S 1.5 220 29.27 60.38 2.34 3.14 0.00 0.00 3 224 26.36 65.02 2.35 4.15 0.00 0.00 5 224 3.64 96.40 4.73 9.28 0.00 0.00 7 224 -15.89 117.94 7.98 13.17 0.00 0.00 9 224 -20.33 111.04 9.56 14.90 0.00 0.00 11 224 -21.05 104.87 8.37 13.81 0.00 0.00 13 224 -24.20 106.40 10.54 14.34 0.00 0.00 15 224 -25.65 104.73 9.39 13.69 48.70 0.00 20 224 -31.20 112.20 10.46 14.00 50.94 0.00 25 217 -35.15 118.66 10.55 15.13 54.83 0.00 J m Cr Mo Ni Al Cu N 1.5 220 0.00 0.00 0.00 0.00 0.00 0.00 3 224 0.00 0.00 -3.39 0.00 3.38 0.00 5 224 8.18 0.00 0.00 0.00 7.03 -194.23 7 224 13.47 16.21 0.00 0.00 10.55 -251.11 9 224 13.58 0.00 6.33 0.00 10.46 -258.53 11 224 13.97 0.00 8.45 0.00 9.19 -205.32 13 224 13.43 0.00 11.16 0.00 8.53 -170.83 15 224 13.91 0.00 11.88 0.00 8.29 -142.09 20 224 14.38 17.75 11.21 0.00 7.34 -115.28 25 217 13.72 20.71 12.73 0.00 9.25 -126.73 506 7 Design of gearing and gear transmissions The surface hardenability and hardness deepening of case-hardened steels in accordance with DIN EN 10084 can be estimated using a calculation-graphic model in accordance with [7.4/37] with a hardenability formula based on [7.4/15]. Accordingly, one arrives at the hardness J [HRC] between the distance from the end face E = 8 to 80 mm in accordance with Equation (7.4/4), dependent on the C content with the known alloying element proportion, taking the austenite grain size K ASTM into account: J 8…80 = 95 C0.5 – 0.00276 E2 C0.5 + 20 Cr + 38 Mo + 14 Mn + 5.5 Ni + 6.1 Si + 39 V + 96 P – 0.81 KASTM – 12.28 E0.5 + 0.898 E – 13 HRC [HRC] (7.4/4) Validity: 0.13 to 0.55% C; 0.15 to 1.40% Si; 0.40 to 1.78% Mn; 0.11 to 2.07% Cr; 0 to 0.25% Mo; 0.01 to 2.02% Ni; 0.01 to 0.12% V; 0.008 to 0.035% P; KASTM = 6.5 to 10.5; R2 = 0.934, SR 3.12 HRC. with C - Element in mass-%, also all other elements J - Hardness in the end-face quenching test [HRC] E - Distance from the end face [mm] KASTM - Austenite grain size [-] R2 - Coefficient of determination SR - Residual variance The case-hardenability estimated with this for the steel 16MnCr5 with a chemical analysis, which results in the hardenability HH min, is depicted in Figure 7.4/9. Compared to those in Figure 7.4/8, the curves obtained for 0.3% C to 0.6% C are idealised. However, they represent a good basis for the explanation of the correlation between chemical composition, hardenability, dimension and quenching. Assumed is oil quenching with H = 0.35 [7.4/37]. The maximum hardness ( hardenability ) dependent only on the carbon content can be determined according to numerous equations [7.4/38]. For this, between the end-face surface and 8 mm distance from the end face, the equation according to [7.4/15], valid for 0.1 %C 0.6, is used: H max = 20 + 60 C 0.5 [HRC] (7.4/5) With 0.6% C the maximum hardness is 66.5 HRC ( 875 HV). It is a measure for the edge hardenability (Figure 7.4/9). The measure for the hardness deepening is the case-hardening depth . According to DIN EN ISO 2639, it is named CHD (formerly Eht ) and defined with a hardness limit of 550 HV1 (52.5 HRC). This applies “for components which, after the final heat treatment, have a hardness of less than 450 HV 1 at a distance from the surface which corresponds to the threefold of the case-hardening depth”. In cases of higher hardnesses, higher hardness limits are recommended. In actual practice, hardness limits of 600 HV1 and 650 HV1, for example, are required on the drawing. These higher values are also recommendable for carbonitrided tempering steels. The hardness deepening, defined with CHD (52.5 HRC), is determined with increasing distance from the end face by the C content. According to Figure 7.4/9, the first point of intersection of the hardness 52.5 HRC for 0.35% C is produced at a distance from the end face of 5 mm, which, according to Figure 7.4/10a, results in the diameter d = 12 mm. The core hardness would be 34 HRC. With increasing distance from the end face and increasing diameter – i.e., with decreasing quenching speed – a higher C content is necessary to achieve the same hardness value. It can be seen that for C = 0.40% ĺ E = 7.5 mm, d = 22 mm, core hardness = 31 HRC. If the edge hardness of 64 HRC is sufficient as quenching hardness for the subsequent tempering process, a C content of approximately 0.43% is necessary ĺ E = 9 mm, d = 30 mm, core hardness = 28 HRC. Consequently, the possibly insufficient hardness deepening of the steel – i.e., 7.4 Selection of Materials: Strength Values 507 the failure to achieve the required case-hardening depth – is to be compensated for by a longer carburisation time, or the steel must have a better hardenability, or respectively it must be faster quenched. It may be possible to case-harden a component with a diameter of 112 mm, but the edge hard-ness would only be 55.2 HRC, and the carbon content limit would have to be raised to 0.60%. The core hardness would be 20 HRC. Figure 7.4/9 Case-hardenability of the steel 16MnCr5 calculated with the given analysis in accordance with [7.4/15] Figure 7.4/10 illustrates the basic principles for this [7.4/37]. According to this there is, as used for Figure 7.4/9, a close correlation between the quenching conditions in oil, expressed by the H value, the decisive heat-treatment diameter d, and the distance from the end face E. Based on this, in Figure 7.4/9 the distances from the end face E and the limit diameter d resulting from the carbon content limits are given which are possible at a quenching intensity of H = 0.35 for the steel 16MnCr5 having the given analysis. These values are summarised in Table 7.4/10 together with those of surface and core hardness. Table 7.4/10 Correlation between Surface Hardness, Carbon Content Limit, Distance from the End Face, Limit Diameter and Core Hardness for the Steel 16MnCr5 in accordance with Figure 7.4/9 Surface hardness Hmax [HRC] Carbon content limit [%] Distance from the end face E [mm] Limit diameter d [mm] Core hardness HK [HRC] 66.5 0.35 5 12 34 65 0.40 7.5 22 31 64 0.43 9 30 28 55.2 0.60 20 112 20 508 7 Design of gearing and gear transmissions According to this, at diminishing edge hardness with rising carbon content limit, the distance from the end face and thus the limit diameter increases, and the core hardness decreases. a) Quenching conditions in oil H value Formulas Very weak, no motion 0.25 d = E (1.27 + 0.0042 • E) E = d (0.755 - 0.0003 • d ) Increasing motion of oil and/or workpiece 0.35 d = E (1.52 + 0.0028 • E) E = d (0.649 - 0.0001 • d ) 0.45 d = E 1.755 E = d 0.57 0.60 d = E 2 E = d 0.5 Very strong motion of oil and/or workpiece 1.0 d = E (2 + 0.03 E) E = d (0.47 - 0.00015 d) b) Figure 7.4/10 Diameter over distance from the end face for various quenching intensities in accordance with [7.4/37]: a) approximation formulas for the correlations between distance from end face E and the relevant heat-treatment diameter d with comparable cooling conditions in the surface layer for various quenching intensities H; b) correlation between distance at the end face E and the relevant heat treatment diameter d with comparable cooling conditions in the surface layer for various quenching intensities H 7.4 Selection of Materials: Strength Values 509 With the values of the given material analysis, the limiting hardness (550 HV1), the work-piece diameter d, the predeter- mined quenching intensity H and the calculated C profile, a calculation program [7.4/39] was developed early on that is used in carburising plants and is depicted schematically in Figure 7.4/11a and has been developed further [7.4/37]. The basis for the hardness pro-file calculation can be found in Figure 7.4/11b for the carburi-sed steel 16MnCr5, d = 30 mm after oil quenching with H = 0.35. At a carburisation depth of 1.5 mm, with refe-rence to 0.35% C and labelled as At 0.35, the case-hardening depth is only 1.4 mm. The rea- son for this is the insufficient hardness deepening, which was already discussed in relation to Figure 7.4/9. This theory has been experi-mentally tested in a very com-prehensive project, with eight different case-hardened steels of differing hardenability [7.4/40]. Six round rods with diameters between 12.5 and 250 mm and target case-hardening depths of 1.0 and 2.0 mm were carbu-rised together, quenched in oil with H = 0.3, and tempered. With a known C profile, the measured hardness profiles were compared to the calcula-ted ones and analysed. Figure 7.4/12a, b contains two very informative, summarised results. a) b) Figure 7.4/11 a) Diagram of hardness profile calculation (method according to Wyss - Weissohn); b) deducing the hardness curve in the ca se-hardened layer (d) from the hardness curve on the end-face quenching sample (a) for oil hardening of a diameter of d (b) and the C curve (c) in the case-hardened layer, from [7.4/39] 510 7 Design of gearing and gear transmissions a) Carbon content limits according to diameter, regression lines, from [7.4/40] b) Case-hardening depth according to diameter (target CHD = 1.0 mm), from [7.4/40] Figure 7.4/12 Influence of carbon content limit and case-hardening depth on the hardenable diameter (constant carburisation and hardening conditions) As Figure 7.4/12a shows, based on the carbon content limit (this is the C content, which, after quenching, leads to the limiting hardness of 550 HV1 as measure for the case-hardening depth; refer to Table 7.4/10) three steel groups can be differentiated according to low, medium and high hardenability (refer to Figure 7.4/7). 7.4 Selection of Materials: Strength Values 511 At high hardenability, depending on the diameter, less carbon is required for the carbon content limit, whereas at a comparable diameter, low hardenability requires a high carbon content limit. This means that with lower hardenability the case-hardening depth decreases with increasing diameter (Figure 7.4/12b). For cases of normal requirements in gear manufacturing, it is sufficient in actual practice also for economic reasons [7.4/41] when one steel each is selected from the three groups of hardenability, where appropriate with slight modifications to the contents of S, Al, Ti, N, B, for example. Examples are • 16MnCrS5 (low hardenability), • 20MnCr5 or respectively 17CrNi6-6 (15CrNi6) (medium hardenability), and • 18CrNiMo7-6 (high hardenability). For gears the surface hardness is 56 to 64 HRC. For medium and high requirements it is 58 to 64 HRC (ISO 6336-5). The hardness is achieved through tempering (refer to Section 10.3.2.4). For very high requirements the upper limit of the hardness range is aimed for. Refer to Section 10.3.2.4 for case hardening. 7.4.3.3 Carbonitrided Gears ISO 6336-5 makes no special statements regarding the load capacity of carbonitrided gears (only regarding nitrocarburising). For carbonitriding, which, according to its definition, belongs to the area of case hardening (refer to DIN EN 10052), alloyed case-hardened steels in accordance with DIN EN 10084 and unalloyed and low-alloyed tempering steels in accordance with DIN EN 10083-2 and -3, are suitable, predominantly with C contents < 0.45% (unalloyed) or respectively < 0.4% (low-alloyed). Frequently, the low-alloyed steels 16MnCr5 and 34Cr4 are used for gears with smaller dimensions and moduli of up to 6 mm. In the attempt to use carbonitriding also for gears of larger dimensions because of the better control of the carbonitriding process, steels with higher alloy contents such as 17CrNiMo5, 18NiCrMo14, 23MnCrMo5, 30CrMo4, 34CrMo4, 33MnCrMo6-3, and 32CrMoV13-10 are in-creasingly being investigated for their suitability as carbonitrided gears. The carbonitriding hardenability can be used to differentiate between the various steels. This comprises the surface hardenability, the hardness deepening and the core hardenability (see case hardenability in Section 7.4.3.2). The carbonitriding hardenability is determined according to the same principle as is shown in Figure 7.4/8 for determining case-hardenability. Carbonitriding hardenability curves are known for unalloyed case-hardening steel [7.4/42]. Due to the nitrogen which is additionally introduced to the existing carbon, the surface hardenability and hardness deepening are highly improved. Just as in case hardening, for a given case-hardening or temper-ing steel, in addition to the carbon content and the quenching speed, only the carbonitriding temperature has an effect on the core hardenability, which, when a high tooth endurance limit is required, should be above A c3 of the core material. Refer to Section 10.3.2.5 for carbonitriding. 7.4.3.4 Nitrogen Case-Hardened Gears In special cases, gears made of high-alloyed non-corroding steels in accordance with DIN EN 10088 or respectively DIN EN 10250-4 (open-die forgings) are needed. The standards divide these steels into ferritic, martensitic, precipitation-hardening, austenitic and austenitic-ferritic steels. Through quenching and tempering, the martensitic, corrosion-resistant steels achieve 512 7 Design of gearing and gear transmissions the strength values of tempering steels ( Rm-min between 700 and 900 N/mm2). The ferritic, non- magnetic austenitic and ferritic-austenitic steels are not quenched and tempered, but rather used in solution-annealed condition, and thus they have low strength. This results in a very low load capacity for the gear. One way to raise the strength is surface-layer hardening through nitrogen diffusion. Due to the high contents of alloying elements such as chrome and molybdenum present in all these steels, when diffusing nitrogen in the temperature range of classic nitriding (500 to 590 C), very thin compound layers develop and special nitrides are released in the diffusion layer. Because of the very high alloy content, the nitriding hardness depth is very shallow. Here the surface layer becomes poor in special nitride formers such as Cr and Mo, and the corrosion resistance drops. One solution is a nitrogen content increase of the non-corroding ferritic-martensitic and martensitic steels in accordance with DIN EN 10088 at very high temperature, with the goal of suppressing the nitride release in the surface layer and, while maintaining the corrosion resistance, producing a high degree of hardness. This was described in [7.4/43] to [7.4/48], and the method was named case-hardening with nitrogen . The steels X6Cr17, X10Cr13, X14CrMoS17, X15Cr13, and X20Cr13 are used. The patented method is called SOLNIT M [7.4/44]. To differentiate when selecting steels, the nitriding hardenability can be used. One can refer to the findings concerning how to determine case and carbonitriding hardenability. Any research results regarding nitriding hardenability are not known yet. At the same time as case hardening with nitrogen, using oxinitritiding in gas as a basis, studies were also carried out on high-temperature nitrogen case-hardening of non-corroding steels [7.4/49], [7.4/50]. On austenitic steels such as X5CrNiMo17-13-2, a nitrogen-content increase at very high tem-peratures is also suitable for forming surface layers enhanced with nitrogen with high corrosion resistance and with low hardness increase (up to 150 HV compared to the core material). This method is called SOLNIT A [7.4/44]. 7.4.3.5 Nitrided/Nitrocarburised Gears Many steels as well as steel and iron casting materials are used for nitrided gears. Principally, in addition to nitriding steels (DIN EN 10085), all tempering steels (DIN EN 10083), and analogue steels for open-die forgings according to DIN EN 10250-2 and -3, the case-hardening steels (DIN EN 10084) and the AFP steels according to DIN EN 10267 can be used. Also, many low-alloyed cast steels in accordance with DIN EN 10293, as well as lamellar-graphite cast iron (DIN EN 1561), malleable iron (DIN EN 1562) and spheroidal graphite cast iron (DIN EN 1563) belong to this group. In addition to the nitriding steels according to DIN EN 10085, in actual nitriding practice three steel groups that differ in terms of carbon content are preferred for gears. These are a) Cr-Mo-(V)-alloyed steels with a C content < 0.2%, such as X6CrMo4, 14CrMoV6-9, 15CrMoV5-9, 17CrMoV10, Ovako 225A (0.16% C; 0.30% Si; 0.85% Mn; 1.85% Cr; 0.55% Mo; < 0.30% Ni 16CrMo8-6), b) Cr-Mo-V-alloyed steels with a C content of about 0.3%, such as 33CrMoV12-9 in accordance with DIN EN 10085, 32CrMoV13-10 (GKH – melted in electric furnace and vacuum degassed) and 32CrMoV5 (GKP – melted in electric furnace or GKH YW melted in vacuum furnace and remelted ĺ highest purity), as well as c) CrMoV-alloyed steels with a C content of about 0.4%, such as the steel 40CrMoV13-9 standardised in DIN EN 10085, and in special cases the hot-working steel X40CrMoV5-1. 7.4 Selection of Materials: Strength Values 513 The Cr-Al-Mo-alloyed or Cr-Al-Ni-alloyed steels have been so far rarely used for gears. There is no reason for this, because with the controlled nitriding technologies in gas or plasma, layer formation (especially the thickness and the type of the compound layer) is completely under control. The steels named under a) to c) as well as the case-hardened and tempering steels, the steels for open-die forgings and the low-alloyed sorts of cast steel are all quenched and tempered before being nitrided. (Refer to Section 10.3.2.1 for quenching and tempering.) As comprehensive tests with the AFP steel 27MnSiVS6 (Al content between 0.05 and 0.1 % and total contents of Cr+V+Mo+Ti of around 0.5%) showed, excellent nitriding results were achieved. With that, substitutes are available for expensive low-alloyed steels [7.4/51]. The current state of the art of nitriding and nitrocarburising ferrous materials is summarised in [7.4/52]. Reference is made to this in the following. The selection of materials has to correspond to the nitridability . Table 7.4/11 depicts the definition of nitridability. The table lists selected parameters of nitridability and the effects that material and technology tend to produce. Their enormous variety indicates the large potential for specific modification of properties [7.4/52, particularly pp. 241–258]. In most cases a composite layer forms, consisting of a compound layer and a precipitation layer , (refer to Section 10.3.2.6). As in the methods based on transformation hardening through the formation of martensite (surface layer, case and carbon-nitride hardening as well as nitrogen-content increase), the tooth endurance limit is the result of the hardening process, which is described by the parameters of surface or edge hardness, nitriding hardness depth, and core hardness. With its positive influence on the fatigue strength during nitriding, the residual stress curve, characterised by steel composition, nitriding temperature and nitriding time, plays a major role [7.4/52, particularly pp. 75–80]. The strongest influence on the surface-layer hardening of the tooth is exerted by the nitride- forming alloying elements , such as aluminium, chrome, molybdenum and vanadium, known as special nitride formers . They are present in varying amounts in the ferrous materials suitable for nitriding and have an influence on the surface hardness, the nitrogen diffusion in the depths, and thus the achievable nitriding hardness depth. At the same time, in the sense of hardenability, as special carbide formers (with the exception of aluminium) they act decisively on the amount of core hardness. Thus, each nitrided steel has its own characteristic hardness profile, which is expressed by the hardness-depth curve (refer to Section 7.4.4.9). The hardness profile is influenced by the initial structure, the nitriding temperature and the nitriding time [7.4/52, in particular pp. 254–256]. The parameters are determined in a metallographic transverse microsection by measuring the compound layer thickness and the hardness profile. Because of the multitude of factors influencing the parameters for the endurance limit of the gear, it is difficult for the person in charge to select the right information. Table 7.4/12 provides information for selecting layers appropriate for the strain. The type and thickness of the compound layer as well as the nitriding hardness depth are adjusted via the nitriding parameters. The surface hardness (refer to Table 7.4/13a, b) depends on the type of steel, the initial structure, the nitriding temperature and the nitriding time, as well as the measuring force. It is impossible to specify anything concretely. The core hardness results from the strength after quenching and tempering, influenced by how hot the tempering temperature is. 514 7 Design of gearing and gear transmissions Table 7.4/11 Schematic Depiction of the Effect of the Nitriding Conditions and the Basic Material on the Nitriding Result, from [7.4/52, p. 257] VS RH ΔH NHD Rht ΔX/NHD ε Nitriding conditions T Ĺ t Ĺ KN Ĺ Ĺ = Ļ Ļ Ĺ Ĺ = Ļ Ļ Ĺ = Ļ Ļ Ĺ Ĺ = Ļ Ļ Ĺ = = = = = = Material Composition Cr Ĺ Al Ĺ C Ĺ =Ļ Ĺ Ĺ ĻĻ Ĺ Ļ Ĺ Ĺ ĹĹ ĹĹ =Ļ Ļ Ĺ ĹĹ =Ĺ Ļ Ļ = Ļ Ļ Ĺ Ļ Structure N V(T AĹ) = Ĺ Ĺ = Ĺ Ļ = Ĺ = Ļ Ļ = Ļ Ĺ Ļ Ĺ - increasing; Ļ - decreasing; = unchanged; = Ĺ - unchanged or increasing; = Ļ - unchanged or decreasing. Definitions: VS – compound layer; RH – surface hardness; KH – core hardness; ΔH – hardness increase = RH - KH; NHD – nitriding hardness depth; Rht – surface hardening depth; ΔX – NHD minus Rht ; ε - nitride; T – nitriding temperature; t – nitriding time; KN – nitriding factor; Al, C, Cr – alloying components; N - normalised; V - quenched and tempered; TA - tempering temperature. Nitridability : Material property - Activation of ferrous materials for material and structural changes of surface layer formation dependent on defined nitriding conditions Table 7.4/12 Information for Selecting Nitriding Layers Appropriate for the Strain, According to [7.4.52, p. 242] Strain Compound layer VS Diffusion layer Type Thickness [ μm] Abrasion ε(γ’)-oxidised, ε(γ’), γ’, without VS >10 subordinated, high surface hardness Adhesion ε(γ’), γ’ >10 subordinated Contact fatigue ε(γ’), γ’, without VS 0 < 10 high surface hardness (> 600 HV 0.5), nitriding hardness depth > 0.35 mm, depth of the surface hardness > 0.15 mm, high ductility 1), high residual compression stresses Volume fatigue subordinated high surface hardness, hardness increase, nitriding hardness depth, core hardness, residual compression stresses, ductility 1) 1) High ductility ĺ compound layer: without VS; γ’; ε ↓; h(CL) < 10 μ m ĺ diffusion layer: quenched and tempered original condition; decarburised surface (Fe 3C) ↓ 7.4 Selection of Materials: Strength Values 515 Table 7.4/13a Hardness Values after Gas Nitriding or Plasma Nitriding1) Steel Distinguishing dimension s [mm]3) Tensile strength (core) Rm-min [N/mm2] Core hardness [HV10]2) (values rounded) Surface hardness (after nitriding) [HV10] min. 16MnCr5+N 100 700 220 500 20MnCr5+N 800 250 20MoCr4+N 700 220 30MnVS6+P (formerly 27MnSiVS6) >30 100 700 220 650 15CrMoV5-94) 17CrMoV10 100 > 100 250 900 850 280 265 750 Ovako 225A ( 16CrMo8-6) 100 > 100 160 1100 1000 340 310 24CrMo13-6 100 > 100 160 950 900 295 280 C45E, C45R 40 650 205 350 38Cr2 40 700 220 450 34Cr4 40 > 40 100 800 700 250 220 500 450 34CrMo4 40 > 40 100 > 100 160 900 800 700 280 250 220 550 500 500 42CrMo4 40 > 40 100 > 100 160 > 160 250 1000 900 800 700 310 280 250 220 550 550 500 500 34CrNiMo6 > 40 100 > 100 160 > 160 250 1000 900 800 310 280 250 600 550 500 30CrNiMo8 > 100 160 > 160 250 1000 900 310 280 600 550 36NiCrMo16 > 160 250 1000 310 500 31CrMoV9 > 40 100 > 100 160 > 160 250 1000 900 850 310 280 265 750 700 700 33CrMoV12-9 > 40 100 > 100 160 > 160 250 1050 950 900 325 295 280 800 750 700 32CrMoV5 > 40 100 > 100 160 900 850 280 265 750 700 32CrAlMo7-10 > 40 100 > 100 160 > 160 250 980 930 880 305 290 275 900 850 800 40CrMoV13-9 > 40 100 > 100 160 > 160 250 900 870 800 280 270 250 750 750 700 X40CrMoV5-1 250 1200 375 800 1) Temperature range 500 to 550 C 2) According to DIN EN 18265, Table A1: Hardness H [HV10] = 0.31 Rm (valid for R m 1385 N/mm2 with R2 = 1); at core hardnesses of 310 HV, depending on the machining process, higher costs could be incurred. 3) At characteristically smaller dimensions the tensile strength/core hardness is larger. This has a positive effect on the hardness after nitriding. 4) All following steels in the state +QT 516 7 Design of gearing and gear transmissions Table 7.4/13b Hardness Values after Gas, Plasma, or Salt-Bath Nitrocarburising1) for Information Purposes Steel Distinguishing dimension s [mm]3) Tensile strength Rm-min [N/mm2] Core hardness [HV10]2) (values rounded) Surface hardness [HV10] min. 16MnCr5+N 100 700 220 450 20MnCr5+N 800 250 20MoCr4+N 700 220 C45E, C45R+QT 40 650 205 350 30MnVS6+P 30 < s 100 700 220 650 38Cr2+QT 40 700 220 450 34Cr4+QT 40 > 40 100 800 700 250 220 500 450 34CrMo4+QT 40 > 40 100 > 100 160 900 800 700 280 250 220 550 500 500 42CrMo4+QT 40 > 40 100 > 100 160 1000 900 800 310 280 250 550 550 500 34CrNiMo6+QT > 40 100 > 100 160 1000 900 310 280 600 550 31CrMoV9+QT > 40 100 > 100 160 1000 900 310 280 750 700 30CrNiMo8+QT > 100 160 1000 310 600 1) Temperature range 500 to 590 C 2) According to DIN EN 18 265, Table A1: Hardness H [HV10] = 0.31 Rm (valid for Rm 1385 N/mm2, coefficient of determination: R2 = 1); at core hardnesses of 310 HV, depending on the machining process, higher costs could be incurred. 3) At characteristically smaller dimensions the tensile strength/core hardness is larger. This has a positive effect on the hardness after nitriding. DIN EN 10085 contains guide values only for the surface hardness. For the Al-free Cr-Mo-(V-) steels the value 800 HV 1 is given; for the Cr-Al-Mo-(Ni) steels this value is 950 HV 1. The footnote for this states: “Values serve as guide/information. The actual surface hardness can differ, depending on nitriding treatment and original tempering conditions.” According to ISO 6336-5 (DIN 3990-5), for nitrided gears the values of the pitting endurance limit σ Hlim and the tooth root strength σFE for the levels ML, MQ and ME for certain steel groups in specific areas of surface hardness are constant; refer to Section 7.4.4.9, Figure 7.4/20. With this in mind, Tables 7.4/13a and 7.4/13b contain hardness values [HV 10] for gear ma-terials which can be achieved through gas or plasma nitriding and nitrocarburising. Many firms have their own data, so the person in charge will receive advice when commissioning the work. It must be remembered that higher hardness values will be measured if lower measuring forces (HV 5 to HV 1) are chosen because, depending on the method, the Vickers diamond registers layers of different hardness, depending on the test load. DIN ISO 15787 (formerly DIN 6773) offers general information for selecting the hardness measurement method for surface layer hardened steels, produced through surface-layer hardening, case hardening and nitriding/nitrocarburising. Refer to Section 10.3.2.6 for nitriding/nitrocarburising. 7.4 Selection of Materials: Strength Values 517 7.4.3.6 Borided Gears Borided gears are only used in exceptional cases. Boride layers are beneficial in cases of high abrasive load, insufficient lubrication, high thermal load (up to 700 C), and at high sliding speed, when with point contact of the tooth flanks (crossed helical gears) nitride layers aren’t adequate anymore and when tooth root and tooth flank load are low. Concerning their boridability, expressed in terms of the structure, composition (goal: Fe 2B) and geometry of the boride layer, as well as surface-layer hardness and residual stresses in the boride layer [7.4/53], studies have been carried out [7.4/53], [7.4/54], [7.4/55] on unalloyed and low-alloyed case-hardening and tempering steels such as C15, 15Cr3, 18NiCrMo4, Ck45, 41Cr4, 42CrMo4, 50CrV4, and 34CrAlNi7. The gear pairing for an oil pump drive is borided in standard production in a fluidised bed (steel 41Cr4 quenched and tempered, boride layer thickness 20 to 50 μm) [7.4/54]. Further technologi- cal developments have led to such gear pairings now also being borided with plasma support in gas (BCl 3-H2-Ar mixture) [7.4/56]. The surface-layer hardness is dependent on the alloy character of the steel, the boride layer thickness and the force used for the Vickers hardness measurement [7.4/53]. Depending on the type of steel, the hardness of the boride layer (measured with microhardness HV 0.05 in the transverse microsection) is between 1600 and 2100 Vickers units [7.4/54]. Increasing carbon content raises the hardness, just as do carbide-forming alloying elements such as Cr, Mo, and V if the carbides are not released during boriding, whereas austenite-stabilising alloying elements such as Mn and Ni have no influence on the hardness [7.4/57]. With increasing carbon content and alloying element content (carbide formers), the growth of the boride layer on unalloyed and low-alloyed steels is inhibited [7.4/57]. Depending on the sort of steel, boride layers of up to 150 μm are possible. The wear properties are sufficient at layer thicknesses of approximately 50 μ m. Refer to Section 10.3.2.7 for boriding. 7.4.3.7 Coated Gears To increase scuffing load capacity and hinder the formation of micropitting, surface-layer- hardened gears can be subsequently coated with hard materials such as metal carbide, metal nitride, metal oxide or combinations of these as well as with carbon (diamond). Suitable as basic materials are nitrided tempering and nitriding steels of high core strength (Section 7.4.3.5) and case-hardened steels, particularly the steel M50NiL (AMS 6278, remelted in vacuum) and 13MoCrNi42-16-14 (Section 7.4.3.2). Because of the high degree of hardness of the so-called hard material layers (1000 to 3000 HV 0.1, depending on the layer type and thickness), the steel to be coated must have a high hardness, particularly in the surface layer, in order to ensure support for the hard material layer. The combination of hard material layer and support material must be carefully chosen [7.4/58]. With regard to gears, the combinations of nitriding-coating with hard material, such as with TiN [7.4/59], [7.4/60], for example, and case hardening-coating with hard material with WC/C have been studied [7.4/61]. The increased scuffing load capacity of the WC/C layer with several lubricants was verified, and the improved pitting load-bearing capacity was mentioned [7.4/62]. Suitability exists for mass production [7.4/63]. The choice of the coating method (in the vacuum as PVD or PACVD) depends on the tempering resistance of the basic material. Refer to Section 10.3.2.8 for coating. 518 7 Design of gearing and gear transmissions 7.4.3.8 Gears Made of Special Ferrous Materials The following examples illustrate the potential of special materials and strengthening treatments, the development of which has not yet been fully utilized. Maraging Steel (High-Strength Martensite-Hardening Steel) To manufacture gears and pinions, low-carbon steels (<0.03% C) with nickel (approx. 18%), cobalt (up to 15%), and molybdenum (up to 12%), as well as with low titanium and aluminium contents such as X2NiCoMo18-9-5 and X2NiCoMoTi18-12-4, are used. The heat treatment is simple. It consists of dissolubility treatment with air cooling and retaining at 400 to 480 C for 3 h. Dimensional changes and distortion are extremely small. These steels are characterised by very high strength and elongation values (even at low temperature). Because of the high alloy content, the achieved values of R p0.2 to 1900 N/mm2 and R m to 2200 N/mm2 are independent of the cross section. The steels have good corrosion resistance and high temperature stability. In addition to the high price, their disadvantage is the low wear resistance [7.4/64]. A drastic improvement in wear resistance can be achieved through plasma-nitriding, whereby the nitriding at around 400 C can simultaneously serve as a retaining period [7.4/65]. The wear resistance is better than the resistance of a case-hardened layer [7.4/66]. Sintering Steel From sintering steels, primarily gears with small characteristic dimensions are manufactured. Sintering steels/metals for moulded parts are standardised, for example, in DIN ISO 5755 (and DIN 30910-4). Their large variety in connection with the technologies of sintering makes the selection complicated. Their use for gears should be accompanied by consultations with competent firms. The disadvantage of sintered steels compared to rolled and forged steels is their low density. Principally, appropriate alloyed sintering steels (0.3 to 0.9 % C, Cu, Mn, Cr, Ni, Mo) are comparable based on their hardenability. Just as with rolled steels, they can be quenched and tempered. Inductive surface layer hardening is possible. As thermochemical methods, plasma-nitriding, vacuum-carburising and vacuum carbonitriding combined with high-pressure gas quenching are predominantly used [7.4/67]. Tooth root strength values were ascertained [7.4/68] on solely sintered and case-hardened gears ( m n = 1.6 or 2 mm, b = 10.6 or 12 mm) made of alloyed sintering steels. A density of >85% is a requirement for case hardening. Values around and over 90% were confirmed through density measurements. Carburisation was done in gas. Compared to the sintered state, through case hardening it is possible to increase the strength at the root by between 10 and 40 %, depending on the sintering steel and sintering technology. The values of compact, case-hardened gears are not achieved. 7.4.4 Strength Values for Gears The following information on values for pitting and tooth root endurance strength make reference to the given calculation equations, particularly in Section 6.5. They are mainly based on ISO 6336-5 (DIN 3990-5) and originate from tests on test gears with moduli m n = 3 to 5 mm and centre distances a = 91.5 mm (125 mm, 160 mm). Their applicability to larger dimensions is taken into account approximately through the size factors ZX, YX and a size-dependent sensitivity factor Yδ. A distinction is made between gears without surface layer hardening (refer to Sections 7.4.4.1 to 7.4.4.4) and gears with surface layer hardening (refer to 7.4.4.5 to 7.4.4.9). Whereas in the first group the endurance limit values are determined by the hardness, the endurance limits of the 7.4 Selection of Materials: Strength Values 519 second group result from the hardness profile, including the residual stress profile. The endu- rance limit is therefore reasonably increased. (refer to Section 7.4.5, “Comparison of Endurance Limit Values”). It should be noted that the tooth root strength values ı FE are local stress values and include the sensitivity effect of the test gear. For that reason they can only be roughly compared to the values which were ascertained on unnotched test pieces. 7.4.4.1 Gears Made of Unalloyed Steels and Cast Steel and Unalloyed Cast Iron The guide values for calculating the pitting endurance limit σ Hlim and the tooth root strength σFE for the material quality MQ in accordance with ISO 6336-5 (DIN 3990-5) are, depending on the hardness, summarised in Appendix 12 for the structural steels in accordance with DIN EN 10025, the unalloyed tempering steels in accordance with DIN EN 10083-2, the unalloyed cast steel sorts in accordance with DIN EN 10293 (all in normalised condition) as well as the unalloyed types of cast iron in cast condition in accordance with DIN EN 1561, DIN EN 1562 and DIN EN 1563. Because of the lower material costs, in actual practice it is common to thermally or thermo-chemically harden the surface layer of gears made of unalloyed steels and unalloyed types of cast iron in the cast condition (refer to Table 7.4/2). Strength values are not known for this. 7.4.4.2 Gears Made of Bainitic Cast Iron A widely varying range of older information exists about tooth flank and tooth root endurance strength. For the hardness range between 290 and 380 HB, an allowable Hertzian contact stress of σ H = 1100 to 1300 N/mm2 is given [7.4/69] (an increase compared to the cast state of more than 100%). According to [7.4/70, p. 168], the Hertzian contact stress reaches values of up to 1250 N/mm2. The American information sheet from 2007-01, “AGMA 939-A07: Austempered Ductile Iron for Gears”, includes information about permissible contact stress, permissible tooth root stress dependent on the Brinell hardness, and the associated S-N curve factors. Permissible contact stress σ HP: σHP = 2.45 H [HBW] + 376 (hardness of 240 to 510 HBW) (7.4/6) Life factor for the pitting strength Z N: ZN = 3.82 N - 0.0756 for N 5 107 (7.4/7) Permissible root stress σFP: σFP = 1300 – 2.0 H [HBW] (shot peened) σFP = 509 – 0.552 H [HBW] (machined) (hardness of 240 to 510 HBW) (7.4/8) The permissible tooth root stress drops with increasing hardness. High values can be achieved through shot peening. Life factor for the tooth root strength Y N: YN = 10.98 N - 0.1606 for N 3 106 (7.4/9) The hardness ranges given above refer to the American qualities AD 750 to AD 1600 in accordance with SAE J2477-2004-05 [7.4/71]. All characteristic mechanical values are included in this. The sorts EN-GJS-800-8 to EN-GJS-1400-1 standardised in DIN EN 1564 have hardnesses between 260 and 480 HBW. 520 7 Design of gearing and gear transmissions 7.4.4.3 Gears Made of AFP Steels For gears made of AFP steels, the standards still don’t indicate any values for pitting strength and strength at the root. When gears are manufactured from AFP steels, it is possible to use as a guide the approximate values of flank and root strength for tempering steels with comparable tensile strength. Potential candidates, for example, are the steels 30MnVS6 with R m-min = 700 N/mm2, 38MnVS6 with Rm-min = 800 N/mm2 and 46MnVS6 with Rm-min = 900 N/mm2 (compare with Table 7.4/14a). It has to be noted that in DIN EN 10267 (refer to Table 7.4/6a) the static strength values for rods are limited to dimensions between 30 and 120 mm. For forged pieces the static strength values must be agreed upon. To increase gear load capacity, surface layer hardening (refer to Table 7.4/8b for surface hardness) and nitriding (refer to Table 7.4/13a, b for surface hardness) can be applied, whereby a small Al content (0.05 to 0.1 %) and other microalloying elements (Cr+V+Mo+Ti 0.5%) have a positive effect on the nitridability [7.4/51]. Refer to Section 10.3.2.7 for nitriding/nitrocarburising. 7.4.4.4 Gears Made of Quenched and Tempered Steels and of Quenched and Tempered Iron Casting Materials The pitting endurance limit and tooth root endurance limit can be calculated from the hardness after quenching and tempering. Figure 7.4/13a, b (ISO 6336-5, DIN 3990-5) gives the ranges of variation of test results of the pitting endurance limit and tooth root endurance limit of quenched and tempered gears made of alloyed tempering steels in relation to the hardness. The tooth root strength is the tooth root endurance limit multiplied with the stress concentration factor (stress correction factor). Thus, σ FE = σFlim YST with YST = 2 for the geometry of the test gear. Entered are the equations for the three material qualities ML (low), MQ (medium), and ME (high) in accordance with ISO 6336-5, DIN 3990-5. Figure 7.4/13a Ranges of variation of pitting endurance limit σ Hlim of quenched and tempered gears made of alloyed tempering steels in relation to the hardness; the basis for the strength values is DIN 3990-5; ISO 6336-5 7.4 Selection of Materials: Strength Values 521 Figure 7.4/13b Ranges of variation of the tooth root endurance limit of quenched and tempered gears made of alloyed tempering steel, in relation to the hardness; σFlim tooth root endurance limit; σFE basic tooth root strength (local stress); the basis for the strength values is DIN 3990-5; ISO 6336-5 Table 7.4/14a depicts the endurance limit values σHlim and σFE of the material quality MQ for tempering steels in accordance with DIN EN 10083. The steels were chosen with increasing alloy content in such a way that in each group of dimensions the minimum tensile strength between 700 and 1000 N/mm 2 rises by 100 N/mm2 each. For quenched and tempered gears with very large characteristic dimensions (up to s 660 mm), Table 7.4/14b contains the endurance limit values. According to that, at the same static strength, the composition of the steel has no influence on the load capacity of quenched and tempered gears if the ranges of the characteristic dimensions are taken into account. In analogy to tempering steels, low-alloyed cast steel materials in accordance with DIN EN 10293 can be chosen according to hardenability (refer to Section 7.4.2.4). As Table 7.4/14c shows, despite the same chemical composition and hardness, the pitting endurance limit σ Hlim and the tooth root strength σ FE are lower than those of the tempering steels (refer to Table 7.4/14a). To increase the static and cyclical strength and to prepare an optimal structure for surface layer hardening, unalloyed lamellar-graphite cast iron in accordance with DIN EN 1561 (e.g., EN-GJL-350), malleable iron in accordance with DIN EN 1562 (e.g., EN-GJMB-700-2) and spheroidal graphite cast iron in accordance with DIN EN 1563 (z.B.EN-GJS-800-2), with modified chemical analysis, can likewise be quenched and tempered. Determinations for this are to be agreed upon with the manufacturer of the cast piece, as well as for the design (wall thickness) and verification of the properties on the sample piece. 522 7 Design of gearing and gear transmissions Table 7.4/14a Approximate Values for the Pitting Endurance Limit σHlim and the Tooth Root Strength σFE for Quenched and Tempered Steels up to s 250 mm in Accordance with DIN EN 10083, Material Quality MQ Type of steel Rp0.2 min. N/mm2 Rm min. N/mm2 Hardness on tooth HO min. HV 10 1) σHlim in N/mm2 2) (for NHlim = 5ǜ107 qH = 10 to 13.2) 3) σFE in N/mm2 2) (for NFlim = 3ǜ106 qF = 6.2) 4) s 16 mm: C45E, C45R 38Cr2 34Cr4 34CrMo4 490 550 700 800 700 800 900 1000 220 250 280 310 560 700 740 785 430 585 605 625 16 mm < s 40 mm: 38Cr2 34Cr4 34CrMo4 42CrMo4 450 590 650 750 700 800 900 1000 220 250 280 310 655 700 740 785 560 585 605 625 40 mm < s 100 mm: 34Cr4 34CrMo4 42CrMo4 34CrNiMo6 460 550 650 800 700 800 900 1000 220 250 280 310 655 700 740 785 560 585 605 625 100 mm < s 160 mm 34CrMo4 42CrMo4 34CrNiMo6 30CrNiMo8 500 550 700 800 700 800 900 1000 220 250 280 310 655 700 740 785 560 585 605 625 160 mm < s 250 mm: 42CrMo4 34CrNiMo6 30CrNiMo8 36NiCrMo16 500 600 700 800 700 800 900 1000 220 250 280 310 655 700 740 785 560 585 605 625 1) In accordance with DIN EN 18265, Table A1: Hardness H [HV 10] = 0.31 ǜRm (valid for Rm 1385 N/mm2 with R2 = 1) 2) Values determined according to ISO 6336-5, DIN 3990-5 3) qH = 13.2 in accordance with ISO 6336-2, DIN 3990-2; qH = 10 according to our own results 4) qF in accordance with ISO 6336-3, DIN 3990-3 At N Nlim a further drop in fatigue strength is to be expected. Assumption: refer to Table 6.5/13, No. 11 or respectively Table 6.5/14, No. 10. 7.4 Selection of Materials: Strength Values 523 Table 7.4/14b Approximate Values for the Pitting Endurance Limit σHlim and the Tooth Root Strength σFE for Quen- ched and Tempered Steels up to s 660 mm, Forgings in Accordance with DIN EN 10250-3, Material Quality MQ Type of steel Rp0.2 min. N/mm2 Rm min. N/mm2 Hardness on tooth HO min. HV 10 1) σHlim in N/mm2 2) (for NHlim = 5ǜ107 qH = 10 to 13.2) 3) σFE in N/mm2 2) (for NFlim = 3ǜ106 qF = 6.2) 4) 160 mm < s 330 mm 36CrNiMo4 30CrMoV9 40CrMoV13-9 36NiCrMo16 500 590 720 800 700 800 900 1000 220 250 280 310 655 700 740 785 560 585 605 625 330 mm < s 660 mm 34CrNiMo6 490 700 220 655 560 30CrNiMo8 590 800 250 700 585 40CrMoV13-9 720 900 280 740 605 36NiCrMo16 800 1000 310 785 625 1) In accordance with DIN EN 18265, Table A1: Hardness H [HV 10] = 0.31 ǜRm (valid for Rm 1385 N/mm2 with R2 = 1) 2) Values based on ISO 6336-5, DIN 3990-5 3) qH = 13.2 in accordance with ISO 6336-2, DIN 3990-2; qH = 10 according to our own results 4) qF in accordance with ISO 6336 -3, DIN 3990-3 At N Nlim a further drop in fatigue strength is to be expected. Assumption: Refer to Table 6.5/13, No. 11 or respectively Table 6.5/14, No. 10. Table 7.4/14c Approximate Values for the Pitting Endurance Limit σHlim and the Tooth Root Strength σFE for Quenched and Tempered Cast Steel in Accordance with DIN EN 10293, Material Quality MQ Type of steel Rp0.2 min. N/mm2 Rm min. N/mm2 Hardness on tooth HO min. HV 10 1) σHlim in N/mm2 2) (for NHlim = 5 107 qH = 10 to 13.2) 3) σFE in N/mm2 2) (for NFlim = 3ǜ106 qF = 6.2) 4) t 50 mm G28Mn6 +QT2 550 700 220 565 491 50 mm < t 100 mm G34CrMo4 +QT1 G42CrMo4 +QT1 G30CrMoV6-4 +QT2 G32NiCrMo8-5-4 +QT2 540 600 750 950 700 800 900 1050 220 250 280 325 565 607 650 715 491 511 530 560 100 mm < t 150 mm G42CrMo4 +QT1 G35CrNiMo6-6 +QT1 G32NiCrMo8-5-4 +QT1 550 650 650 700 800 850 220 250 265 565 607 630 491 511 521 150 mm < t 250 mm G42CrMo4 +QT1 G35CrNiMo6-6 +QT1 G32NiCrMo8-5-4 +QT1 350 650 650 650 800 820 205 250 255 544 607 615 482 511 514 1) Acc. to DIN EN 18265, Table A1: Hardness H [HV 10] = 0.31 ǜRm (valid for Rm 1385 N/mm2 with R2 = 1) 2) Values based on ISO 6336-2, DIN 3990-5 3) qH = 13.2 in accordance with ISO 6336-2, DIN 3990-2; qH = 10 according to our own results 4) qF based on ISO 6336-3, DIN 3990-3 At N Nlim a further drop in fatigue strength is to be expected. Assumption: Refer to Table 6.5/13, No. 11 or respectively Table 6.5/14, No. 10. 524 7 Design of gearing and gear transmissions 7.4.4.5 Surface-Hardened Gears The values for pitting endu- rance limit and tooth root endurance limit are dependent on the surface hardness that can be achieved. This is shown by the ranges of variation in Figure 7.4/14a and b for sur-face-hardened gears in relation to surface hardness [HRC]. The description in relation to surface hardness in HRC was chosen because in actual prac-tice the hardness is measured primarily using the Rockwell method. Figure 7.4/14 Ranges of variation of contact stress endurance limit ıHlim (a) and tooth root endurance limit ıFE (b) of surface-hardened, quenched and tempered gears made of alloyed tempering steels in relation to the surface hardness [HRC]; ıFlim nominal bending stress number; ıFE reference strength of tooth root (local stress) 7.4 Selection of Materials: Strength Values 525 Table 7.4/15 lists the approximate values for pitting endurance limit and tooth root strength for gears (up to s 660 mm) made of surface-hardened steels of the material quality MQ, which were previously quenched and tempered. The steels chosen are those with medium carbon contents of 0.34%, whose surface hardness is between 50 and 56 HRC after surface layer hardening and tempering (180 C/1 h). Because of the mentioned hardness requirement, it isn’t possible to specify the same steels as in Table 7.4/14a, b in every case. The structure of the surface layer shoul d be finely grained martensite, and the core structure low on ferrite, as required for alloyed tempering steels. The surface hardness is verified on the toothed part primarily with mobile measuring instruments. The hardening depth after surface layer hardening is determined in accordance with DIN EN 10328 on the gear/tooth segment after detaching one or more teeth. The necessary characteristic values, such as surface hardness, hardening depth, and core hardness, can be ascertained with the help of the hardness-depth curve (refer to Figure 7.4/15). The darkly etched layer in the macro etching figure is merely a quality attribute and not a measure of the hardening depth. Figure 7.4/15 Hardness-depth curve after surface-layer hardening (schematic) with the specifications of surface hardness, hardening depth, core hardness and possible measurement points In the course of the conversion of the DIN standards into international standards, it is expected that the former term hardening depth after surface layer hardening , abbreviated as DS, will be replaced by the term surface hardening depth , abbreviated as SHD (refer to DIN ISO 15787). DIN EN ISO 18265, Table A.1 can be used to convert the Rockwell hardness to the Vickers hardness and vice versa. 526 7 Design of gearing and gear transmissions Table 7.4/15 Approximate Values of Pitting Endurance Limit ıHlim and Tooth Root Endurance Strength ıFE for Quenched and Tempered, Surface-Hardened Steels, Material Quality MQ; Requirement: 50 HOmin 56 HRC ıFE 2) 4) (for NFlim = 3 106, qF = 8.7) 5) [N/mm2] 733 740 740 740 740 740 740 720 740 720 720 740 720 720 740 720 720 720 720 1) According to DIN EN 18265, Table A1: H [HV10] = 0.31 Rm (valid for Rm 1385 N/mm2 with R2 = 1) 2) Values according to ISO 6336-5 (DIN 3990-5) 3) qH = 13.2 according to ISO 6336-5 (DIN 3990-5); qH = 10 based on our own results 4) The values for ıFE are valid for a hardened tooth root. If the tooth root is not hardened, a root strength reduced up to 60% of quenched and tempered steels of same hardness at tooth has to be used (see Table 7.4/14a). 5) qF according to ISO 6336-5 (DIN 3990-5) In case of N Nlim an additional decrease of the endurance limits is expected. Assumption: see Table 6.5/13.2, No. 11 or Table 6.5/14, No. 10. ıHlim 2) (for NFlim = 5 107, qH = 10 to 13.2) 3) [N/mm2] 1180 1200 1190 1190 1200 1190 1190 1160 1190 1160 1160 1190 1160 1160 1190 1160 1160 1160 1160 Hardness at surface-har- dened tooth HOmin [HRC] 52 54 53 53 54 53 53 50 53 50 50 53 50 50 53 50 50 50 50 Hardness at quenched and tempered tooth Hmin [HV10] 220 250 280 310 205 250 280 310 250 280 310 220 250 310 220 235 310 220 310 Min. tensile strength Rm,min [N/mm2] 700 800 900 1000 650 800 900 1000 800 900 1000 700 800 1000 700 750 1000 700 1000 Min. yield strength Rp0.2,min [N/mm2] 450 550 660 750 400 560 650 800 550 700 800 500 600 800 460 540 800 490 800 Steel type 38Cr2 46Cr2 41Cr4 42CrMo4 46Cr2 41Cr4 42CrMo4 34CrNiMo6 42CrMo4 34CrNiMo6 36NiCrMo16 42CrMo4 34CrNiMo6 36NiCrMo16 42CrMo4 34CrNiMo6 36NiCrMo16 34CrNiMo6 36NiCrMo16 Distinguishing dimension 1) s [mm] 16 < s 40 40 < s 100 100 < s 160 160 < s 250 250 < s 330 330 < s 660 7.4 Selection of Materials: Strength Values 527 Between H = 20.3 and 64 HRC and H = 240 to 800 HV the equations [][ ] () [] ()223 HRC 14500 HV 100 HRCH H H⋅+ = − (7.4/10) [][] () []()100 HV 14500 HRC HV 223H H H⋅− = + (7.4/11) from [7.4/72] can be used. The mathematical difference from DIN EN ISO 18265, Table A.1 is a maximum of 2 HV or respectively 0.3 HRC. The limiting hardness HG is derived from the minimum surface hardness H O-min [HRC or respectively HV] (H G = 0.8 ǜ HO-min). Table 7.4/16 (excerpt from DIN ISO 15787) shows the correlation. Table 7.4/16 Connection between Minimum Surface Hardness According to Vickers or Rockwell C and Limiting Hardness (Corresponding to 80% of the Minimum Surface Hardness), Excerpt from DIN ISO 15787 for the Range 48 to 58 HRC Limiting hardness HG HV 2) Minimum surface hardness HO-min in HV and HRC 1) HV HRC 400 485 to 515 48 to 50 425 520 to 545 51, 52 450 550 to 575 53 475 580 to 605 54, 55 500 610 to 635 56, 57 525 640 to 665 58 1) Measurements with other measuring force, such as HRA HR 15 N, HR 30 N, HR 45 N, refer to DIN ISO 15787. 2) The limiting hardness is used to determine the hardening depth after surface-layer hardening DS. Note: The table can be used as a guide (it may not be used as an official table of comparative hardness). In the guidelines, the recommended hardening depths for gears are given based on the normal module mn. For tooth-by-tooth gear hardening, values from 0.15 ǜ mn to 0.3 ǜ mn and for single shot gear hardening with fillet, values up to 0.4 ǜ mn, are recommended (refer to Table 7.4/17). Table 7.4/17 Hardening Depths after Surface-Layer Hardening for Flame- and Induction-Hardened Gears DS = 0.2 ǜ mn for m n < 10 mm; DS = 0.15 ǜ mn for mn ≥ 10 mm; Criterion: Tooth Root Strength Normal module mn in mm Hardening depth DS in mm 3 to 3.5 0.7 + 0.7 4 to 4.5 0.9 + 0.9 5 to 6 1.2 + 1.0 7 to 8 1.6 + 1.3 9 to 12 2.0 + 1.6 14 to 16 2.7 + 2.2 18 to 22 3.5 + 2.8 25 to 28 4.5 + 3.6 For flame hardening, beginning at mn = 8 mm, a check must be done on whether the plus tolerances of approximately 80% of the nominal value are sufficient. For this the hardness depth is defined as the distance from the surface to the point at which the hardness is 80% of the surface hardness (hardness in HV). Principally it must be noted that different minimum and optimum values apply for the tooth root and pitting strengths. 528 7 Design of gearing and gear transmissions In Table A.5, DIN ISO 15787 specifies SHD values and limiting deviations among others for induction and flame hardening, which are similar to those in Table 7.4/17. Induction- and flame-hardened gears have so far only been studied [7.4/73] in detail up to m n = 8 mm concerning the load capacity of the tooth flank and root, that is, in the range of hardening depths up to 2 mm. Comparative results on induction- and case-hardened gears with m n = 8 mm (thickness of diffusion layer = 1 to 2 mm) are given, with and without shot-peened tooth root. Through shot peening the tooth root endurance strength is raised by approximately 50% after induction hardening, without reaching that of case hardening and shot peening. There is a prospect of load capacity research on induction-hardened gears with m n = 10 to 30 mm and corresponding hardening depths [7.4/74]. In cases of unhardened tooth root fillets (flank hardening), another way to raise tooth root endurance strength is the mechanical hardening of the tooth root surface [7.4/73]. In special cases, to prevent excessive distortion, large carburised gears made of case-hardened steel have been flame or induction hardened [7.4/75]. In order to raise the hardness depth and thus especially the pitting endurance limit, nitrided gears made of tempering steel are subsequently induction hardened in some cases [7.4/76]. A summary of results on the influence of the material and surface layer hardening through flame and induction hardening on the load capacity of the flank and root can be read in [7.4/73]. Reports on test rig results with gears made of the steel 16MnCr5 – both carburised and non-carburised – whose flanks were remelted using an electron beam, can be found in [7.4/77]. The increase in wear resistance (reduction in weight loss) with a remelted as opposed to a case-hardened gear surface is impressive. According to [7.4/78], gears with m n = 4 mm were studied after case hardening (0.2CrNi case- hardening steel) and laser-beam hardening (0.4CrNiMo tempering steel) with regard to allowable contact stress. Here the laser-beam-hardened gears achieved 92% of the tooth flank endurance limit of the case-hardened gears. Refer to Section 10.3.2.3 for surface layer hardening. 7.4.4.6 Case-Hardened Gears When choosing materials, the requirements on materials and heat treatment (ISO 6336-5; DIN 3990-5) are to be used as a starting point, as well as Figure 7.4/16, which lists the ranges of variation for pitting and tooth root endurance limits for gears made of case-hardened steels. There the endurance limit in the three material qualities is determined from ranges of the core hardness H K, independent of the surface hardness (between 56 HRC and 64 HRC). There are three ranges for the tooth flank and four for the tooth root (Table 7.4/18). Table 7.4/18 Ranges of Core Hardness HK for Tooth Flank and Tooth Root Material quality Tooth flank Tooth root ML (low) H O = 56 to 64 HRC 20 HRC HK < 34 HRC 20 HRC HK < 28 HRC MQ (medium) Steels with nickel 1.5% H O = 58 to 63 HRC 34 HRC HK < 40 HRC 28 HRC HK < 34 HRC 34 HRC HK < 40 HRC ME (high) Steels with nickel 1.5% H O = 59 to 63 HRC HK ≥ 40 HRC HK ≥ 40 HRC 7.4 Selection of Materials: Strength Values 529 However, an influence is exerted by the surface hardness, and because reliable results are lacking, it cannot be taken into account. Specifically, when there is an increase in surface hardness from 56 HRC to 64 HRC, the flank and root strength also increases. In a comprehensive analysis of the fatigue load capacity of case-hardened gears, taking into account multiple influences, such as surface oxidation depth and residual surface stresses, it was shown that the strength increases remarkably in the given hardness range of case-hardened gears [7.4/79]. According to the experience of the authors, as opposed to ISO 6336-5, the values for pitting endurance limit σ Hlim = 1500 N/mm2 above MQ (dashed, framed area in Figu- re 7.4/16a) and the values for the tooth root nominal bending stress number σ Flim > 500 N/mm2 (dashed area in Figure 7.4/16b) are not reliably achieved. a) Figure 7.4/16 Ranges of variation for pitting endurance limit σ Hlim (a) and tooth root endurance limit (b) of case-hardened gears in relation to the surface hardness (ISO 6336-5, DIN 3990- 5); σ Flim nominal bending stress number (shape strength); σFE reference strength of tooth root (local stress) (basic tooth root strength) b) 530 7 Design of gearing and gear transmissions For gears that should have core hardnesses of ≥ 34 HRC, case-hardened steels with nickel contents of ≥ 1.5% are used (recommendation of DIN 3990-5). For this, according to DIN EN 10084, three steels of medium and high hardenability each are suitable (refer to Figure 7.4/7): • Medium hardenability: 17CrNi6-6+HH, 15NiCr13+HH, 20NiCrMoS6-5+HH, • High hardenability: 18CrNiMo7-6+HH, 14NiCrMo13-4+HH, 20NiCrMo13-4+HH. In [7.4/80] it is pointed out that, through the use of steels alloyed with nickel, the root strength (static and limited life strength) rises with increasing core hardness to an optimum, but with brittle material characteristics (e.g., with Mn-Cr alloyed steel) a drop can occur already above 300 HV. At low notch effect ( Y S 2.3), the bearable maximum loads are dependent on the core strength (yield stress), and incipient cracks only form in the tooth root at higher load after plastic deformation of the tooth [7.4/81]. The lower limit of load capacity of case-hardened gearing, due to an incipient crack in the tooth root, is at stress concentrations of Y S > 2.3. The characteristic values of surface/surface layer hardness, case-hardening depth and core hardness are to be verified after case hardening. The surface layer and core structure obtained are to be documented. To do this, in addition to the destruction of a part to take samples, the representative test sample or increasingly the coupon sample is used (refer to ISO 6336-5 and DIN 3990-5, Table 5). The dimensions of these round samples are to be agreed upon with the customer. In actual practice, samples with identical material and a diameter of 35 mm and a length of 70 mm are used. The result on the gear, thereby, especially in cases of large characteristic dimensions, is not always comparable [7.4/82]. The case-hardening depth CHD is determined according to DIN EN ISO 2639, taking into account the limiting hardness and test load (e.g., 550 HV1). In cases of mass production with small characteristic dimensions, it is determined on the gear or tooth segment after extracting one or more teeth. The core hardness should be measured at the same time. ISO 6336-5 / DIN 3990-5 recommend as a measuring point a vertical distance to the 30 tangent of 5 CHD and a minimum of 1 m n. The associated hardness value can then only be determined using the hardness-depth curve (low force hardness HV1) (Figure 7.4/17). In actual practice, it has become common to measure the core hardness at the centre of the tooth root (refer to Figure 7.4/17). With the results of macro hardness measurement in accordance with Rockwell or Vickers (e.g., HV 10 or HV 30), the conversion to the tensile strength can be done (see Appendix 11). For larger gears, for which the destruction for the measurement of core hardness would be too expensive, a simulation model for predicting the core hardness with a high degree of accuracy has been developed [7.4/83]. It takes into account the influencing factors of the selection and costs of the material, the hardenability, case hardening, quenching and gear geometry. From the tests it is clear that the core hardness measured beneath the tooth root fillet is higher than the core hardness inside the tooth. For the case-hardened surface layer structure (material quality MQ), the goals are a finely-grained martensite, no visible mesh or bone carbides at an enlargement of 500 times (ISO 6336-5: photo micrograph 20b, maximum length of any carbide < 0.02 mm), less than 30% residual austenite (ISO 6336-5: < 25%), no visible surface decarburisation at an enlargement of 500 times (ISO 6336-5: a maximum drop in hardness of 40 HV to a depth of 0.1 mm), and only negligible surface oxidation (ISO 6336-5: surface oxidation depth dependent on the case-hardening depth). 7.4 Selection of Materials: Strength Values 531 Figure 7.4/17 Hardness-depth curve after case hardening (schematic) with surface hardness, case-hardening depth, core hardness and potential measuring points specified The requirement of a ferrite-free martensite/bainite structure then refers to the chosen measuring point of the core hardness. If the ferrite-free structure aimed for exists in the middle of the tooth, it certainly also applies for the measuring point beneath the tooth root fillet. The occurrence of ferrite in the non-carburised core material is dependent on the quenching speed and on the hardenability-increasing alloying element content in the steel. From continuous Time-Temperature-Transformation diagrams (see Figure 7.4/5), with the help of the cooling time t 8/5 from 800 to 500 C, or respectively with the help of the cooling parameter λ = (cooling time t8/5 in s)/100, (7.4/12) it is possible to estimate the resulting structure for the given content of the chemical analysis. From a multitude of Time-Temperature-Transformation diagrams for conventional case-hardening and tempering steels, the relationship between the chemical composition, the cooling parameter λ and the associated hardness was derived. The following applies according to [7.4/18] for ferrite-free cooling: ln λ F = 3.802 C + 2.216 Mn + 2.344 Cr + 1.028 Ni + 1.835 Mo – 7.036 (7.4/13) Coefficient of determination R2 = 0.80 Residual variance SR = 0.851 From this the hardness HF of the ferrite-free structure results: HF = 686 C + 54.4 Si + 105 Cr + 20.6 Ni + 615 V –194 C Cr – 2169 C V – 12.6 ln λF + 33.1 [HV] (7.4/14) Coefficient of determination R2 = 0.69 Residual variance SR = 41 HV Validity of both equations for th e chemical elements in mass-%: C = 0.10 to 0.65; Si = 0.06 to 2.00; Mn = 0.20 to 1.90; Cr to 2.70; Ni to 4.75; Mo to 1.0; V to 0.3. 532 7 Design of gearing and gear transmissions With the values from the chemical analysis corresponding to the lower limit HH min from Figure 7.4/9, the following is true: λF = 0.157 (cooling time of 800 to 500 C = 15.7 s), and the hardness HF of the ferrite-free structures would be 245 HV ( 21 HRC). The result of the comparative rating of the case-hardened steels based on hardenability assumes that the case-hardening steels alloyed with nickel are allocated the corresponding distances from the end face for the hardness of 34 HRC and 40 HRC on the lower limit +HH min in accordance with DIN EN 10084 (refer to Figure 7.4/7), and the diameter is calculated with the relationship d = f( E) for H = 0.35 in accordance with Figure 7.4/10a. Table 7.4/19 shows the result. Table 7.4/19 Distances from the End Face and Associated Diameter for Nickel-Alloyed Case-Hardening Steels (H = 0.35) in Accordance with DIN EN 10084 Steel +HH min Distance from the end face E [mm] for 34 HRC Diameter d [mm] Distance from the end face E [mm] for 40 HRC Diameter d [mm] 17CrNi6-6 11 41 4 8 20NiCrMoS6-5 13 54 7 20 15NiCr13 15 69 9 30 14NiCrMo13-4 25 167 13 54 18CrNiMo7-6 30 234 11 41 20NiCrMo13-4 >> 50 > 1000 50 (estimated) 661 From this and with reference to Figure 7.4/16, Table 7.4/20 was generated. It lists the main characteristic values of case-hardened, nickel-alloyed steels of the material quality MQ, taking the characteristic dimension into account. Additionally the steels 16MnCr5+HH and 20MnCr5+HH were included in the table. It is assumed that all steels are hardened from the core-hardening temperature according to Table 10.3/2b (refer to Section 10.3.1). The surface hardness H O = 58 + 5 HRC for the material quality MQ is achieved through tempering. The tooth endurance limit values in accordance with Table 7.4/20 apply, taking minimum case- hardening depth values into account. The characteristic dimension s min is an approximate value, which makes reference to the lower limit of hardenability HH with oil quenching and with the quenching intensity H = 0.35. In actual practice, gears with larger characteristic dimensions are also manufactured, whereby in some cases lower σ Flim and σ FE values are then possible. Module-dependent case-hardening depths can be selected according to Table 7.4/21 (minimum values: root ≈ 0.1 mn, flank ≈ 0.15 mn). In the case of large diameters and small modules in particular, it is recommended that the strength depth profile is compared with the stress depth profile (refer to Appendix 7). This means that the module (strength at the root) and the gear diameter and the depth profile of the contact stress (pitting strength, flank fracture) are determining factors for the case-hardening depth. 7.4 Selection of Materials: Strength Values 533 Table 7.4/20 Approximate Values of Pitting Endurance Limit ıHlim and Tooth Roo t Endurance Strength ıFE for Case-Hardened Alloyed Steels according to ISO 6336-5 (DIN 3990-5), Material Quality MQ Limit load cycle number NFlim 3 106 1) +HH min means the lower limit of the hardenability scatter band, which is valid for the distinguishing dimension s. 2) Approximate values for un-carburised, hardened tensile specimen ( d = 10 … 12 mm); Rp0.2/Rm 0.6 3) Values according to ISO 6336-5 (DIN 3990-5), valid for surface hardness depths CHD root 0.1 m n and CHD flank 0.15 mn 4) qH according to ISO 6336-2 (DIN 3990-2) 5) qF according to ISO 6336-3 (DIN 3990-3) 6) H = 0.35 7) According to practical experiences, rather lower than listed in ISO 6336-5 (DIN 3990-5) In the case of N Nlim an additional decrease of the endurance limits is expected. Assumption: see Table 6.5/13.2, No. 11 or Table 6.5/14, No. 10. Limit load cycle number NHlim 5 107 ıFE 3) qF = 8.7 5) [N/mm2] (850) 7) 920 ıHlim 3) qH = 13.2 4) N/mm2 (1450) 7) 1500 Min. yield strength 2) Rp0.2 N/mm2 680 760 740 800 820 760 800 840 Surface hardness at tooth HO [HRC] 58 to 63 Core hardness of tooth HK [HRC] 34 Distinguishing dimension 1) s [mm] for oil quenching 6) 25 45 65 80 105 240 330 1000 Type of steel 1) 16MnCr5+HH min 20MnCr5+HH min 17CrNi6-6+HH min 20NiCrMoS6-4+HH min 15NiCr13+HH min 14NiCrMo13-4+HH min 18CrNiMo7-6+HH min 20NiCrMo13-4+HH min 534 7 Design of gearing and gear transmissions Table 7.4/21 Recommended Case-Hardening Depths for Gears (Root Strength Aspect) Normal module mn in mm Case-hardening depth CHD in mm (drawing information) Tooth flank Tooth root 1.5 to 2.25 0.3 + 0.2 0.3 + 0.2 2.5 to 3.5 0.5 + 0.3 0.5 + 0.3 4 to 5.5 0.8 + 0.4 0.8 + 0.4 6 1.0 + 0.4 1.0 + 0.4 7 1.2 + 0.5 8 9 1.5 + 0.6 10 1.2 + 0.5 11 1.8 + 0.7 12 13 2.1 + 0.8 1.5 + 0.6 14 15 2.4 + 1.0 16 1.8 + 0.7 18 2.7 + 1.1 20 3.0 + 1.2 2.4 + 1.0 22 3.3 + 1.3 24 3.6 + 1.4 25 3.8 + 1.5 3.0 + 1.2 28 4.2 + 1.7 30 4.5 + 1.8 32 4.8 + 1.9 3.2 + 1.3 Principally it must be noted that different minimum and optimum va-lues apply for the tooth root strength and pitting strength [refer to Equations (7.4/15), (7.4/16), and (7.4/16a) and Appendix 7]. Stock allowances must be taken into consideration. The plus tolerances are specified in agreement with DIN ISO 15787. The suitability of these was confirmed in a compre-hensive study assessing process suitability on heat-treated compo-nents [7.4/84]. ISO 6336-5 specifies ranges CHD = 0.15 m n to mn = 10 mm and beyond that CHD = 0.083 m n + 0.67 (refer to Figure 7.4/18). Figure 7.4/18 Values of case-hardening depth CHD ; ISO 6336-5 (root strength aspect) 7.4 Selection of Materials: Strength Values 535 When selecting the case-hardening depths, attention must be paid to the fact that the value in the tooth root is lower, because of the slower C diffusion there compared to the tooth flank. Optimal values for pitting and load capacity of the tooth root are given in [7.4/85], taking the radius of curvature into account. Influencing factors for consideration of a deviating case-hardening depth are named. Their validity must still undergo further testing (refer to Sections 6.5.1.3 and 6.5.2.3). For the flank load capacity (pitting strength), it is therefore recommended to prove whether the depth profile of the local strength always is above that of the local stress. According to Appendix 7, this test can be done in particular for the steels 16MnCr5 and 18CrNiMo7-6. According to the von Mises theory (refer to [7.4/86]), for steel the depth of the equivalent stress maximum due to Hertzian contact stress is ()2 Hwt 1 t bıvG maxı Nm m tanĮ0.62 cosĮ1000 100 1 cos ȕ =⋅ ⋅ ⋅ + d uzu (7.4/15) with σH - Hertzian contact stress d1 - Reference diameter of the pinion u - Gear ratio ( u = z2/z1, z2 z1) (other quantities according to ISO 6336, DIN 3990). With wt t ȕ0 and ĮĮ 20 ≈≈ ≈D the following can be used as an approximation: ()2 H1 ıvG maxı Nm m 0.211000 100 1 ≈⋅ ⋅ + duzu (7.4/16) The depth of the equivalent stress maximum zσvGmax is a basic guiding quantity for determining the case-hardening depth CHD to ensure flank strength. To avoid flank damage that originates below the surface because of Hertzian flank contact, ISO 6336-5 recommends the approximate value Hw 1 w t 2 Hb 2 1ı sinĮ cosȕ=⋅+dzCHDUz z (7.4/16a) with UH = 66000 N/mm2 for the quality MQ/ME. The recommended CHD value is larger than the depth of the equivalent stress maximum Z ıvGmax . In more precise analyses, the stress depth profile must be compared with the strength depth profile (refer to Appendix 7). The selection of the right steel and achieving the right values for surface hardness, case-hard-ening depth, and core hardness are just one part of the job of ensuring the tooth flank and tooth root load capacity of case-hardened gears, given their very broad range of industrial applications, as opposed to surface layer-hardened or nitrided gears. The case hardening of gears leads to residual stresses σ E in the surface layer, which are contained globally in the fatigue strength values. A mathematical analysis of strength for the fatigue strength of case-hardened gears, taking residual stresses of the surface layer into account, was carried out and confirmed its positive influence [7.4/79]. 536 7 Design of gearing and gear transmissions According to [7.4/87], the σE profile from the local case depth curve can be roughly estimated after the case-hardening process: σE = – 1.25 ( Hx – H K) [N/mm2] for ( Hx – H K) 300 (7.4/17) σE = 0.2857 ( Hx – H K) – 400 [N/mm2] for ( Hx – H K) > 300 (7.4/18) with Hx - Respective hardness at a distance x from the surface HK - Core hardness . To calculate the load capacity of the tooth root in accordance with ISO 6336-3 / DIN 3990-3, taking into account the increased endurance limit achieved through shot peening, the factor YRS is suggested [7.4/88] (refer to AGMA 938-A05, “Shot Peening of Gears”). With the further development of known and new surface layer hardening and manufacturing methods, even more ways to increase load capacity in case-hardened gears are produced. The following are included: • There are the well-known methods of cold surface layer hardening (refer to Table 7.4/2) through shot peening or deep rolling, with their endurance-limit-raising effect from generating residual compression stresses in the surface layer [7.4/89]. The positive effect of cold hardening on the limited life and endurance strength through peening is more manifest [7.4/89, 7.4/90] than the effect on the tooth flank. More detailed information about shot peening of gears can be found at [7.4/91] (refer also to AGMA 938-A05, “Shot Peening of Gears”). • Another method of hardening the surface layer of case-hardened gears is high-pressure water jets [7.4.92]. Because of the generation of residual compression stresses, which do not reach as deeply as shot peening but do not modify the surface topography, water jets are advantageous in certain cases. • Rolling the tooth root surface of case-hardened gears was studied [7.4/93], but because of the high costs of tool production, it is out of the question except in cases of mass production. • Mechanical finishing methods such as honing [7.4/94], vibro-honing, CASE finishing, and superfinishing [7.4/95] are practiced on case-hardened gears with the aim of smoothing the surface, preventing micropitting, and in some cases generating residual compression stresses. • The combination of case hardening with subsequent WC/C coating in the region of typical tempering temperatures (200 C) has been introduced into practice to improve scuffing and micropitting load capacity [7.4/61]–[7.4/63], where case-hardening steels are used in accordance with DIN EN 10084. • If the high-temperature case-hardening steel 13MoCrNi42-16-14 is used with 1.0 to 1.3 % V in accordance with DIN EN ISO 683-17 (also called X12MoCrNiV4-3-3), subsequent gas nitriding (500 C) or plasma nitriding (400 C) after case hardening leads to a further increase in the load capacity of the tooth root and flank [7.4/31]. • The optimisation of the tooth root geometry (16MnCr5 steel) has been successfully introduced to some extent [7.4/96, 7.4/97]. With additional shot peening, the load capacity of the tooth root can be increased by up to more than 30% (refer also to AGMA 938-A05, “Shot Peening of Gears”). Use of many of the additional methods described for increasing the load capacity of case-hardened gears is mainly a question of cost. Influences from the case-hardening process on the load capacity of case-hardened gears, such as surface carbon content, surface oxidation, and residual austenite content, are discussed in Section 10.3.2.4. 7.4 Selection of Materials: Strength Values 537 7.4.4.7 Carbonitrided Gears The material strength values of carbonitrided gears reach or in special cases exceed case-hardened gears, as is known from past research [7.4/98], {7.4/99]. For carbonitrided gears made of case-hardening steel, case-hardening depths of 0.25 m n are recommended. When tempering steel is used, because of the higher core hardness the hardness depth can be reduced to (0.07 to 0.09) m n in some cases [7.4/98], whereby the depth of the equivalent stress maximum must be taken into account (refer to Appendix 7). For tooth flank and tooth root, just as for case-hardened gears, different case-hardening depths CHD are optimal [7.4/100]. To raise the tooth root endurance strength, shot peening is a possibility. Refer to Section 10.3.2.5 for carbonitriding. 7.4.4.8 Nitrogen Case-Hardened Gears Because this method has only been in development for a short time, there are still no results on gear load capacity. However, from the characteristic values which exist for nitrogen case-hardened ferritic or martensitic non-corroding steels, it can be estimated that it is possible to achieve the pitting endurance limit and root strength values of case-hardened gears. It is possible to achieve surface hardnesses of between 54 and 61 HRC. The hardness profiles of the surface layer are similar to those of case-hardened steels (X20Cr13, X15Cr13) or nitrided steels (Figure 7.4/19). Figure 7.4/19 Hardness-depth curve of the ferritic steel X6Cr17 and the martensitic steels X14CrMoS17, X15Cr13 and X20Cr13 after case hardening with nitrogen, from [7.4/47] The core hardness can be varied in wide ranges (e.g., between 250 and 500 HV1) through the selection of steel. The residual compression stresses close to the surface are higher than in case- hardened, tempered surface layers. Nitrogen-martensite has a significantly higher high-temperature stability than carbon-martensite, so nitrogen case-hardened gears are suitable for use at high temperatures (up to approx. 350 C) [7.4/50]. 538 7 Design of gearing and gear transmissions Yet to be answered is the question of hardenability of nitrogen case-hardened steels and thus the question of limits of use regarding the characteristic dimension. A definition of hardening depth is also still unresolved. The term “nitrogen case-hardening depth” which is used today is oriented towards the achievement of core hardness in the hardness profile [7.4/47]. 7.4.4.9 Nitrided/Nitrocarburised Gears The properties of nitrided gears are mostly compared with those of case-hardened gears. With regard to tooth flank and tooth root load capacity, nitrided gears are inferior to case-hardened gears. Only the scuffing load capacity is higher for nitrided gears [7.4/101]. Other advantages of nitrided gears over case-hardened gears are as follows: Nitriding with a compound layer produces better running-in characteristics. The modulus of elasticity of the pore-free γƍ compound layer of a gas- or plasma-nitrided steel can be slightly larger than 206000 N/mm 2 [7.4/102]. Regarding adhesive wear, the empirical structure factor is larger in compound layers formed through nitriding. With a formed compound layer the coefficient of friction is lower. The hot hardness of CrMoV alloyed tempering or nitriding steels is high if nitrided for a long period at low temperature [7.4/103]. With a formed compound layer there is temporary corrosion protection. A compound layer that is subsequently oxidised improves the corrosion resistance in aqueous media, for example. Due to the transformation-free nitriding (below 590 C), distortion is low. Due to nitrogen diffusion, dimensional changes are unavoidable. Since they are constant for a given type of steel and technology, they can be taken into account in the pre-machining. After low-distortion nitriding, generally no finishing is necessary. One disadvantage of nitrided as opposed to case-hardened gears is that it is not permitted to straighten nitrided workpieces because of the risk of crack formation in the compound layer. The core hardness of nitriding and tempering steels, produced by quenching and tempering ( R m = 900 to 1200 N/mm2 ≈ 29 to 40 HRC), is hardly lower than after case hardening of low-alloyed case- hardened steels. Following the mechanical pre-machining, additional stress-free annealing with subsequent di-mensional correction is usually necessary prior to the nitriding. As a result of the nitriding at temperatures below 590 C, the treatment time is generally much longer than for case hardening. With gas- or plasma-nitrided and nitrocarburised gears it is possible to bridge the gap in characteristic values between quenched and tempered and case-hardened gears (compare Figure 7.4/20 with Figure 7.4/13 and Figure 7.4/16). Figures 7.4/20a, b list the strength values in accordance with ISO 6336-5 and the ranges of variation of pitting and tooth root endurance limit in accordance with DIN 3990-5 in relation to surface hardness. From this the following can be concluded: • The highest values are achieved with gas- and plasma-nitrided quenched and tempered Al- free nitriding steels, and nitrocarburised gears are only usable for low-strain situations, which has to do with the layer formation technology. • Where the materials and the group of methods and the material qualities are the same, above 450 HV1 the endurance limits are constant, which must be questioned, however, just as in the case of the case-hardening values (refer to Section 7.4.4.6). • Through gas- or plasma-nitriding the case-hardening values with regard to the tooth root are achieved, but for the tooth flank they are lower. 7.4 Selection of Materials: Strength Values 539 Figure 7.4/20 Values for pitting endurance limit σHlim (a) and tooth root endurance limit (b) for nitrided and nitrocarburised steels in relation to surface hardness (based on ISO 6336-5) and ranges of variation (according to DIN 3990–5); σFlim nominal bending stress number, σFE reference strength of tooth root (local stress). One reason for the lower pitting strengths could be the higher residual compression stresses from nitriding as opposed to those from case hardening (refer to [7.4/107]). Through the suitable selection of nitriding and tempering steels and the optimal nitriding tech-nology, it is possible to achieve the characteristic values of tooth flank and tooth root endurance strength of case-hardened gears, as was proven long ago [7.4/104] (which, however, is only achievable in borderline cases and not for shock overloading). 540 7 Design of gearing and gear transmissions Remarkable about this information is that the steels used have high strength values after quenching and tempering (880 to 1200 N/mm2), and that the nitriding was done at low temperature (500 to 530 C) and long duration (up to 120 h) and primarily with a γ ƍ compound layer, but also without a compound layer. Conclusion: For heavy-duty gears, at high core strength it is necessary to produce thin compound layers (< 10 μm) or no compound layers at all and to restrict or prevent cementite precipitation on the grain boundaries in the diffusion layer. ISO 6336-5 recommends a compound layer of 25 μ m with predominantly ε nitride and little γƍ nitride. This is technically possible, but due to the method of processing, the nitriding hardness depth is also limited. For alloyed and unalloyed steels the surface hardness is specified with H < 500 HV2 or respectively H < 300 HV2. The gears used should be quenched and tempered. Verifiable by X-ray diffraction analysis or metallographically, the ε compound layer should be 10 to 30 μm thick and contain little γ ƍ phase. Compound layers and precipitation on the grain boundaries reduce the ductility [7.4/105] and have a negative influence on the flank strength [7.4/106]. By lowering the nitriding factor it is possible to limit the thickness of the compound layer or prevent its formation (refer to Section 10.3.2.6). A low nitriding factor, a low carbon content in the steel or a decarburising of the surface layer are limiting the unwanted precipitation at a given content of nitride formers. For certain dimensions, the following requirement (material quality MQ) is placed on the steels suitable for nitriding: After quenching and tempering (+QT): R m 900 N/mm2, KV > 50 J The steels are listed in Tables 7.4/22a and 7.4/22b. Table 7.4/22a Tempering Steels acc. to DIN EN 10083-3: C < 0.45%, Cr < 1.5%, Mo, V (Suitable for Nitriding) Steel Dimension d 1) [mm] Rp0.2 min [N/mm2] Rm [N/mm2] KV min [J] 34Cr4 ≤ 16 700 900 to 1100 - 37Cr4 ≤ 16 750 950 to 1050 - 41Cr4 16 to 40 660 900 to 1100 35 25CrMo4 ≤ 16 700 900 to 1100 - 34CrMo4 16 to 40 650 900 to 1100 40 42CrMo4 40 to 100 650 900 to 1100 35 34CrNiMo7-6 100 to 160 700 900 to 1100 45 30CrNiMo8 160 to 250 700 900 to 1100 45 Remark: None of the tempering steels achieves the KV value > 50 J at Rm > 900 N/mm2 in accordance with the standard. 1) Refer to Table 7.4/22b The notch impact work values can be higher than 50 J if the tempering is done in such a way that a high ratio of yield point to tensile strength R p0.2-min/R m-min is aimed for; for example, for the steel 24CrMo13-6, 700/900 = 0.78. Then a nearly ferrite-free tempered structure is possible (refer to Section 7.4.2.4). 7.4 Selection of Materials: Strength Values 541 Table 7.4/22b Nitriding Steels acc. to DIN EN 10085: C < 0.35%, Cr < 3.5%, Mo, V, no Al Steel Dimension d 1) [mm] Rp0.2-min [N/mm2]Rm [N/mm2]KV min [J] 24CrMo13-6 100 to 160 700 900 to 1100 30 31CrMoV12 100 to 160 735 930 to 1300 30 31CrMoV9 100 to 160 700 900 to 1100 35 33CrMoV12-9 160 to 250 700 900 to 1100 45 40CrMoV13-9 40 to 100 720 900 to 1100 25 Remark: None of the nitriding steels achieves the KV value > 50 J at Rm > 900 N/mm2 in accordance with the standard. 1) Dimensions below the given limit have higher strength and lower notch impact work, and above the given limit they have lower strength and higher notch impact work. In the use of plasma-nitrided gears, the tendency internationally is towards steels with a very high core strength (up to Rm = 1550 N/mm2), including the simultaneous application of long nitriding times (40 to 80 h) at different temperatures (500 to 560 C) [7.4/107]. Specified as the reason for the high endurance limit found in the rotating bending test with nitrided samples, as opposed to case-hardened samples, are the high residual compression stresses in regions close to the surface. For the gas nitriding of the ultra-pure steel 32CrMoV13 ( R p0.2 = 1060 N/mm2, Rm = 1250 N/mm2; Rp0.2/Rm = 0.85; KV up to 130 J) – used among others for gears – nitriding hardness depths ranging from 0.6 to 1 mm are common, whereby the nitriding time can be up to 200 h at 530 C. This process is called “deep nitriding” [7.4/108]. Here, of course, thick compound layers (up to 35 μm) are produced, the grinding off of which is taken into account. The approximate values for the pitting endurance limit σHlim and tooth root strength σFE are listed in Table 7.4/23a for gas- or plasma-nitrided nitriding steels and in Table 7.4/23b for nitrided and nitrocarburised tempering and case-hardening steels. With few exceptions, for gas- and plasma-nitrided nitriding steels, the ranges of the characteristic dimension are chosen in such a way that the minimum tensile strength is R m 900 N/mm2 and the yield stress ratio Rp0.2 / R m is between 0.76 to 0.85, depending on the steel. In the nitrocarburised condition, the load capacity values for the multiple-alloyed tempering steels tend to be too low in comparison with case-hardened steels which have been normalised. The minimum values for surface hardness [HV10] are listed in Table 7.4/13a. The Al-alloyed steel 32CrAlMo7-10 achieves the highest hardness value. For the Cr-Mo-V-alloyed nitriding steels, the values are between 700 and 800 HV 10. DIN ISO 15787 provides information on the selection of measuring force in relation to the layer thickness. The core hardness/strength results from the preceding quenching and tempering. The core structure should be practically ferrite-free and stabilised for the subsequent nitriding through sufficiently long tempering at a temperature of 50 K above the nitriding temperature (refer to Section 10.3.2.6). The thickness of the compound layer can be determined according to DIN 30902. The acronym CLT (compound layer thickness) is to be used for this. CLT np or respectively CLT p can be used to label the pore-free and porous areas. CLT values and their limiting deviations which are technologically achievable are listed in DIN ISO 15787. Figure 7.4/21 depicts the analysis of the hardness-depth curve (DIN 50190-3 hardness depth of heat-treated parts; determination of nitriding hardness depth) using the parameters of nitridability. 542 7 Design of gearing and gear transmissions The main difference between nitriding hardness depth NHD and case-hardening depth CHD is obvious. Since “nitriding hardness depth” is defined differently than “case-hardening depth”, it cannot be compared with it. According to Figure 7.4/21 the following are true: CHD 0.45 mm, NHD 0.85 mm. Figure 7.4/21 Hardness-depth curve after nitriding (schematic) with surface hardness, nitriding hardness depth, core hardness, and possible measuring points specified; comparison of nitriding hardness depth NHD and case-hardening depth CHD With the equations from [4.7/87] it should be possible to roughly estimate the residual stress profile after nitriding as well as after case hardening. The diverse influences on the hardness profile after nitriding (e.g., type of steel, tempering condition, temperature, duration and amount of nitrogen) also have an effect on the residual stress profile. With these equations, for example for the difference between the hardness H x (that is, the respective hardness of the hardness-depth curve starting at the surface and going to the core hardness HK), at (H x,(HV) - HK(HV) ) > 300 the value of the residual compression stress, with σ E = –500 N/mm2, is constant [compare with Equation (7.4/19)]. With that it isn’t possible to depict real residual stress profiles after nitriding (refer to Section 10.3.2.6). It is to be expected that the former acronym Nht will be replaced by the acronym NHD (nitriding hardening depth) (refer to DIN ISO 15787). The module-dependent nitriding hardness depths can be found in Table 7.4/24 (refer to Figure 7.4/21 for determining the nitriding hardness depth). The values are based on information from ISO 6336-5 (DIN 3990-5): “The endurance limit values given for nitrided test gears apply for nitriding depths of 0.4 to 0.6 mm” (module 3 to 5 mm). In the table, the ratio NHD /m is a minimum of 0.1. More precise tests with regard to damages originating beneath the surface must be done by comparing the strain depth profile with the case depth curve (refer to Appendix 7). 7.4 Selection of Materials: Strength Values 543 Table 7.4/23a Approximate Values of Pitting Endurance Limit ıHlim and Tooth Root Endurance Strength ıFE for Quenched and Tempered gas or plasma nitrided 1) Nitriding Steels, Material Quality MQ according ISO 6336-5 (DIN 3990-5) ıFE [N/mm2] NFlim = 3 106 4) 850 qF = 17 7) 1) Temperature range of nitriding: 500–550 C 2) The tensile strength/core hardness is larger for a smaller distinguishing dimension. This is positive for the hardness after ni triding. 3) According to DIN EN 18 265, Table A1: H [HV10] = 0.31 Rm (valid for Rm 1385 N/mm2 with R2 = 1) in the case of core hardness 310 HV, the machining costs can be higher. 4) Values according to ISO 6336-5 (DIN3990-5) 5) All steels in status +QT 6) qH according to ISO 6336-2 (DIN 3990-2) 7) qF according to ISO 6336-3 (DIN 3990-3) In the case of N Nlim an additional decrease of the endurance limits is expected. Assumption: see Table 6.5/13.2, No. 11 or Table 6.5/14, No. 10. ıHlim [N/mm2] NHlim = 2 106 4) 1200 qH = 11.4 6) Min. hardness after nitriding [HV10] 750 750 700 700 800 750 700 750 700 900 850 800 750 750 700 800 Min. core hardness [HV10] 3) (values rounded) 280 265 340 310 295 280 310 280 265 325 295 280 280 265 305 290 275 280 270 250 375 Tensile strength Rm [N/mm2] 900 850 1100 1000 950 900 1000 900 850 1050 950 900 900 850 980 930 880 900 870 800 1200 Min. yield strength Rp0.2 [N/mm2] 700 650 930 900 750 700 800 700 650 850 750 700 700 650 835 735 675 720 700 625 1000 Distinguishing dimension s [mm] 2) 40 < s 100 100 < s 250 40 < s 100 100 < s 160 40 < s 100 100 < s 160 40 < s 100 100 < s 160 160 < s 250 40 < s 100 100 < s 160 160 < s 250 40 < s 100 100 < s 160 40 < s 100 100 < s 160 160 < s 250 40 < s 100 100 < s 160 160 < s 250 s 250 Type of steel 15CrMoV5-9 5) 17CrMoV10 Ovako 225A ( 16CrMo8-6) 24CrMo13-6 31CrMoV9 32CrMoV13-10 33CrMoV12-9 32CrMoV5 32CrAlMo7-10 40CrMoV13-9 X40CrMoV5-1 544 7 Design of gearing and gear transmissions Table 7.4/23b Approximate Values of Pitting Endurance Limit ıHlim and Tooth Root Endurance Strength ıFE for Quenched and Tempered or Normalised Nitrocarburised 1) and Quenched and Tempered Gas or Plasma Nitrided 2) Tempering and Case-Hardened Steels, Material Quality MQ according to ISO 6336-5 (DIN 3990-5) ıFE [N/mm2] NFlim = 3 106 5) gas or plasma nitride d 640 qF = 84 7) 740 qF = 17 7) 1) Temperature range: 550–590 C 2) Temperature range: 500–550 C 3) The tensile strength/core hardness is larger for a smaller distinguishing dimension. This is positive for the hardness after ni triding. 4) According to DIN EN 18265, Table A1: H [HV10] = 0.31 Rm (valid for R m 1385 N/mm2 with R2 = 1) in the case of core hardness 310 HV, the machining costs can be higher. 5) Values according to ISO 6336-5 (DIN3990-5) 6) qH according to ISO 6336-2 (DIN 3990-2) 7) qF according to ISO 6336-3 (DIN 3990-3) 8) All steels in status +QT In the case of N Nlim an additional decrease of the endurance limits is expected. Assumption: see Table 6.5/13.2, No. 11 or Table 6.5/14, No. 10. Nitrocarbu- rised 640 qF = 84 7) ıHlim [N/mm2] NHlim = 2 106 5) Gas or plasma nitrided 800 qH = 31.4 6) 1000 qH = 11.4 6) Nitrocarbu- rised 800 qH = 31.4 6) Min. hardness after nitriding [HV10] 500 350 650 450 500 450 550 500 500 550 550 500 500 600 550 500 600 550 500 Min. core hard- ness [HV10] 4) (values rounded) 205 235 205 205 220 220 250 220 280 250 235 310 280 250 235 310 280 250 310 280 310 Tensile strength Rm [N/mm2] 650 750 650 650 700 700 800 700 900 800 750 1000 900 800 750 1000 900 800 1000 900 1000 Min. yield strength Rp0.2 [N/mm2] 300 350 300 430 450 450 590 460 650 550 500 750 650 550 500 800 700 600 800 700 800 Distinguishing dimension s [mm] 3) 100 16 < s 40 30 < s 100 16 < s 40 16 < s 40 40 < s 100 16 < s 40 40 < s 100 100 < s 160 16 < s 40 40 < s 100 100 < s 160 160 < s 250 40 < s 100 100 < s 160 160 < s 250 100 < s 160 160 < s 250 160 < s 250 Type of steel 16MnCr5+N 20MnCr5+N 20MoCr4+N C45E, C45R 8) 30MnVS6 38Cr2 34Cr4 34CrMo4 42CrMo4 34CrNiMo6 30CrNiMo8 36NiCrMo16 7.4 Selection of Materials: Strength Values 545 The nitriding hardness depths in Table 7.4/24 are also to be checked, just as is recommended for determining case-hardening depths, to ensure that the local strain is always lower than the local strength. This check is easy to achieve as an approximation for the outermost surface (depth z = 0, ı Hlim for nitriding) and for the transition to the core (with ıHlim for the core hardness after quenching and tempering); see also Appendix 7. ISO 6336-5 specifies two limiting hardness values for definitions of nitriding hardness depth. The effective nitriding hardness depth is defined with the limiting value of 400 HV ĺ NHD400 . At a core hardness of 380 HV the determination of NHD (H K + 50 HV) in accordance with DIN 50190-3 is appropriate for use. For preselection, Table 7.4/25 provides information on struc-tural variations of the nitriding layers. To prevent damage originating from areas beneath the surface due to Hertzian flank contact, ISO 6336-5 recommends the approximate value c H w1 wt 2 521bı sinĮ 1.14 10 cos ȕ=⋅+ ⋅Ud zCHDzz (7.4/16b) where UC = 1.0 at ıH = 900 N/mm2 and H = 300 HB; refer to ISO 6336-5 for further information. Refer to Section 10.3.2.6 for nitriding/nitrocarburising 7.4.4.10 Sintered Gears In order to raise the load-bearing capacity of sintered gears, the processes of induction hardening, carburisation and hardening/carbonitriding in gas at normal pressure and in a vacuum or plasma, as well as nitriding/nitrocarburising in the same atmospheres, are used. For this the gears should be densely sintered. Tooth root strength values were ascertained on hot-pressed and combined hot-cold-pressed sintered and case-hardened gears ( m n = 1.6 or respectively 2 mm, b = 10.6 or respectively 12 mm) made of alloyed sintering steels. For case hardening a density of > 85% of the compact steel is required. Values around and over 90% were confirmed through density measurements. Carburisation was done in gas. Compared to pressing and sintering, through case hardening it is possible to increase the tooth root strength by between 10 and 40 %, depending on the sintering steel and sintering technology. The values that exist in case-hardened gears are not realised [7.4/109]. Through a special compression method on gears which have already undergone a conventional compression process, it is possible to achieve considerable increases in strength. In a rolling mechanism the sintered gear is rolled between two or three mating gears. Here only the tooth flank undergoes compression, which tapers off toward the tooth tip. Teeth which are compressed in this way ( m n = 3, b = 20 mm) demonstrate a pitting endurance limit after case hardening that is comparable to the case-hardened gear made of the steel 16MnCr5. The tooth root strength of the sintered gear only amounted to approximately 80% of the steel gear [7.4/110]. Because of the enormous effort required in terms of machinery, the manufacture of such gears is only economical in large quantities. Table 7.4/24 Recommended Nitriding Hardness Depths for Gears in Accordance with ISO 6336-5 (Main Criteria: Tooth Root Endurance Strength, Scuffing, Wear) Normal module mn in mm Nitriding hardness depth NHD in mm 1) 1 2 3 4 5 6 8 10 0.1…0.2 0.17…0.3 0.22…0.35 0.27…0.4 0.3…0.45 0.35…0.5 0.45…0.6 0.50…0.65 1) Nitriding hardness depths > 0.7 mm can only be achieved with very long nitridingtimes and in some cases higher nitridingtemperatures. 546 7 Design of gearing and gear transmissions Table 7.4/25 Recommendations for Nitriding Layers on Teeth Appropriate for the Strain Material examples 15CrMoV5-9 31CrMoV9, 33CrMoV12-9, 40CrMoV13-9 Ovako 227 C45E, 16MnCr5, 20MnCr5, 42CrMo4, 31CrMoV9 C45E, 16MnCr5, 20MnCr5, 20MoCr4, 30MnVS6, 38Cr2, 34Cr4, 42CrMo4 1) CL – compound layer, İ- nitride Fe 2-3N, Ȗƍƍ- nitride Fe 4N 2) CHD 1.5 mm for gas and plasma nitriding, CHD 0.5 mm in case of salt bath, gas or plasma nitrocarburising 3) H O – surface hardness; values depend on material, initial structure, and nitriding parameter 4) HK – core hardness; values depend on material and characterising dimension 5) The nitriding hardness depth is subordinate in the case of low pressure. 6) Values depend on material, initial structure, and nitriding parameter. Suited nitriding methods gas nitriding plasma nitriding salt bath, gas and plasma nitrocarburising, gas nitriding, plasma nitriding, possibly with post-oxidation salt bath, gas and plasma nitrocarburising, gas nitriding, plasma nitriding HK4) 280 205 to 310 205 to 310 HO3) 700 to 800 350 to 750 350 to 650 Nitriding hard- ness depth NHD [mm]2) 0.1 m n 0.08 to 0.1 mn 0.1 m n 5) 0.08 m n Compound layer CL Thickness [ m] 10 20 20 10 Type1) Ȗƍ or without CL İ(+Ȗƍ) İ(+Ȗƍ) or Ȗƍ Tooth property high contact stress endurance limit ı Hlim 1200 N/mm high tooth root basic strength ı FE 850 N/mm high resistance to abrasive and adhesive wear, high resistance to corrosion medium contact stress endurance limit 6) ıHlim = 800 to 1000 N/mm medium tooth root basic strength 6) ıFE = 640 to 740 N/mm 7.4 Selection of Materials: Strength Values 547 With DIN EN ISO 4507 there is a standard for determining and testing case-hardening depth for ferrous materials for sintering. The case-hardening depth is defined at 550 HV 0.1. The analysis takes porosity into account. Testing requires more effort than for compact steels. 7.4.4.11 Gears Made of Thermoplastic Materials Information on the materials can be found in the VDI Guideline 2736 “Thermoplastic gear wheels”. Typical stress limits include wear, scuffing, as well as the time and temperature dependency of the strength values. 7.4.5 Comparison of Endurance Limit Values In Tables 7.4/14a to c, 7.4/15, 7.4/20 and 7.4/23a, b, approximate values of pitting endurance limit σHlim and tooth root strength σFE for material quality MQ are listed for specific steels, in accordance with the principles described for material selection. They apply for certain areas of the characteristic dimension, in conjunction with minimum values of hardness (influence of hardenability, nitridability), and are an addition to ISO 6336-5 (DIN 3990-5). The material and treatment-dependent value ranges displayed in Figure 7.4/22 are taken from these tables for the material quality MQ. Figure 7.4/22 Comparison of the value ranges of pitting endurance limit σHlim (a) and tooth root strength σFE (b), material quality MQ: 1 - Alloyed tempering steels, quenched and tempered; 2 - Tempering and case-hardening steels, normalised or quenched and tempered, nitrocarburised; 3 - Tempering and case-hardening steels, quenched and tempered, gas-nitrided; 4 - Nitriding steels (without Al), quenched and tempered, gas-nitrided; 5 - Tempering steels, quenched and tempered, flame or induction-hardened; 6 - Alloyed case-hardening steels, case-hardened 548 7 Design of gearing and gear transmissions Compared to the minimum values of the quenched and tempered gears, for case-hardened gears the pitting endurance limit is to be increased to 238% (630 to 1500 N/mm2) and the tooth root strength to 204% (540 to 1100 N/mm2). The ranking of the heat-treatment methods, listed with rising rate of increase, is as follows: quenching and tempering, nitrocarburising, gas-nitriding of tempering and case-hardening steels, surface layer hardening, gas-nitriding of Al-free nitriding steels, and case hardening. A conversion of the characteristic endurance limit values which are valid for 1% failure probability to 10% or 50%, for example, is possible [7.4/111], [7.4/112]. It should be noted, however, that these failure probability or survival probability values that are frequently denoted are merely approximate values, which do not apply for each series individually or each individual manufacturer. Moreover they are not purely material quantities, but rather are associated with the calculation method. Table 7.4/26 Increasing the Load Capacity in a Systematic Way through Material and Treatment Variations Gear load capacity (general) Load capacity of the tooth flanks Load capacity of the tooth root Use of open-die forgings or application of cold or hot rolling of gearing Increase in steel purity through vacuum melting or vacuum remelting Use of fine-grain steels with high core strength for case hardening, as well as tempering steels with high core strength for nitriding Optimisation of the composition of alloys with regard to the hardenability, the case-hardenability, and the quantity of nitride formers with regard to the nitridability Optimisation of the residual stress profile, for example through cold-hardening, the correct selection of alloy, and optimal cooling for case hardening, and the correct selection of alloy and the ideal parameters for nitriding Reduction in roughness of the tooth root and/or the tooth flank through grinding, polishing or lapping, through surface finishing by rolling Pitting strength Optimisation of the hardening curv e for the given gear geometry ( mn, ρ), taking the distribution of both residual and load-induced stress into account Systematic generation of residual austenite during case hardening and during austempering (ADI) Avoiding grain boundary cementite during case hardening, suppression of cementite precipitation in the diffusion layer during nitriding Optimisation of the case-hardening and surface- hardening depths as well as the nitriding hardness depth, optimisation of thickness, structure and composition of the compound layer during nitriding Boundary layer fatigue (micropitting) Systematic generation of residual austenite during case hardening (considerable oil influence) Abrasive wear (a) and adhesive wear (b) a) Nitriding with compound layer, boriding, combination of nitriding without compound layer and hard material layer b) Preventing residual austenite during case hardening, nitriding with compound layer, chemical phosphate or galvanic copper layer on case-hardened steel (considerable oil influence) Tribo-oxidation Oxidised compound layer on nitride layer; hard material layers on case-hardening or nitriding steel, spray coat on tempering steel Cold-hardening of the tooth root surface after surface layer hardening without tooth root surface, after case hardening, rolling after nitriding Avoiding or minimising surface decarburisation, surface oxidation Countermeasure: Cold- hardening (shot peening) Optimisation of the carbon content of the surface layer during case hardening Optimisation of the hardening curve for the given gear geometry ( m n) taking the distribution of both residual and load- induced stress into account 7.4 Selection of Materials: Strength Values 549 The load capacity values calculated for “standard” conditions change in terms of size and distribution if other parameters describing the material condition are generated with other steels and with specific surface hardening, case hardening and in particular nitriding technologies. The load capacity can be increased beyond the “standard” characteristic values (Table 7.4/26) through the use of other material and treatment variations. The influence of the lubricant is not taken into account. The possibilities listed are interconnected with the requirements on the properties of the surface layer, which are presented in relation to strain in Table 7.4/1. For alloyed steels with heat treatment appropriate for the strain, Table 7.4/27 includes approximate values of pitting endurance limit σ Hlim and tooth root strength σFE only for the purposes of rough calculation . They are between the values of the material quality ML and MQ in accordance with ISO 6336-5 (DIN 3990-5). For recalculation, the approximate values in Tables 7.4/14a to c, 7.4/15, 7.4/20 and 7.4/23a, b can be used, or (following further, systematic research) strength values determined mathe-matically in advance, ascertained based on material, treatment, size and form of the component. 7.4.6 S-N Curves From the general S-N curve equation σ =σ qN N\ ' \ ' (7.4/19) with Ny Load cycle number ND Load cycle number for the endurance limit σy Limited life strength [N/mm2] σD Endurance limit [N/mm2] q S-N curve exponent taking the life factors [tooth flank ZNT in accordance with ISO 6336-2 (DIN 3990-2); tooth root YNT in accordance with ISO 6336-3 (DIN 3990-3)] into account, it is possible to construct the S-N curves for the six main material and treatment groups. Figure 7.4/23 shows the result for the tooth flank (a) and the tooth root (b), for material quality MQ and 1% probability of failure. The case-hardened gears show the highest limited life strength and endurance limit. Deviations from the “standard” exponents q H and q F result for example when there are variations in the heat-treatment technologies, with consequences for the structure and residual stresses, or variations in the tooth manufacturing technologies, with changes to structure and micro and macro-geometry (e.g., tooth root notch) [7.4/113, Section 2]. 550 7 Design of gearing and gear transmissions Table 7.4/27 Approximate Values of Pitting Endurance Limit ıHlim and Tooth Root Endurance Strength ıFE for Selected Gear Materials 1) for Approximate Calculations Properties, UHPDUNV ) wear sensitive, high overload resistance 4) high wear resistance, very sensitive to overload 4) 5) Problem: tooth root har-dening, contour conform hard- ness profile 4) wear resistant, sensitive to overloading 4) 5) Problem: distortion 4) 1) For approximate values see Tables 7.4/14a, 7.4/15, 7.4/20, 7.4/23a, and 7.4/23b. 2) For hardness depths see Tables 7.4/17, 7.4/21, and 7.4/24. 3) If s < 40 mm or s > 120 mm, then the hardness and strength values of Tables 7.4/14a, 7.4/15, 7.4/20, 7.4/23a, and 7.4/23b have to be used. 4) Criteria are described in Section 10.3.3. 5) Properties see Table 7.4/25 6) Without hardening of the tooth root, the strength ıFE can lie below the values for quenched and tempered steels. 7) For carbonitrided gears with s 70 mm the values ı Hlim and ıFE of case-hardened steels can be achieved. In the case of N Nlim an additional decrease of the endurance limits is expected. Assumption: see Table 6.5/13.2, No. 11 or Table 6.5/14, No. 10. qF 6.2 84 8.7 17 8.7 NFlim 3 106 3 106 3 106 3 106 3 106 ıFE N/mm2 560 585 605 625 640 740 6) 7406) 7406) 7206) 740 900 qH 13.2 31.4 13.2 11.4 13.2 NHlim 5 107 2 106 5 107 2 106 5 107 ıHlim N/mm2 655 700 740 785 800 1200 1190 1190 1160 1000 1450 Min. yield strength Rp0.2 N/mm2 460 550 650 800 350 460 550 650 400 560 650 800 550 800 800 700 670 760 800 Min. hardness at tooth HV10 or HRC HO (surface) 220 HV 250 HV 280 HV 310 HV 500 HV 450 HV 550 HV 550 HV 54 HRC 53 HRC 53 HRC 50 HRC 750 HV 600 HV 750 HV 750 HV 58 HRC HK (core) 235 HV 220 HV 250 HV 280 HV 205 HV 250 HV 280 HV 310 HV 280 HV 250 HV 310 HV 280 HV 34 HRC Status of heat treatment quenched and tempered quenched and tempered, nitrocarburised quenched and tempered, surface hardened quenched and tempered, nitrided case hardened, carbonitrided 7) Type of steel 34Cr4 34CrMo4 42CrMo4 34CrNiMo6 20MnCr5 34Cr4 34CrMo4 42CrMo4 46Cr2 41Cr4 42CrMo4 34CrNiMo6 34CrMo4 34CrNiMo6 31CrMoV9 15CrMoV5-9 16MnCr5+HH 20MnCr5+HH 18CrNiMo7-6+HH Distinguishing dimension s mm 40 to 1003) 30 45 330 7.4 Selection of Materials: Strength Values 551 Figure 7.4/23 S-N curves for the material quality MQ with the failure probability of 1%; (a) pitting strength; (b) tooth root strength: 1 - Alloyed tempering steels, quenched and tempered; 2 - Tempering and case-hardening steels, normalised or quenched and tempered, nitrocarburised; 3 - Tempering and case-hardening steels, quenched and tempered, gas-nitrided; 4 - Nitriding steels (without Al), quenched and tempered, gas-nitrided; 5 - Tempering steels, quenched and tempered, flame or induction-hardened; 6 - Alloyed case-hardening steels, case-hardened 552 7 Design of gearing and gear transmissions 7.4.7 Symbols and Symbol Explanations of Section 7.4 A % elongation at fracture Ac3 C temperature at which the conversion of ferrite to austenite stops in heating (DIN EN 10052) At0.35 mm carburisation depth for a limited carbon content of 0.35% a mm centre distance B % coefficient of determination bH mm half width of Hertzian deflection zone b mm face width CHD mm case-hardening depth for hardness limit of 550 HV1 D mm diameter DS mm precipitation layer or diffusion layer after nitriding, hardness penetration depth after case hardening, according to DIN EN 10328 d mm reference diameter, diameter of relevant heat-treatment section (DIN EN 10083-1), part diameter (DIN EN 10083-2), specimen diameter (DIN EN 1562) E mm face distance in Jominy specimen H, HH, HL - symbols of hardenability in face quenching curve H max - upper limit of hardenability confidence lines +H Hmin - lower limit of hardenability confidence lines +H HH max - upper limit of restricted hardenability confidence lines +HH HH min - lower limit of restricted hardenability confidence lines +HH HL max - upper limit of restricted hardenability confidence lines +HL HL min - lower limit of restricted hardenability confidence lines +HL H m-1 quenching intensity (heat flow equivalent) H HBW, hardness HV, HRC H F HV hardness of ferritic structure HG HV1, hardness limit HV0.5 H K HRC, core hardness HV10, HV1 H max HRC maximum possible hardness after quenching Hmeasured HRC hardness measured after quenching HO HV, hardness, surface hardness HV1, HV0.5 H R HV1, case hardness HV0.5 Hx HV hardness at distance x from surface h(CL) m compound layer thickness ΔH HV0.5 hardness increase = HR – HK J HRC hardness in face quenching test KV J notch impact energy l mm length, wave length ML, MQ, ME symbols (ISO 6336-5) of indicating requirements of material and heat treatment M s C martensite start point mn mm normal module N, N y - number of load cycles ND - number of load cycles for endurance limit Nht mm nitriding hardness depth according to DIN 50190-3 NHD mm nitriding hardness depth according to DIN ISO 15787 NFlim - number of load cycles at break of slope of S-N curve for tooth root stress NHlim - number of load cycles at break of slope of S-N curve for contact stress q - exponent of S-N curve qF - exponent of S-N curve for tooth root bending qH - exponent of S-N curve for pitting R - degree of cure Re MPa upper yield strength R2 - coefficient of determination Rht mm surface hardening depth Rm MPa tensile strength Rp0.2 MPa 0.2% offset yield strength s mm characteristic dimension, nominal thickness (DIN EN 10025-2) SHD mm surface hardening depth S R - residual spreading TA C austenitising temperature TAnl C tempering temperature t mm part thickness (DIN EN 10083-2), thickness (DIN EN 10293), decisive wall thickness (DIN EN 1563) t 8/5 s duration of cooling from 800 to 500 C tR mm thickness of decisive section (DIN EN 10250-2) 7.4 Selection of Materials: Strength Values 553 VS - compound layer u - number of teeth ratio YNT - life factor for tooth root stress for standard conditions YRS - factor for endurance increase due to shot peening YST - stress correction factor of standard reference test gear YX - size factor (tooth root stress) Yδ - notch sensitivity factor Z % percentage reduction of area after fracture ZNT - life factor for contact stress for standard conditions ZʍvGmax mm depth of the maximum of equivalent stress due to Hertzian pressure ZX - size factor (contact stress) ɸ - iron nitrite Fe 2-3N Ȗƍ - iron nitrite Fe 4N Ȝ s/100 cooling parameter ȜF s/100 cooling parameter for ferrite-free structure ࣁ ,ȡ mm radius of curvature ı MPa strength, stress ıA MPa fatigue strength ıD MPa endurance strength ıE MPa residual stress ıF MPa tooth root stress ıFE MPa tooth root basic strength ıFlim MPa tooth root endurance limit ıFP MPa permissible tooth root stress ıH MPa contact stress ıHlim MPa contact stress endurance limit ıHP MPa permissible contact stress Additional symbols: see ISO 6336, (DIN 3990) and Section 10.3 554 8 Ensuring the Accuracy of Cylindrical Gears 8.1 Introduction 8.1.1 Standardization As an important factor influencing the **backlash**, the function and interchangeability of the gears are ensured by requirements for the accuracy of the single gears in the tolerance system and for their tooth thickness in the system of fits . The terms for the deviations of single gears and gear pairs, as well as the allowable values for the accuracy and **backlash** allowance parameters, are defined in standards. Specifications in accordance with ISO 1328 [8/18], [8/19] and informative amendments in ISO TR 10064 Part 1 to 5 [8/20] to [8/24] apply internationally. In Germany, the key terms were defined in DIN 3960 [8/1], the tolerance system in DIN 3961 [8/2], DIN 3962 [8/3] to [8/5], DIN 3963 [8/6] and DIN 3964 [8/7] and the system of fits in DIN 3967 [8/8]. A further revision of the standards will take place continuously in the coming years and will gradually lead to the standardisation of specifications. The international standard ISO 21771 [8/10] in the German version DIN ISO 21771 [8/11] and the national amendment DIN 21772 [8/12] are a basis for further development. The essential difference in the calculations is that, when using the equations in accordance with DIN ISO 21771, all geometric variables have a positive result, even with internal gearing (despite the negative number of teeth, which remains as an identifying feature), whereas, when using the equations in accordance with DIN 3960 for internal gearing, the negative number of teeth produces negative geometric parameters (diameter, pitch angle, tooth thickness angle, tooth space angle and centre distances). This negative sign does not mean that it is not a matter of real variables. In order to ensure the consistency of the calculations throughout the book, the new definitions and equations according to ISO 21171 have also been dispensed with in Section 8. Therefore, in the case of internal gearing, absolute values must be used as a basis for the testing (calculation of actual properties and comparison with the target properties). The revision of ISO 1328 in light of the AGMA specifications, e.g., [8/33], [8/34], [8/35] and [8/36], represents a possible basis for the unification of the allowable deviations and tolerances. From the standpoint of measurement implementation, the above-mentioned standards are supplemented by VDI/VDE guidelines [8/26] to [8/32]. 8.1.2 Tolerance System The accuracy grades of ISO and DIN gear tooth tolerance system correlate to the manufacturing costs for the individual types of deviation. They contain 12 grades of accuracy, where accuracy grade 1 is assigned to the smallest and accuracy grade 12 to the largest allowable deviations [8/2]. 8.1 Introduction 555 The specification of test parameters and the grade of accuracy is usually carried out empirically on the basis of many years of experience. In order to facilitate the tolerance specifications, functional groups for specific operating characteristics and requirements are defined in DIN 3961: • uniformity of the transmission of motion ( G), • quiet running and dynamic load capacity ( L), • static load capacity (load distribution) ( T) and • no indication of the function ( N). For each functional group (G, L, T, N) three test groups (A, B, C) are defined (Table 8/1). These test groups contain up to 3 measurement quantities, which are suitable to evaluate in order to qualify gears regarding their accuracy grades. For example, testing of the pitch deviations, the transverse profile deviations and the flank line deviations is recommended for the functional group L (quiet running and dynamic load capacity) in the test group A for the accuracy grade 6. It is also possible to assign two or more functional groups of a different accuracy grade to one gear. This is referred to as tolerance families . Example: G 8, L 7 means: specification factors for uniformity of the transmission of motion ( G) in accuracy 8, specification factors for the quiet running ( L) in accuracy 7. In many cases no functional group is assigned in the tolerance indication. Then, the specifications of the manufacturer, the requirements of the client or jointly reached agreements apply to the testing. The specification of tolerances or grades of accuracy is increasingly carried out for heavily loaded gears according to dynamic calculations, calculations for load capacity and running uniformity on the basis of numerical analysis methods. The types of deviation to be tested (tolerances) should be defined in accordance with the information contained in each measurement quantity, the manufacturing conditions, the operating conditions, and possibly taking into account statistical analyses. 8.1.3 Gear: System of Fits The system of fits includes the relationships between **backlash** and toleranced fit sizes. To ensure the desired **backlash** between two meshing gears, reductions in the tooth thicknesses of both gears compared to the nominal tooth thicknesses are required. In DIN 3967 [8/8], tables are shown with recommended values for upper limit devi-ations E sns and tolerances Tsn of the normal tooth thickness. This results in the lower limit deviations E sni (Figure 8/1). It should be noted that, in practice, further influence quantities (e.g., centre distance, axis position deviations, tooth system deviations, as well as heating and the elastic effects of the load) influence the **backlash** too. Figure 8/1 Tooth thickness limit deviations and **backlash** 556 8 Ensuring the Accuracy of Cylindrical Gearing Scope of the test procedures indicated using abbreviated symbols Required quantities within tolerances: Fȕ, fHȕ, ffȕ Fα, fHα, ffα Fp, Fp z/8 fp, fu fpe Test procedure Flank line test Transverse profile test Cumulative pitch test Individual pitch test Base pitch test Symbol Fȕ Fα Fp fp fpe Required quantities within tolerances: Fi′ and f i′ Fi′′, fi′′, aL (if not T s) fi′′ Rs TRA Test procedure Single-flank roll test Double-flank roll test Double-flank roll test Tooth thickness variation Contact pattern test Symbol Fi′ Fi′′ fi′′ Rs TRA Table 8/1 Test Groups for the Functional Groups and Grades of Accuracy (Selection according to DIN 3961) N No indication of function C fpe Rs TRA fpe Rs TRA fpe Rs TRA fpe Rs fpe Rs fpe fpe Rs Rs B Fi′′ F α Fȕ Fi′′ F α Fȕ Fi′′ F α Fȕ Fi′′ TRA Fi′′ TRA Fi′′ Fi′′ fpe fpe A Fi′ Rs Fi′ Rs Fi′ Rs Fi′′ F α Fȕ Fi′′ F α Fȕ Fi′′ TRA Fi′′ TRA Fi′′ Fi′′ T Static load-bearing capacity C Fi′′ TRA Fi′′ TRA Fi′′ TRA fp TRA fp TRA fp TRA fp fp B fpe TRA fpe TRA fpe TRA Fi′′ TRA Fi′′ TRA Fi′′ TRA fi′′ fi′′ A fpe Fα Fȕ fpe Fα Fȕ fpe Fα Fȕ fpe TRA fpe TRA fpe TRA fpe fpe L Quiet running and dynamic load- bearing capacity C fpe Fα TRA fpe Rs TRA fpe Rs TRA fpe Rs fpe Rs fpe Rs fpe Rs B Fp Fα Fȕ Fp Fα Fȕ Fi′′ F α Fȕ Fi′′ F α Fȕ Fi′′ F α Fȕ Fi′′ F α Fȕ Fi′′ F α Fȕ A Fi′ Fȕ Fi′ Fȕ Fp Fα Fȕ Fp Fα Fȕ Fp Fα Fȕ G Uniformity of the transmission of motion C fpe Fα Fȕ fpe Fα Fȕ fpe Fȕ fpe Fȕ B Fp Fα Fȕ Fp Fα Fȕ Fp Fα Fȕ fpe Fα Fȕ fpe Fα Fȕ A Fi′ Rs Fi′ Rs Fi′ Rs Fp Fα Fȕ Fp Fα Fȕ Functional group Test group 4 5 6 7 8 9 10 11 12 Grade of accuracy 8.2 Metrological Basic Principles 557 8.2 Metrological Fundamentals 8.2.1 Limit Deviations and Tolerances The accuracy grade of a gear will be determined by assigning of measurement results of agreed quality features into the system of defined limit deviations or limit dimensions of the measured quantity. Limit deviations are the upper respectively lower allowable deviations (ISO 286 [8/13], [8/14]). They describe the difference between the maximum and the nominal value or between the mini-mum and the nominal value. In the determination of limit deviations for the various quality features, different approaches are common: • Runout, double-flank and single-flank composite deviations, total pitch deviation as well as total and form deviation of the transverse profile, flank line and the generator are unilateral parameters. Only the upper limit deviation is determined. The lower limit deviation is zero. Therefore, the nominal value corresponds to zero, but its specification is uncommon. • In the case of the circular pitch, base pitch, transverse pressure angle, helix angle and centre distance, the upper and lower limit deviations for the measured quantity are usually symmet- rically arranged relative to the nominal value. In the case of the tooth thickness and the tooth thickness test dimensions, the upper and lower limit deviations are usually negative (except as stock allowance in the pre-machining). Deviations are indicated by the letter f or F , and in the case of tooth thickness test dimensions by the letter E. The reference to the quality feature is marked by indices. The assigment of the upper and lower limit deviation is carried out by the additional index letters s and i (e.g., E sni = lower limit deviation of the normal tooth thickness). The tolerance is the algebraic difference between the maximum and minimum value or between the upper and lower limit deviation. The letter T is the symbol for the tolerance (e.g., T sn = toler- ance of the normal tooth thickness). The limit deviations and allowances are specifications of the designer. They do not represent actual properties. 8.2.2 Actual Values and Actual Deviations The measurement values are referred to as actual values . The actual deviation (in short: devia- tion) is the algebraic difference between the actual value and the nominal value of a measured quantity. The letter f or F is the symbol for deviations. In order to distinguish limit deviations, the index act can be used for actual [8/18], if necessary. The variation is the difference between the largest and the smallest actual value of one measured quantity in a specified area, at all teeth or flanks of the gear. If the number of measurements is not limited by test instructions, this area covers the entire gear circumference. The letter R is the symbol for the variation of measured values (e.g., R sn = variation of the normal tooth thickness). The tooth to tooth deviation is the difference between the values of the same quality feature at two immediately successive measuring points on the gear circumference. It is called adjacent pitch regarding pitch measurement and tooth to tooth single- or tooth to tooth double-flank composite deviation considering single- or double-flank roll testing. It is always indicated without a sign. 558 8 Ensuring the Accuracy of Cylindrical Gearing 8.2.3 Referencing of Measured Quantities and Measurement Methods The reference of measurement results is essential to their information content. Measurement methods and parameters which relate to the gear axis are referred to as axis-related. If the meas-urement of toothing deviations is carried out through probing of the tooth flank and with simul-taneous reference to the tooth flanks themselves (i.e., alignment of the measuring instrument position), then this is referred to as a flank or toothing axis-related measurement method. If the measurement of toothing deviations is carried out with reference to the tip circle or tip cylinder, then this is referred to as tip circle-related measurement methods and quality parameters. The axis, around which the gear has rotated during the tooth flank machin-ing on the gearing machine, is referred to as the toothing axis or manufactur- ing axis . The toothing axis is identical with the axis of the actual base cylin-der of the manufactured gearing. The gear axis (axis of the gear) is defined by the bearing zones of the gear. These bearing surfaces are the geometry reference elements, indicated in the workpiece drawing according to DIN 3966-1 [8/16], on the gear main body. A position deviation of the toothing axis occurs when the toothing axis and gear axis do not coincide (e.g., due to a misalignment when the gear body is clamped onto the processing machine). This position deviation can be described by the distance between the two axes in the middle of the usable face width b F, the eccentricity of the gearing fe, and by their angle of intersection – wobble of the gearing fı (Fig- ure 8/2). If the gear is mounted on a rotary table for a measurement and the rotary axis is used as the datum axis for the measurement, then this rotation axis becomes the measurement axis . In gen- eral, the measurement conditions should be designed in such a way that the measurement axis and gear axis coincide. The axis of the tip cylinder usually results from the pre-machining. Its location relative to the subsequently realised toothing axis and the tip cylinder form deviation are therefore particularly problematic. The axis position deviation effects gear axis-related or tip circle-related quality parameters as periodic fluctuation at the gear circumference. It especially influences the maximum value of the measured quantities. Flank-related measurement methods are not effected by the deviation of axes. Figure 8/2 Displacement of the toothing axis 8.2 Metrological Basic Principles 559 8.2.4 Uncertainty of the Measurement Results The machining process is per- formed on the basis of the de-signed specifications of the main gearing geometry and the tooth modification. Here, the selected method, the specified technology, the deviations of machine, tool, apparatus and operation, as well as thermal, static and dynamic de-formations, lead to deviations at the manufactured gear (Figure 8/3). Thus, for example, deviations between the gear axis and manu-facturing axis lead to sinusoidal variations of gear axis-related parameters on the gear circum-ference. In the case of hobbing, local deviations on the tooth flanks result from the contact traces left by the hob and the generat-ing cut deviations [8/42] to [8/44]. Profile grinding of crowned helical gears causes, undesired profile twist deviations, result from the contact conditions between the grinding wheel and workpiece. Additional random deviations can be caused by the action of internal forces, excited vibrations and wear on the tool, for example. (Also compare with [8/69].) The gearing manufactured with systematic and random devia-tions is measured metrological-ly. It should be noted that the results are influenced by the mea-surement uncertainty, which results from the measuring in-strument, software, clamping tool and environmental condi-tions, as well as from the meas- uring object, the predefined measurement strategy and the execution of measurements (Figure 8/4). This influence is not negligible, particularly at small-sized tolerances. Especially with high accuracy requirements (small accuracy class), a large discrepancy shows up between the measurement capabilities and the requirement to maintain a ratio of 11 to 10 5U T= between tolerance T and measurement uncertainty U. Based on the measured values, it is checked whether the requirements of the design have been met. The gearing will be assessed. At the same time, production-oriented but also function-oriented parameters are taken into account. Furthermore, on the basis of the measurement data obtained, the process or machine can be evaluated and quality-oriented control or correction parameters for the machining can be derived. Figure 8/3: Relationships between design, manufacture and tooth system deviation Figure 8/4 Objectives of the measurement and measurement uncertainty 560 8 Ensuring the Accuracy of Cylindrical Gearing While the operator of the measuring device must be empowered through training for the quali- fied execution, evaluation and analysis of measurements, the measuring instruments must be maintained, monitored, inspected and certified [8/38] in order to ensure the traceability of meas- urement results to the national standard. For the inspection and continuous monitoring of the measuring instruments, special gearing standards are available (e.g., involute and flank line standards, ro lling artefacts), or master gears [8/9], [8/57] can be used. Essential requirements for the evaluation of instruments for the mea-surement of gears are summarised in DIN ISO 18653 [8/25]. The software used also has a signif-icant impact on the results. Their accuracy can be demonstrated by a certificate of the Physikalisch-Technische Bundesanstalt (PTB), Braunschweig [8/38]. Work is successfully fin-ished on the use of the virtual coordinate measuring machine VCMM [8/51], [8/37] for the determination of uncertainties for gearing measuring instruments. Tooth flank waviness stand- ards are in development for the examination of the dynamic properties of flank testing devices [8/39]. 8.3 Quality Parameters 8.3.1 Flank Deviations 8.3.1.1 Preliminary Remarks Flank deviations are deviations from the nominal tooth flank of the gear. They can be based on cross sections (Figure 8/5) • transverse profile or profile (in the transverse plane, perpendicular to the gear axis), • flank line (at the intersection with a cylinder coaxial to the gear axis), • generator (at the intersection with the plane tangential to the base cylinder) or on the entire flank surface. Special flank measuring instruments (Figure 8/6), which measure the nominal flank surface when controlled mechanically or by computerized numerical control (CNC), are suitable for the determination of these deviations [8/54], [8/55], [8/59], [8/63]. Alternatively, universal coordi-nate measuring instruments (Figure 8/7) can be used with special software [8/45], [8/60], [8/61], [8/62], [8/67]. The scope of application of the currently available flank measuring instruments is limited by the module m = 0.5(0.2) to 32 mm, the tip diameter d a up to approximately 3 m and the face width of the gear b up to approximately 1.5 m. A further limit of application is the weight and therefore the manageability of the gearing. With the use of coordinate measuring machines, the scope of application is enlarged. The mea- surement of miniature gearing can be carried out with optical probing (spur gearing) or by using miniature probes (e.g., fibre optic probes [8/50]). Large gearing can be measured using bridge-type measuring machines [8/60] with a fixture positioned at the border area of the measuring range, or in segments. Figure 8/5 Flank cross section 8.3 Quality Parameters 561 Figure 8/6 Four-axis flank testing machine Figure 8/7 Coordinate measuring machine When measuring gears with the highest requirements (e.g., standards), the uncertainty in the measurement direction can be reduced by using a laser tracer [8/40]. When using attached units or integrat-ed measurement systems on gearing machines, it is important to note that many of the manufactured deviations of the gearing (e.g., deviations be- tween the gear axis and the manufac- turing axis or runout deviations of the fixture) cannot be measured due to the identical axis (manufacturing axis = reference (measurement) axis). Flank deviations are gear axis-related quality parameters. Thus, the determination of the axis is of essential importance. The reference for the measurement can be given if the gear is mounted on an arbor or on the pinion shaft and clamped between the centers between the tips (Figure 8/8a). The alignment of the centers and its location to the subsequent bearing surfaces are crucial. In particu-lar, a position deviation of the counterholder center tip to the rotary axis results in increased devia-tions. If the part is mounted in a chuck or by other clamping devices, mechanical alignment is possible in principle but, because of the accuracy requirements, barely adequate. Figures 8/8b and c show two possibilities to determine the gear axis by measurement. For this purpose, the surfaces of the workpiece, which also form the support surfaces (e.g., bearing seats) during subsequent use, are to be preferred. The two circles required for the determination of the axis should be as far apart as possible. With flat (disc-shaped) workpieces (Figure 8/8d), the measurement of a combination of a plane surface – (the normal vector describes the direction of the rotary axis), and a circle (for the centre determination) – is more reliable. In order to recognize deviations the effect of variations between the gear axis and the toothing axis (eccentricity, wobble) in the measurement of flank deviations, a measurement of at least three, preferably four, teeth or tooth spaces evenly distributed around the gear circumference is required. Figure 8/8 Reference gear axis 562 8 Ensuring the Accuracy of Cylindrical Gearing The measured deviations in the transverse profile, on the flank line and on the generators can be represented in the form of flank test diagrams. In the test pattern of most flank test-ing machines, the unmodified nominal trans-verse profile, the unmodified nominal flank line or unmodified nominal generator appears as a straight line. The deviations of the flank surface must be spatially measured in a direct geometric rela-tionship (common defined basis). Its representa-tion is referred to as tooth flank topography. It is composed of several transverse profiles and/or several flank lines on one flank, or of measurement points recorded in the form of a grid. The data collected provide the possibility of an external further processing in the sense of real spatial flank or modification evaluations. In order to estimate the effect of flank deviations occurring during subsequent use of the gear, their detection or evaluation must always be performed in the transverse section (perpendicular to the gear axis) and within the plane tangential to the base cylinder. The influence of probe radii, filters and measuring point intervals must be taken into account in the measurement of flank deviations [8/48]. Figure 8/10 on the left shows a profile test diagram, which was measured with a small probe ball radius and without subsequent filtering of the measured values. The figure on the right shows the same measuring result after the application of a low pass filter. The modified information con-tent is clearly visible. A regression analysis is required in order to determine dimensional deviations and defined quali-ty parameters from the measured deviations. For this purpose only measurement points will be considered in each defined regression zone separately. The regression analysis must be carried out according to GAUSS, with (8/1) The result of the regression analysis is referred to as a regression profile or a regression flank surface. For the quality assessment (classification of gearing into accuracy classes), the maximum values of the calculated total, angular, and form deviations distributed over the gear circumference are used. Its variations allow additional statements on quality. The periodic changes in the angular deviations of the transverse profile and the flank line over the gear circumference result from axis position deviations (eccentricity and wobble deviation). A regression analysis based on a linear combination of sine and cosine elements is ideal for the determination of the amplitude and phase of the local eccentricity [8/53]. From these results, conclusions can be drawn about the size of the =→n if 12i MINFigure 8/9 Flank measurement Figure 8/10: Transverse profile deviation under differen t conditions 8.3 Quality Parameters 563 associated axis position deviations and the specific tooth, at which the largest deviations on the circumference can be expected, and their maximum size can be approximately predicted. The mean values of all determined slope deviations of the profile and flank line distributed around the gear circumference are used to derive the parameters that control the manufacturing process. Further information can be obtained, for example, from waviness parameters. 8.3.1.2 Deviations of the Transverse Profile Transverse profile deviations are the detected deviations from the nominal transverse profile in the transverse section within the active (in us e) flank. The working length of the active trans- verse profile L AE is the length of path of contact by pairing of the gear (index 1) with its mating gear (index 2): (8/2) To ensure a wider safety margin in the direction of the tooth root of the utilised flank, or if the mating gear is unknown, the working length L AF, which results from the pairing of the cylindri- cal gear (index 1) with a rack, is used instead: (8/3) Usually, the mean transverse section is used for the transverse profile test. With very wide gears, several transverse sections can be agreed upon for testing. The regression zone results from the working length assigned to the active transverse profile (definition range) reduced by 8% based on the usable tip cylinder. The limitations of the measurement ranges, regression zones and evaluation ranges can be defined by specifying the diameter, the working lengths and the corre-sponding roll angle. Figure 8/11 Parameters for the determination of the transverse profile deviations ( )22 22 2 AE Na1 b1 Na2 b2 wt 212s i nĮ2zLd d d d az =− + − − ( ) tn 1 aP t b12 b12 Na1 AFĮsinĮtan21 m x hd d d L−+ − − = 564 8 Ensuring the Accuracy of Cylindrical Gearing A distinction is made between • profile slope deviation fHĮ, • profile form deviation ffĮ and • total profile deviation FĮ. The profile slope deviation fHĮ is the distance between the two nominal profiles, which, at the starting and ending points of the active profile LAE, intersect the previously extended regression profile (Figure 8/12). In some specifications LĮ is used as a reference length for fHĮ instead of LAE. The profile slope deviation is considered positive if the regression trans- verse profile in the direction of the tooth tip ascends towards the material-free side compared to the nominal profile. Deviations of the tool profile angle or base circle have an impact as a linear increase in the test diagram of the transverse profile and thus as a mean profile slope deviation. An eccentricity between the gear axis and manu-facturing axis superimpose these on the gear circumference through the periodic variation of the slope deviation (Figure 8/13). The eccentricity can be calculated approximately from its amplitude a fHĮ [8/53]: (8/4) The base circle deviation fb is the difference between the actual base diameter and the nominal base diameter. The pressure angle deviation fĮt is the difference between the actual transverse pressure angle and the nominal transverse pressure angle. If necessary, both can be calculated from the mean profile slope deviation f HĮm: (8/5) (8/6) The profile form deviation ffĮ is the distance between two regression profiles (Figure 8/14 left) enveloping the actual transverse profile within the active profile LAE. Deviations outside LĮ but within LAE should only be taken into account if they occure towards the material-free side (Fig- ure 8/14 right). Periodic characteristics of the form deviations, the wavelength of which within the test diagram corresponds to a normal base pitch and which, in the case of measurements distributed around the gear circumference, do not change their position in the test diagram, generally result from the runout of the tool. Figure 8/26 shows this effect in the test diagram of the hobbing trace valley measurement. AEfHĮb e 2Lad f≈ Hm AEbbffdLα= [] AE tHĮm Įt radtanĮffL=− Figure 8/12 Profile slope deviation without eccentrcity with eccentricity Figure 8/13 Transverse profile test diagrams 8.3 Quality Parameters 565 The total profile deviation F Į is the distance between two nominal transverse profiles (Fig- ure 8/15 left) enveloping the actual transverse profile within the active profile L AE. Deviations outside LĮ but within LAE should only be taken into account if they occure towards the material- free side (Figure 8/15 right). Figure 8/14 Profile form deviation Figure 8/15 Total profile deviation 8.3.1.3 Deviations of the Flank Line Flank line deviations are the deviations from the nominal flank line detected within the usable face width bF. The usable face width bF is the axial distance between two transverse sections, enveloping the zone of full profile; refer to Figure 8/16. For the estimation of the por-tion effective in operation, consideration of the effective face width b w and the position of the face width that comes into contact with the mating gear is required. For the flank line measurement a flank line on the V-cylinder is normally used. The regression zone results from the usable face width, reduced by the smaller amount of 5% of b F and once the module from each side. A distinction is made between • flank line slope deviation fHȕ, • flank line form deviation ffȕ and • total flank line deviation Fȕ. The flank line slope deviation f Hȕ is the distance between two nominal flank lines, which inter- sect the extended regression flank line at both face sides of the usable face width bF (Fig- ure 8/18). Deviating from this, in some specifications Lβ is used as a reference length for fHȕ. The flank line slope deviation is considered positive if the deviation increases the helix angle. In the case of spur gearing, it is considered positive if the deviation occurs in the sense of a right-hand helix. The opposite signs apply for internal gearing. Figure 8/16 Parameters for the determination of flank line deviations 566 8 Ensuring the Accuracy of Cylindrical Gearing The helix angle deviation f ȕ is the difference between the actual helix angle and the nominal helix angle. If necessary, it can be approximately calculated from the mean flank line slope de- viation fHȕm: (8/7) Inclinations between the machine frame and the toothing axis lead to linear deviations in the test diagram of the flank line and, thus, to constant flank line slope deviations on all teeth of the gear. Tangential inclinations cause indentical directions at right and left flanks. Radial inclinations cause opposed directions of deviations for both flanks. A wobble bet-ween the gear axis and the manufactur-ing axis leads to periodic variations in the slope deviations on the gear circum-ference (Figure 8/17 right). Figure 8/17 Flank line test diagrams The flank line form deviation f fȕ is the distance between two regression flank lines enveloping the actual flank line within the usable face width bF (Figure 8/19). Within bF but outside Lȕ, only deviations in the direction of the material-free side should be taken into account (see transverse profile). The total flank line deviation F ȕ is the distance between two nominal flank lines enveloping the actual flank line within the usable face width bF (Figure 8/20). Within bF but outside Lȕ, only deviations in the direction of the material-free side should be taken into account (see transverse profile). Figure 8/18 Flank line slope deviation Figure 8/19 Flank line form deviation Figure 8/20 Total flank line deviation 8.3.1.4 Deviations of the Generator Deviations of the generator are the deviations from the nominal tooth flank detected in a meshing position in a plane tangential to both base cylinders.They are expressed as distances perpendicular to the axis. The usable face width bF or the active root and tip cylinder dNa and dNf (Figure 8/21) limit the portion of the generator to be tested. Usually, the longest generator located in the centre of the flank surface is used for the measurement. If the usable face width limits the generator (position A), the measurement should be carried out in accordance with the flank line evaluation. If the active []2 FtH m ȕcos radcosĮffb≈ without wobble deviation with wobble deviation 8.3 Quality Parameters 567 tip and active root cylinder limit the generator (position B), the measurement should be carried out in accordance with the profile evaluation. For the transition position of the generator that can be identified as both A and B, the measurement should be carried out in accordance with the flank line evaluation. There may be also positions where the generator is limited by an end face and the active tip or root cylinder. Here it is up to the user to define clear rules depending on the objective. A distinction is made between • generator slope deviation f HE, • generator form deviation ffE and • total generator deviation FE. Generator deviations are considered for functional evalua- tions, because the contact with the mating gear occurs on the generator. The benefit of these conclusions for the ma-nufacturing control is rather limited, because the effect contains mixed contributions of both directions, the profile line and the flank line. 8.3.1.5 Hobbing Trace Valley Measurement Particularly during hobbing as pre-machining, the axial feed of the hob affects the tooth flank as a clearly visible mark (wave) (Figure 8/22). This wave is intersected during a measurement in the transverse plane and in the flank line. The effect is highly dependent on the geo-metrical data of the gearing (especially on the helix angle). Figure 8/23 shows the location of a transverse profile measurement on a flank with respect to the feed mark. Figure 8/24 shows the measured flank lines, and Figure 8/25 shows the transverse profiles. The meas-urement was carried out on four teeth evenly distributed around the circumference. In the evaluation, it should be noted that the form and total deviations from the profile and flank line largely affected by the chosen technology parameters (axial feed f a) and not caused by tools or machines. Because the transverse section in the example shown intersects the feed mark so that ap-proximately one wave can be seen in the test diagram (Figure 8/25), and this wave occurs out of phase on the gear circumference. Therefore the feed mark heavily influences the profile slope deviation and its variation on the gear circumference. Thus, no reliable con- Figure 8/21 Position of the generator Figure 8/22 Infeed meshing mark: hobbing Figure 8/23 Transverse profile and hobbing trace valley 568 8 Ensuring the Accuracy of Cylindrical Gearing clusions, for example regarding tool deviations, are possible. In order to prevent this influence, the measurement can be carried out in the hobbing trace valley (within the contact trace; refer to Figure 8/23). Figure 8/26 shows the resulting test diagram, in which a profile slope deviation and a runout of the hob can be clearly identified. Figure 8/24 Flank line measurement Figure 8/25 Transverse profile measurement Figure 8/26 Hobbing trace valley measurement More particularities must be considered when using a multi-thread hob [8/42], [8/43]. 8.3.1.6 Deviations of the Total Flank Area Both unmodified and modified tooth flanks can be evaluated on the basis of a 3D flank measure- ment. The measurement results describe the measured flanks much more reliably than separate lines. While, in the use of conventional flank testing machines, the deviations with regard to the unmodified nominal involute helical surface are recorded and therewith regression surfaces identified, universal coordinate measuring instruments also provide measuring coordinates [ x, y, z, (ĳ)], which then require a different approach in the regression analysis. This 3D data can be used to determine individual re-gression tooth flanks [8/49] and, if points are present from four (at least three) flanks distributed evenly around the circumference, also to calculate a mean imaginary gear from regressions (including devia-tions between the gear axis and toothing axis). On this basis, evaluative parameters can further describe usual flank deviations, but also the entire tooth flank. Figure 8/27 Imaginary gear from regressions Deviations of the tooth flank are the deviations from the nominal tooth flank measured within the flank area. This area is limited by the usable respectively effective face width and the form re-spectively active diameters. The regression zone is obtained by reducing the evaluation range on the tip and the limiting end faces, as with the profile measurement and flank line measurement. A distinction is made between • tooth flank form deviation f fȈ and • total tooth flank deviation FȈ. 8.3 Quality Parameters 569 The tooth flank form deviation ffȈ is the distance between two regression flank surfaces envelop- ing the actual flank measured within the plane tangential to the base cylinder and in the trans- verse section (perpendicular to the gear axis) (Figure 8/28). The total tooth flank deviation F Ȉ is the distance between two nominal flank surfaces enveloping the actual flank measured within the plane tangential to the base cylinder and in the transverse section (perpendicular to the gear axis) (Figure 8/29). In both cases, only de-viations in the direction of the material-free side should be considered (see profile and flank line) out-side the regression zone. Figure 8/28 Tooth flank fo rm devia tion Figure 8/29 Total tooth flank deviation 8.3.1.7 Measurement of Modified Flanks The fundamental modifications are the slope modi- fication, the crowning and the tip and/or root relief in the transverse profile, and the slope modi- fication, the crowning and end reliefs on the flank line. In the case of modified tooth flanks, the slope, form and total deviations to be determined are supplemented by measurements and description characteristics of the modifications. In the rest of this section, only profile modifications will be explained. For modifications of the flank line, the explanations apply correspondingly. The required regression approach is always based on the specification of the designer. In general, in the case of a slope modification, a straight line should be used for regression; in the case of crowning, unless otherwise specified, a parabola should be calculated. In the case of reliefs, the recorded data associated with the modification zone or the unmodified part have to be analysed separately (Figure 8/30, zones L αa, Lαf, Lαm). Measurement points in the transition zones are not included in the regression analysis. Alternatively, a complete regression of all measuring points in the L α zone can be carried out, even if the transition zones are included. In the case of linear tip and/or root reliefs (which should be avoided if possible due to disadvantageous wear charac- teristics), straight lines are used in the zones as regression approaches. Otherwise, parabolic (e.g., quadratic) approaches may be required in the relief areas according to the specification of the designer. Furthermore, it may be necessary to reduce the evaluation range, not by 8%, but by a lower percentage in order to leave a sufficient number of measurement points in the relief area. This must be separately determined and specified [8/46], [8/47]. Figure 8/30 Tip and root relief regression zones with indications of tolerance 570 8 Ensuring the Accuracy of Cylindrical Gearing The actual values of the modifica- tions are calculated from the actual parameters that describe the regres- sion zones. In this case, the defini-tions of the parameters and their reference range must be considered. If the modification has been designed for the active ( L AE, bw) or usable ( LAF, bF) flank, it must be extrapolated accordingly despite a reduced regres-sion zone. In the case of reliefs, two different approaches are possible. In the first variant (Figure 8/32 left), the regression zone of the unmodified area is extended to the starting and finishing point of the active profile L AE. The actual value of the modifica- tion refers to the resulting points of intersection. This vers ion should be preferably used when all flank areas have been machined in a single operation (e.g., grinding with a correspondingly dressed tool) and when the main objective of the measurement is the derivation of parameters controlling manufacturing. Variant 2 (Figure 8/32 right) is advantageous mainly for evaluating the modification and in the manufacturing of the modification and remaining flank in separate production steps. It is important to reach agreement between the customer and sup-plier and to clearly identify the variant used in advance. The calculation of slope, form and total deviations is carried out using the same definitions as in the case of unmodified gearing (Figure 8/33 to Figure 8/35). In the case of reliefs, it must be determined whether the zones should be considered separately or together (Figure 8/34). Figure 8/34 Form deviation in the case of tip an d root relief Figure 8/35 Total deviation in the case of profile crowning In addition to the modifications from the profile and flank line, diagonal reliefs (triangulary shaped reliefs), flank twist and topographic modifications can be measured and evaluated on the flank surface (Figure 8/36). The approach of the regression analysis is defined by the description of the theoretical flank surface. The result of the regression analysis is referred to as the fitting surface. In addition to the actual parameters of the modifications, evaluation parameters, such as the tooth flank form deviation and the total tooth flank deviation, are once again available. Figure 8/33 Slope deviation in profile slope modification Figure 8/31 Actual values in slope modification and crowning Figure 8/32 Actual values in profile reliefs 8.3 Quality Parameters 571 Figure 8/36 Flank surface modifications 8.3.1.8 Flank Twist In addition to the determination of the twist of the flank surface SȈ (Figure 8/36 left) through 3D regression, twist can be defined as a combination of defined cross sections. For this purpose, a mini- mum of three transverse profiles and three flank lines should be measured on one flank. Figure 8/37 shows a profile and flank line measurement, in which a total of four flanks were measured. The testing of the twist was only carried out on the first measured flank. The twist of the transverse profile is calculated as the difference S Į between the slope deviations of the profiles that envelope the usable face width [1d (down) and 1t (top)], and the twist of the flank line is calculated as the difference S ȕ between the slope deviations of the flank lines at the useable tip and root cylinder (1t and 1r). 8.3.1.9 Waviness The waviness is a periodically repeating component in the course of the form deviation, which can be described in terms of wavelength and wave height. In profile or flank line sections, wave heights f wĮ or f wȕ result for specified wavelengths LwĮ or L wȕ, respectively. More comprehensive results can be obtained when the form deviation curve is analyzed by means of a frequency analysis. Significant waviness in critical frequency ranges under operational conditions should lead to further studies. spatial representation false-colour image frequency analysis Figure 8/38 Spatial waviness measurement (probe radius: 0.5 mm, measuring point distance: 0.02 mm, no filter) flank twist generated relief topographic modification Transverse profile Flank line Figure 8/37 Twist measurement 572 8 Ensuring the Accuracy of Cylindrical Gearing In order to estimate the effect of the waviness on the noise excitation , a spatial measurement and analysis of the tooth flank form deviation is required. Figure 8/38 shows a section of the form deviation of a tooth flank spatially and in a two-dimensional false-colour representation, as well as the results of the frequency analysis represented logarithmically. In addition to the wavelength respectively the number of waves for each normal base pitch and the amplitude, the position respectively direction of the waves on the tooth flank and its assignment to the mating gear are determining the functional characteristics. [8/48]. 8.3.1.10 Tolerance Field (K Diagram) By means of a tolerance field (Figure 8/39, also referred to as the K Diagram), deviations of the transverse profile and flank line can be evaluated in their entirety by testing whether each test diagram is within the defined tolerance field. The tolerance field and test diagram must have the same scales. In profile tests, the roll path, and in flank line tests, the usable face width of the tolerance field must be clearly assigned to the test diagram. The K Diagram must be shifted so that it is tangent to the upper limit of the tolerance field. The part is then considered to be within tolerances when the measurement curve does not leave the tolerance field. Tolerance fields are suitable for the evaluation of the achieved manu-facturing results. The deduction of parameters for quality-oriented manufacturing control is therefore not possible or possible only through a visual judgement on the basis of experiences. Figure 8/39 Examples of tolerance fields (K Diagram) 8.3.2 Pitch Deviations 8.3.2.1 Circular Pitch Deviations Special pitch-measuring devices or the afore- mentioned gear measuring machines are used for the detection of the circular pitch devia- tions. With gear measuring machines, the gear is rotated around the pitch angle τ, and the position of the tooth flank is determined based on this angular position (Figure 8/40 left). The result corresponds to the individual values of the cumulative pitch deviation curve. Special pitch-measuring devices can be designed as stationary or mobile devices [8/66]. Pitch-measuring devices often use two probe systems for measuring (Figure 8/40 right). In this case, one serves as a reference and the other is used to measure the relative position deviation of the current and preceding tooth flank, and thus the single pitch deviation. Mobile devices can be used directly on the gear cutting machine. It should be noted that deviations of the machine also influence the measurement, and so the in-formation content is limited. Figure 8/40 Pitch measurement 8.3 Quality Parameters 573 The single pitch deviation fpt is the algebraic difference between the actual value and the nominal value of a single pitch pt in the transverse plane (Figure 8/41). It is recommended that the pitch measurement be carried out on the V circle. For the quality evaluation, the largest single pitch deviation is used separately for the right and left flanks. Figure 8/41 Single pitch deviation The adjacent pitch difference f u is the difference between the actual sizes of two consecutive single pitches of the right and left flanks. It is indicated without a sign. The adjacent pitch differ-ence is of particular interest in gearing that is manufactured using discontinuous profile machin-ing, because single pitch deviations of the machine add up over the circumference of the gearing, and thus problematic pitch deviations between the first and last machined flank may occur. Figure 8/42 Test diagram of the pitch deviations The cumulative pitch deviation curve (Figure 8/42) is calculated from the continuous addition of all single pitch deviations from k = 1 to z . The total pitch deviation Fp is the biggest difference in the course of the cumulative pitch deviation curve . It is evaluated separately for right and left flanks and indicated without a sign in each case. The cumulative pitch deviation over k pitches Fpk is the difference between the actual value and the nominal value of the circular arc over k pitches. It is equal to the sum of the single pitch deviations over the same k pitches. The number of k pitches can be selected between k = 2 and k = z/2. The cumulative pitch deviation Fpz/8 (i.e., k = z/8) should be preferred for the evaluation of gearing. In Figure 8/43, Fpk for k = 3 is shown. On a gear with z teeth, there are z cumulative pitch deviations of the right flanks and z cumulative pitch deviations of the left flanks (of pitch 1 to k, 2 to k + 1, 3 to k + 2, and so on, respectively). Figure 8/43 Cumulative pitch deviation over k = 3 pitches 574 8 Ensuring the Accuracy of Cylindrical Gearing Especially for large numbers of teeth ( z > 50 or 100), pitch-span deviations can be measured over S adjacent single pitches, instead of the single pitch deviations, in order to reduce the effort required. There are then no measured values from the intermediate teeth. Analogous to the single pitch deviations, z/S pitch-span deviations fpS are obtained in each case for right and left flanks, and, after its addition to the cumulative pitch-span deviation curve, the total pitch-span deviation F pS is obtained. The Figures 8/44 to 8/46 show the effect of hobbing on the measure-ment results. The circular pitch measurement provides quality para-meters related to the gear axis. Thus, the axis position deviation between the gear axis and toothing axis results in a sinusoidal com-ponent with one period at the gear circumference in the course of the single and cumulative pitch devia-tions. The resulting total pitch de-viation corresponds to twice the eccentricity f e. In the case of hobbed gearing, hobbing marks spiralling around the gear also have an effect, as the probe contacts these marks during a pitch mea-surement (taking place in transverse plane). Both influences combine and are mathematically inseparable (Figure 8/44). Figure 8/45 shows the result of a pitch measurement in the hobbing mark valley. This takes place spirally at the gear circumference corresponding to the axial feed. As a result, the component of the total pitch deviation, which results from the axis position deviation can be identified and influenced when setting up the machine. Figure 8/46 shows the effect of a three-start hob on the single pitch deviations. 8.3.2.2 Base Pitch Deviation The direct measurement of the base pitch deviation on the path or plane of contact is carried out with simple hand-held measuring instruments (Figure 8/47) or with blade probes on pitch-meas-uring devices (Figure 8/48) [8/41]. Hand-held instruments are supported directly on the flank, and the measurement results are flank-related. Figure 8/47 shows an instrument model in the measurement on a spur gearing. The cylinder in the left tooth space supports the instrument and, by tilting the instrument, the dead centre of the indication is determined. This shortest distance between the two left-hand tooth flanks represents the actual base pitch on the path of contact. The measurement is also possible in the normal section of a helical gearing with the application of two successively arranged support balls in place of the cylinder. In addition to the actual pitch between the tooth flanks, the deviation of the pressure angle or the base diameter is included in the result. Figure 8/44 Cumulative pitch-span variation in hobbed gearing ( = 29 , fa = 6 mm), measurement in transverse section, Fp = 40 m Figure 8/45 Cumulative pitch-span variation in hobbed gearing ( = 29 , fa = 6 mm), measurement in hobbing trace valley, Fp = 12 m Figure 8/46 Single pitch deviation (three-start hob) 8.4 **Backlash** Allowance Parameters 575 Figure 8/47 Base pitch mea- surement on the path of contact (hand-held instrument) Figure 8/48 Base pitch measurement on the path of contact with blade probes Figure 8/49 Deviation of trans-verse base pitch on the path of contact A base pitch deviation on the path of contact fpe is the difference between the actual value and the nominal value of a base pitch pe. Deviations of the transverse base pitches are indicated by fpet (Figure 8/49), and deviations of the normal base pitch are indicated by fpen. The correspond- ing mean pressure angle deviation can be calculated from the mean normal base pitch deviation of the right and left flanks: (8/8) The transverse base pitch deviation on the base cylinder fpbt is the algebraic difference between the actual value and the nominal value of a single base pitch on the base cylinder pbt in the trans- verse section. It corresponds theoretically to the transverse base pitch deviation on the path of contact, but it is not directly measurable. 8.3.3 Runout Deviation The runout deviation Fr is the variation of the posi- tion of a probe (sphere or cylinder, rider or V head), inserted successively in all tooth spaces and simulta-neously contacting the respective right and left flank, relative to the gear axis. The probe element should contact the tooth flanks around the V-cylinder (Fig-ure 8/50). The runout deviation is identical to the variation of the radial single-ball measurement. Due to the gear axis reference, the double eccentricity f e is reflected in the runout deviation. 8.4 **Backlash** Allowance Parameters 8.4.1 **Backlash** Figure 8/50 Runout testing Figure 8/51 Runout diagram ( z = 16 The **backlash** is present between the non-working flanks of the teeth of a gear pair when the working flanks are in mesh (Figure 8/52). A distinction is made between meshing (base normal) **backlash** j bn, circumferential **backlash** jwt and radial **backlash** jr. The meshing **backlash** jbn is the shortest distance between the non-working flanks of the teeth of a gear pair when the working flanks contact. It can be tested simply but with great uncertainty in [] nnpenm Įnm radʌ sinĮffm=− rF 576 8 Ensuring the Accuracy of Cylindrical Gearing large gearing by placing a lead wire between the non-working flanks and rotating it through the space of the mating gear. The thickness of the deformed wire shows the **backlash**. Anoth-er approach is to test using a thickness gauge. The **backlash** angle ĳ j can be measured by recording the angle by which the gear, in the case of a fixed mating gear, can be rotated from the contact to the right flanks to the con-tact with the left flanks. From this, the meshing **backlash** can be calculated using The circumferential **backlash** j wt is the length of the working pitch circle arc, one gear in case of the fixed mating gear, can be rotated from the contact with the right flanks to the contact with the left flanks. Its quantity is calculated as follows: (8/10) or, based on the length of the reference circle arc, as follows: (8/11) In order to measure the **backlash** angle and circumferential **backlash** at the gear pair (in the transmission), specially adapted fixtures can be used in combination with angle-measuring in-struments and length-measuring instruments, which are arranged on a defined measuring circle. The portion of the circumferential **backlash** resulting from one gear (e.g., in the testing of gauges) can be determined with special circumferential **backlash**-measuring instruments [8/58]. The radial **backlash** j r is the difference of the centre distances between the operating condition and zero-**backlash** engagement of the flank pairs. The variation of the **backlash** parameters on an assembled gear pair can be used as additional quality parameter. 8.4.2 Tooth Thickness Testing the tooth thickness at the single gear, is the basis for compliance with the **backlash**. The tooth thickness is defined as a circular or helical arc. Methods based on coordinate measuring tech-nology can be used to test the tooth thickness. In the case of spur gears with profile shift, measurements close to the V-cylinder are preferable to measure-ments on the reference cylinder. 2Įcos n n j bnz m jϕ = bn b wtwtȕcosĮcos1j j= bn ntĮcosȕcos1j j=Figure 8/52 **Backlash** (shown in transverse plane) Figure 8/53 Tooth thickness, normal chordal tooth thickness and height above the normal chordal tooth thickness on the Y cylinder 8.4 **Backlash** Allowance Parameters 577 The tooth thickness half angle ψ respectively ψy (Figure 8/53) and tooth space half angle η re- spectively ηy, which are frequently used to determine the **backlash** allowance parameters, are obtained for the reference circle d and for an arbitrary circle dy using yt y t ȥȥ invĮinvĮ =+ − (8/12) yt y t ȘȘ invĮinvĮ =− + (8/13) 8.4.3 Tooth Thickness Test Dimensions 8.4.3.1 Preliminary Remarks Indirect measuring methods are common for the evaluation of tooth thickness dimensions. In order to convert the deviations of the tooth thickness into the desired test dimensions, use is made of so-called deviation factors, which describe the relationship between the two. These are summarised in Table 8/2. Because the deviation factors consider only purely geometric relation-ships, they apply strictly speaking only to deviation-free gearing. The relatedness of the mea-sured quantity does not affect the deviation factor! Table 8/2 Tooth Thickness Test Dimensions and Calculation of Deviation Factors Measured quantity Equation for deviation factor Comment tooth thickness on any circle tooth thickness in the transverse section base tangent length diametral two-ball measurement even number of teeth odd number of teeth radial single-ball measurement centre distance of double-flank composite testing normal chordal tooth thickness over z = 12 normal chordal tooth thickness tip diameter n ʌ4 tanĮȥ2 x z+= n ʌ4 tanĮȘ2 x z−= yd ββ = =cos cos y y snsny \*snydd EE E β= =cos1 snst \*stEEE kW n snW \*W cosα = =EEE dKMβ αα≈ =cos sincos Ktt snMdK \*MdKEEE z EEE 2cos cos sincos Ktt snMdK \*MdKπ β αα≈ = rKMβ αα≈ =cos sin 2cos Ktt snMrK \*MrKEEE Laβ αα≈ =cos sin 2cos wtt snaL \*aLEEE ns1 snn s \*n s = =EEE nys ββ ≈ =coscos y y snny s \*ny sdd EE E aMd wn0 snda \*da cotα ≈ =EEE 578 8 Ensuring the Accuracy of Cylindrical Gearing Because some deviation factors include approximations, it is recommended to first calculate the mean tooth thickness allowance Esnm and from this to calculate the mean generating profile shift coefficient xEm: and (8/14) The tolerance mean dimensions of the different tooth thickness test dimensions can be calculated directly as a reference value by using the mean generating profile shift coefficient xEm in the corresponding calculation equations instead of the profile shift factor x. The corresponding upper and lower limits of size can then be determined using the deviation factors. For example, the limits of size for the base tangent length result from and (8/15) Alternatively, the allowable maximum and minimum dimensions can be determined if, in the calculation of the data, the respective associated generating profile shift coefficient x Ei or x Es is used instead of x . 8.4.3.2 Base Tangent Length In external gearing, the base tangent length Wk is the distance between two parallel planes over k teeth, contacting a right and left flank in the involute part of the tooth flanks (Figure 8/54). The lines of contact between the measuring instrument and tooth flank are in a plane tangential to the base cylinder. On internal gearing, the base tangent length is measurable only in the case of spur gearing and then on k tooth spaces. For this purpose, measuring cylinders or measuring balls are used in place of flat measuring surfaces. The base tangent length is usually measured with base tangent length calipers. These are sup-ported only on the tooth flank. The resulting measured quantity is flank-based and therefore independent of position deviations between the gear axis and toothing axis. If the base tan-gent length measurements result from a calculation based on coordinate meas-urements, the changed information con-tent (reference base, influence of local flank deviations) must be noted. In order to prepare a base tangent length measurement, the measured number of teeth k is calculated so that the contact of the measuring instrument takes place as close as possible to the V-cylinder (close to half the tooth height): vt tn 2 btanĮ 2INT invĮ tanĮ 1ʌcosȕz xkz =− − + with b vt vcosd dα= (8/16) sn sns snm21T E E− = n nsnmEmtan 2 α+ =mEx x \*Wsnkm ks2ETW W + =\*Wsnkm ki2ETW W − = Figure 8/54 Base tangent length measurement, external gearing, measured number of teeth k = 3 or k = 5 8.4 **Backlash** Allowance Parameters 579 Usually different numbers of teeth come into consideration in the measurement of the base tan- gent length. Figure 8/54 shows a spur gear. There numbers of teeth between k = 3 to k = 5 may be used. However, in this example, the most advantageous contact conditions in the vicinity of the V-cylinder are obtained with k = 4, which is not shown. With the determined value of k, the base tangent length is given by kn n t n n cosĮʌ invĮ 2s i nĮ2zWm k z x mz =− + + (8/17) Figure 8/55 Representation of the profile length available for a sufficient contact of measuring surface Figure 8/56 Representation of the required face width for the base tangent length measurement In any case, the applicability must be checked. For this purpose, the following questions must be answered: • Is the measuring range of the existing base tangent length caliper applicable? If no measur- ing instrument is available, you can try to increase or decrease the measured number of teeth. The applicability with the new measured number of teeth must then be checked. • Do the measuring surfaces contact sufficiently the usable tooth flank? The resulting measur- ing diameter in the case of a symmetrical contact is calculated by (see dM3 and dM5 in Figure 8/54). The measuring instrument can be tilted with respect to this symmet- rical position as long as the contact remains on the usable flank. Figure 8/55 shows the possi-ble tilting range. The larger the tilting range, the easier and safer the operation. • Is the minimum face width available? This condition is only relevant for helical gearing. Figure 8/56 shows the necessary overlap between the measuring instrument and tooth flank. The minimum face width required for a secure contact is calculated by . The limiting value defined in [8/1] is established empirically with [mm]. 8.4.3.3 Radial Test Dimensions for the Tooth Thickness A distinction is made between the diametrical two-ball dimension and the radial single-ball di- mension. The points of contact between the measuring ball and the right and left flanks should be located close to the V-cylinder. The required ball diameter can be calculated or determined ap-proximately by nomogram or experimentally. Because the measuring balls need to touch the tooth flanks only in the vicinity of the V-cylinder, measuring balls with diameters that are slightly dif-ferent from the determined value can be used. It must be ensured that the selected measuring balls contact the flanks in the usable part and do not have contact with the tooth root surface. 2 k2 b Mk W d d+ = Mb b M b k Fmin cos sin β +β = b W b Mk= 1.2 + 0.018 bW 580 8 Ensuring the Accuracy of Cylindrical Gearing For the measuring balls actually used with the actual diameter DM, the profile angle ĮKt for the ball centre circle is calculated with η as space width half angle according to the equation (8/18) The diameter dK of the circle, on which the measuring ball centre is located, is given by (8/19) With an external gear, the diametral two-ball dimension M dK is the largest outer measurement over two balls (Figure 8/57). With an internal gear, it is the smallest inner measurement between two balls (Figure 8/58). The ball diameter is DM. It contacts the flanks in two tooth spaces that are farthest away from each other on the gear. In small gearing, it can be measured with micro-metres or universal length measurers with inserted balls as probe elements, and in large gearing it can be measured with special support structures and dial indicators. The centre points of the two measuring balls must be in the same transverse section of the gearing. This can be achieved most simply by tilting along the tooth space. The shortest recorded measurement is the two-ball dimension. With an even number of teeth (Figure 8/57), the nominal value is calculated as follows: (8/20) With an odd number of teeth (Figure 8/58), the nominal value is (8/21) Figure 8/57 Diametral two-ball dimension with an external spur gear with an even number of teeth Figure 8/58 Diametral two-ball dimension with an internal spur gear with an odd number of teeth With very large gearing or if the range of the measuring instrument is too small, the tangential two-ball dimension can alternatively be used. It should be noted that the measurement uncer- tainty with a smaller tooth sector rises sharply. The radial single-ball dimension MrK is the distance between the gear axis and, in external gearing, the outermost point, or in internal gearing, the innermost point of a measuring ball with the diameter D M, which has contact in a tooth space on both tooth flanks (Figure 8/59). The nominal value is (8/22) The diametral two-ball measurement is a flank-related and the radial single-ball measurement is a gear axis-related test dimension. The resulting difference in information content has to be taken MM Kt t b nn b binvĮȘ invĮȘ cosĮ cosȕDD zm d=− + = − Ktb KttKcos coscos α=αα=dd d M K dK D d M+ = M K dK 2cos Dzd M + =π ()KM rK1 2Md D =+ 8.4 **Backlash** Allowance Parameters 581 into account in the assessment of the measurement results. For external gearing and internal spur gearing, measuring cylinders with the diameter D M can also be used in place of measuring balls. This results in the dimension over two pins MdZ and the radial single-cylinder dimension MrZ. Figure 8/59 Radial single-ball dimension on an internal spur gear 8.4.3.4 Chords The chordal tooth thickness sժn or sժyn (Figure 8/53) is the shortest distance between the tooth traces of a tooth at the reference cylinder (8/23) or at any cylinder (8/24) The height hժy above the chord sժyn (or hժa over sժn) required for the setting of the measuring instrument is or (8/25) Figure 8/60 schematically illustrates a measuring instrument with support at the tooth tip. Due to this reference, measurement and form deviations of the tip cylinder and the position deviation of its axis with respect to the gear axis are included in the result. This circumstance is very disad-vantageous, because in most cases the processing of the tip cylinder is completed after the blank machining and, thereby, only low quality requirements are met. This disadvantage can be limited by using the actual tip cylinder diameter to calculate the height above the chord. The position deviation of the tip cylinder axis is shown through a sinusoidal deviation curve along gear cir-cumference. 8.4.3.5 Centre Distance of Double-Flank Composite Testing The centre distance of double-flank composite testing a L is the centre distance of a test pair in a zero-**backlash** mating. The test pair is a gear pair consisting of the gear to be tested and a master ()3 n2 sinȥ cosȕ cosȕd s= ()3 yyy yn2 ysinȥcosȕ cosȕd s= 24 ay y y y y ȥ cosȕ 24dd dh−≈+24 a a ȥcosȕ 24dd dh−≈+Figure 8/60 Chordal tooth thickness 582 8 Ensuring the Accuracy of Cylindrical Gearing gear. The test pair is not to be confused with a gear pair in the transmission. For a test pair con- sisting of a gear with the number of teeth z and a master gear with the number of teeth zL, the profile shift factor xL, and with the known tooth thickness actual value EsnL, the value for the associated test dimension can be calculated from () Ln t L LcosĮ 2c o sȕcosĮzzma+= (8/26) with (8/27) Figure 8/63 shows the measurement setup. If the instrument has no absolute scale for the centre distance, the nominal centre distance can be set as the basis by placing gauge blocks between the arbors, and the associated output of the measuring system can be registered (Figure 8/64). The measuring range must cover the setting and measurement. 8.4.3.6 Tip Diameter in Overcut Cylindrical Gears If both tooth flanks and the tip surface of the gearing are produced in one operation, the diameter daM of the overcut tip cylinder can be used as a test dimension for the tooth thickness. In the case of hobbing (and similar methods on external gearing), daM is given by (8/28) When using a shaper cutter, daM is determined by the shaper cutter root diameter df0 and centre distance a0: (8/29) 8.5 Roll Deviations 8.5.1 Preliminary Remarks In the roll test, gear teeth are paired with counter gear teeth, and the joint impact of their individ- ual geometric deviations on the rolling action as roll or composite deviations are measured. This test is therefore also referred to as a composite test. The composite deviations can be associated with a gear (the test object) when a master gear is used as mating gear. The deviations of this master gear are negligibly small compared to the deviations of the test gear. If the deviations of the mating gear are not negligibly small (e.g., in the roll test of two production gears), then the composite deviations can only be associated with the gear pair. 8.5.2 Single-Flank Roll Test In the single-flank roll test, the difference between the actual rotating angle and the theoretical rotating angle, resulting from the transmission ratio in the pairing of a test object with a master gear, is measured. For this purpose, the two gears, with the test centre distance corresponding to the subsequent working centre distance but with no load, are meshed, where either the right flanks or left flanks remain in constant engagement (single-flank engagement); refer to Figure 8/61. EL Lt n Linv Į inv Į2t a n Įxx zz+=++ aM E n fP0 2 2 dd x mh =+ + aM 0 f 0 2 da d=− 8.5 Roll Deviations 583 The single-flank composite deviations of the right flanks of a gear are generally different from those of the left flanks of the same gear. The deviations are usually given as an angle or length based on a measuring circle (e.g., of an V circle) (Figure 8/62). [8/64] Figure 8/61 Single-flank roll test Figure 8/62 Single-flank roll diagram of a gear compared to a master gear The total single-flank composite deviation Fiࡓ is the variation of the deviations in the angle of rotation within one revolution of the test object. The tooth to tooth single-flank composite de- viation f iࡓ is the biggest difference between the angular position deviation, which occurs within a rotating angle corresponding to one mesh. The long-wave component f lࡓ of the single-flank com- posite deviation curve is the diffe rence observed in the ordinate direction between the highest and deepest point of the mean line within one test gear revolution. The short-wave components fkࡓ of the single-flank composite deviation result from the differences between the recorded test trace and the “mean line” that describes the long-wave component (Figure 8/62). The periodicity of the short-wave components on the gear circumference coincides in many cases with the number of teeth of the test object. But influences from helix form deviations also influence these compo-nents. Much more detailed information can be obtained when the recorded deviation in the angle of rotation undergoes a frequency analysis. The deviation components occurring at defined fre-quencies are also called cyclical deviations. Through a special design of a roll master, which is used instead of the master gear, the entire gear can be scanned in transverse planes, whereby the influence of the individual flanks is sepa-rated [8/56]. This variant is interesting due to the rapid recording of all tooth flank surfaces. The data collected can be evaluated spatially (flank topography) and in terms of transverse profile, flank line, tooth thickness deviations and runout. 584 8 Ensuring the Accuracy of Cylindrical Gearing 8.5.3 Double-Flank Roll Test In the double-flank roll test, the centre distance alteration of a gear meshing **backlash**-free with the master gear is measured. Under the influence of a force acting in the direction of the centre dis-tance, at least one left and one right flank of the gears are always engaged simultaneously (two-flank engagement); refer to Fig-ure 8/63. The self-adjusting centre distance in the double-flank roll test is denoted by a L. The radial composite deviation Fi޵ is the variation of the centre distance of double-flank compos- ite testing within one revolution of the test object. The tooth to tooth radial composite deviation fi޵ is the biggest difference of the centre distance of double-flank composite testing within a rotating angle corresponding to one mesh. The radial composite runout Fr޵ is the variation of the long-wave component of the total radial composite deviation curve (Figure 8/64) [8/65]. Figure 8/64 Double-flank roll diagram of a gear compared to a master gear 8.5.4 Deviations in Transmission Gear Pairs 8.5.4.1 Axis Position Deviations of the Gear Pair The centre distance deviation fa is the difference between the actual and nominal centre distances of the gear pair in the centre of the effective face width. The deviation from parallelism between the gear axes of gears 1 and 2 can be determined on the basis of the ef-fective face width b w or housing width L g. A distinction is made between axial inclination fȈį and axial skew fȈȕ (Figure 8/65). Figure 8/65 Centre distance, axial inclination and axial skew Figure 8/63 Do uble-flank roll test 8.6 Contact Pattern 585 Centre distance, axial inclination and axial skew can be calculated with methods based on coordi- nate measuring technology, as well as with simple length- or angle-measuring instruments and appropriate tools. 8.5.4.2 Roll Deviation of Manufactured Gear Pairs The testing of the roll or composite deviation of a manufactured gear pair is carried out over the entire overrun period (at least one revolution of the larger gear) under the same conditions and with the same evaluations as in the single-flank roll test. The tolerance of the total transmission deviation results from the superposition of the individual deviations. The individual deviations are considered random parameters and superimposed by the least-squares method: for example, total transmission deviation (8/30) The testing of manufactured gear pairs under actual or approximate operating conditions, referred to as the roll test, is a convenient tool for evaluating the running performance and noise behaviour of gearing. The evaluation of the measurement results is similar to the single-flank roll test, but it is important to note the conditions deviating from the procedure according to the standards. When measuring the transmission deviation of a multi-stage gear transmission , the transmission accuracy between input and output shaft is determi ned. It includes the single-flank composite deviations of the single gears and the effects of the mounting de viations and, if necessary – when measured under load – the effects of elastic deformation and bearing clearance. 8.6 Contact Pattern As a result of the toothing deviations, deviations of the gear rotating axis and operating influ-ences, it is possible that a tooth flank is not contacted at all points by the mating flanks during meshing. The contact pattern is the area of a tooth flank at which contact actually occurs. A contact pattern is formed by the meshing of a tooth coated with a contrast medium with a tooth of a mating gear (Fi- gure 8/66). Different variations of contact pattern can be dis- tinguished depending on the chosen conditions (e.g., meshing with one or several mating gear teeth, load conditions, test run or normal operation of the transmission, running times). In the case of a mixed contact pattern , several of these conditions were combined. The wear contact pattern is a contact pattern that occurs under operating conditions and is identifiable through wear on the flank surface. In any case, the chosen conditions must be documented, as they significantly affect the results. We usually restrict ourselves to a qualitative evaluation of the contact pattern. For a quantitative description, its width b T and height hT must be determined, as well as the position of its centre with respect to the centre of the flank. The contact pattern centre is defined as the centre of gravity of the loaded area. If necessary, individual portions of the contact pattern can be described by ( bTn hTn). The values obtained are generally subject to considerable uncertainty. The contact pattern size TRA is the percentage share of the area of the contact pattern with respect to the surface of the active tooth flank. 2'22'1'2 , 1 i i iF F F+ = Figure 8/66 Contact pattern photo (loaded areas are light) 586 8 Ensuring the Accuracy of Cylindrical Gearing 8.7 Roughness Measurement Typical microstructures result from the selected manufacturing process, which should not be underestimated, particularly in the load capacity, lubrication and wear conditions and noise excitation. The microstructures of the tooth flank surfaces resulting from discontinuous generat-ing grinding, profile grinding and honing are shown by way of an example in Figure 8/67. In order to better illustrate the microstructures, the form of the actual involute helicoid was re-moved from just the (1 mm by 1 mm) flank section through regression analysis of the measure-ment points, and only the overlying structure was shown. Due to the different profile heights, the magnifications were adjusted so that the structure is clearly visible in all cases. discontinuous generating grinding profile height 12 m profile grinding profile height 3 m honing profile height 4 m Figure 8/67 Microstructure of helical tooth flanks after various manufacturing processes (sections 1 mm by 1 mm) The testing of the surface quality of the tooth flank and the tooth root fillet is usually carried out using 2D contact stylus instruments [8/32]. This is problematic in both cases, because the mea-surement is made difficult by the curved surfaces and the small space within the tooth space. A distinction is made between skid and reference surface probe systems. Measuring instruments with a circular or involute reference are uncommon. Skided probe systems need a much smaller measuring range, as the skid follows the form of the actual involute. But the size of the mecha-nism is mostly a disadvantage. Reference surface probe systems are based on an internal planar reference surface. This is to be aligned approximately parallel to the middle position of the surface to be measured. In the measure-ment of the flank roughness, the probe detects the profile of the involute helicoid as the distance from this reference surface (Figure 8/68). Therefore, because the curvature of the flank is included in the primary profile, a greater measuring range is required than with the use of a skid probe system. The sampling direction must be selected so that it is as perpendicular as possible to the machining marks and thus depends on the manufacturing method. In the tooth root area, machining marks generally face towards the face width. Therefore, the measurement must be carried out in the transverse section. Due to small root fillet radii, this is only partly possible (Figure 8/69). Figure 8/69 Measurement of the rough- ness in the tooth root filletFigure 8/68 Flank roughness 8.9 Use of Spline Gages 587 The choice of the measuring conditions (e.g., cutoff wavelength, tracing length, probe ball ra- dius, measuring point distance) must be made according to [8/15] and [8/17]. If there is insuffi-cient measuring distance or the measuring range of the probe is too small, the number and not the length of individual measurement distances (the cutoff wavelength) must always be reduced. The altered conditions must be pointed out in the results . Alternatively, but with greater uncer- tainty, the roughness can be determined by means of a mould. 8.8 Noise Noise analyses are recognised as test methods that complement the manufacturing process and conclude the manufacturing of a transmission. A distinction is made between the structure-borne sound and airborne sound test. The structure-borne noise test supports the error diagnosis objectively. In an acoustically defined e nvironment (sound booth), no ticeable noise can be identified with the often subjective airborne sound evaluation and, thus, complaints about noise can be avoided. Both complement each other. Microphones and vibration sensors or laser Doppler vibrometers are used in the detection of signals. Single gears meshing with specially selected gears or master gears, individual sets of gears or manufactured transmissions can be tested. The recording of speed and the associated noise, if necessary for all gears of the transmission, supports the subsequent analysis. Final conclusions are possible only with real operating and environmental conditions. In a subjective test, the experience of the tester has to be relied upon. In an objective test, signals can undergo a frequency analysis (FFT), among other things, so as to separate single components and assign possible causes. Special versions of representation, such as frequency spectra recorded depending on the speed (Campbell diagram), are also helpful in the evaluation [8/52], [8/68]. 8.9 Use of Spline Gauges The determination of the described quality and **backlash** allowance parameters is insufficient particularly in the evaluation of splines. Their function (fitting behaviour) can be tested only through the use of gauges. There are toothed ri ng gauges and mandrel gauges for this purpose. Because gauges allow only yes/no decisions, no direct conclusions can be drawn about manu-facturing deviations from their application. 8.10 Symbols and Symbol Explanations a mm centre distance, amplitude aL mm centre distance for double-flank engagement b mm tooth face width bM mm contact line overlap (for measuring base tangent length) bT mm width of contact pattern CHĮ m amount of transverse profile slope modification CHȕ m amount of flank line slope modification CĮ m amount of profile crowning (barrelling) CĮa m amount of tip relief CĮf m amount of root relief Cȕ m amount of flank line crowning CȕI,II m amount of end relief d mm reference diameter da mm tip diameter db mm base diameter df mm root diameter dFa mm tip form diameter dFf mm root form diameter dK mm diameter of circle through centre of ball dM mm diameter of measuring circle 588 8 Ensuring the Accuracy of Cylindrical Gearing DM mm measuring ball diameter dNa mm active tip diameter dNf mm active root diameter dv mm V-circle diameter dw mm working pitch diameter E deviation limit (allowance) f deviation (individual deviation) F deviation (composite deviation) fa m center distance deviation fa mm axial feed fe m center offset FE m total generator deviation ffE m generator form deviation ffĮ m profile form deviation ffȕ m helix form deviation ffȈ m tooth flank form deviation fHE m generator slope deviation fHĮ m profile slope deviation fHȕ m helix slope deviation fiƍ m tooth to tooth single-flank composite deviation Fiƍ m total single-flank composite deviation fiƎ m tooth to tooth radial composite deviation FiƎ m total radial composite deviation fkƍ m short wave component of single flank composite deviation flƍ m long wave component of single flank composite deviation curve fp m single pitch deviation Fp m total pitch deviation fpb m single base pitch deviation on the base circle fpe m single base pitch deviation on the path of contact Fpk m cumulative sector pitch deviation fpS m single pitch span deviation FpS m total pitch span deviation Fpz/8 m sector pitch deviation (k = z/8) Fr m runout FrƎ m radial composite runout fu m adjacent pitch difference fw m wave depth fĮ pressure angle deviation FĮ m total profile deviation fȕ helix angle deviation Fȕ m total helix deviation fį declination angle FȈ m total tooth flank deviation fȈȕ m axial skew fȈį m axial inclination h mm tooth height hթ mm height above chord (chordal addendum) hT mm height of contact pattern jb mm meshing **backlash** jr mm radial **backlash** jw mm circumferential **backlash** k - number of teeth/spaces or pitches in a span LAE mm profile evaluation length (A-E) roll length with mating gear L AF mm profile evaluation length (A-F) roll length with basic rack L CĮ mm roll length of tip or root relief Lg mm width of gear box Lw mm wave length LĮ mm transverse profile fitting area Lȕ mm flank line fitting area m mm module MdK mm dimension over balls MdZ mm dimension over cylinders MrK mm radial single-ball dimension MrZ mm radial single cylinder dimension p mm pitch pb mm base pitch on the base circle pe mm base pitch on the path of contact R - variation Rp m variation of pitch deviation RS m variation of tooth thickness s mm tooth thickness sթ mm chordal tooth thickness SĮ m twist of the transverse profile Sȕ m twist of flank line SȈ m twist of tooth flank T - tolerance TRA mm2 area of contact; contact pattern U - measurement uncertainty Wk mm base tangent length over k teeth or spaces x - profile shift coefficient xE - generating profile shift coefficient xL - profile shift coefficient of master gear z - number of teeth zL - number of teeth of master gear Į pressure angle ȕ helix angle ȕb base helix angle įw pivoting angle (base tangent length measurement) Ș space width half-angle ȟ involute roll angle Ĳ pitch angle ĳj **backlash** angle ȥ tooth thickness half-angle Ȧ rad/s angular velocity 8.10 Symbols and Symbol Explanations 589 Indices: a tip act actual, measured b base circle E generating, generator f root F form circle, maximum usable flank area i lower deviation limit k number of teeth, spaces, pitches or spans K ball dimensions L master gear m mean M measurement max maximum min minimum n normal section P basic rack tooth profile R roll dimensions s upper deviation limit t transverse section v V-circle w working pitch circle w wave y user-defined circle Z cylinder dimensions Į contact, transverse profile ȕ helix (flank line) Ȉ tooth flank 0 generating tool, generating gear 1 pinion (smaller gear of a pair) 2 wheel (larger gear of a pair) I reference face II non-reference face \* coefficient (relative value) 590 9 Drawing Data 9.1 Overview The drawing data can be generally divided into • Geometric data The geometric drawing data can be divided into three complexes: o the graphic representation of the main body of the gear with details of the subsequent bearing (supporting) points and their admissible form and positional deviations, o the representation of the gearing itself with details of the gearing parameters, their tolerances and information on the quality, and o tolerances required on the surface structure, such as roughness and waviness, particularly at the bearing positions and on the tooth flank. • Material details These include the material itself, the heat treatment, the surface protection as well as pre- and supplementary treatments. • Organisational-technical data This means, among other things, determining the checking and acceptance according to the type and scope or, in special cases, technological regulations, compliance with which is necessary for the function and/or load capacity. These include details such as “drilled together with component ...” and “chamfering before heat treatment”. 9.2 Geometric Data Figure 9.2/1 Arrangement of the data in a gear drawing; diagram FRENCO GmbH Gear + Spline Technology Jakob-Baier-Str. 3 • 90518 Altdorf • Germany Phone.: +49 9187 9522 0 • frenco@frenco.de www.frenco.comUltra High Precision H • Master Gears • ^ĞƫŶŐ DĂƐƚĞƌƐ • Spline Gauges • Gear Artefacts • IC Gear Artefacts • WƌŽĮůĞĚ ůĂŵƉŝŶŐ ^ǇƐƚĞŵƐ • Gear Forming Tools • Gear and Spline Manufacture /ŶƐƚƌƵŵĞŶƚƐ ĨŽƌ ^ŝǌĞ /ŶƐƉĞĐƟŽŶ s • /ŶƐƚƌƵŵĞŶƚƐ ǁŝƚŚ 'ƵŝĚŝŶŐ WƌŽĮůĞƐ • Measuring Pins and Ball Inserts • Instruments Rocking Type • Instruments with Face Stop ZŽƚĂƟŽŶ DĞĂƐƵƌŝŶŐ ^ǇƐƚĞŵƐ Z • ŽƵďůĞ &ůĂŶŬ 'ĞĂƌ ZŽůů /ŶƐƉĞĐƟŽŶ • Instruments with Measuring Circles • DƵůƟƉůĞ /ŶƐƉĞĐƚŽƌ • Gear Flank Analyser • Linear Gear Flank Analyser Rack 'ĞĂƌ Θ ^ƉůŝŶĞ /ŶƐƉĞĐƟŽŶ W • tŽƌŬƉŝĞĐĞ /ŶƐƉĞĐƟŽŶƐ • ŶĂůǇƐŝƐ ŽĨ ĞǀŝĂƟŽŶƐ • KŶůŝŶĞ /ŶƐƉĞĐƟŽŶ ĞƌƟĮĐĂƚĞ ͻ ŬŬ^ Ͳ ĂůŝďƌĂƟŽŶ • s Ͳϱ Ͳ ĞƌƟĮĐĂƚĞ • DŽŶŝƚŽƌŝŶŐ ŽĨ /ŶƐƉĞĐƟŽŶ ƋƵŝƉŵĞŶƚ • ůůͲdŽŽƚŚͲDĞĂƐƵƌĞŵĞŶƚ͞ ǁŝƚŚ Z Ez ǀĂůƵĂƟŽŶ Know-How Transfer K • ^ŽŌǁĂƌĞ ͻ dƌĂŝŶŝŶŐ͕ ^ĞŵŝŶĂƌƐ͕ tŽƌŬƐŚŽƉƐ • ŽŶƐƵůƟŶŐ ĂŶĚ ĂůĐƵůĂƟŽŶƐ • >ŝƚĞƌĂƚƵƌĞ ĂŶĚ ŽĐƵŵĞŶƚĂƟŽŶƐ • EĂƟŽŶĂů ĂŶĚ /ŶƚĞƌŶĂƟŽŶĂů ^ƚĂŶĚĂƌĚƐ 'ĞĂƌ DĞĂƐƵƌŝŶŐ /ŶƐƚƌƵŵĞŶƚƐ ŵĂĚĞ ŝŶ 'ĞƌŵĂŶǇ Future-orientated technologies make great ĚĞŵĂŶĚƐ ŽŶ ƚŚĞ ƋƵĂůŝƚǇ ŽĨ ŐĞĂƌƐ ĂŶĚ ƐƉůŝŶĞƐ͘ &Z E K ŝƐ ĐŽŵŵŝƩĞĚ ƚŽ ƚŚĞ ĨƵůů ƐƉĞĐƚƌƵŵ ŽĨ ƋƵĂůŝƚǇ ĂƐƐƵƌĂŶĐĞ ĨŽƌ Ăůů ƚǇƉĞƐ ŽĨ ŐĞĂƌƐ ĂŶĚ ƐƉůŝŶĞƐ͘ KŌĞŶ͕ ƐƚĂŶĚĂƌĚ ƐŽůƵƟŽŶƐ ĚŽ ŶŽƚ ŵĞĞƚ ƚŚĞ ƌĞƋƵŝƌĞŵĞŶƚƐ ŽĨ ŚŝŐŚůǇ ƐĞŶƐŝ- ƟǀĞ ĂŶĚ ĐŽŵƉůĞǆ ĐŽŵƉŽŶĞŶƚƐ ĂŶĚ ƐǇƐƚĞŵƐ͘ ^ƉĞĐŝĂůŝƐĂƟŽŶ͕ ŝŶŶŽǀĂƟǀĞ ĚĞǀĞůŽƉŵĞŶƚƐ ĂŶĚ ŝŶĚŝǀŝĚƵĂů ƐŽůƵƟŽŶƐ ĂƌĞ ƌĞƋƵŝƌĞĚ͘ &Z E K adopted this strategy from the beginning. WƌŽĚƵĐƟŽŶ DĞƚƌŽůŽŐǇ ĨŽƌ 'ĞĂƌƐ ĂŶĚ ^ƉůŝŶĞƐ ^ŝŶĐĞ ϭϵϳϴ͕ &Z E K has developed from a ŵĂŶƵĨĂĐƚƵƌĞƌ ŽĨ ƐŝŵƉůĞ ŵĞĂƐƵƌŝŶŐ ĞƋƵŝƉŵĞŶƚ for splines to a system supplier and specialist ŝŶ ƚŚĞ ĐŽŵƉůĞǆ ĂƌĞĂ ŽĨ ƉƌŽĐĞƐƐ ŝŶƚĞŐƌĂƚĞĚ ƋƵĂůŝƚǇ ĂƐƐƵƌĂŶĐĞ ŽĨ ŐĞĂƌƐ ĂŶĚ ƐƉůŝŶĞƐ͘ dŚĞ ĐŽŵƉĂŶǇ ĨŽĐƵƐĞƐ ŽŶ ŇĞǆŝďŝůŝƚǇ ĂŶĚ ŝƐ ŽƉĞŶ ƚŽ ĚŝƐĐƵƐƐŝŽŶƐ͕ ŝĚĞĂƐ ĂŶĚ ŶĞǁ ǁĂǇƐ͘ ĂƐŝĐ ƌĞƐĞ- ĂƌĐŚ͕ ĐŽŽƉĞƌĂƟŽŶ ǁŝƚŚ ŝŶƐƟƚƵƚĞƐ ĂŶĚ ĂƐ ǁĞůů ĂƐ ĐŽůůĂďŽƌĂƟŶŐ ǁŝƚŚ ƉĂƌƚŶĞƌ ĐŽŵƉĂŶŝĞƐ͕ ĐŽŵďŝŶĞĚ ǁŝƚŚ ŝŶƚĞƌŶĂů ĞǆƉĞƌƟƐĞ ĂƌĞ ƚŚĞ ĨŽƵŶĚĂƟŽŶƐ ĨŽƌ Ă ĨƵƚƵƌĞͲŽƌŝĞŶƚĞĚ ƚĞĐŚŶŽůŽŐǇ͘Quality is the Measure of Success 9.2 Geometric data 593 The geometric data ensues both directly in the graphic representation and in a special gearing table. Figure 9.2/1 shows an example for the arrangement of the data in the drawing. Figures 9.2/2a to 9.2/2d and Table 9.2/1 provide information on the usual data, including on the basis of DIN 3966-1 [9/1] and in general terms DIN ISO 128. Figure 9.2/2 Drawing data for gears: a) cylindrical gear with borehole, externally toothed; b) cylindrical gear, internally toothed; c) cylindrical gear with shaft-connecting joints, externally toothed; d) tolerancing of the roughness on the tooth root surface. Note: Recently, instead of the well conceivable mean (maximum) roughness depth R Z, the arithmetic mean roughness R a – safer from a metrological perspective – has often been preferred The following parameters are included in the graphic representation of the gear: • Tip circle diameter da The tip diameter is generally toleranced with of h 9 to h 11. For larger case-hardened gears, the dimensional changes, which may be significant in individual cases, should be noted. If the tip cylinder is required as a technological basis, or if it is a critical functional aspect, then a narrower tolerance must clearly be required (and subsequently realised by, for example grinding). • Face width b The face width is generally specified with no special tolerance (general tolerance in accordance with DIN ISO 2768 [9/2]). • Root diameter d f The root diameter emerges automatically from the cutter module and the positioning of the tool (this to realise the tooth thickness). This information is therefore frequently disregarded. However, if a risk of interference exists (for example, with major dimensional changes as the result of the heat treatment of large gearings), then this information is useful. Notwithstanding this, though, a sufficiently large tolerance should be allowed for, and attention should also be paid to any influence of the tooth thickness deviations on the root diameter. • Reference basis, datum axis The reference basis (or the reference element of the gear) is generally the gear axis. The bearing positions envisaged can be, for example marked by the representation of the bearings as adjoining parts in accordance with DIN ISO 128-24 [9/3]. Entering the form, positional, and running tolerances required ensues in accordance with DIN EN ISO 1101 [9/4]. • Roughness The information on the maximum roughness of the tooth flank permissible ensues at the line marking the centre of the flank (Figure 9.2/2a and b) by details of the mean (maximum) rough- ness depth R z or the arithmetic mean roughness R a. For ground gears, the permissible roughness 594 9 Drawing Data is approximately Rz = ( 4 t o 6 . 3 ) ȝm or Ra = (0.3 to 0.8) ȝm. For milled gears, this value is approximately Rz = 16 ȝm or Ra = 1.6 ȝm [9/6]. Although details of the roughness for the tooth root surface are normal in the case of high bending, it is only useful if the fillet has no grinding steps since their influence is then pre-dominant. In limiting cases, polishing of the tooth root surface is required. When using normal protuberance tools, values up to R z = 25 ȝm are permissible. Shot peening generally removes the influence of any roughness on the tooth root strength of Rz ≤ 25 ȝ m (and – in individual cases – of greater values). The significant data relating to the gearing are summarised in a table. Included are the description para-meters as well as details of the gear tooth quality in accordance with ISO 1328-1, -2 (DIN 21772; formerly DIN 3962 to 3964) and the determination of the tooth thickness deviations (upper and lower allowances). The values can be freely chosen or determined in accordance with DIN 3967 (compare ISO 21771 Annex A and Section 8). The information listed in the gearing table (Table 9.2/1) can also be supplemented by • the basic rack for the tool • the tooth height h • the chordal tooth thickness s թ and the chordal height • the radial or diametral test dimension and the measuring ball diameter • the two-flank working distance La • special details for geometrical deviations Tooth flank modifications Modifications of the transverse profile and the tooth trace are generally described in separate repre- sentations of diagrams (refer to Figure 9.2/3). A distinction is made between crowning, reliefs and angle modifications in each case. Reliefs have been hitherto generally realised in linear or involute form (linear in the diagram). The length and amount of the modification are then sufficient as drawing data (refer to Figure 9.2/3a). Recent investigations have revealed that the transition from a non-modified to modified flank part can be realised with increasing accuracy by improving the Table 9.2/1 Gearing Table Cylindrical gear externally toothed 1) internally toothed 1) Module mn Number of teeth z Basic rack ISO 53 (DIN 867)1) Helix angle β Flank direction Reference diameter d Base diameter db Profile shift coefficient x Addendum alteration k m n Gear tooth quality, tolerance field 2) Base tangent length 3) over k teeth Wk k = Mating gear Item number number of teeth z Centre distance in housing with tolerances a Length of path of contact gα Supplementary information (as required) 1) Delete as appropriate. 2) In accordance with ISO 1328 (DIN 21772, formerly DIN 3962, 3963, 3967; enter as appropriate). 3) Alternatively: details for the direct checking of the tooth thickness or other test dimensions, for example over balls. 9.2 Geometric data 595 manufacturing technology, thereby creating a kink on the flank, which leads to an increase in contact stress at this transition point and to greater flank pressure. This should be avoided by setting a parabolic (instead of a linear) relief, whose transition connects to the non-modified flank part without a kink (refer to Figure 9.2/3b). Details of the mathematical function or (approximately at suitable intervals) the size of the modification are then required in the drawing data. Crowning has hitherto been designed in a circular form or corresponding to a parabola (refer to Figure 9.2/3c). New approaches in this area also offer interesting possibilities. The design of a higher-order exponential curve (refer to Figure 9.2/3d) combines, for example, the benefits of crowning and end relief. Figure 9.2/3e depicts a linear angle modification of the tooth trace with superposed crowning. In each case, the mathematical function should be unambiguously specified in order to allow a safe implementation of the manufacturing and measuring technology. Figure 9.2/3 Description of modifications: a) linear profile tip and root relief (sharp transition still common but extremely unfavourable due to contact stress!; this method should no longer be used); b) parabolic profile tip and root relief; c) circular tooth trace crowning (p arabola); d) tooth trace crowning (higher-order exponential curve); e) tooth trace angle modification (to compensate for misalignments) with superposed crowning (to compensate for fluctuations) (to select modification sizes and tolerances refer to Appendix 3.2) The modifications of the total flank area of the wheel as a whole are differentiated into intended profile twist, diagonal (triangular or generated) reliefs in the rolling direction at the beginning and end of contact, and topographic modifications, for which the desired position of the points on the tooth surface in relation to the deviation-free involute helicoid is specified for a grid in the plane of action (refer to Figure 8/36). The corresponding drawing data must unambiguously describe the modification parameters (limits and depths) and possibly also the mathematical function or the individual points. The following additional details may be necessary or useful for special cases : • Working flank For the non-working flank, larger deviations are possible and should possibly be recorded in separate guidelines specified in the drawing. • Material removal when balancing Dynamic balancing is required at greater speeds. For highly stressed gears, the region permis- sible for material removal (type, location, surface, depth) should be specified. 596 9 Drawing Data • Place of marking For reasons of expediency and to avoid dama ge to highly stressed points, the place and style of marking should be specified. • End-face and tooth tip chamfer For highly stressed gears, not only the size of the chamfering (or rounding) but also the technological classification (e.g., “chamfered before carburisation”) should be specified, in order to avoid unfavourable effects on the local residual stress condition. A fast, secure exchange of geometric data between the construction, manufacture and metrology stages should be realised by means of a data exchange format (GDE format) defined in the VDI Guideline 2610 [9/5]. Here the use of XML (amongst other features) allows simple use on the Internet and the further processing in databases. Inspection regulations , in which the type, scope, performance and documentation of the inspections are recorded, are normal with well-developed manufacturing processes and are marked in the manufacturing documents. Figure 9.2/4 provides a rough overview of the qualities achievable depending on the manufacturing method. Pressed, stamped Hobbed, shaped Shaved Ground Quality in accordance with ISO 1328 (DIN 3962 to DIN 3964) 1 2 3 4 5 6 7 8 9 10 11 12 decreasing quality Figure 9.2/4 Gear tooth quality achievable in accordance with ISO 1328 (DIN 3962 to 3964) in association with industrial manufacture depending on the manufacturing method. (The dark regions denote the accuracy achievable with “normal” effort. For special requirements, such as test or master gears, the unmarked qualities of higher accuracy are also realised in individual cases.) Some special standards are compiled in the literature of Section 9 and in Appendix 17.1. 9.3 Information on Heat Treatment and Surface Coating 9.3.1 Information on Heat Treatment The geometric data on the gear should be extended to include the material properties ensuring the manufacture and function (surface, edge layer, volume). Standardised specifications exist both for the method of entering the terms and procedures and for the choice of parameters determining these properties. The content of some proven German standards (DIN) has been taken over into international stand-ards (ISO, EN). In some cases new symbols have been in troduced. These international standards will be adopted as DIN ISO or DIN EN and replace the former DIN standards. The terms used in heat treatment of ferrous materials are contained in DIN EN 10052. The heat treatment description ensues in accordance with DIN ISO 15787, “Technical Product Doc-umentation – Heat-Treated Ferrous Materials Components – Presentation and Details”. The resulting specifications have to be agreed on for the manufacturing process. This has led to ex-tensive modification of DIN 6773. One of the principal changes concerns the abbreviations used for layer depths (refer to Table 9.3/1). 9.3 Information on Heat Treatment and Surface Coating 597 Table 9.3/1 Comparison of the Abbreviations Used for Layer Depths, in Accordance with DIN ISO 15787. This Standard Also Includes Remarks Abbreviation in accordance with DIN 6773:2001 Abbreviations in accordance with ISO 15787 Designation in accordance with DIN ISO 15787 Eht CHD Case-hardening depth At CD Carburisation depth VS CLT Compound layer thickness Nht NHD Nitriding hardness depth Rht SHD Surface hardening depth The drawing data can consist of the following: • General information: Here machining allowances can be defined for depth data. • Information on the material: The material (with as-received condition) is specified in the text field of the drawing. • Heat-treatment condition: The terms used to describe the condition after heat treatment must be in the form of the past participle of the appropriate verb (e.g., “hardened”). • Hardness details: The surface hardness and – as required – the core hardness must be stipulated in accordance with standardised measuring methods and with the greatest possible tolerances meeting functional requirements. • Marking the measuring points: If measuring points are necessary, they must be marked and di- mensioned in conformity with the standards. • Hardness depth: The terms and abbreviations listed in Table 9.3/1 apply to all depth data. The relevant standards on their determination must be complied with (refer to Sections 7.4 and 10.3). The values as well as the following depth data should be specified with the greatest possible tol-erance meeting functional requirements. The lower limit deviation must be zero. • Carburisation depth: This is generally defined with a limiting carbon content. • Compound layer thickness: This only refers to the layer formation in association with nitriding. • Details on strength: Strength values should only be specified when necessary. The definitions on the sampling location in the material standards, e.g., DIN EN 10083, should be noted. Microstructure: Information on the hardness and hardness depth can be supplemented by the mi-crostructure after heat treatment. With the last five sets of data, it should be noted that the values can only be obtained by destroying a workpiece or a dimension-distinguishing sample of equal type of steel. DIN ISO 15787 contains comprehensive information and examples on the graphic representation. “Heat-treatment images” are used considering for regions necessitating heat treatment, regions which may be heat treated, and regions which may not be heat treated. Different line types have been defined for their marking. Heat-treatment data or a heat-treatment image together with heat-treatment details are expediently entered on the left side of the text field (refer to DIN ISO 15787). 598 9 Drawing Data In addition to comprehensive specification of heat-treatment details for methods appropriate for the stress, DIN ISO 15787 also provides information on annealing procedures, whereby the “normal-ised” specification, for example, can be supplemented by a toleranced hardness value and differen-tiated details on the microstructure. The importance and unambiguity of these individual details is underlined by the fact that they have to provide the designer with the guarantee of the characteristics for the durability of the gear wheels in the gear unit. It is assumed that all the parameters specified can be tested, either directly on the component or indirectly on treated samples that are representative of the components [refer to the requirements on the quality of the material and on heat treatments in accordance with ISO 6336-5 (DIN 3990-5)]. Product-related material properties in intermediate states (e.g., stress-free annealing following rough machining) should be entered in the heat-treatment order (refer to DIN 17023) with the abbreviation HTO , in the heat-treatment schedule with the abbreviation HTS, or in the production plan . The information necessary for gears are depicted in Table 9.3/2a, b (arranged by heat-treatment method; refer to Section 10.3) by means of examples based on DIN ISO 15787. Annealing : In addition to the name of the procedure (e.g., “normalised”), a hardness range (e.g., in Brinell units) can be specified, as contained in standards (e.g., DIN EN 10083). If, in addition, a micro-structure requirement has been set, then its possible realisation must be carefully checked. A dimensionally-equal sample/gear is necessary, which is identical in relation to the chemical compo- sition and has been treated with the annealing batch. Note: For annealing procedures, re fer to Section 10.3.1.1 to 10.3.1.4. Quenching and tempering: The procedure description is contained in DIN 17022-1. The information on the mechanical pro- perties to be achieved ensues mainly by means of the hardness in Brinell, Rockwell or Vickers units (for reassessing hardness in tensile strength refer to Appendix 11). Details of the tensile strength Rm (toleranced), yield stress Rp0.2 (minimum value) and fracture elongation A (minimum value) should only ensue if a gear or a similarly dimensioned test part for taking samples from a specific geometric region (sampling location) can be destroyed (refer to Table 9.3/2a, Figure 9.3/1). The hardness, tensile strength and other mechanical properties are approximately expressed for each kind of steel, depending on its hardenability, the quenching and tempering parameters (refer to Section 7.4.2.4), and the size and the overall design of the gear, by the distinguishing dimension (refer to Section 7.4.1.2). Note: For quenching and temperi ng, refer to Section 10.3.2.1. Austempering : The procedure description is contained in DIN 17022-1. Due to the technological complexity of the procedure with the resulting characteristics, the heat-treatment information for steels is restricted to the term “austempered” and narrowly toleranced hardness value in HRC together with a reference to a heat-treatment instruction. For the austempering of spheroidal graphite cast iron (ADI), details of the tensile strength, yield stress, strain and Brinell hardness are possible (refer to DIN EN 1564). Note: For austempering, refer to Section 10.3.2.2. Surface hardening: The procedure description is contained in DIN 17022-4. Edge-layer hardening comprises the flame and induction hardening procedures relevant for gears, as well as the still rarely applied laser and electron beam hardening. 9.3 Information on Heat Treatment and Surface Coating 599 As only the toothing – and not the entire component surface – is treated with the edge-layer harden- ing procedures, the heat-treatment image is required. The regions to be partially treated are marked with a wide dotted line outside the body contour (refer to Table 9.3/2a, Figure 9.3/2). The text for the details of the heat treatment is next to the dotted line, while the properties to be achieved are be-low the dotted line. As a general rule, this specification applies to all the following edge-layer hard-ening methods. The test load to measure the surface hardness (HRC or HV) should be selected de-pending on the depth of the edge-layer hardening in accordance with DIN ISO 15787 (for minimum achievable hardness refer to Section 7.4.3.1, Table 7.4/8). The abbreviation SHD with the readjusted numerical value of the limiting hardness in Vickers units (for limiting hardness values, refer to DIN ISO 15787) denotes the edge-layer hardening depth (DIN EN 10328:2005-04, currently undergoing revision). For technologically realisable depths of surface-layer hardening (including the upper limiting deviations), refer to DIN ISO 15787. Surface-layer hardening depths for gears may be selected from Section 7.4.4.5, Table 7.4/17. Note: For edge-layer hardening, refer to Section 10.3.2.3. Carburisation : The procedure description is contained in DIN 17022-3 (refer to Section 10.3.2.4). If, in special cases, carburisation is performed on gears without subsequent hardening, the carburisation depth (abbreviated as CD) should be marked with a limiting carbon content, usually 0.35%, denoted as CD 0.35. For technologically realisable carburisation depths (including the upper limiting devia- tions), refer to DIN ISO 15787. In this regard, proof is required in the form of a sample piece or gear having the same batch of melted mass and the same dimensions, which has also been treated with the carburisation batch. Partial carburisation is possible (here a heat-treatment image is necessary). Note: For carburisation, refer to Section 10.3.2.4. Case hardening: The procedure description is contained in DIN 17022-3 (refer to Section 10.3.2.4). Surface and core hardness are dependent on the hardenability of the case-hardened steel in question, primarily on the characteristic dimension and the quenching medium. For MQ-quality gears, the surface hardness should be specified with 58 to 63 HRC (corresponding to a range between 655 and 770 HV) [re-evaluation of the hardness values in accordance with DIN EN ISO 18267 and (as orientation) Appendix 11]. The test load to measure the surface hardness (HRC or HV) should be selected depending on the depth of the edge-layer hardening in accordance with DIN ISO 15787. The abbreviation CHD denotes the case-hardening depth (refer to DIN EN ISO 2639), which also applies to carbonitriding. It is also possible to specify various case-hardening depths (refer to Table 9.3/2b, Figure 9.3/3), as well as partial carburisation. The heat-treatment image is necessary. Partial carbonitriding (with direct hardening) is not common. For technologically realisable case-hardening depths (including the upper limiting deviations), refer to DIN ISO 15787. For case-hardening depths for gears, refer to Table 7.4/21 in Section 7.4.4.6. Note: For case hardening refer to 10.3.2.4; for carbonitriding refer to Section 10.3.2.5. Nitrogen content increase: The term selected denotes the nitriding of corrosion-resistant steels at temperatures around 1100 C with subsequent quenching, also referred to as case hardening with nitrogen [7.4/46], [48]. Here details are required in analogy to classic case hardening. As the treatment represents a special case, the heat-treatment details should be coordinated with the hardening plant. 600 9 Drawing Data Nitriding/Nitrocarburising: The procedure description is contained in DIN 17022-4. The minimum hardness achievable by nitriding is specified in Section 7.4.3.5 in Table 7.4/13. The surface hardness is a subordinated factor for nitrocarburising. The test load to measure the surface hardness (in Vickers units only) should be selected depending on the nitriding hardness depth in accordance with DIN ISO 15787. For nitrided gears, the compound layer thickness (abbreviation CLT) should be 10 ȝm. The nitro- carburising process results mainly in thicker compound layers. For compound layers produced by nitriding or nitrocarburising, DIN 30902 seeks to distinguish between the thickness of the porous component CLT p and the thickness of the non-porous component CLT np. For technologi- cally realisable compound layer thicknesses (including the upper limiting deviations), refer to DIN ISO 15787. The abbreviation NHD denotes the nitriding hardness depth (refer to DIN 50190 T 3). Although partial nitriding is possible here, it is mostly not appropriate. Technologically realisable nitriding hardness depths (including the upper limiting deviations) are contained in DIN ISO 15787. For nitriding hardness depths for gears, refer to Section 7.4.4.9, Table 7.4/24. The oxidisation (as appropriate) ensuing following the nitriding can be additionally entered with “oxidised” as well as layer type and thickness – e.g., “oxide layer (Fe 3O4) 4 m” – below the nitriding details. Note: For nitriding/nitrocarburising, refer to Section 10.3.2.6. Boriding: This process is used in exceptional cases for gears for the purpose of enhancing the wear resistance. Alongside the term “borided”, a minimum value of the surface hardness (Vickers microhardness), the thickness of the boride layer, and, as appropriate, the layer composition with FeB/Fe 2B or Fe 2B should be specified. The choice of the layer parameters is a matter of experience. The heat-treatment details should be coordinated with the hardening plant. Note: For boriding, refer to Section 10.3.2.7. Surface coating Quenched and tempered, case-hardened or nitrided gears are surface coated with hard materials using the PACVD process (temperatures between 150 and 700 C) and the PVD process (tem-peratures between 200 and 500 C) to improve the load capacity [7.4/61], [7.4/62]. The high de-gree of layer hardness (up to over 2000 HV 0.05) is provided by the coating material. The layer thicknesses are in the range up to 10 μ m. As this kind of treatment still represents a special case, the particular details should be coordinated with the company carrying out the surface coating. In this regard, chemical and electroplated layers, such as that produced by phosphating and copper-ing, are worth considering. Note: For surface coating, refer to Section 10.3.2.8. 9.3 Information on Heat Treatment and Surface Coating 601 Table 9.3/2a Examples of Drawing Details According to DIN ISO 15787 for Thermal Gear Treatment Procedures Standards for drawing details 1) DIN EN 10084 2) DIN EN 10083-3 DIN 17022-1 3) DIN EN 1564 DIN 17022-1 4) DIN EN 10083-3 DIN 176022-1 5) DIN EN 10084 DIN 17022-3 6) DIN EN 10085 DIN 17022-4 7) DIN EN 10083 Figure 9.3/2 Heat treatment picture and marking of measuring position; Surface hardened, entire part tempered Surface hardening 42CrMo4+QT 4) surface hardened, entire part tempered (ͷʹ ଴ା଺ HRC) see Figure 9.3/2 according to DIN ISO 15787 refer to Figure 9.3/2 according to DIN ISO 15787 Austempering EN-GJS-400-18 3) to EN-GJS-1000-5 austempered 300 to 360 HBW and/or Rp0.2 700 MPa Rm 1000 MPa n/a n/a n/a Quenching and tempering 34CrMo4+A 2) quenched and tempered 40 mm < s 100 mm 235 + 50 HBW and/or Rp0.2 550 MPa Rm = 800 + 150 MPa n/a according to DIN ISO 15787 refer to Figure 9.3/1 according to DIN ISO 15787 Figure 9.3/1 Heat treatment picture and marking of sample taken with measuring position; quenched and tempered par t Annealing 16MnCr5+U 1) normalised (+N) 138–187 HBW n/a n/a n/a Decision of the designer Material and condition as delivered Heat-treatment procedure (final state) Hardness and/or strength Depth in the case of surface layers Denomination of the position of measuring or sample taken Picture for heat treatment 602 9 Drawing Data Table 9.3/2b Examples of Drawing Details According to DIN ISO 15787 for Thermal Gear Treatment Procedures Coating 16MnCr5 case hardened DLC coated 1500 HV 0.05 layer thickness min. 3 m n/a n/a Boriding 41Cr4+N 7) boron treated 1600 HV 0.05 boride layer thickness n/a n/a Figure 9.3/3 Heat-treatment picture and marking of measuring position; surface-hardened part Nitriding Nitrocarburising 31CrMoV9+QT 6) nitrided NHD = ͲǤͷ ଴ା଴Ǥଶହ n/a n/a Case hardening Carbonitriding 18CrNiMo7-6 5) case hardened and annealed refer to Figure 9.3/3 according to DIN ISO 15787 refer to Figure 9.3/1 according to DIN ISO 15787 Carburising 20MnCr5+N 5) carburised n/a n/a n/a Decision of the designer 1) Material and condition as delivered Heat-treatment procedure (final state) Hardness Depth in case of surface layers Denomination of the position of measuring Picture for heat treatment 1) For numbers 5) to 8) refer to Table 9.3/2a. 9.3 Information on Heat Treatment and Surface Coating 603 9.3.2 Information on Surface Coating According to Table 7.4/2, fatigue-proof coating, wear-protection coating and corrosion-preven- tive coating used for gears belong to surface coating (refer to DIN 8580: Manufacturing method, Terms, Classification). They are described in Section 10.3.2.8. Surface coating always ensues following heat treatment in the finished condition. As a matter of principle, it is necessary to differentiate between two types of layers: • layers with an extremely high degree of hardness, the hard material layers, primarily produced by PVD treatment, and • layers with low to medium hardness, arising from electroplating or chemical treatment. 9.3.2.1 Hard Material Layers DIN standards on the surface coating of gears with hard material layers do not as yet exist. For this purpose, VDI Guideline 3198 in conjunction with VDI Guideline 3824 may serve as a reference point for the information in the drawing. The following information must be specified in the drawing: • the PVD method primarily used, e.g., PVD treated • the coating material to be applied, e.g., WC/C (tungsten carbide/carbon) • the layer hardness required, e.g., 1100 HV0.05, and • the layer thickness to be achieved, e.g., 2 to 4 m. The particular details should be coordinated with the company carrying out the surface coating, taking account of the temper resistance of the base material and, as appropriate, any intermediate layer present (e.g., nitrided layer). 9.3.2.2 Galvanically Applied Layers Galvanic coppering is the normal method used to improve the running-in characteristics. In contrast to the terminology used in accordance with DIN 8580, these layers are known as coat- ings. Galvanic coatings are specified in DIN EN 12540, including the copper-nickel coating. The designation of coatings in technical documents ensues in accordance with DIN EN ISO 27830, while the drawing details ensue in accordance with DIN 50960-2. The DIN EN ISO 2064 (Defini-tions) and DIN EN ISO 1463 (Layer Thickness Measurements) standards apply to measuring the layer thickness. In accordance with DIN 50960-2, the coatings are expressed by a graphic symbol of surface quality in accordance with DIN EN ISO 1302. case hardened and tempered 750 + 100HV30 CHD = 0.5 + 0.3 ISO 1456-Fe//Ni30b/Crr Figure 9.3/4 Drawing details for the combination of case hardening and surface coating In addition to all-over coating, partial coating is also possible, whereby different line types are used, similar to the heat-treatment details in accordance with DIN ISO 15787. The particular details should be coordinated with the company carrying out the surface coating . 604 9 Drawing Data 9.3.2.3 Chemically Applied Layers The use of phosphating in order to improve the frictional resistance or provide temporary corrosion protection is described in DIN EN ISO 9717. Zinc phosphate (Znph) or manganese phosphate (Mnph) coatings are worth considering. The area-related mass (g/m2) is used to indicate the quality of the coating. Manganese iron phosphate coatings with 5 to 20 g/m2 are re- commended for gears. It is appropriate to use the graphic symbol of galvanic surface coating for the drawing details and use the designation in the sequence as laid down by DIN EN ISO 9717 in Item 4 (refer to Figure 9.3/5). Partial surface coating is not described. ISO 9717–Fe /Mnph15 (Base material) (Coating) Figure 9.3/5 Drawing details for phosphating; entries in addition to heat-treatment details or surface roughness or on the part of the component contour marked for the surface coating or as per Figure 9.3/4. Note: The explanations provided in parentheses in Figure 9.3/5 are not part of the drawing specification 9.4 Symbols and Symbol Explanations At mm carburisation depth CD mm carburisation depth At0.35 mm carburisation depth for a limit carbon content of 0.35% CLT mm compound layer depth CLT p mm porous proportion of compound layer depth CLT np mm non-porous proportion of compound layer depth dw mm shaft diameter HTO - heat-treatment order HTS - heat-treatment schedule NHD mm nitriding hardening depth Nht mm nitriding hardening depth Rht mm surface hardening depth Rz mm average surface roughness SHD mm Surface hardening depth s mm distinguishing dimension sթ mm tooth root chord VS mm compound layer depth (15 g/m2 area- related mass ) 605 10 Manufacturing of Cylindrical Gearings 10.1 General Manufacturing Process for Cylindrical Gears As the main functional element of cylindrical gears or cylindrical gear shafts, the toothing presents high demands on the entire manufacturing process [10.1/1], [10.1/2]. The main reasons for this are as follows: • A high degree of geometric complexity and assignment of the three-dimensionally structured tooth flanks and tooth root surfaces to other functional elements • Different functional requirements on the material properties in the region of the tooth flanks, the tooth root and the workpiece core where the surface-layer characteristics with regard to the course of the microhardness, internal stresses and surface microstructure are of particular importance • High precision requirements in fulfilling different requests in relation to load capacity, quiet running and kinematic precision of the paired gears as well as a high degree of dependency of the accuracy of the toothing on the precision of the functional datum axis The fulfillment of all these requirements is generally ensured by means of a multi-stage manufac-turing process [10.1/3]. A variety of processes have been developed for the manufacture of cylin-drical gears, which may represent differing optimal solutions depending on the technical charac-teristics of the toothing, such as geometry, material, structure, and precision, plus the batch size of the workpieces as well as the manufacturing processes used in each case. In order to obtain a generalised working schedule for the production of cylindrical gears, it is advantageous to refer to the main workpiece form-related process steps . The technological similarity in this case results in a working schedule , which includes the main basic process steps and is modified according to technical-economic requirements and possibilities (Table 10.1/1). Key modifications to the working schedule in special cases are as follows: • Depending on the complexity and precision of the processes used, it is possible to combine several main process steps into one single step. • The (fine) finishing can be fully or partly identical with the pre-machining or combined with it. • For technical-economic reasons, the modification of the material characteristics by hardening or surface-layer hardening may only take place following the completed (fine) finishing. However, the requirement in this regard is low distortion, which does not result in any significant loss of quality of the already-finished cylindrical gears. The special features involved in machining cylindrical gears compared to other mechanical engi-neering workpieces primarily consist in • the high degree of precision required when manufacturing the datum surfaces for the workpiece centring and clamping in the toothing manufacture when performing the basic and subsequent shaping, both during its pre-machining (p roduction-related) and its finishing/(fine) finishing (function and production-related), and • the geometrically and kinematically complicated processes of tooth-shape forming. 606 10 Manufacturing of Cylindrical Gear Units Table 10.1/1 Generalised Sequence of Working Schedule for the Manufacturing of Cylindrical Gears Main process steps Special features of cylindrical gears only exist with regard to tooth-shape forming, since the technological solutions for the manufacture of datum surfaces from functional and production-oriented requirements are not limited to cylindrical gearing manufacturing processes. To that extent, past, present and future tooth-shape forming is the crucial element in the manufacturing of cylindrical gears, in which the influences of the other process steps – particularly the manufacture of the datum surfaces – must be taken into account. Callenberger Str. 52 | 96450 Coburg | Germany Phone: +49 9561 866-0 | Fax: +49 9561 866-1003 E-Mail: info@kapp-niles.com | Internet: www.kapp-niles.com The KAPP NILES SURGXFW OLQH LQFOXGHV JHDU DQG SUR OH JULQGLQJ PDFKLQHV DV ZHOO DV JULQGLQJ DQG GUHVVLQJ WRROV :H RIIHU RXU FXVWRPHUV IROORZLQJ SURFHVVHV IRU KDUG QLVKLQJ RI JHDUV DQG SUR OHV generating grinding SUR OH JULQGLQJ bore and end face grinding in combination with gear processingthousandth millimetre and up to a diameter of eight metres. Specialists optimise each system solution LQGLYLGXDOO\ IRU WKH FXVWRPHU V UHTXLUHPHQWV DQG SURYLGH VXSSRUW WKURXJKRXW LWV OLIH F\FOH 0DFKLQHV DQG WRROV IURP .$33 1,/(6 JXDUDQWHH ERWK SUHFLVLRQ DQG FRVW HIIHFWLYHQHVV ZKHQ PDQXIDF WXULQJ VRSKLVWLFDWHG FRPSRQHQWV ,Q VL[ ORFDWLRQV WKH NQRZ KRZ DQG TXDOLW\ RI 0DGH LQ \*HUPDQ\ LV SUHVHQW ORFDOO\ LQ DOO LPSRUWDQW PDUNHWV ,Q WKLV ZD\ .$33 1,/(6 FXVWRPHUV VHW WKHLU FRQFHSWV DQG SURGXFWV LQ SUHFLVH PRWLRQ RQ ODQG LQ WKH ZDWHU DQG LQ WKH DLU precision for motion 10.2 Tooth-shape Forming Methods 609 10.2 Tooth-shape Forming Methods 10.2.1 General Structure of the Tooth-shape Forming Methods The processes used for tooth-shape forming can be assigned in accordance with Figure 10.2/1a to the principal methods, that is, primary shaping, forming and separating with the chipping method, or represent a combination of them. Of the many technically possible methods availab le, those ensuring the manufacture of the too thing in the least number of steps possible in association with low production costs have proven themselves as the most suitable. The key criteria in the choice of a suitable method (or combination of methods) are the quantity to be produced (large-scale or small series production, piece production), the quality requirements (application) as well as the dimensions (mass) of the gears to be produced. From geometric-kinematic perspectives, the machining methods of tooth-shape forming with chip removal are divided into (refer to Figure l0.2/1b) • forming processes (point-by-point or line-by-line), • profile-forming processes, • rolling processes (with rack or gear), • cross-helical rolling processes (with cylindrical worm), and • screwing and moulding processes. Figure 10.2/1a Tooth-shape forming: a) processes and their useful combinations [10.2/2] The rolling and screwing processes reproduce the kinematics between the functional elements of differing types of gears in association with superposed functional movement, as appropriate [l0.2/l]. The following process modifications, in particular, are known in (fine) finishing: • simultaneous profile machining of all or a specific number of tooth spaces, • tooth (space) profile machining with simultaneous generation of both tooth flanks and the tooth root surface, • double-flank machining with simultaneous generation of both tooth flanks, • single-flank machining with phased generation of both tooth flanks. 610 10 Manufacturing of Cylindrical Gear Units Figure 10.2/1b Tooth-shape forming: b) functional principles 10.2.2 Primary Shaping Manufacture of Cylindrical Gears Primary shaping tooth-shape casting as the original technical casting process group for producing metal cylindrical gears is currently important for the manufacture of cylindrical gear blanks of large dimensions for low strain. These include the • hand or machine moulding process (lost mould with permanent die), • shell moulding process (lost mould with permanent die), and • investment casting process (lost mould with lost pattern), which are only economically viable when applied to (very) large modules. On the other hand, efforts have been recorded – particularly in the area of high-volume and mass production – to produce cylindrical gear toothing using the precision casting process that fulfils the usual precision requirements of pre-machining for subsequent (fine) finishing [10.2/2] to [l0.2/9]. These primarily include • gravity die-casting (high-temperature-resistant steel or graphite permanent mould) for cylindrical gears with modules m < 3–4 mm and diameters up to 500 mm from heavy metals (and alloys), such as aluminium-bronze alloys for slow-runni ng gears and actuator transmissions; • precision casting (lost mould with lost-wax casting) (see Table 10.2/2 number 1) for the manufacturing of toothed workpieces up to approx. 100 kg in weight, as well as complex geometry in association with large quantities. In most cases, the lost-wax casting technique is used, whereby a wax model is first produced in a primary shaping. The tolerances that may be produced are between 0.3 and 0.7 % of the nomin al value. A finish machining with chip removal usually ensues for cylindrical gears subject to low loads only; • pressure die casting (permanent hot-working steel model) for cylindrical gears with module m < 1.5 mm, 0.l to 40 kg in weight, and quantities between 1000 and 5000 from low-melting materials (such as aluminium, copper, zinc and magnesium alloys), for drive and positioning purposes, as well as cylindrical gear segments with tolerances in the range of a few tenths of a millimetre (quality of fit: IT 11–13). Zinc pressure die casting ensures the highest precision; • injection moulding casting (permanent metal mould) for cylindrical gears made from polymers. The shrinkage of the material occurring on cooling has a significant impact on the precision of cylindrical gears in all casting methods. In relation to their technical-economic applicability, primary shaping techniques lead to the substitution of entire process steps (sequences), enabling cylindrical gears to be produced in one 10.2 Tooth-shape Forming Methods 611 main operation cycle, to which, however, production steps concerning the heat treatment appro- priate for the strain and finishing with integrated machining of the toothing modifications are added. From the technical moulding perspective, the principal disadvantages of primary shaping casting are to be found in the less favourable structural properties in relation to the load capacity, as well as the lower precision. Similar precisions, as achieved with the precision casting process, can also be obtained using powder metallurgic techniques. The high productivity obtained using powder metallurgic methods (sintering) – see Table 10.2/2 number 2 – are offset by the disadvantages of the high powder and tool costs incurred as well as the generally lower load capacity of the workpieces, such that cylindrical gears produced by powder metallurgical techniques are primarily used as mass-produced parts with smaller dimensions and weights ranging from 0.1 to 1 kg, maximum lengths of 50 mm, length/diameter ratios < 2.5 and wall thicknesses > 2 mm for low-stress loads. Typical areas of application for sintered cylindrical gears are the household products and electrical industries as well as in office engineering and vehicle construction in association with quantities exceeding 10000. 10.2.3 Primary Shaping and Separating Tooth-shape Forming Whereas the moulding process can always be applied from a primary-shaping perspective due to the amorphous base material [10.2/27], the entire spectrum of generation principles is possible using the principal forming and cutting processes. Alongside the chip-removing processes, the separating and ablating processes are included in the principal cutting methods. For the separating processes , a distinction can be made between • blanking (punching) with a mould insert and forming pin with a cutting gap of 0.2 to 0.3 mm, • precision cutting, preferably with a V-ring, also with a mould insert and forming pin with a cutting gap of approx. 0.01 mm, and • cutting with a laser beam or high-pressure water jet from thin sheet metal with a thickness of about 6 mm. With blanking (punching), the relatively large cutting gap and the lack of position fixing results in a cutting surface which only corresponds to a third of the sheet thickness, while the remaining two-thirds of the sheet thickness is separated by shearing in association with rough scoring. With precision cutting and the prevention of yielding or plastic flow of the material through the V-ring located on the tool in this procedure, the small cutting gap leads to smooth cuts. Precision-cut, face-gearing workpieces are used in the manufacture of office machines, toothed discs and multi-plate clutches. Ablating processes include • chemical or electrochemical ablation, • laser beam ablation, • electro-erosion and wire-cut EDM. The solid forming in association with translational interaction motion allows the manufacture of cylindrical gears with sometimes complicated workpiece geometry from mainly cylindrical blanks (slug sections). Preferred methods include the drawing/moulding processes [10.2/38], [10.2/39], [10.2/42] (see Table 10.2/2 number 3) and the extrusion methods – each as cold (room temperature) and hot working processes (above the recrystallisation temperature) [10.2/15]. 612 10 Manufacturing of Cylindrical Gear Units For the cold drawing [10.2/15] of unalloyed steel pinions, round steel up to a diameter of 25 mm is drawn through a drawing nozzle with the negative profile of the gear. The cutting to the required length as well as the bearing seat machining takes place subsequently, for example on lathes. These processes can be used to manufacture spur pinions in the aforementioned dimensional range with allowance for a few tenths of a millimetre for a mostly chip-removal finish machining. High-volume or mass production is essential for the operational efficiency. The same dimensional ranges of such pinions can also be produced by means of extrusion moulding , which corresponds to cold drawing except for the change in force application and process temperature (above the recrystallization temperature). With hot pressing , it is possible to manufacture cylindrical gears on an individual basis and therefore within the range of mean batch production. In this regard, the powder forging and pre-cision forging method s are particularly important. The powder forging method [10.2/11], [10.2/12], [10.2/40], [10.2/41] is based on the die-moulding process in a closed die, that is, a moulding process, the benefits of which are burr-free forging in association with a high degree of contour accuracy. The disadvantage of this, however, is the precise volume dosing necessary to achieve on one hand a good mould filling and thorough forging, while on the other hand avoiding overloading and potential tool breakage. The necessary requirement for this is provided by the use of sintered preforms, which are sorted into weight categories – each with a maximum 0.5% tolerance – before powder forging, and graphitised to prevent decarburisation in the forging process. Following the heating of the moulding to the forging temperature of approx. 1050 C and the forging die to between 200 and 300 C, the homogenisation and hardening of the material structure ensues in the forging process and subsequently the cooling under inert gas. When designing powder-forged cylindrical gears, it is necessary to comply with rules that essentially apply to moulding processes with a high application of force and result from the mould manufacture and the attainment of a high tool life as well as from the mould filling with the material. Powder-forged cylindrical gears are generally case hardened or quenched and tempered. This and their isotropic mechanical properties make them suitable for high degrees of strain, which is demonstrated, for example, by a higher tooth r oot endurance strength. The precision attainable is first and foremost dependent on the size, form, specificity and strength of the material, as well as the dimensional and weight tolerances of the preform. For high technical production quality of the moulds and optimal process control, it is possible to attain precisions in the 7 to 9 quality range in accordance with ISO 1328 (DIN 3962/63). Precision forging [10.2/13], [10.2/15], [10.2/33]–[10.2/35] (Table 10.2/2 number 4) enables the manufacture on an individual basis of straight-toothed cylindrical gears or beveloid gears (with variable profile shift across the face width) in the die with high accuracy, that is, in the moulding process. Precision forging ensures an accuracy that requires no (or only a low amount of) chip- removal machining. Depending on the size and accuracy of the workpiece and the execution of the process, the required tolerance is 0.05–0.4 (1.2) mm. Straight-toothed cylindrical gears can be manufactured with a diameter range up to 110 mm and cylindrical gears with Konozyl toothing (a conical design) up to 240 mm. It is not possible to manufacture helical cylindrical gears. The precision attainable, generally related to the precision-forged main functional surfaces, is in the IT 8–10 tolerance range, that is, less than 0.2 mm. The precision forging of cylindrical gears ensues in two to three working steps at temperatures of (950) 1150 to 1200 C. The pre-forging, in which approx. 90% of the shape is realised, is preferably undertaken on eccentric bushes, and the final moulding on friction screw presses – in each case in closed dies where all principal functional surfaces are housed in only one side of the die. The forging and cooling must ensue using an inert gas to prevent scale formation and surface decarburisation. 10.2 Tooth-shape Forming Methods 613 Precision forging, however, is only economically applicable for cylindrical gears if they have already been designed considering forging processes, and thus considerable savings in material are possible in comparison with chip-removal finishing, higher load capacity due to the more favourable fibre orientation, optimal curves in the tooth root region or at vulnerable cross-section transitions or the realisation of additional functional surfaces, which cannot (or only with great effort) be machined on the same component [10.2/32] to [10.2/35]. The use of cold extrusion [10.2/16] to [10.2/18], [10.2/22], [10.2/26], [10.2/27] allows the manufacture of solid and hollow parts with internal and external toothing made from bar cuttings or cut-out circular blanks. A die open on one side with positive counter gear teeth is used as a tool. Generally speaking, several working steps involving intermediate treatment, such as recrystallisation annealing, surface treatment, and so on, which enhance the formability, ensue for the purpose of limiting the cold hardening of the basic material depending on the degree of forming. The advantages offered by cold-extrusion workpieces are the uninterrupted fibre orientation with a hardened material as well as a high t ool precision, particularly determined by the precision and elastic deformations of the tool, which can be between ISO (DIN quality) 6 and 12, depending on the toothing geometry and process parameters involved. Depending on the basic material and the available pressing forces of the machines, the high forming forces limit the range of application of this process to a low degree of forming (sma ll workpiece dimensions). The use of cold extrusion particularly allows the manufacture of cylindrical gears with piece weights of 1 to 35 kg with length/diameter ratios < 3 (steel) or < 6–8 (12) (aluminium) up to a maximum length of 600 mm in large-scale production with production volumes > 5000–10000 pieces. The precision generally attainable in the pressing direction (length) is in the IT 15–16 range, transverse to the pressing direction (diameter) in the IT 10–12 range and for centre deviations in the IT 13 range. Because, in contrast to cold forming, no hardening occurs in the hot forming as a result of the constantly superposed re-crystallisation of the material in the forming process, and therefore the permissible material stress applicable cannot be exceeded, hot forming allows virtually unlimited degrees of forming. As far as the process sequence is concerned, hot extrusion pressing corresponds to cold extrusion pressing. Nevertheless, the significantly larger process variations result in allowances in the millimetre range, so in most cases intermediary and (fine) chip-removal finishing is necessary. Depending on the fibre directi on, cylindrical gears produced by pressing have differing fatigue strength values, which can vary up to a factor of two. A favourably acting fibre direction occurs, for example, with the drawing and pressing methods [10.2/4]. At the same time, the more favourable fibre orientation in comparison to chip removal machining also leads to a 20–30% reduced hardening distortion with regard to case hardening. The costs of pressed or extrusion-pressed cylindrical gears are primarily determined by the tool costs. In comparison to chip-removal machining, for which universal tools can be used in many cases, high fixed costs are involved, so economic batch sizes are invariably above 2000–5000 workpieces. Nevertheless, the high material utilisation of approximately 75–85% is an advantage. The high costs of tools and the chip-removal machining that is often required, however, have hitherto stood in the way of a wide application of this technical forming process. In general, its field of application is restricted to those cases where the procedural modifications simu ltaneously allow constructive benefits, such as significant savings in material. In contrast to the moulding processes listed above, which enable the integration of the tooth-shape forming in the manufacture of the basic and extended form, the technical forming rolling process [10.2/19] to [10.2/25], [10.2/28] to [10.2/32], [10.2/43] simultaneously achieves rolling, helical pitch or cross-helical gear principles with a rotary tool and workpiece motion. With this, only a locally limited forming zone with consecutive generating of the tooth form occurs. The benefits are to be found in a generally higher workpiece precision, which for cold-rolling processes is in the ISO (DIN) quality range 6–8, and for hot-rolling processes in the ISO (DIN) quality range 10–12. 614 10 Manufacturing of Cylindrical Gear Units By combining a pre-machining (by hot rolling) and a subsequent finishing (by cold rolling), it is possible to incorporate the benefits of both process variants to a certain degree. The heating of the workpiece for hot rolling preferably ensues inductively up to a heating depth that approximately corresponds to the tooth depth of the toothing to be manufactured. The initial diameter of the blank d Roh is calculated taking into account the cross section parity of the basic material to be displaced in the region of the bottom of the tooth gap and the tooth tip to be generate d. Good results can be achieved with () () Roh a f n1 0.3 to 0.42dd d m =+ + (10.2/1) where da is the tip diameter of the toothing in mm df is the root diameter of the toothing in mm mn is the normal module of the toothing in mm In accordance with [10.2/21], the size of the rollable toothing is for • Hot rolling: module m = 1 to 10 mm, diameter d = 30 to 600 mm • Cold rolling: in discontinuous profile milling: module m < 3 mm, number of teeth z = 6 to 100 in the continuous process (pre- and finishing): module m < 4 mm, diameter d = 20 to 125 mm With cold rolling, a hardening of the material occurs, which results in an increase of the bending strength of the teeth, the surface-layer hardening and thus the wear resistance and fatigue strength of the teeth [10.2/21], [10.2/27], [10.2/28], [10.2/31]. In this regard, cold-rolling processes are used to increase the precision of pre-machined gears in mass production, whereby a desired effect is the increase in strength. With all rolling processes, there remains the problem both of controlling the material flow and of the amount of the rolling forces required, which necessitate an adaptive tool design, all this making these processes only economically viable for very large piece numbers. The rolling-drive principle is realised by transverse rolling processes . These can be classified by the type of gear teeth on the workpiece into • transverse rolling with flat-die (rack) tools, • transverse rolling with circular-die (cylindrical gear) tools, and • transverse rolling with internally profiled (internal cylindrical gear) tools. In all these methods, the workpiece toothing in the tool is specified inclusive of the relative rotation between the tool and the workpiece. For transverse rolling with flat-die tools , a rolling process with rack-shaped tools (preferably used as a cold-rolling process), the workpiece slaving ensues by means of frictional connection. The rack toothing of the tool is performed with the two- to four-fold number of workpiece teeth, whereby their toothing possesses an increasing tooth height (starting from zero) from the running-in side, distributed approximately on a 1.5-fold number of workpiece teeth. This is followed by approx. 12 calibrating teeth. Modifications of the process consist of • a stationary and a movably mounted rolling die, and • two movably mounted rolling dies (ROTO-FLOW process ). Generally speaking, all unalloyed and low-alloyed case-hardening and tempering steels as well as specific types of cast iron, malleable iron and aluminium are rollable. The maximum rollable gearing parameters and – particularly – the maximum module are to a large extent dependent on the material, the hardness and the pressure angle of the gearing. Usually, however, gears are mostly transverse rolled in the module range m < 3 mm. To enhance the cold formability, forged or extrusion-pressed blanks are soft annealed beforehand. For transverse rolling, hardness values between 175 and 235 HB are preferable, since these values provide both a sufficient heat treatability and the yield strength required for a high level of precision. 10.2 Tooth-shape Forming Methods 615 Transverse rolling with externally toothed circular dies (Figure 10.2/2), a rolling process with cylindri- cal-gear-shaped tools, is performed as cold rolling for small modules – or, in rare cases, in association with sufficient production quantities as hot rolling for larger modules – in the process variants: • with two axially parallel tools, which are driven in through-feed, infeed and tangential processes, or work on an idling basis; • with three tools, each coaxially driven and arranged at an angle of 120 degrees with the workpiece. Generally speaking, the heating and rolling ensue directly one after the other on the rolling machine. The process parameters can most effectively be influenced for this rolling process, thus enabling the highest qualities to be achieved when rolling (Figure 10.2/3). Due to the generating principle, however, high qualities can only be achieved for helical gearings having a sufficient contact ratio. On the basis of the material flow possible in the circumferential direction, different pressure angles on the left and right flanks are possible, particularly with low contact ratios. In addition, this results in tooth tip radii of slightly differing sizes, which in unfavourable cases increase the allowance for subsequent machining. Transverse rolling with internally toothed tools , also known as the WPM process , belongs to the same generating principle as that with externally toothed tools. The two tools arranged coaxially to the workpiece and driven by eccentric shafts form the toothing along the entire circumference in the course of a single stroke. In this process, the material flow is only controllable to a limited extent, and the rolling forces are so high as to result in less precision. Figure 10.2/3 Gear tooth quality in accordance with ISO 1328 (DIN 3962/63) based on cold rolling with circular dies: a) cumulative pitch deviation Fp; b) tooth-to-tooth pitch deviation fu; c) runout Fr [10.2/24] Figure 10.2/2 Transverse rolling with externally toothed circular dies as per the rolling principle [10.2/19]: a) process principle, b) process sequence, c) schematic representation of the forming process 616 10 Manufacturing of Cylindrical Gear Units Cold rolling has attained a significant importance in discontinuous profile rolling based on the helical principle, also known as the GROB process; see Figure 10.2/4 (Table 10.2/2 number 5). Figure 10.2/4 Axial rolling with forming tools (GROB process) [10.2/103] In accordance with the generating principle, profile tools in the form of two opposing, eccentrically arranged profile rolls are used. They form the toothing in combination with the axial workpiece feed motion and a planetary tool motion together with a workpiece rotary motion mostly permanently coupled with the gear ratio. As a consequence of this process principle, accuracies in the ISO (DIN) qualities 6 to 8 range are attained. Low tool costs, in combination with high tool lifetimes, result in tool-related costs in the order of a few cents, thus making this process economical even for medium-series production. Toothing devia-tions can be simply compensated for in tool manu-facture (or reworking). The axial flow of the material results in considerable burr formation on the face sides, which is simple to eliminate. 10.2.4 Chip-removal Tooth-shape Forming Using Tools with Geometrically Defined Blades 10.2.4.1 Technological Basics The requirement for chip-removal tooth-shape forming is a relative motion between the workpiece and a V-shaped and wear-resistant tool penetrating into the material. This relative motion consists of the cutting movement, which would result in a one-off chip removal, and the feed motion. In this process, either a repeated or continuous chip removal is achieved. The cutting cross section of a tool blade into the material in the plane, on which the cutting velocity vector ȣ c is in a vertical position (refer to Figure 10.2/8), is referred to as the chipping cross section A , its dimension in the cutting plane as the chipping width b and in the plane perpendi- cular to the cutting plane as the chipping thickness h. The product of the chipping cross section and cutting speed represents the chip-removal rate Q w in mm3/s, the quantity of which represents the productivity of the process. As a first approximation, this chip-removal rate is proportional to the effective chip-removal power P z and chipping force Fz, whose components in the cutting direction are referred to as the cutting force Fs and – perpendicular to it – as the passive force Fp. This passive force, in turn, is resolved into the feeding force Fv and the radial force Fr (Figure 10.2/5). While the cutting force is the decisive factor for the power of the tool drive, as a result of the working direction oriented to the cutting direction as per (10.2/2) where Ps = cutting power Fs = f (chip cross section, cutting-blade geometry, material, and so on), SS CPFυ=Figure 10.2/5 Blade forces 10.2 Tooth-shape Forming Methods 617 the other force components are of particular interest for the purpose of determining the load perpendicular to the workpiece surface causing the effective process deformations and deviations. The friction between the tool blade and the workpiece material leads to wear marks on the cutting tool blade. These marks have the width VB on the blade and show a scouring on the cutting face of the tool, along which the separated chip runs off (Figure 10.2/6). The reasons for these types of wear are the shearing off of pressure-welding particles, mechanical abrasion, scaling and diffusion processes. This order also represents the degree of occurrence with increasing cutting speed and thus cutting temperature (Figure 10.2/7). Figure 10.2/6 Forms of wear on the hob tooth Figure 10.2/7 Causes of wear Figure 10.2/8 Manufacturing of toothing by chip removal in rolling processes [10.2/2]: a) process principle; b) tool-workpiece rolling kinematics; c) chipping geometry in rolling processes 618 10 Manufacturing of Cylindrical Gear Units In the past, cutting materials have been developed that ever better satisfy the requirements of the chip-removing process and even allow the machining of case-hardened materials with a geometrically defined blade. This requires super-hard materials, which enable the generation of extremely small cutting edge radii in order to keep the relationship between the cutting edge radius and the minimum chipping thickness h min, which is about 0.01 mm for chip-removing processes with geometrically defined blades, within the technically acceptable limits of chip removal, thus avoiding excessive chip-removal forces. The following cutting materials are of importance in gear technology: • High-speed steels (also high-performance rapid steel [HRS]). These are high-alloy steels with hardnesses from 60 to 67 HRC and a tempering resistance up to approx. 600 C. The tempering resistance, hardness and wear resistance are determined by the martensite and the special carbides embedded (particularly Mo-W double carbides, Cr and V carbides). In contrast to the aforementioned high-speed steels that are manufactured using melt-metallurgical techniques, PM-HRS cutting materials are produced using powder metallurgic processes. These materials generally have a higher alloy content, while their fineness of grain, high degree of purity, the uniform distribution of the carbides as well as the lack of segregation provide further benefits. The ductility of the material resulting from the high degree of purity makes it particularly suitable for gear technology applications with geometrically complicated tools subjected to abrupt loads, such as hobs, shaper cutters and shaving cutters. • Hard metals are sinter materials and consist of special carbides (WC, VC, TiC, TaC and NbC) embedded in a bonding phase. Compared to high-speed steels, the hard metals offer a higher degree of hardness and thermal stability thanks to their high hard-material content (70–95%). Although from a technical chip-removal perspective they are beneficial at cutting temperatures up to 1000 C, these benefits can be reduced due to the risk of diffusion and the low bending strength. Attempts have been made to compensate for these disadvantages by the use of smaller grain sizes, particularly for gear cutting tools. Hard metals containing ultra-fine grains (grain size from 0.4 to 0.5 m) have a strength some 30% higher than the standard grain size (0.9 to 1.4 m). As a consequence of their limited machinability compared to high-speed steels, they are particu-larly used in the manufacture of inserts, soldered or clamped into the main tool body, with two to four blades on the edges. When using inserts, the maximum four possible blades can be used in succession by turning the cutting inserts on the main body, provided that this is not prevented by geometrical reasons, such as an unsymmetrical blade design in association with modified gear tooth profiles (Figure 10.2/9). In this regard, similar tools (such as hobs) also differ as to whether they are designed with high-speed steels or hard-metal steels. In a similar manner to high-speed steels, the wear resistance of hard metals can be improved with a surface coating applied by means of the CVD technique, due to the lack of structural transformation. Figure 10.2/9 HM cutting inserts on hobs [10.2/106] 10.2 Tooth-shape Forming Methods 619 • Polycrystalline boron nitride (PKB) . After diamond, this is the hardest available material (cubic boron nitride) and is used at the present time as a cutting insert in a limited dimensional range or on hard-metal supporting plates. Despite its extremely high impact sensitivity, this cutting material can be used for the machining of hardened steels in finishing conditions. The machinability depends largely on the pairing of cutting material, basic material, and auxiliary substance. As a result of the only limited influence on the chipping force, the machinability is mainly measured by means of the wear characteristics. It is recommended to use the cutting speed as the control variable because this represents the parameter with the highest dependence. In this regard, its size is corrected in the process design by the value of the relative machinability. The production accuracy for chip-removal processes is characterised by the following principles [10.2/44], [10.2/45]: • The generation of the toothing ensues by means of discrete cutting blades and individual cuts, whereby the theoretical flank surfaces are approximated by a succession of individual lengths (per cut) of the geometric tool blade-flank surface pair engaged. Depending on the position of the main feed motion (the principal component of the complex feed motion), this results in process-related profile deviations generated when approximating the profile (main feed motion in the flank direction, e.g., in shaping) or infeed meshing marks in the flank direction (principal component in the invo lute direction, e.g., in hobb ing) (Figure 10.2/10). • For rolling, helical pitch, and general screw rolling processes, the rolling kinematics require a mechanical generation via an appropriate drive that – in the case of its electronic design – contains mechanical links, which generate an inevitable profile form deviation occurring as waviness in the profile. Their wave number corresponds to the ratio of the length of the path of contact to the base pitch multiplied by the relation derived from the interference frequency and the tool rotational frequency. Figure 10.2/10 Flank area with process-related geometrical deviations [10.1/1]: a) diagram of process-related profile form deviations ffav as a result of profile deviations generated when hobbing; b) process-related flank surface form deviations ff v of hobbed cylindrical gears as a result of axial feed marking; c) flank shape of hobbed and shaper-cut cylindrical gears 10.2 Tooth-shape Forming Methods 620 Figure 10.2/11 The influence of geometrical deviations of the tool on the toothing accuracy for a hob with wrongly negative cutting angle on the tip blades • The tool with its functional surfaces deviating in flank and tip clearance angle, as well as its flank and tip rake angle from the basic rack in the face width direction, has the design profile in one plane only. Inaccurate positions of the free surfaces or cutting faces thus lead to differences between the actual and nominal cutting position, which result in deviations of the workpiece toothing (for example, refer to Figure 10.2/11). • The tool wear in the form of wear marks leads directly to an offset of the actual cutting position compared to the nominal cutting position and – in the form of crater wear via the orthogonal clearance angle – leads directly to a change in the cutting position compared to its rol-ling or nominal position required based on the helical principle. So both forms of wear cause wear-related deviations of the toothing, which, in turn, may lead to different deviations depending on the type, position and uniformity of the wear formation. • Under the effect of the chipping force associated with the chipping process and its component perpendicular to the surface, tool-fixture-workpiece deformations arise due to the finite stiffness of the tool machine system, which lead to systemic deviations depending on deformation size and uniformity along the line of contact (example: skiving). • Along with those deviations caused by chipping-related technical reasons, the generally applicable principles relating to the influence of the precision of the tool, its clamping, the datum surfaces and the accuracy of the workpiece clamping also have an effect, leading not only to runout and cumulative pitch deviations but also to a wobble of the profile and of the tooth trace on the workpiece circumference (Figure 10.2/12). Figure 10.2/12 Workpiece eccentricity as per [10.2/1]: a) clamping deviations of a wheel; b) effects of an eccentricity e on profile, tooth trace and pitch • Chipping processes are based on a discontinuous (indexing after cutting of a tooth space) or continuous (toothing pitch stored in the tool) process principle. Whereas with the single-pitch process the indexing accuracy of the tool machine determines the accuracy of the circular 10.2 Tooth-shape Forming Methods 621 pitch, with continuous processes it is prima- rily dependent on the tool-pitch accuracy. Multiple-start, worm-shaped gear cutting tools in particular (e.g., hobs) exhibit greater devia-tions, which in the event of integer divisibility of the number of teeth of the workpiece by the number of starts result in pitch and profile form deviations (Figure 10.2/13). Figure 10.2/13 Profile form deviation of the tooth flank with multiple-start hobbing 10.2.4.2 Plane by the Generating Method Generating planing (Table 10.2/2 number 6), from its technical process and gear principles involved – namely the plane/shaping process and the rack-cylindrical gear rolling principle – represents a chipping process with simple process-engineering geometry and kinematics as well as manageable principles. It is therefore not surprising that this process was developed as far back as 1913 by the Swiss MAAG Company and has proven its worth ove r many decades, particularly in the manufacture of large gear units in the form of single-unit production (Table 10.2/2 number 6). The constant further refinement and increasingly better mastering of the hobbing process, including the development of ever more complicated – but also more powerful – hobbing tools, has resulted in less importance of generating planing. Generating planing is suitable not only for spur and helical gears but also for chain wheels, splined shafts as well as other machine elements with profiles that can be rolled. It represents an universal process, albeit with a lower productivity, suitable also for gears with small space for tool overrun. Figure 10.2/14 Generating planing process principle [10.2/103] The reason for the low productivity is the fact that the return stroke of the rack tool has to be performed as an idle cut, and the generation of the stroke sets mechanical engineering limits with regard to the stroke rates attainable. It is, of course, possible to work with generating process machines by not including the rolling motion in the profiling process, thus also allowing the manufacture of internally toothed gears. 622 10 Manufacturing of Cylindrical Gear Units The generating planing process principle corresponds to the rolling-off of a cylindrical gear on a rack, whereby the rack is embodied by the tool geometry in the cutting face, including the cutting motion (Figure 10.2/14). The cutting motion is carried out in the form of a stroke by the rack-type cutter (several planing teeth in parallel) or by the single tooth planing tool, so generating planing represents a shaping process in the technical chipping sense as a result of the moved tool. The teeth of the rack-type tool are relieved at the flanks to achieve the clearing angle necessary from a chipping technical perspective (to avoid friction on the flanks) and to contain a grinding of the cutting face on the face, which can be identical with the cutting face, in order to achieve a favourable chip removal. As this cutting face grinding results in geometrical deviations from the nominal face of the toothing, an appropriate correction of the tool profile, which otherwise corresponds to the basic rack, is necessary, taking into account the angles at the planing tooth [10.2/50]–[10.2/52]. The relatively uniform chipping force and the diverse possibilities to influence its size and progress along the rolling path are what makes it possible to achieve a relatively high degree of precision using the generating planing process, attaining a quality of 5 (4) in accordance with ISO 1328 (DIN 3962/63). 10.2.4.3 Gear Shaping Gear shaping (Table 10.2/2, number 7) is a continuous gear cutting process that operates accor- ding to the generating principle. Consequently, in the case of cylindrical gearing, the axes of the workpiece and the tool are coaxial in space. A spur or helical shaper cutter is used as the tool, the teeth of which are relieved in order to obtain the relief angle at the flanks and the tooth tip that is required for the chipping process [10.2/53]. During the chipping process, the tool and workpiece intermesh analogously to the meshing in the transmission, which results in a combined feed motion for the purpose of chipping. Figure 10.2/15 Process geometry and kinematics of gear shaping (Note: ram stroke/shaping amplitude is in the z direction) Due to the relief grinding of the shaper cutter teeth and in order to produce the cutting motion, the shaper cutter performs a stroke motion, which is superimposed on a rotary motion in the case of helical gears (Figure 10.2/15). The stroke motion in the z direction in combination with the rotary motion is performed by the ram of the machine, which contains the so-called lead cartridge for realizing the helical motion. This guide cartridge, which is interchangeably mounted in the machine, has a helical guide groove of a constant lead H for the tappet. The combined feed motion resulting from the rolling motion is generated through the kinematic coupling of the rotary motion of the workpiece and shaper cutter. 10.2 Tooth-shape Forming Methods 623 Conventional gear shaping ma- chines are equipped with a mecha-nical transmission, in which the coupling of the generating motion between the shaper cutter and workpiece is produced by change gears. Modern gear shaping ma-chines use CNC controls, which not only eliminate the costly exchange of the change gears but, due to the performance of the computerised control systems, enable multiple tasks to be carried out, thus increa-sing the flexibility and efficiency of these shaping machines. The ram motion is generally gene-rated via a crank drive so that the cutting motion is variable due to the adjustable stroke length. The disadvantage of the non-uniform speed along the stroke length of the mechanical crank drive can be avoided with an additional crank drive. With this design solution, the operating speed is reduced to about 70% of the maximum speed, but it is kept constant over almost the entire stroke length while the maximum return stroke speed is increased by approximately 40% (Figure 10.2/16). The required stroke length results from the face width and the starting and overrun path of the blades of the shaper cutter on the workpiece face. For spur workpieces, the stroke length 1 h is (10.2/3) and for shaper cutters with step cut for helical workpieces () hn 0.6 to1.0 sin ȕ+cosȕ 14bblm=+ ⋅ (10.2/4) with b face width ȕ helix angle mn normal module Compared to the other manufacturing methods of gears, shaping requires a small distance to the adjacent flanges. This property and the generation principle of gear shaping resulted in the typical application areas of the method (Table 10.2/2, number 7.): • internal straight cylindrical and helical gears, • external straight and helical cylindrical gears, while ensuring a low tool overrun (such as block wheels and switching shafts in vehicle transmissions), • racks, • special gearing such as tooth segments, crown wheels, gear hubs, spline shafts, sprocket wheels, among others. h1.14lb=⋅Figure 10.2/16 Gear shaping: a) normal stroke drive, b) drive for constant cutting speed; a = working stroke, b = face width of wheel, c = return stroke, lh = stroke length, ȣA = tool speed in working stroke, ȣAmax = max. tool speed in working stroke, ȣRmax = max. return stroke speed 624 10 Manufacturing of Cylindrical Gear Units Thus, gear shaping is the cutting method most com- monly used after hobbing. The design of the shaper cutter depends on the workpiece geometry and the cutter arbor of the spindle of the gear shaper. In principle, distinctions are made between (DIN 1825, 1826, 1828, 1829; refer to Annex 15.1) • disc shaper cutters for external gearing and large internal gearing with sufficient tool runout for the fastening nut, • bell-type shaper cutters for external and large internal gearing, where there is a risk of collision of the workpiece and the fastening nut of the shaper cutter, and • shank shaper cutters with tapered shank for internal gearing with a small reference diameter. Shaper cutters as solid shaper cutters are commonplace, where high-speed steel dominates as the cutting material. In order to improve the cutting ability and tool life (total cutting time between two required regrindings), shaper cutters are surface treated and PVD coated in particular. Regrinding is done on the cutting face (facial normal plane of the tool) and causes on one hand a reduction of the usable length of the tool and so by the relieving of teeth to a change of the profile shift, and on the other hand to a change of the edge-holding capacity of the shaper cutter. Other than on the machinability of the workpiece ma- terial, the applicable cutting parameters mainly depend on, among others, the parameters of the chipping process, such as the pairing of the cutting material, workpiece material, and auxiliary substance, and the geometric intersection per stroke between the shaper cutter and workpiece. That means the wear-tool life performance of the shaper cutter is decisive for the productivity of the process. In this respect, the process design is subject to continuous development [10.2/54] to [10.2/62]. In the case of gear shaping, there is a disadvantageous machining process at the tip edge to the trailing pinion cutter flank because the smaller chipping thicknesses at the trailing pinion cutter flank lead to material hold-up and consequently to increased wear. Special improvements in the design of the cutting process could be achieved by computer-aided control of the machining process. A continuous but degressive infeed while rolling several revolutions of the workpiece at a high generating feed allows for a remarkable increase in performance when the requirements of high rigidity of clamping of workpiece, tool, and machine tool are met. Another increase of machining power is possible by implementing the controlled cutting process (CCP). This method allows for the design of optimized machining data (number of strokes, radial infeed, generating feed), while a more uniform chip is formed from leading to trailing flank. The chip formation is controlled and will increase cutter life (Figure 10.2/17). The CCP requires high rigidity and dynamic load capacity of the machine. At the same time, the more advantageous cutting conditions lead to improved toothing accuracy, which is limited in the case of conventional cutting conditions, particularly due to the influence of the springback between tool and workpiece during the completion of a cut (revolution). Springback is caused as a result of the roll-out of the shaper cutter in the roll-in area, and the resulting reduction in cutting force. When using the CCP method in Figure 10.2/17 Co mparison of chip cross sections in the conventional and the CCP method; h sp = radial infeed per workpiece revolution; isp = number of workpiece revolutions 10.2 Tooth-shape Forming Methods 625 conjunction with shaper cutters coated in titanium nitride, it is possible to achieve the quality 6 to 7 according to ISO 1328 (DIN 3962) while maintaining a surface roughness of Rz < 10 m. This has particular importance mainly in the manufacturing of internal gearing with subsequent nitriding, because finishing in order to improve the accuracy or the surface quality can be very costly. 10.2.4.4 Skiving Skiving corresponds to the pairing of shaper cutter and workpiece, but both perform a continuous rotary motion, and the tool also executes a continuous feed motion in the axial direction (Table 10.2/2, number 8). It is a working cutting method by chip removal according to the helical pitch principle that was invented at the begin-ning of the last century (Figure 10.2/18). However, it has only been ready for ap-plication in recent decades because there are extraordinary requirements for the accuracy and rigidity of the machine and tool, including the clamping [10.2/64]. Today, skiving is a highly productive method for the machining of external and internal gearing. As a special variant of the method, the method in the form of hard skiving is also suitable for the machining of case-hardened gears with small dimensions. The sum of the helix angle of the tool ( 0) and workpiece ( 2) are in the range 0 Ș = Ȉ0 = Ňȕ0 + ȕ2Ň 90 (10.2/5) Prefix rule: External gearing: right-handed + left-handed – Internal gearing: right-handed + left-handed – Thus, the cutting edges of the skiving wheel also move in the lead direction of the tooth so that chip removal can be carried out. Consequently, the components of the relative motion between tool and workpiece resulting from the method principle, in the form of the feed motion (1) and the additional differential rotation (2) (Figure 10.2/18), are in fixed proportion to each other. Only the axial feed motion results as an independently selectable motion component. The skiving wheel corresponds to a spur or helical gear with a cylindrical or conical outer contour, whereby the conical outer contour has the required relief angle for the chipping process. In order to achieve an effective clearance angle, the cylindrical skiving wheels must have a cutting face offset from the axis-crossing point. Generally, skiving wheels are executed in a step cut analogous to the shaper cutters of the gear-shaping method in order to produce cutting-edge geometries that are advantageous for the chipping process. The cutting face lead angle generally corresponds to the helix angle of the skiving wheel ȕ 0. With regard to the rolling kinematics, because the individual points of the cutting edges do not meet the meshing conditions for a deviation-free rolling action, corrections of the involute geometry of the skiving wheel are inevitable. The achievable accuracy Figure 10.2/18 Skiving 626 10 Manufacturing of Cylindrical Gear Units when skiving non-hardened gearing is in the quality range 7 to 8 in accordance with ISO 1328 (DIN 3962/63) and at quality 6 for hard skiving with carbide skiving wheels. 10.2.4.5 Hobbing Hobbing (Table 10.2/2 number 9), a gear rough-machining and gear-finishing process of external and internal gears as well as other rollable profiles, invented in 1897 by Pfauter in Chemnitz based on preliminary work by Schiele (1856) and Grant (1887), represents the predominant process for manufacturing cylindrical gears, being suitable both for the highly productive pre-machining and finishing of non-har-dened gears and – in its special form of skiving – for the finishing of case-hard-ened gears (Figure 10.2/19). It encom-passes the dimensional range of gears from a few millimetres up to ten metres and a range of moduli from m = 0.1 to 50 mm. Its advantages include not only a high degree of flexibility but also high productivity, while past experience has shown this process to be continually ev-olvable. For the manufacture of cylindrical gearing units, this process principle corresponds to the hobbing chipping method and the helical pitch principle (Figure 10.2/1b). In this process, the tool is embodied by a cylindrical worm, whose normal section corresponds to a basic rack which is moved in the direction of the tooth by means of a component of the feed motion, so that the counterpart rack is enveloped from the basic rack of the hob and the feed motion component. The swivel angle of the hob Ș 0 is derived according to the generating principle from the geometry of the tool and the workpiece and is given by (10.2/6) where ȕ is the helix angle, Ȗ0 is the lead angle of the hob; lower sign (–) with the same lead direction, upper sign (+) with the opposite lead direction of tool and workpiece Figure 10.2/20 Method according to Grant Method according to Pfauter 0 0 γ β= η Figure 10.2/19 Hobbing a cylindrical gear 10.2 Tooth-shape Forming Methods 627 The chipping-technical relative motion is determined from the kinematics of the process, whereby the rotary motion of the hob represents the cutting motion. The feed motion is composed of several components. The rolling feed motion ȣ fw is produced by the rotary motion of the workpiece toothing (permanently coupled with the rotary motion of the hob) in conjunction with the motion of the tool basic rack tangential to the toothing as a result of the lead of the cutter worm, whereby the tangential feed rate ȣ ft (in the Y direction) is proportional to the pitch, the number of starts of the hob (cutter), and the number of revolutions of the hob. The axial feed components v fa required to roll out the entire face width (in the Z direction) can be determined via the axial feed sa and is given by (10.2/7) where sa is the axial feed, n0 is the tool rotary speed, z0 is the number of threads of the tool, z is the number of workpiece teeth With helical gears, the lead direction for hobbing according to Grant is produced by shifting the hob parallel to the tooth direction of the gear. As in this case where the workpiece does not perform any additional rotation, no differential is required from a machine-technical perspective. For differential hob-bing according to Pfauter, the axial feed motion (1) is superposed by the helical motion (2) (Figure 10.2/20). Different hobbing variants can be distinguished depending on the implementation of the feed motion, as follows: • Axial hobbing for radial feed outside the gearing • Radial-axial hobbing as a combined plunge-axial feed process • Diagonal hobbing with simultaneous axial and tangential hob shift A distinction can be made between the hobbing process alternatives climb hobbing (cut takes place on the outer diameter with larger chipping thickness) and conventional milling (cutting starts on the root diameter with zero chipping thickness) (Figure 10 .2/21). Climb hobbing allows for a higher performance in association with a higher machine load and lower surface quality, while with conventional milling the load on the machine is reduced and a better surface quality can be achieved. Figure 10.2/22 A hob representing a generating worm Figure 10.2/23 Tooth segment of an insert hob [10.2/106] 0 fa a 0zsnzυ= Figure 10.2/21 Process variants: a) conventional up-cut milling; b) climb hobbing 628 10 Manufacturing of Cylindrical Gear Units For this reason, hobbing for the purpose of achieving long tool life and good workpiece precision is commonly performed in two stages: the roughing ensues in the climb milling process and the finishing in the conventional hobbing process. For helical gears, there are other process variants, like uni-directional and contra-directional hobbing, in which the hand of the lead of the tool and workpiece are in the same direction or opposed. The hob represents a cylindrical worm, which is interrupted by chip grooves, so that the number of blades per pitch on the hob corresponds to the number of chip grooves (Figure 10.2/22). The interrupted worm thread forms the cutting edges between the chip grooves. The number of threads of the worm determines the number of starts of the hob (cutter) z 0. The hob teeth are designed so that favourable chipping technical angles are created. The hob is relieved or backed off in analogy with shaper cutters to generate the clearing angle on the flanks and the tooth tips. The position of the cutting face formed by the chip grooves determines the cutting angle appropriate for the chip-removal process. In this regard, hobs are designed as straight and inclined grooved, possessing tip rake angles either equal to (or deviating from) zero depending on the position of the cutting face to the axial cross section of the hob. As the hob represents the basic rack, the mating geometry (e.g., the tool/workpiece generating angle Į w0) is not changed when re-sharpening the cutting face. Hobs are usually manufactured from high-speed steel (uncoated and coated), but also increasingly with carbide inserts • in a compact design as block hobs, • with inserted cutting parts as blade-guide hobs and spline hobs, and • as carbide insert hobs ( m 5 mm) (Figure 10.2/9 and Figure 10.2/23). Hob basic racks are standardised in DIN 3972 for various application scenarios such as finish hobbing, roughing (for shaving) or tooth-flank grinding and differ in their tool addenda. With basic rack I (DIN 3972) for the finish hobbing of rough-machined workpieces (e.g., skiving), the addendum coefficient is set to the value 1.167 m, and with basic rack II for finishing to 1.25 m, and for basic racks III and IV to (10.2/8) where q is the machining allowance per flank and is calculated from the cube root of the normal module m multiplied by a factor. Tools for the pre-machining of highly stressed cylindrical gears and large gearings for grinding are commonly designed differently from the standardised basic racks with protuberances for the purpose of generating an undercut. In large-scale production, profiles are often used to also generate a tooth tip chamfer to prevent damage to the tooth tip edges. Occasionally, a modi-fication of the tooth profile in the form of a circular or parabola-shaped deviation from the standard profile is useful to achieve an optimal distribution of the allowance along the profile for the subsequent finishing. These requirements, however, are associated with a restriction in the application of the tool with regard to the number of teeth or the diameter and helix angle. The necessity of such modifications should be carefully checked in the design phase, taking the high hob costs into account. Hobs for the manufacture of large internally toothed gears (generally m > 8 mm, d > 750 mm) have special profiles for kinematic reasons: each hob blade has the theoretical position required (from a rolling kinematic perspective) only in the zone of contact with the workpiece flank to be generated, while all other blade areas are recessed and thus do not cut into active workpiece flanks. A special feature of hobs for internally toothed gears is the setting-up tooth E, which is 0aP0sin25 . 1αqm h + ⋅ = 10.2 Tooth-shape Forming Methods 629 necessary to align the hob to the centre of the machine M in the Y direction. This centre position enables the symmetrical formation of the lines of action of the right and left flanks. These tools should only be used for internal gears of the same module and a range of teeth where the profile shift coefficient is assigned to each number of teeth. Figure 10.2/24 Hobs special profiles: a) prot uberance profiles; b) tip chamfer profile: dFa - form diameter of addendum, dFfV - form diameter of dedendum, FStV - root undercut in the transverse section, hF f P0 - utilized dedendum, qt - machining allowance, ĮKP0 - tip chamfer angle, Įpr P0 - protuberance angle A high tool precision is of great importance for the workpiece quality (see Section 10.2.4.1). In accordance with DIN 3968, hobs are divided into five quality classes (A, B, C, D and the special class AA). The precision of the tool is changed during the life cycle by the repeated sharpening of the cutting faces. Here the following measured quantities should be specified: • Form and position deviation of the cutting faces • Single pitch of the chip grooves • Cumulative pitch of the chip grooves • Direction of the chip grooves It should be taken into account that the quality achievable on the workpiece is affected not only by the deviations on the tool but also by positional errors of the hob and workpiece, as well as the internal forces. Tools of AA quality class enable quality grades of 7 to 8 in accordance with ISO 1328 (DIN 3962), and B quality class tools enable quality grades of 10 to 11 [10.2/106]. For more information on processes and tools, refer to [10.2/63] to [10.2/72]. 10.2.4.6 Form Cutting In the case of form cutting with disk-type milling cutters or end mills, the tooth spaces are milled individually by form-cutting tools. The oldest method for manufacturing gearing was employed 630 10 Manufacturing of Cylindrical Gear Units before the introduction of the hobbing method. For spur and helical gears with a small helix angle, the axial cross section of the hob largely corresponds to the tooth space profile in normal section. Form-cutting tools are used in the following ways (Figure 10.2/25): Figure 10.2/25 Methods of form cutting • End mills made of high-speed steel, which can be reground when worn and the profile of which matches the tooth space profile through the corresponding dressing. Due to their small diameter, which is determined by the tooth space geometry, they wear quickly. Their low tool life and their low rigidity make them the least productive of the form-cutting tools. • Side-milling cutters made of high-speed steel with a ground profile, which can also be re- ground on the cutting face when worn. Their larger diameter and the far more rigid tool holder on milling arbors, which is comparable to hobbing, allow for higher productivity while obtaining the largely exact flank surface. In the case of helical gearing, the pre-offset and post-offset of the chip formation zones compared to the axial cross section profile of the hob located in the normal plane of the gearing lead to an undercutting of the theoretical flank surfaces on the left and right flanks, which must be taken into account in the tool profile calculation. Modern calculation methods allow an accurate simulation of the contact conditions between the tool and the workpiece, thus enabling the optimisation of the cutting conditions. • Side-milling cutters with carbide (cutting) inserts as a more efficient version of the form side-milling cutter. They have either a matching profile, which enables the use of continuous carbide inserts for the tool profile, or they approximate the gear tooth profile with a few generated cuts so that several carbide inserts with different profile angles are arranged on the tool profile height. They are the most efficient type of form-cutting tools, although they also have the same disadvantages as side-milling cutters, but these are of minor significance due to their prevalent use as roughing cutters. Profile cutting with carbide side-milling cutters is important for the roughing of large-module external and especially internal gearing with a preferably large face width for subsequent finishing with tooth grinding or shaving. An economic comparison between the method of pre-machining through form cutting with finishing in the generating process and the method of complete machining in the generating pro-cess is used to decide whether form cutting would be useful, also taking into account all additional costs and time, such as the additional tool and the tool change. Hobbing will normally be the economical solution even in the case of large external gearing, so the field of application of form cutting is mainly restricted to the pre-machining of large internal gearing. Profile cutting is generally carried out on gear hobbers while deactivating the rolling motion, as long as the hobbers are suitable for the setup of these tools with large diameters (approx. 400 to 450 mm). 10.2 Tooth-shape Forming Methods 631 10.2.4.7 Gear Shaving Gear shaving (Table 10.2/2 number 10) is a chip- ping process, designed for the finishing of mostly non-hardened external and internal gears, which have been cut in pre-machining, to improve both the toothing accuracy and the surface quality [10.2/72], [10.2/75], [10.2/79]. Its application ranges from precision engineering to heavy engineering for workpiece diameters from 12 to 6000 mm, face widths from 6 to 500 mm and modules from 0.6 to 20 mm under the conditions from single piece through mass production (Table 10.2/2 number 10). As gear shaving is not suitable for the finishing of hardened gears on a commercial basis, a subsequent case hardening or nitriding is generally required in modern gear manufacturing, which as a result of the distortions determines not only the geometrical precision and surface quality but also the material-technical properties in the surface layer. Due to the high requirements on the precision of the gears, the use of shaved gears is essentially limited to quenched and tempered (or nitrided) cylindrical gears or – in the case of case hardening – to low component dimensions, whereby the heat treatment (nitriding or case hardening) mostly takes place after shaving. In the other cases, after-hardening processes with kinematic enforced movement are most often used. Tooth-flank shaving is a helical thread rolling process: the tool and workpiece are arranged with crossed axes with axis-crossing angles of Ȉ 0 = ȕ0 + ȕ2 90 , whereby there is no kinematic enforced movement between the shaving cutter and the workpiece (Figure 10.2/26). The generally driving larger wheel of the pairing of shaving cutter and workpiece takes the mating cylindrical gear along with it. The helical thread rolling principle leads to a point contact of the tooth flanks of both cylindrical gears, which, however, spreads out into a zone of contact as a result of the penetration of the tool flanks into the workpiece flanks. Kinematically, this helical thread rolling principle results in • a sliding movement in the tooth-tip direction ȣ cr and • a sliding movement in the face width direction ȣca, which, added together, give a resulting speed of ȣc. This speed changes, depending on the different sizes of both components, their quantity, and the direction on the workpiece flank. The result is – in analogy to gear skiving – an arch-shaped contact trace across the workpiece flank (Figure 10.2/27) [10.2/72] to [10.2/74], [10.2/79]. In order to machine the workpiece across its entire width, it is necessary to shift this contact trace across the face width of the workpiece, which leads to the gear shaving process variants according to the direction of this feed motion: • For parallel shaving , the workpiece is axially shifted across its entire face width. Although this results in a large feeding path, it is nevertheless possible to shave gears of any given width. Figure 10.2/27 Sliding move- ments when shaving Figure 10.2/26 Tool and workpiece assignment for gear shaving [10.2/103] 632 10 Manufacturing of Cylindrical Gear Units • For diagonal shaving , the feed ensues with the crossed axes angle, resulting in a shorter feeding path and a lower machining time. If the crossed axes angle į = 90 , then we have what is known as transverse shaving . The maximum workpiece face width results from the maximum shaving gear width of about 50 mm, which means that a face width up to about 45 mm can be shaved. • For plunge-cut shaving , only a feed motion is executed. This process variant, however, is subject to the condition that the shaving cutter width b corresponds at least to the workpiece face width. In this regard, it is possible to machine face widths up to approx. 40 mm. The components of motion relating to the shaving cutter represent the motion required from a chipping kinematic perspective. These components are • in the face width direction, the combined rolling and sliding motion of the shaving cutter (the axial cutting speed ȣca); • in the tooth-tip direction of the shaving cutter, the radial cutting speed ȣcr; • in the face width direction, the shift (depending on the process variant) as longitudinal ( į = 0 ), diagonal (0 < į < 90 ) or transverse feed ( į = 90 ); • radial to the workpiece, the feed motion with the feed rate ȣfr. The variant-dependent minimum distances of the gear to the side-restricted surfaces, such as shaft shoulders, which can be very small for diagonal shaving and particularly for plunge-cut shaving, result both from the feeding path of the process variant in question and the geometric conditions. The precision achievable with shaving is essentially determined in the design phase since the meshing geometry between the shaving cutter and the workpiece, as well as – via the workpiece form – the degree of distortion, is influenced in the course of the case hardening that commonly takes place subsequently. For the prevailing double-flank shaving, the shaving cutter simultaneously engages the workpiece at several right- and left-hand flanks. A chipping force forms at each meshing point, which can be divided into a radial component, deforming the tool and workpiece clamping, and a tangential component [10.2/72]–[10.2/74]. The resulting tangential force arising from the opposing tangential components of the chipping force on the right- and left-hand flanks leads to dynamic meshing disturbances during the shaving process, which result in profile form deviations in the form of a waviness in the tooth profile. Assuming practically equal chipping forces on the meshing flanks, the size and direction of the resulting tangential force is dependent on the quotient of the number of meshing right- and left-hand flanks. The value of this quotient becomes equal to one when an identical number of right- and left-hand flanks make contact. This means that the transverse contact ratio is an integer value. The quotient mainly deviates from the value of one for contact ratios İ Į = i + 0.5 ( i is an integer) and for a small number of flank contacts. In this respect, it is possible to change the precision achievable for shaving by means of the meshing geometry. Large contact ratios are necessary to achieve low deviations when shaving. The geometry of the shaving cutter and the workpiece should be coordinated such that an identical number of right- and left-hand flanks are always in contact, since the quotient is then exactly (or nearly) equal to one. The influence parameters determining the quality of the shaving precision are [10.2/73] • the transverse contact ratio, • the module (minimum requirement: m > 1.5 mm), • the number of teeth (minimum requirement: z > 17), • the helix angle ( ȕ > 10 is favourable), • the profile shift, • the axis-crossing angle ( Ȉ0 = 15 is favourable; as a minimum Ȉ0 = 8 ), • the rough-machining precision (runout and pitch deviations should be a maximum of one quality worse than the shaving quality). 10.2 Tooth-shape Forming Methods 633 Figure 10.2/28 Shaving cutter tooth The shaving cutter represents a helical cylindrical gear, whose flanks are interrupted by grooves. This results in the blades with cutting edges required from a chipping-technical perspective between the grooves (Figure 10.2/28), whereby the shaving cutters (and thus the blades) are manufactured from high-speed steel. The width of the shaving cutters depends on the process variant. For gear shaving, the precision of the machining depends to a large extent on the accuracy of the pre-machining. This is due to the lack of enforced movement. Since, however, the radial position of the shaving cutter to the workpiece is specified, runout deviations from the pre-machi-ning are translated into pitch deviations after the shaving, while – in a similar manner – pitch deviations from the pre-machining are translated into runout deviations after the shaving. In order to avoid either of the two effects dominating, only small allowances (in the order of one-hundredth of the module) are planned, which, in addition, also have the advantage of lower machining times. By taking account of a high degree of pre-machining precision, it is possible with gear shaving to achieve a toothing accuracy in the range of the qualities (5) 6 to 7 (9) in accordance with ISO 1328 (DIN 3962/63) [10.2/76] to [10.2/79]. 10.2.4.8 Profile Broaching Profile broaching represents a highly efficient method of manufacturing straight or helical inter- nal and external toothing, albeit requiring complicated and expensive tools. In this process, the broaches exhibit cutting teeth over their entire length, which begin with the pre-machining profile, with the tooth height zero. As the cutting tooth height increases from one row of teeth to the next, they correlate in the endpiece (rows of calibration teeth) to the negative of the gear tooth profile to be manufactured, that is, the tooth height of the gear. Broaches are thus invariably workpiece-specific and have a guide piece for the exact centring of the broaches with respect to the workpiece. The cutting motion (linear for spur gears and helical twisted for helical gears) and the feed motion generated by the staggering of the teeth (increasing tooth height) are specified as the relative movement between the broach and the workpiece. The toothing is completed after pulling the active length of the broach (or broaches if the profile staggering is divided into two or three broaches) through the workpiece. Broaches are manufactured up to 150 mm in diameter from massive high-speed steel. For a diameter in excess of 150 mm, broaches are equipped with toothed bushes for each series of teeth, since the manufacturing process of such massive broaches results in a loss of control in the process, particularly with regard to the hardening, and these expensive tools can be better finished in the event of wear. A disadvantage, however, is the lower precision of these composite broaches. Vertical broaching machines with the highest possible guiding precision should be used as broaching machines for the manufacture of gears because only these ensure both the centring of the broaches and their exact movement. 634 10 Manufacturing of Cylindrical Gear Units The precision of the broached gear is fundamentally determined by the accuracy of the broaches and the clearance between the guiding surfaces on the broaches and the workpiece. The broach-ing precision should therefore be 1 to 2 qualities better than the manufacturing precision of the gear. Moreover, the guide clearance should be at a minimum, in which regard an exact pre-ma-chining of the workpiece-guiding surface is necessary. A greater loss of workpiece toothing quality in relation to the broach toothing occurs with larger modules, which is due to the increase in the spring-loaded deformation as well as the thermal deformation. On the face sides of the gears, helix deviations are to be expected, particularly in the outlet area of the broaches, since the chipping force will decrease (and thus the springback comes into effect) due to the ever lessening number of rows of teeth cutting. Thus, in this region, the cutting into the toothing invariably be-comes deeper. The disadvantage caused by only non-hardened gears being broached as a result of the use of high-speed steel tools has been eliminated by the development of diamond-tipped broaches. These can also be used to finish case-hardened internal gears. The improvement to the precision of the gears, however, is slight, due to the limited stiffness of the broaches. 10.2.5 Chip-removal Tooth-shape Forming Using Tools with Geometrically Undefined Blades 10.2.5.1 Technological Basics Shape forming with geometrically undefined blades, which includes the grinding, honing and lapping processes, corresponds in principle to shape forming with geometrically defined blades if the chipping process is due to the single cutting edge. In contrast to the geometrically defined blade, however, the single cutting edge possesses special characteristics, as it has only an extremely limited expansion transverse to the cutting direction and exhibits low cutting edge radii, which reduces the minimum chipping width to values in the range of 0.01 ȝm. This enables cutting of the grinding wheel into the basic material at the smallest contact magnitudes, resulting in a reproducible accuracy in the micrometre range. A further feature is the fact that the abrasive grains are spatially arranged in association with the volumetrically structured grinding and honing wheels. This layout allows machining with self-sharpening tools in combination with the tendency to split the exceptionally hard abrasive grains, which primarily requires a dressing process in the form of jiggering or regrinding, which, however, in contrast to re-sharpening the tools with geometrically defined blades, takes place on the processing machine. As a result of the spatial layout and the limited width of the abrasive grain blades perpendicular to the direction of cutting, shape forming with geometrically undefined blades resembles a surface-scratching process, the productivity of which is significantly lower and whose energy requirement is considerably higher than shape forming with geometrically defined blades [10.2/80]. Therefore, this method is primarily used for fine finishing both non-hardened and case-hardened gears, whereby the machinability of case-hardened materials is in general better than that of non-hardened materials. The high energy requirement leads to a generally high level of heat generation in the grinding process, which can be influenced by the process design, the pairing of cutting material, material, and auxiliary substances, and the intensity of the cooling lubrication. The limited cutting width across the cutting direction results in a transverse roughness that is individual for all grinding processes, while the symmetrical (on statistical average) configuration of all blades in this direction leads to a chipping force composed solely from the components parallel and perpendicular to the workpiece surface. Due to the operating 10.2 Tooth-shape Forming Methods 635 direction, which is the same as the cutting motion, the cutting force parallel to the workpiece surface Fc determines the necessary cutting power Pc, while the component perpendicular to the workpiece surface Fn determines the deformation between the tool and workpiece. In this regard, it influences the precision of the method. Here the relationship is Fn/Fc > 1. The following cutting materials for shape forming with geometrically undefined blades (generally referred to as abrasives) play a key role in the manufacture of gears: • Corundum is the abrasive traditionally used for machining non-hardened and case-hardened steels. Its chemical composition, A1 2O3, is temperature-resilient up to approx. 1425 C, to which specific components are added for the purpose of achieving various characteristics. For precision grinding of gears made of steel materials, white corundum (A1 2O3 with a 99% degree of purity) is particularly favoured, although monocrystal corundum and alloyed corundum (e.g., in the form of ruby corundum (white corundum with chromium oxide additive)) are also used. • Sintered corundum possesses (in comparison to fused corundum) a higher fracture toughness in association with approximately the same hardness. Furthermore, its fracture behaviour differs significantly from fused corundum because no preferential crack planes ensue, due to the extremely small individual crystals (< 0.5 m). The edge rounding of sintered corundum results in particles breaking off, which are markedly smaller than the flaky fractures occurring along the preferential crack planes when fused corundum is used. Here it is appropriate to speak of a self-sharpening process. • Silicon carbide is only of minor importance as an abrasive for gears since it is primarily suitable for the machining of short-chipping materials such as lamellar-graphite cast iron, non-ferrous metals, carbides, glass and stone. These materials, however, are not typically used in the manufacture of gears. • Cubic boron nitride (CBN), with its extremely high degree of hardness, a temperature resilience of up to approx. 1125 C and a thermal conductivity 30 to 100 times higher than corundum, is used both as a galvanically bonded abrasive on a steel main body and as a ceramically bonded abrasive in dressing grinding discs. For galvanically bonded tools that cannot be dressed, the geometrical precision of the profile is in the main body and cannot subsequently be changed. However, it does possess a high for m constancy over its time of use. In order to maintain the ability of the tool cutting, a constant wear caused by microcrystralline grain chips is striven for. This self-sharpening effect is not uniform along the work area and has to be supported by targeted dressing using a diamond tool. The dressing and contact conditions between the dressing tool and the grinding disc have a critical influence on the surface roughness and microstructural damage at the tooth flank. The grindability of gear materials [10.2/80], [10.2/81] is primarily determined by • the influence of the cutting force or cutting performance and thus the cutting energy generated as a performance- and wear-determining criterion for the grinding process, • the influence of the perpendicular force as a deformation-causing parameter and thus pre-cision-determining material criterion, • the thermal durability of the surface layer of the material in relation to the prevention of grinding burn or undesirable residual tensile stresses with the consequences of reducing the surface-layer hardening as a service-life-determining criterion for gears or even the occurrence of grinding cracks. The heat created in the machining process that penetrates into the workpiece leads to an increase in temperature in the edge layer of the workpiece, which, in turn, results in an increase in volume 636 10 Manufacturing of Cylindrical Gear Units and thus to compressive stresses, which cause permanent form changes on exceeding the yield strength, so a complete reversal of this process is no longer possible on subsequent cooling. The consequences are residual tensile stresses, which reverse and tend to decrease with the increasing surface distance due to the equilibrium. These stresses superpose each other in the stresses already existing in the workpiece, resulting in the progression depicted in Figure 10.2/29. Alongside the heat load of the material, a high pressure load is recorded when the single cutting edges make contact with the surface of the workpiece, which primarily result from the elasto-plastic material deformation upstream of the chip formation, corresponding to a scratch process, and which, when paired with the cutting motion, lead to tensile stresses during the cutting proc-ess. By analogy with the thermal stresses, compressive stresses develop on reversing this defor-mation process after the cutting contact. This compression hardening of the surface layer of the material thus ensues without a corresponding chip removal, so the mechanical hardening remains in effect with decreasing cutting values (or material removal) while the thermal material strain, however, is decreased. The resulting residual stresses represent the superposing of the stresses mechanically and ther-mally caused (Figure 10.2/29), so both residual tensile stresses and residual compression stress are possible as the result of grinding processes, depending on technological conditions and cut-ting parameters. While residual compression stresses are a desired effect and lead to an increase in the load ca-pacity of cylindrical gears, residual tensile stresses have the opposite effect and may lead to grinding cracks in the event that they exceed the tensile strength of the surface layer of the mate-rial. In this regard, it is often noted that these grinding cracks are not immediately visible when the grinding process is ended because an edge layer subject to compressive stress only a few mi-crometres thick initially prevents the grinding cracks from reaching the surface of the workpiece. Together with this stress formation, high temperatures in the edge layer of the material also result in the decarburisation of the material, leading to the known localised burning, this constituting a soft layer that also reduces the capacity of cylindrical gears to withstand stresses. The overall material-technical state following the grinding is often referred to using the term “material integ-rity”. The geometric precision of the workpiece, which is an evaluation criterion of the grinding process just as much as the material integrity, is subject to multiple causal influences. The most important phenomena can be summarised as follows [10.2/81]: Figure 10.2/29 Internal stress and hardness curve in the surface layer after the case hardening 10.2 Tooth-shape Forming Methods 637 • For rolling and screw rolling processes, the geometrical-kinematic approximation of the toothing causes the profile and infeed meshing marks already known from the gear-cutting processes that use a geometrically defined blade. These marks represent the form deviations of the profile or the tooth trace. It is possible to influence them by changing the mutual amount of the feed components. • The kinematic deviation of the drive train of the grinding machine causes geometrical devia-tions of the profile and – with helical gears – also of the tooth trace, which are generally wave-like in nature. Their amplitude is determined by the kinematic accuracy of the gears of the grinding machine and is simultaneously excited and damped in the grinding process. In general, the precision-determining transmission elements are first and foremost the last elements acting directly on the workpiece drive. • The imbalance of the grinding wheel leads, in turn, to forced vibrations, which, depending on the process concerned, manifest themselves as a wave-like geometrical deviation of the profile or tooth trace. These deviations are predominantly mastered by the balancing level of the grinding wheel. • The normal force Fn perpendicular to the tooth flank, interacting with the finite stiffness of the machine-tool-workpiece-clamping device system, results in deformations, which lead to geometrical deviations of differing characteristics according to the number of tooth flanks meshing at the same time. For single-flank machining, this results in a form deviation similar to crowned gearing. For multi-flank machining, by analogy with hobbing or gear shaving, this creates alternating directions of force and deformations and, as a consequence, wave-like form deviations of the profile. It is possible to influence these form deviations by reducing the normal force, namely by means of improved cutting properties of the grinding wheel, lower cutting parameters or a sparking-out following the rough grinding. • As a consequence of the smaller width of the single cutting edge of an abrasive grain in relation to the width of the grinding wheel, a transverse roughness occurs and – as a result of the generation of the surface of the workpiece by successive individual steps in the direction of feed – a longitudinal roughness occurs, which together create the workpiece-surface topography typical of grinding processes. The size of the roughness can be influenced by the selection of a suitable grain since a natural relationship exists between the roughness following sparking-out and the grain size of the grinding wheel. A tooth flank, typical for the process, is formed from the superposition of these phenomena, which can have a major influence on the functional properties of the gear. The grinding, honing and lapping processes are derived from the technical principles specific to shape forming with geometrically undefined blades and the technical gear-generating principles of the rolling, helical thread rolling and screwing processes, whereby the grinding processes pos-sess the greatest importance because they are able to eliminate the distortions, particularly from the predominating case hardening with high productivity; that is, they work in allowances in the tenths of a millimetre range. Honing primarily serves to improve the precision of the form and the position of the tooth flanks, as well as the fine surface structure, by this means enhancing the noise behaviour of the gears that are almost finished following the case hardening. An overview of the gear-grinding processes is depicted in Figure 10.2/30. All gear-grinding processes operate with a kinematic enforced movement between the grinding wheel and the workpiece, such that the final precision after grinding is determined by the technological conditions and the tool and workpiece positioning on the grinding machine, while the deviations arising in pre-machining are eliminated. In this regard, the deviations arising in pre-machining primarily have an effect on the size of the allowance and thus the productivity, but less significantly on the final precision. An overview of the precisions achievable with tooth grinding is depicted in Figure 10.2/31. 638 10 Manufacturing of Cylindrical Gear Units Figure 10.2/31 Quality achievable with tooth grinding 10.2.5.2 Discontinuous Generating Grinding with Plate-shaped Grinding Wheel In discontinuous generating grinding with plate-shaped grinding wheels (MAAG method), the involute profile of the tooth is produced by rolling the toothing on two plate-shaped grinding wheels, which correspond to the normal section of the basic rack. The rolling motion is produced by a generating system, consisting of a pitch block and roll band. The enveloping of the face width of the basic rack is achieved through the longitudinal feed of the machine’s sliding carriage, which means that the current feed direction is the result of both components. A sinusoidal path of the feed trajectory forms due to the alternating direction of the generating feed component over the tooth flank. Figure 10.2/30 Overview of the gear grinding process 10.2 Tooth-shape Forming Methods 639 The cutting motion results, as in all grinding methods, from the rotary motion of the grinding wheel. Depending on the arrangement of the axial position of the grinding wheel relative to the tooth-ing, a distinction is made between two variations of the method (Table 10.2/2, number 11): • In the 0 grinding method the grinding wheels are arranged coaxially and tangentially in relation to the toothing. The work surface is the small ring area on the face side of the relieved, plate-shaped grinding wheel. Since the grinding wheel pressure angle Į0 equals or almost equals the value of zero and corresponds to the pressure angle during production, the generating motion must be done by rolling off the base circle. • In the Į grinding method both grinding wheels are inclined towards one another by twice the pressure angle Į , which means that each face side stands under the pressure angle Į to the common radial section of the gearing. The work area can correspond to the face area in total, or it limits itself to the small ring area of the grinding wheel, as in the 0 method. Different surface structures are produced on the ground tooth flank, depending on the angle of the inner cone. A distinction is made between the plane grinding for Į = 0 and the crisscross and the surface grinding for Į > 0 . In both variations the profile is produced continually, and the tooth trace in discrete longitudinal feed steps. To avoid deviations resulting from wear on the typically small grinding edge, it is probed, and when a mismatch with the nominal position is detected, it is axially adjusted via the grinding spindle. This measure, combined with the kinematics of the method and the low machining power, ensures extremely high accuracy and surface quality with this method. The accuracy achieved corresponds to quality 2 according to ISO 1328 (DIN 3962/63) and a surface roughness of R z < 2.5 m. Corundum, silicon carbide and CBN grinding wheels are used as grinding tools. Silicon carbide grinding wheels are particularly well-suited for processing cast materials, and CBN grinding wheels for tool, high-speed and nitrided steels. The grain size is adapted to the material group in order to achieve uniformly high surface quality. Corundum and silicon carbide grinding wheels are used with porous ceramic binding, and CBN grinding wheels, in many cases, with resin binding. Grinders for discontinuous generating grinding with plate-shaped grinding wheels are manu-factured with a horizontal (small type series) or vertical workpiece axis (large type series for m 36 mm, d 4734 mm) and enable both the machining of conventionally modified gearing using templates, as well as such with topological flank modification [10.2/82] to [10.2/85], using an electronic topological modification mechanism for the generating kinematics. Here the topological modifications are entered into the controls oriented to the tooth heights and face width. The controls convert the modification data into the planes of action of both flanks and plot this field in three dimensions if needed. Then the data are implemented in the control field of the machine’s kinematics. 10.2.5.3 Discontinuous Generating Grinding with Disc-type Grinding Wheel The generation of the toothing geometry in discontinuous generating grinding with disc-type grinding wheels (NILES method), (Table 10.2/2, number 12) is done according to the principle of rolling. The cylindrical gear (workpiece) rolls on a grinding wheel profile (grinding disc). This partly embodies the rack. The grinding wheel (the grinding disc) executes a stroke motion (Figure 10.2/32) [10.2/86] to [10.2/88]. Depending on the width of the grinding wheel and the processing of the flanks, it is possible to differentiate between the variations: 640 10 Manufacturing of Cylindrical Gear Units • Single-flank grinding: The tooth flanks of a tooth space are successively generated with a grind-ing wheel profile. As opposed to the basic rack, this has a smaller tooth thickness. The bridging of this differ-ence in tooth thickness to grind the other flank is done periodically through an additional axial motion of the grin-ding wheel. • Double-flank grinding: Contact is made on both tooth flanks of a tooth space. The grinding wheel profile partially corresponds to the basic rack of a rack in the normal section. Figure 10.2/32 Principle of discontinuous generating grinding with disc-type grinding wheels [10.2/90] The generating motion is produced as combination of a rotary and a longitudinal motion of the cylindrical gear opposite the basic rack. Technically, these motions, which are in a fixed relation depending on the workpiece toothing data, are realized in different ways. What has been estab-lished in particular is as follows: • the electronic gear unit as stepless transmission between rotary and longitudinal motion, • the threaded spindle-worm-change gear-gear unit (TWC) principle with adaptation of the speeds between rotary and longitudinal motion by way of a change-gear gear unit, • the pitch block-roll band principle in analogy to the generation principle of the firm MAAG (refer to Section 10.2.5.2.) with interchangeable pitch block, • the pitch block-roll band principle with adjustment lever for stepless adjustment of the speed ratio between rotary and longitudinal motion. The cutting motion corresponds to the grinding wheel rotary motion. Depending on the face width and the helix angle of the gear teeth on the workpiece, the length of the stroke and, depending on the position of the toothing relative to the machine table surface, the stroke position are adjusted. The generation of the grinding wheel profile is primarily realised with diamond dressing tools acting in a point-shaped way, whose motion is path-controlled using templates or CNC controls, so that modifications to the profile of the toothing can be generated. Tooth trace modifications are achieved by influencing the kinematics over the length of the stroke, either by changing the centre distance or by an additional rolling motion. On conventional machines, the movements are specified by sometimes separately acting templates for the right-hand and left-hand flanks, but on modern CNC machines by the controls. Since the flank surfaces of the workpiece teeth come into contact with the basic rack along the generating line almost exactly matches the generating contact trace during a stroke, the manufacturing of topological flank modifications is enormously difficult. The generation of defined grinding wheel radii to guarantee the grinding of predeter-mined tooth root radii has only been possible to date using CNC dressers. Alternatively, the grinding wheel can be chamfered to achieve grinding in the tooth root while avoiding a stress-increasing grinding notch. Similarly, the generation of a tip chamfer during grinding is limited to machines with CNC dressers. 10.2 Tooth-shape Forming Methods 641 Figure 10.2/33 Flank shape for di scontinuous generating grinding with disc-type grinding wheel [10.2/90]: a) schematic view of flank surface deviations, b) profile measurement, c) helix measurement In terms of the machining kinematics, the sequence of movements involved in discontinuous gener- ating grinding with a disc-type grinding wheel consists of the generating of both workpiece tooth flanks and the indexing to position the grinding wheel in the adjacent tooth space to be ground, such that the grinding process repeats itself corresponding to the number of teeth of the gearing and the number of grinding cycles. As a result of the kinematics particular to this method, the intrinsic deviations of the machine, the forces active in the grinding process and the resulting deformations, a flank shape typical of this method is produced, which impacts the contact pattern and moreover the functional behaviour of the gearing (Figure 10.2/33). Mainly corundum and sometimes CBN grinding wheels are used as grinding tools. In addition to the high values of the main axial feed velocity, the typical cutting depths of 0.01–0.03 mm that exist in productive grinding according to this method in the plane of contact lead to high thermal material strain and can lead to residual tensile stresses, grind- ing cracks and even fires, which is why comprehensive provisions have been taken to master these phenomena [10.2/86] to [10.2/89]. Aside from opti-misation of the method with regard to these aspects, this especially includes the intensive use of cooling lubrication. Sometimes intermediate processing of large case-hardened gearings is carried out using skiving-hobbing, for example, before the finish grinding is done. The accuracy achievable with this grinding method corresponds to the qualities 3 to 4 in accordance with ISO 1328 (DIN 3962/63), but experienced personnel are required in order to ensure consis-tency in a series because there are a variety of influences, including subjective, on the outcome. To achieve high quality in the gearing, a complex grinding process is necessary, which differs in the roughing, sparking-out, and smoothing phases [10.2/89], [10.2/90]. 10.2.5.4 Continuous Generating Grinding In the case of continuous generating grinding (REISHAUER method) (Table 10.2/2, number 13), a grinding worm with tooth rack profile is used as a tool, which is inclined by the angle Ș 0 = ȕ Ȗ with respect to the workpiece axis, which means that this method operates according to the helical pitch principle and, with respect to the kinematics, thus also resembles hobbing. The toothing is generated through the relative motion of the rotating pair of grinding worm and workpiece, as well as the axial feed motion in the direction of the face width. As such, the infeed motion consists of the purely axial component (axial to the toothing) and, in the case of helical gearing, the differential motion to generate the helix angle [10.2/91] to [10.2/94] (Figure 10.2/34). 642 10 Manufacturing of Cylindrical Gear Units The cutting speed corresponds to the speed tan- gential to the thread of the grinding worm. As a variation of the method, it is possible to work with continuous shifting, in analogy to hobbing. More-over, the variations of single- and double-flank grinding are possible, even if double-flank grin-ding is the usual variation of the method. The generation of the rolling motion, derived from the rotary motion of the grinding worm, is made possible by a (formerly) mechanical, but today elec-tronic, gear mechanism of the machine. With the use of the mechanical mechanism, the drives of the grinding worm and the workpiece are coupled electrically by the alternating current frequency. The dressing (dressing and sharpening) of the grinding worm is accomplished alternatively using • diamond dressing rolls for - linear dressing, in which form rolls generate one after the other the grinding worm profile and nearly all possible modifications above the tooth height, or - profile dressing, whereby profile rolls embody the grinding worm profile including pos- sible profile modifications. For that reason they may be workpiece-dependent; • single-point dressing diamonds in a turning process, whereby the grinding worm profile is partially stored in the diamond dresser and is partially generated by shifting the dressing tool along the nominal grinding worm flank. This method is still only used on old machines. Modifications and corrections along the face width (lead direction), which can be realised for right-hand and left-hand flanks differently on contemporary machines, are generated by changes to the profile shift, or in other words by varying the centre distance of the grinding worm and the gearing, and in the case of different specifications regarding the right-hand and left-hand flanks, by an additional rotary motion of the workpiece relative to the grinding worm. Profile modifications and corrections are generated by changing the pressure angle and lead angle of the grinding worm. In the case of profile dressing, modifications to the profile form necessitate a modified dressing tool profile. For linear dressing, the grinding worm profile geometry can be generated path-controlled with a largely constant dressing tool profile. If modifications to the tooth trace are generated by changing the radial position of the grinding worm, in the case of helical gearing, profile twists occur caused by the arc-shaped contact paths between grinding worm and workpiece flanks. These twists can be avoided by changing the pressure angle or the lead of the grinding worm coordinated with the axial position in diagonal generating grinding. In these cases, the grinding worm has a variable profile across its width. In the same way it is also possible to produce gearing with a specific nominal twist that is incorporated in the actual design, which is intended for optimising the load capacity or noise. The meshing between grinding worm and workpiece gear teeth takes place on the planes of action of the right-hand and left-hand flanks. Se veral flanks in the same and opposed directions are constantly engaged in this meshing, so, as a result of alternating directions of tangential force along the height of the profile, variations in force result. Because of the nature of the system, they lead to the deviations in the profile form (profile waviness) which hobbing is known for. However, because of the overall low levels of force, they remain small. Moreover, feed markings form in the axial direction. In interaction with an excitation of vibrations caused by a grinding wheel imbalance, these Figure 10.2/34 Principle of the method of continuous generating grinding (REISHAUER system) 10.2 Tooth-shape Forming Methods 643 deviations can lead to an inadequate grinding pattern in the fine-finished gearing produced in continuous generating grinding. If tight requirements on the balance quality of the grinding wheel are ensured, with this method it is possible to achieve levels of toothing accuracy corresponding to the qualities 2 to 3 in accordance with ISO 1328 (DIN 3962/63) for dressable grinding worms and the quality 4 for non-dressable grinding worms. To avoid the so-called washboard effect, which results from continuous scratch marks across the face width, the “low-noise shifting method” was developed. This involves a specific shifting in which the roughness scratch marks are interrupted in the direction of the tooth trace, and a more-or-less stochastic roughness profile forms. Through low-noise shifting, the excitation of noise can be considerably reduced, particularly in gearing with small modules. The rolling of the workpiece teeth on the flank of the grinding worm has the effect of not producing a formation of generated cuts, but rather a continuous generating along the profile. Because of this, with the exceeding of the minimum chip thicknesses, the abrasive cutting edges – as a balancing reaction to the acting motion of the grinding worm work surface – and the tooth profiles begin to mesh so that the formation of chips occurs especially under mechanical load. As a result, if there is sufficient chip space in the form of grinding wheel pores which are not clogged, the desired residual compression stresses are produced on the surface layer of the gearing, which means that the load capacity of gearing produced by continuous generating grinding is high. Used as grinding tools here are generally single and multi-thread ceramically bonded corundum grinding worms, as well as galvanically and ceramically bonded CBN grinding worms [10.2/93], [10.2/94]. To achieve high material removal rates and high surface qualities in the gearing, in the case of galvanic CBN grinding wheels, it is necessary to work with grinding worm sets consisting of a coarse-grained, rough grinding worm and a fine-grained, finish-grinding worm or a polishing worm, which are arranged coaxially (Appendix 15.6). Aside from that, combinations with profile grinding are also possible (Appendix 15.7). In order to raise the effectiveness of the method, on modern machines, continuous shifting is also applied in the roughing and in the smoothing phase, which in effect is equal to continuous dressing. Between the rough and the finish grinding, usually a large shift step is carried out right into the finishing area, which is compensated for again after the finish grinding. As such, the finish grinding, which is crucial for determining quality, is always carried out with a freshly dressed thread of a grinding worm. The productivity achievable with this method combined with its high level of accuracy has led it to become an important gear-grinding method for both small series and large series production [10.2/107]. Machines that work according to the continuous generating grinding principle are usually manufactured with a vertical workpiece axis and are suitable for the dimensional ranges of m ≤ 8 (10) mm and d ≤ 1000 mm. The NC axes correspond to those of hobbing, whereby additional motion axes are necessary for the dr essing of the grinding worm. 10.2.5.5 Discontinuous Profile Grinding In the broadest sense, profile grinding (Table 10.2/2, number 14) draws the geometry and kinematics of its method from the screw principle. Accordingly, the geometry of the gearing must be embodied by the grinding wheel geometry, at least on one plane and one pitch (Table 10.2/2, number No. 14). Based on this principle, all of the variations of the method of profile grinding are derived; the geometry of the positive reference – internal – gearing is largely saved in the grinding tool and to some extent realised by the kinematics of the method [10.2/95] to [10.2/97]. 644 10 Manufacturing of Cylindrical Gear Units In discontinuous profile grinding, at least one right- hand and one left-hand flank are ground simultane-ously or sequentially in an operation corresponding to a stroke or double stroke in the axial direction. This means that this operation represents a basic unit of the method, and it is repeated as often as the number of teeth of the workpiece and the technologically determ ined number of grinding cycles require. As such, the repetition of this basic operation is executed by the machine moving the part by the pitch angle of 2 ʌ/z. Variations of the method of profile grinding are achieved from variab le arrangements of the grin- ding disc which embodies the tooth space profile and the possible infeed directions tangential, radial or rotational relative to the gearing. Depending on the respective methodological or machine require-ments, profile grinding can be carried out accor-ding to the climb principle or the up-cut principle, using the stroke and the return stroke for infeed or using one stroke for sparking out as well as with indexing in one or both stroke return positions. The generation of the grinding wheel profile, the process of which is related to the type of grinding wheel and the method of dressing, has a significant influence on the productivity and flexibility of discontinuous profile grinding. It is necessary to differentiate between the following [10.2/98], [10.2/99]: • Grinding wheels which are non-dressable or non-redressable, which are therefore workpiece- dependent for their entire lifetime (refer to Appendix 15.10 and 15.11) • Grinding wheels which are dressable and therefore can be dressed on the grinder, and which can be re-trued to a different workpiece geometry (refer to Appendix 15.10 and 15.11) Modifications along the face width require a corresponding control along the feeding path and are exactly – and in the case of different modifications on the left-hand and right-hand flanks generally – only manufacturable through finish grinding in single-flank grinding, preferably with rotatory infeed of the gear flank. In discontinuous profile grinding, topological modifications in the sense of variable twists require a combined motion of multiple axes along the length of the stroke. Discontinuous profile grinding with a dressable grinding wheel is offered for external as well as internal gearing in the dimensional range of m ≤ 16 (25) mm and d ≤ 4000 mm (Table 10.2/2, number 14). With non-dressable, galvanically bonded CBN grinding wheels, the method is used in a similar dimensional range as a stand-alone machine solution and, aside from that, for larger moduli, as a complementary setup for hobbing machines for processing external and internal gearing in the dimensional range of m = 1 to 25 mm and d ≤ 4000 mm. 10.2.5.6 Continuous Profile Grinding Just as with continuous generating grinding, continuous profile grinding (Table 10.2/2, number 15) is based on a worm-shaped grinding tool. As opposed to discontinuous profile grinding, the grinding wheel profile simultaneously covers multiple pitches and, corresponding to the line of Figure 10.2/35 Disc ontinuous profile grinding 10.2 Tooth-shape Forming Methods 645 contact between grinding worm and workpiece, a maximum face width which is especially dependent on the helix angle of the workpiece. In gear transmission terms it represents a globoid worm. For swivel angles of the tool of Ș 0 = ȕ Ȗ 0 the principle of this method is based on the expansion of the line of contact between the grinding worm and the workpiece in the axial direction to the workpiece, which means that helical gearing with limited width is grindable in a highly productive plunge-cut process [10.2/100], [10.2/101]. In other words, it involves a radial infeed combined with a rotational motion of the workpiece to make contact between the grinding worm flanks and the workpiece flanks, while the axial infeed is omitted for a limited width. For collision reasons, continuous profile grinding can only be carried out in single-flank grinding. It is helpful to illustrate the contact conditions using the example of contact between a globoid grinding wheel and a cylinder. As can be seen in Figure 10.2/36, a winding contact line results in a certain way around the cylinder, whereby the contact angle between grinding wheel and cylinder can be up to 180 . Because of the undercut which then occurs, larger angles are not possible. On the grinding wheel, face-grinding areas would then occur on the inner contour. The maximum grindable face width corresponds to the axial length of contact between the grinding wheel and the cylinder. This length depends on the axis-crossing angle, plus the two-fold half – that is, the single – chord length of the contact of the inner face side of the grinding wheel with the cylindrical workpiece, which depends on the protrusion of the grinding wheel face areas over the workpiece axial cross section. When the cylindrical workpiece is turned on its axis, it is cylindrically ground on the entire length of contact. Figure 10.2/36 Principle of continuous profile grinding (left) and the grinding operation (right) The dressing of the globoid grinding worm is done with a workpiece-dependent, diamond dressing cylindrical gear that emulates the workpiece geometry and incorporates all modifications of the workpiece. Through radial infeed of the grinding worm against the diamond wheel that is mounted in the same axial position as the workpiece and rotating feed to engage and feed to the right-hand and left-hand flanks, in analogy to the grinding process, its profile is mapped in an angular range in the globoid grinding worm. This angular range is determined by the grinding worm cross section that is perpendicular to the workpiece axis. In order to take into account the larger thickness of the workpiece teeth due to grinding stock and to avoid collisions while grinding, it is necessary to dress an enlarged tooth space width by rotatory feed. The space width of the worm exceeds the tooth thickness of the pre-processed toothing so that 646 10 Manufacturing of Cylindrical Gear Units the tool can be fed to the desired manufacturing centre distance during grinding, without any collision. The grinding is done, as mentioned above, with single-flank grinding with a rotatory infeed motion, whereby the rotatory motion component between dressing and grinding can vary in its size. The geometry and kinematics of the method lead to reaction loads, which cause deviations in the tooth trace. Of particular importance here are dynamic influences. Unsteady force components tangential to the workpiece gear teeth can create form deviations (waviness) in the tooth trace. The continuous profile grinder is known for its extremely rigid structure, the purpose of which is to achieve adequate static and dynamic rigidness and thus prevent unwanted form deviations caused by the system, on the tooth trace in particular. It works with an electronic gear unit, which derives the workpiece rotary motion from the grinding worm rotary motion. The centre distance between grinding worm and workpiece corresponds to that between the grinding worm and the dressing wheel. Any deviations in the tooth trace which may occur can be compensated for to a limited extent by changes to the centre distance or corrections to the axes crossing angle, because there the profile deviation that occurs is considerably smaller. As grinding tools, one- to seven-thread ceramically bonded corundum grinding worms are used on continuous profile grinders. Generally these machines are equipped with an automatic workpiece changer, which also allows for their use in automated production lines. To date, continuous profile grinding can be used for small helical gearing in the dimensional range of m = 1.5 to 3 mm, d ≤ 250 mm and b ≤ 40 mm. In plunge-cut grinding, because of the technological parameters derived from the geometry and kinematics of the met hod, arbitrary extensions are not possible. The achievable accuracy is essentially dependent on the accuracy of the diamond dressing wheel, but also on the dressing and cutting conditions and the workpiece geometry. It achieves the ISO (DIN) quality 4. Because of the technological limits of the method and the workpiece dependency of the diamond dressing wheel, which is a main determinant of costs, this highly productive method is mainly suitable for large series and mass production. Generally, to improve the fine structure of the surface and the noise behaviour, a downstream roll-honing procedure is carried out, which, in the case of combined profile-grinding and honing machines, can be carried out in one work cycle on the same machine. In this case the roll honing is done using an externally toothed honing tool. In order not to damage the sometimes complex flank shape generated in continuous profile grinding, the roll honing is done with very small stock allowances (approx. 0.005 mm/flank) so that the removal is mostly in the range of the roughness profile of the workpiece after the grinding process. 10.2.5.7 Roll Honing Roll honing is a fine-machining method that is mainly for case-hardened cylindrical gearing and works according to the helical pitch principle. In addition to improving precision and the surface roughness of the workpiece, it is also used to reduce noise excitation. In terms of the method’s kinematics, roll honing corresponds to roll shaving, whereby, however, primarily internally toothed tools are used (Table 10.2/2, numbers 16 and 17), which, in the case of roll shaving, would only be manufacturable at extremely high cost. It is possible to differentiate between the following roll-honing methods, depending on the type of toothing of the roll-honing tool, and its specification, and the technological conditions: • Externally toothed roll-honing disc - Roll honing with resin-bonded abrasive roll-honing disc, without rolling coupling (touch honing) - Super shaving with galvanically bonded CBN roll-honing disc, in analogy to diagonal shaving without rolling coupling 10.2 Tooth-shape Forming Methods 647 - Hard shaving with galvanically bonded CBN roll-honing disc, in analogy to diagonal shaving with additional guide gear set • Internally toothed roll-honing disc - Ring-honing method with resin or galvanically bonded roll-honing disc, with or without rolling coupling. Corundum, CBN and diamond are used as the abrasive material with resin and ceramic binding. Refer to Appendix 15.5. Since, in analogy to roll shaving, the principles of multiple flank contact apply, which means that in order to achieve the best possible final accuracy a transverse contact ratio of İ Į = 2 and a large contact area between the workpiece and the tool should be strived for, roll-honing with an internally toothed honing wheel represents a preferred solution. In analogy to roll shaving, the chip removal is done on arc-shaped contact paths along the flanks. This results in cutting speeds which, compared to typical honing methods, are relatively high, so in this respect roll-honing methods are also known for their relatively high productivity. Touch honing (predominantly ground gearing) works with a resin-bonded honing wheel with an abrasive grinding material (corundum), which is dressed and sharpened using a diamond dressing wheel. There are different variations of this method (diagonal, transverse and plunge roll-honing, in analogy to the roll-shaving methods) which can be applied, depending on which flank areas should (preferably) be processed. In general, the amount removed on the flanks is less than 0.005 to 0.008 mm, which particularly means that form deviations and roughness are reduced. The final level of accuracy is strongly dependent on the accuracy of the pre-machining. The surface quality that can be achieved is R a = 0.1 to 0.2 m. Super shaving represents the implementation of diagonal soft shaving in a hard-machining process; the shaving cutter was replaced by a CBN-coated honing wheel (a single-layer CBN coating galvanically applied to a cylindrical gear as basic body), which acts like a shaving cutter without rolling coupling in double-flank contact on the hardened workpiece and, as such, reduces in particular the deviations arising from case hardening and pre-machining. As far as the achievable accuracy goes, the principles of roll shaving apply. For super shaving, the main parameters affecting the achievable accuracy are the runout and axial runout behaviour of the cylindrical gears after case hardening. Since the exact pitch of the tool can only form in the overlap area, because of the free running between workpiece and tool, runout deviations falsify the cumulative pitch of the workpiece. Wobbling leads to wobbling profile lines and tooth traces since the cylinders which are effective in generation are not identical to the measuring cylinders and planes of measurement. The amount removed from the flanks is 0.015 to 0.020 mm, and the achievable surface quality R a = 0.2 to 0.4 m. Presently, super shaving has been forced out of the market by generating grinding or honing with an internally toothed tool. Just as with super shaving, hard shaving works preferably with a CBN-coated externally toothed honing wheel, which, however, is roll coupled to the workpiece by a guiding pair of discs affixed to the same shafts as the tool and workpiece. Aside from a CBN-coated honing wheel, it is also possible to use a ceramically bonded corundum wheel. To realise the infeed of the hone, the **backlash** of the guiding pair of discs can be exploited. However, with that, in hard shaving the contact between the tool and the workpiece is single-flank, which means that the two different tooth flanks must be machined one after the other. In the case of the CBN-coated honing wheel, it is possible to efficiently achieve removal of up to 0.15 mm from the flanks, whereby enforced kinematic motion also produces an improvement to the geometric accuracy of the toothing. With hard shaving it is possible to achieve an accuracy of up to ISO 1328 (DIN 3962/63) quality 5, along with a surface quality of R a < 1.25 m. With that, this honing me thod should be an alternative for tooth grinding in some cases, even without having achieved the high grinding qualities, however. As such, hard shaving currently has no relevance on the market. 648 10 Manufacturing of Cylindrical Gear Units Roll honing with an internally toothed honing wheel represents an alternative to the methods named above, but it allows – due to the higher contact ratio between honing wheel and workpiece and with that, a larger contact area in the mesh, as well as due to the use of an electronic gear unit between honing tool and workpiece – an improvement in the machining accuracy with respect to [10.2/102] • runout deviations, • form deviations on the flank surface up to ISO 1328 (DIN 3962/63) quality 1 to 2, • surface roughness to Rz = 1 to 2 m, and • deviations in cumulative pitch by 1 to 2 ISO/DIN qualities. Roll honing with an internally toothed honing ring is used to improve flank shape and surface structure and the quality of case-hardened gearing, mainly to reduce transmission noise in automobile manufacturing. More than in the other fine-finishing methods, the productivity of this method is dependent on machining allowances and, with that, on the machining of the workpiece after hardening. Typical machining allowances for roll honing are approximately 0.05 mm per flank, and for tooth grinding more than (0.08) 0.10 mm per flank. As in the other honing methods, the dressable honing wheel is dressed using a diamond dressing wheel, which incorporates the flank shape of the workpiece with all profile modifications. It is also possible to use universal dressing gears, as long as the module, pressure angle, and helix angle of the workpiece and the dressing gear match and no other profile modifications are required. If the helix angles of the dressing wheel and the workpiece do not match, the axes-crossing angle must be changed between dressing and honing. Generally, honing is carried out with an axial infeed (oscillating motion). Crowned or tapered cylindrical gearing can be produced on NC-controlled honing machines, whereby the crowning can be placed at mid face width or shifted to one face side. The honing wheel is made of white corundum for a workpiece hardness of up to 64 HRC. For more than 64 HRC, silicon carbide is also used. The grain size can be adapted to the desired surface quality. 10.2.5.8 Roll Lapping Roll lapping (run-in lapping) is a fine-finishing method for gears that works according to the rolling principle, improving the surface structure and quality of the flanks in particular. The lapping is done during the running-in of pairs of gear sets, with the continuous supply of a lapping liquid. This consists of oil with a fine-grained abrasive material added. The abrasive material used is primarily diamond or silicon carbide. However, because of the development of roll honing, roll lapping has almost lost its significance for cylindrical gearing, whereas bevel gearing is often roll lapped. Today the area of application of roll lapping is limited particularly to special cases and spiral bevel gearing in automobile manufacturing. In the roll-lapping process, the right and left flanks are worked one after the other, while one gear is driven. The other flank is either worked when the direction of rotation is reversed or the input torque is changed from acceleration to deceleration. The cut volume per time unit depends on the sliding speed of the tooth flanks against each other and the local contact pressure. As such, the toothing geometry and the precision have an effect on the productivity. With increasing contact surface between the tooth flanks at decreasing deviations, the removal speed also drops. Usually a flank allowance of a few microns to a few hundredths of a millimetre is removed. Here, in analogy to roll honing, the better the machining accuracy, the less material removed, so at a minimum only the surface roughness is reduced. As with roll honing, roll lapping can also be used for non-hardened and case-hardened gearing. With roll lapping, crowned gearing requires an additional feed motion. For roll lapping, a vibratory movement boosts productivity and the surface quality. 10.2 Tooth-shape Forming Methods 649 The result of roll lapping is influenced by the following factors: • Lapping conditions (sliding speeds, contact stress, lapping time) • Lapping suspension (abrasive material, grain size, viscosity of the lapping oil, concentration of the abrasive material in the suspension) • Additional feed motion in vibratory lapping (amplitude, frequency, curve form) • Gearing (toothing geometry, material and its structural condition, deviations in the gearing) Since sliding speed and contact stress above the length of path of contact are not constant, non-uniform removal is to be expected, whereby protruding flank parts are removed quicker because of the increased pressure. With larger lapping times, however, uneven removal along the path of contact leads to negative effects on the tooth flank profile. With an optimal coordination of sliding speed, contact stress, and lapping time, the desired lapping result is achieved in gener- ally short lapping times and little removal. The value of roughly 18 m is aimed for as the lapping grain size because medium roughness then takes on a value between 1 and 2 m. On the part of the gearing, in particular the transverse contact ratio and overlap ratio, as well as the profile shift, have an effect on the lapping result. Roll lapping is carried out on special lapping machines. Lapping machines are used particularly in cases where additional feed motions are required or where large-scale productions must be processed. Table 10.2/2 contains a summary of important finishing methods for cylindrical gears. 10.2.5.9 Super Finishing Super finishing – also called vibratory finishing according to DIN 8589 – is a mechanical and chemical method of surface processing mainly for metallic workpieces. The workpieces to be processed are placed in a container lined with rubber or plastic, together with the abrasive pieces (the so-called chips), water, and special chemical solvents (compound). The container is made to vibrate so that a relative motion is established between the abrasive bodies and the workpieces. This causes the removal of material on the surface. For large and heavy gear unit components of up to 2000 kg, devices have been developed in which the workpiece is suspended in a fixed position, and the work container with the chips is made to vibrate in such a way that the abrasive material moves around the workpiece (Fi-gure 10.2/37). In metalworking, the abrasive bodies in a wide range of sizes and shapes are usually made of ceramic. Their abrasive material content (e.g., corundum) determines the removal effect and the achievable surface roughness. In conventional super finishing, sometimes also called trovalisation, a tenside solution is used as a compound which absorbs the wear debris of the abrasive piece and that of the workpieces and takes it away. In chemically accelerated super finishing , in a multi-stage process first a slightly Figure 10.2/37 Super finishing of large gears [10.2/108] 650 10 Manufacturing of Cylindrical Gear Units acidic solution is added, which oxidises the hard surface and makes it somewhat softer, so that more material can be removed [10.2/108]. The operation is repeated over and over again until the desired result is achieved. After that, a cleansing and neutralising agent is added. This treatment lasts roughly 3 hours in total. For case-hardened or nitrided tooth flanks, the surface roughness that can be achieved is at values of around R z = 0.5 m, depending on the method parameters. The extremely low surface roughness has a very positive effect on pitting and micropitting load capacity. 10.2.6 Testing Ground Cylindrical Gears for Damage to the Surface Layer At the foreground of production engineering is the effort to achieve an optimum in terms of efficiency, performance and quality. For cylindrical gears subject to heavy loads, quality is not just defined in terms of the geometric precision of the deviations in form and position, but also considerably in terms of the material characteristics, and the surface layer characteristics in particular, which are strongly influenced by grinding as the finishing method. Grinding burn and grinding cracks are among the kinds of processing damage that are feared the most. Grinding burn is brought about by the heat that is produced on the abrasive grains due to friction on the flank and the cutting face, caused by displacement energy and shear energy (Figure 10.2/38). The amount of heat q introduced is dependent on the grinding power P c and the material removal rate Qw in mm3/s. This heat leads to a change in the structure, which is accompanied by the reorientation of the crystal lattice and the partial precipitation of intercrystalline-bound alloying components. At very high removal rates and unfavourable processing conditions, residual tensile stresses can arise, which lead then to grinding cracks. The following reasons can be listed for thermal damage on the surface layer: • High machining rates Q w due to high infeed or high feed rates. Local grinding burn can also come about because of heat-treatment distortion where the stock allowance is dis-tributed very unevenly. • Too high of cutting speed: The risk of grinding burn rises rapidly at cutting speeds of more than 45 m/s. • Grinding disc - Grinding disc specification: Abrasive grain Fixing Hardness Structure - Condition of the grinding disc: Worn out Insufficient space for chips Sloppy mounting on the grinding spindle Figure10.2/38 Flow of energy in the abrasive grain: a Friction on chips; b Friction on the flank; c Shear; d Abrasive grit; e Fixing; f Chips. Heat dissipation through: A Cooling medium; B Fixing; C Grain; D In the chips; E In the workpiece 10.2 Tooth-shape Forming Methods 651 • Cooling lubricant - Specification, additives - Quantity, pressure - Arrangement and form of the nozzles • Toothing geometry - Large contact surfaces - Small profile angle, for example near the tooth root • Material properties - High residual austenite content - Hardness above 61 HRC Due to the interaction of the different influencing factors, predicting any damages is very difficult. That is why it is necessary to use suitable testing methods to ensure the quality of the component [10.2/104]. The methods available to test surface-layer properties can be divided into methods which take place after the machining process (post-process), close to the process, or during the processing (in process): Ɣ Post-process: The analyses are carried out by trained personnel after the machining process in labs set up intentionally for this purpose. - Metallography for assessing the structure and determining the structural constituents requires lab facilities and machines to produce specimens, as well as microscopes and trained personnel. Tests are done on smaller samples, which are extracted from the workpiece. Thus they are not non-destructive. The test results are clear and reproducible. - Microhardness testing to determine the surface-layer hardness requires suitable hardness testing facilities and trained personnel in the test lab. The tests can usually only be done on smaller samples, which are extracted from the workpiece. The test results are informative and reproducible. - X-ray diffraction analysis is a method that is applied in the lab and also in mobile units on location by specially trained personnel to determine the residual stress and the residual austenite content. The method utilises the property of an X-ray beam striking a crystal lattice, only to be reflected at certain angles of incidence. The angle T under which this diffraction occurs is dependent on the distance between the lattice planes and on the wavelength of the X-ray. If one knows the wavelength, it is then possible to determine the lattice distance by measuring the diffraction angle Ĭ. Mechanical stresses change the distance d between the lattice planes. A comparison of the measured lattice distance with the stress-free status and using the material parameter provides the mechanical stress in the crystal. Ɣ Methods close to the process are used in the direct vicinity of the machining process and do not depend on preparations or the conditions typical of a measurement lab. - Visual inspection can be carried out by trained personnel without the need for any technical equipment. However, it would only be possible to detect obvious discolouration due to grinding burn. The conclusions remain subjective and unclear. - Nital etching (from the term nitric acid) represents the most frequently applied method for detecting grinding burn. The degreased workpiece is submerged in a 2% solution of nitric acid for approximately 3 minutes and then “bleached” in a 3% solution of hydrochloric acid. Here the areas that are coloured brown-black by the nitric acid become light grey to light brown again. Only the parts that have grinding burn remain darkly coloured. After flushing them in a water bath, the surface is neutralised in a sodium hydroxide solution and subsequently dried and conserved. To do this it is necessary to have acid-resistant containers 652 10 Manufacturing of Cylindrical Gear Units of an adequate size and suitable means of transportation for heavy workpieces. In order to come to reliable conclusions regarding grinding damage, it is necessary to have the acid baths constantly supervised by trained personnel. On very large workpieces, which are too big to treat in a dipping tank, the chemicals can also be applied locally using a cotton wool dabber. Because of its low reproducibility, this method should only be used in exceptional cases. Nital etching is standardised in the standard AGMA 2007-C00. Nital etching is often demanded by customers for heavily loaded gearing parts in the areas of shipbuilding, utility vehicles or wind power plants. - Crack inspection is a common method for checking ground surfaces for cracks. Usually magnetic crack inspections are carried out using the leakage flux test. Here a carrier oil, to which a fluorescent iron powder has been added, is channelled over the magnetised workpiece. At a crack the course of the magnetic flux lines is disrupted. Here the flux lines exit the workpiece in an arc-shaped path and form a bridge. This effect is known as leakage flux. At the exit point, poles form, at which the fluorescent iron powder can collect. Since the leakage flux is wider than the crack, under UV light the coloured test substance also makes the smallest crack visible. After completion of the crack inspection, the workpiece is to be carefully demagnetised. The colour penetration method is also used to inspect for cracks, which makes it possible to inspect non-ferromagnetic workpieces. - Micromagnetic surface-layer analysis utilises the parameters of Barkhausen noise in ferritic steels. When ferromagnetic materials are magnetised, the magnetic flux variation creates a hysteresis loop. The curve, which at first glance appears smooth, is actually composed of incremental steps. In a spool, which is placed closed to the surface of the test sample, a pulse-like signal is generated (Figure 10.2/39) [10.2/109]. These impulses, generated in solid material, form a noise-like signal with a wide spectrum, the so-called Barkhausen noise. The residual stresses of a component (toothing) can be analysed by measurements of the intensity of the Barkhausen noise close to the surface. Here, tensional stress raises the intensity of the Barkhausen noise, and compressive stress lowers it (Figure 10.2/40). Figure 10.2/40 The effect of residual stress and hardness on the Barkhausen noise [10.2/110] Figure 10.2/39 Diagram of the measurement system for measuring residual stresses [10.2/110] 10.2 Tooth-shape Forming Methods 653 Depending on the individual case, there is a large quantity of sensors available in which the exciter brace and the sensor are mounted in a housing. To take measurements, the sensor is placed on the test surface and is then guided line by line over the test area, either mechani-cally or manually. This test method is used close to the process to monitor passes in grinding work. It does not provide absolute values, but it clearly shows changes in the material due to the ma-chining. - In addition to the phenomena listed above, micromagnetic, multiparametric microstructure analysis (3MA) utilises other physical factors such as incremental permeability, tangential field intensity and eddy currents [10.2/105]. A total of 13 parameters can be ascertained, which provide insight into the condition of the surface layer. In actual practice, the calibration of the system turned out to be a problem, so one can expect good qualitative results, but one can come to no clear conclusions regarding the quantity. Ɣ In-process monitoring utilises parameters which are determined in the work space of the machine during the processing. - Measurement of effective power and registration of process forces are used in actual practice to monitor processes, but they do not allow any clear conclusions about surface-layer damage. - Acoustic emission utilises the ultrasound which a grinding disc in contact generates. When the abrasive grain wears out or the chip space clogs with material from the workpiece, the frequency spectrum changes and allows one to draw conclusions about how the surface layer of the workpiece is being affected. - Visually monitoring the condition of the grinding disc using a triangulation sensor registers the grinding condition of the grinding disc and this provides indications about potential influences on the surface layer. 654 10 Manufacturing of Cylindrical Gear Units Table 10.2/2 Finishing Methods for Cylindrical Gearing Notes 1. Primary non-chip forming Toothed workpieces with complicated geometry, tolerances between 0.3 and 0.7% of the nominal value, usually finishing by chip removal 2. Methods involving forming Usually finishing or fine finishing by chip removal Usually subsequent machining by chip removal necessary Short cycle times, low tool costs, high precision, increase in strength, improvement in structure Area of application Cylindrical gears with low load capacity, small batch sizes Iron alloys, brass, bronze, Al alloys, low load capacity, mass production > 10000 pieces Pinion made of unalloyed steel, large series and mass production Spur gears, tapered gearing (conical) Gear tooth finishing of pre-milled cylindrical gears and thin-walled cylindrical ring parts with internal and external gearing, medium batch sizes Dimensional range, quality according to ISO 1328 & DIN 3962/63 Cast parts to approximately 100 kg l < 50 mm l/d < 2.5 Wall thickness > 2 mm 0.1 to 1 kg d 25 mm d 110 mm (spur gear) d 240 mm (tapered gearing); quality achievable is 6 da 214 mm z = 10 to 60 lz 900 (1450) mm Quality achievable is 6 Method / mode of operation Precision casting Lost-wax casting technique Typical sintering procedure: 1. Mixing 2. Volumetric dosing 3. Hardening of the moulded blank Extrusion press Extrusion of round steel through negative profile above recrystallisation temperature Precision forging Hot pressing individually in die Cold rolling GROB method No. 1 2 3 4 5 10.2 Tooth-shape Forming Methods 655 Table 10.2/2 (continued) Notes 3. Machining methods with defined geometric cutting edge Area of application Large module, unhardened and hardened gearing, single-item and small series production Unhardened, non-hobbable gearing, single-item to medium series production Hardened gearing, large series production Dimensional range, quality according to ISO 1328 & DIN 3962/63 m < 50 mm d 14 000 mm Quality achievable is 5 m < 12 (14) mm d < 1250 mm Quality achievable is 6 m 3 mm d < 160 mm Quality achievable is 6 Method / mode of operation Shaping with rack-type tool Rolling process with rack-shaped tool Shaping Rolling process with cylindrical gear-shaped tool Skiving Helical pitch process with cylindrical-gear- shaped tool No. 6 7 8 656 10 Manufacturing of Cylindrical Gear Units Table 10.2/2 (continued) Notes High tool costs for skiving of hardened gearing Low-distortion hardening methods necessary; tool design is workpiece or workpiece-group specific Area of application Unhardened and hardened gearing, single-item to large series production Unhardened and hardened gearing, medium to large series production Dimensional range, quality according to ISO 1328 & DIN 3962/63 m = (0.1) 0.2 to 32 (59) mm 0 < d < 5000 (14000) mm Quality achievable is 6 (7) m < 8 (20) mm d < 320 (6000) mm Quality achievable is 5 dependent on the accuracy of pre-machining Method / mode of operation Hobbing Helical pitch process with worm-shaped tool Roll shaving Helical pitch process with cylindrical-gear- shaped tool Axial position along lines of No. 16/17 No. 9 10 10.2 Tooth-shape Forming Methods 657 Table 10.2/2 (continued) Notes 4. Machining methods with undefined geometric cutting edge 4.1. Grinding methods Low productivity due to point contact between grinding wheel and workpiece and because of dry grinding Pay attention to possible grinding burn or grinding cracks Pay attention to possible grinding burn or grinding cracks Area of application Unhardened and hardened precision gearing, single-item and small series production Unhardened and hardened gearing, predominantly large dimensions, single-item to small series production Unhardened and hardened gearing, small to large series production Dimensional range, quality according to ISO 1328 & DIN 3962/63 m 36 mm d < 4734 mm Quality achievable is 2 m 40 mm d < 4000 mm Quality achievable is 3 (4) depending on the size of the gearing m 8 mm d < 820 mm Quality achievable is 3 Method / mode of operation Discontinuous generating grinding MAAG discontinuous generating grinding with plate-shaped grinding wheels in 0 method, Į method Discontinuous generating grinding Discontinuous generating grinding with conical grinding wheel (NILES, H FLER) Continuous generating grinding Generating grinding with cylindrical grinding worm (REISHAUER) No. 11 12 13 658 10 Manufacturing of Cylindrical Gear Units Table 10.2/2 (continued) Notes Galvanically bonded CBN grinding wheel and dressable, ceramically bonded CBN grinding wheel High-productivity method, particularly in automotive gear manufacturing Area of application Unhardened and hardened gearing, small to large series production Hardened helical gearing, large series to mass production Dimensional range, quality according to ISO 1328 & DIN 3962/63 m 12 (18) mm d 1250 mm Quality achievable is 2 As additional setup: m 25 mm d 4000 mm Quality achievable is 4 (5) m 3 mm d 200 mm Quality achievable is 4 Method / mode of operation Discontinuous profile grinding helical process with radial- tangential- profile grinding disks profile grinding disks (KAPP, NILES, PFAUTER) (NILES, KAPP) Continuous profile grinding Plunge grinding in helical process with globoid grinding worm (REISHAUER) No. 14 15 10.2 Tooth-shape Forming Methods 659 Table 10.2/2 (continued) Notes 4.2. Honing methods With CBN honing wheels, machining allowances of up to 0.15 mm are removed (compensation for heat- treatment distortion). Roll honing with a corundum honing ring usually leads only to reduced roughness and is done after case hardening, sometimes on ground gearing. The pre-machining before roll honing has a considerable effect on the final accuracy, particularly the cumulative pitch deviation. 4.3. Lapping methods Area of application Hardened gearing, large series to mass production Hardened gearing, primarily to improve surface quality and quiet running, large series to mass production Primarily to improve the quiet running and surface quality of large-scale mounted gears Dimensional range, quality according to ISO 1328 & DIN 3962/63 m 6 mm d 280 mm Quality achievable is 5, dependent on gearing geometry and accuracy of preparations m < 8 mm d < 320 mm Quality 6 to 7 Suitable gear unit dimensions Quality: No significant improvement compared to rough machining quality Method / mode of operation Roll honing (shave grin ding, hard shaving) General screw rolling process with externally toothed honing wheel made of CBN (galvanically bonded) or corundum (resin-bonded) Roll honing (F SSLER method) General screw rolling process with internally toothed honing wheel made of corundum Run-in lapping Roll lapping in the gear unit or on a lapping machine with gear pair of the gear unit No. 16 17 18 660 10 Manufacturing of Cylindrical Gear Units Table 10.2/3 Tool Properties for the Finishing Methods for Cylindrical Gears Tool costs Wear costs per workpiece Dressing/s harpe- ning tool ↓ ↓ ↓ ↓ ↓ ↓ ↓ ↓ - Machi- ning tool ↓ ↑ ↑ ↓ ↓ ↓ ↓ ↓ ↑ Costs for purchase Dressing/ sharpe- ning tool ↓ ↓ - ↓ (↓) (↑) (↓) (↑) ↓ (↓) (↑) - Machi- ning tool ↑ ↑ ↑ ↓ ↑ ↓ ↑ ↓ ↑ Tool geometry Dressing/sharpening tool Work- piece dependent - - - - - - - (+) - Work- piece in- dependent + + + + + + (+) - Machining tool Work- piece dependent (+) - (+) - - - - - (+) Work- piece in- dependent (+) + (+) + + + + + (+) Cutting edge Cutting material HSS HM HM Co CBN-G CBN-K Co CBN-K Co CBN-G Type Geome- trically undefined - - - + + + Geome- trically defined + + + - - - Tool enveloping body External cylindrical gear Cylindrical worm External cylindrical gear Rack, plate disc Rack, conical grinding wheel Cylindrical worm Methods Roll- shaving h Skiving h Hard skiving h Disconti- nuous generating grinding h, w Disconti- nuous generating grinding h, w Continuous generating grinding h, w No. 1 2 3 4 5 6.1 6.2 Swiss Precision Gear Grinding Ever since we have invented continuous gene- rating grinding, we have set the standard in the hard-finishing of gears. Reishauer AG Wallisellen / Schweizwww.reishauer.com 10.2 Tooth-shape Forming Methods 663 Table 10.2/3 (continued) Tool costs Wear costs per workpiece Dressing/ sharpe- ning tool ↓ ↓ - ↓ ↓ ↓ ↓ Cubic boron nitride with galvanic bonding Cubic boron nitride with ceramic bonding Machi- ning tool ↓ ↓ ↑ ↓ ↓ ↑ ↓ Costs for purchase Dressing/ sharpe- ning tool (↓) (↑) (↓) (↑) - ↑ ↑ ↑ ↑ Machi- ning tool ↓ ↑ ↑ ↓ ↓ ↑ ↓ Tool geometry Dressing/sharpening tool Work- piece dependent (+) (+) - + + + + CBN-G CBN-K Work- piece in- dependent (+) (+) - - - - - HSS High-performance high- speed steel HM Carbide Co Corundum Machining tool Work- piece dependent + + + + - + - Work- piece in- dependent - - - - + - + Cutting edge Cutting material Co CBN-K CBN-G Co Co (hard finishing) CBN (hard shaving) Co h Hard machining w Soft machining Type Geome- trically defined + + + + Geome- trically defined - - - - Tool envelo- ping body Single or multiple tooth space profile Globoid worm External gear Internal gear + Applicable - Not applicable () Alternative possible Methods Discontin uous profile grinding h,w Conti- nuous profile grinding h Roll honing h No. 7.1 7.2 7.3 8 9.1 9.2 9.3 664 10 Manufacturing of Cylindrical Gear Units 10.2.7 Symbols and Symbol Explanations of Sections 10.1 and 10.2 10.3 Heat Treatment According to DIN EN 10052:1994-1, “ heat treatment ” is a “ series of heat treatment steps, during which a workpiece is wholly or partly subjected to time-temperature sequences in order to bring about a change in its properties and/or structure. If necessary, the chemical composition of the material can be changed during the treatment ”. The standard is currently being revised, and ISO 4885 is planned as a replacement (Ferrous Products – Heat Treatments – Vocabulary). The phase and structural changes occurring in the ferrous material as a result of heat treatment can be demonstrated in a simplified manner by the iron-carbon diagram, and particularly by the time-temperature transformation graph (TTT graph) and the time-temperature resolution graph. Detailed information can be found in [7.4/11] and [10.3/1]. The method is demonstrated in a schematic temperature-time curve in Sections 10.3.1 and 10.3.2. Of the variety of heat-treatment methods defined by DIN EN 10052, only a few are used for gears. These are used to produce a material condition that is suitable for manufacturing and/or appropriate for the strain . A mm2 area of cutting cross section b mm tooth face width b mm cutting width d mm reference diameter of the gear da mm tip diameter of the gear da0 mm tip diameter of the tool df mm root diameter of the gear dRoh mm diameter of the blank h mm cutting thickness h - hard machining hmin mm minimum cutting thickness e mm eccentricity Fn N normal force Fc N cutting force Fp N passive force Fp ȝm cumulative pitch deviation Fv N feeding force Fz N cutting force fp m single pitch deviation lh mm stroke length m mm module mn mm normal module nAG - number of necessary working steps for manufacturing the particular functional surface nL - number of workpieces per production batch (batch size) n0 min–1 tool speed nP - number of palettes per production batch Pc kW cutting/grinding power Pz kW chip-removal power Qw mm3/s chip-removal rate Ra ȝm arithmetic mean roughness Rz ȝm mean roughness sa mm axial feed vc m/s; m/min cutting velocity vfa mm/min axial infeed velocity vfd mm/min diagonal infeed velocity vfr mm/min radial infeed velocity vft mm/min tangential infeed velocity vfw mm/min generating infeed velocity (table sliding velocity) w - soft machining z - number of gear teeth z0 - number of tool starts Į pressure angle of the gear ȕ helix angle of the gear ȕ 0 helix angle of the tool ȕ2 helix angle of the workpiece Ȗ angle between axis of tool and workpiece in continuous generation grinding Ȗ 0 lead angle of the tool į diagonal angle in shaving İĮ - transverse overlap İȕ - axial overlap Ș0 tilt angle of the hob Ȉ0 axis cross angle Ȧ min–1 angular velocity 10.3 Heat Treatment 665 Each method includes the heat-treatment steps heating, holding, and cooling (refer to DIN EN 10052). Upon heating , the heat is transferred indirectly by radiation and convection, or directly with inductive or conductive methods through high-energy radiation (electron beam and laser beam) or through the ionised gas condition (plasma). A structural change can occur in the workpiece depending on the level of the selected target temperature. An undesired change in the chemical composition, especially in the surface layer, should be avoided. The same applies to internal stresses. The holding sub-step is used to adjust the near-equilibrium structural condition, to reduce resi- dual stresses or, during concomitant element diffusion in the surface layer (e.g., carbon during carburisation), to achieve a desired concentration curve. There are many mathematical models, particularly for the prediction of the carbon profile during carburisation, which are widely used in systems engineering. The real focus is the cooling sub-step because, in most cases, the condition suitable for manu- facturing or appropriate for the strain is adjusted here. While cooling is usually performed slowly, for example, after annealing, tempering and nitriding, accelerated cooling – quenching – is necessary for all hardening methods. The heat extraction is carried out in aqueous solutions, oils and salt baths, increasingly in pressurised gas streams and rarely in fluid beds. The three cooling phases of film boiling, nucleate boiling and convection occur in aqueous solu-tions and oils. The temperature-time behaviour of these three phases has a decisive influence on the result of treatment. In salt baths, the heat extraction is carried out mainly by conduction; in gases it is performed mainly by convection. The advantage of this media is generally the improved distortion behaviour. The three heat-treatment steps are associated with structural changes; the first and last are associated with changes in temperature. The results of this are residual stresses. In particular, in the cooling/quenching step, the structure and the residual stresses are fixed by temperature dif- ferences between the edge and core, which affect the properties and the distortion of the detailed component geometry. The properties can be influenced by further heat-treatment steps, such as tempering. The distortion that generally causes the highest costs is a system property, which results from • the construction, • the material, • the manufacturing by primary shaping and forming or • all manufacturing steps before and during the heat treatment. Extensive studies commissioned by the German Research Foundation (DFG) in the Colla- borative Research Centre SFB 570 “Distortion Engineering” (Foundation Institute of Materials Science, Badgasteiner Str. 3, 28359 Bremen, Germany) are carried out with the aim of mathe-matically predicting the minimisation of the distortion of components. The “distortion” is defined as “the sum of all dimensional changes of a component due to process-related thermal, mechanical and chemical effects” [10.3.2]. In the practice of gear heat treatment, the distortion is measured to determine measures for rational and custom manufacturing. Residual stress measurements are carried out in scientific institutions, specialised companies and operating laboratories. 666 10 Manufacturing of Cylindrical Gear Units 10.3.1 Heat Treatment Suitable for Manufacturing Heat treatment suitable for manufacturing is used primarily to homogenise the structure, improve machinability, reduce residual stresses and adjust the initial condition for the final heat treatment. Annealing methods, quenching and tempering, thermomechanical treatment and, if necessary, combinations thereof, for example, quenching and tempering and stress relieving, are used. Annealing, quenching and tempering or thermomechanical treatment on the blank is performed by the blank manufacturer. Thermomechanical treatment is referred to in Section 10.3.1.1. Quen-ching and tempering are described in detail in Section 10.3.2.1. The heat treatment suitable for manufacturing or related to manufacturing, which is appropriate for the individual groups of materials, is summarised in Table 7.4/3a, b. In the case of minor gear stress, the heat treatment suitable for manufacturing can simultaneously represent the final state and the heat treatment appropriate for the strain. After each heat treatment suitable for manufacturing that is not carried out under an inert gas, the blanks or workpieces must be descaled before the subsequent mechanical machining. 10.3.1.1 Normalising According to DIN EN 10052, “ normalising is heat treatment consisting of austenitising and followed by cooling in still air ” (refer to ISO 4885). Mainly unalloyed tempering steels (DIN EN 10083-2) and all case-hardened steels according to DIN EN 10084 are supplied by the blank manufacturer normalised (condition +N). The aim of normalising is to eliminate the cast and rolled structure through a repeated structural transformation and to produce ferrite grains (light) and pearlite grains (dark) that are as equal in size and as evenly distributed as possible (black and white structure, similar to Figure 10.3/1b), and also to reduce any residual stresses. According to Figure 10.3/1a, heating is performed slowly up to 650 C. The holding time approximately 50K above A c3 is short at 30 minutes. The normalising temperature TN can be cal- culated from the Ac3 temperature with the known chemical analysis of case-hardened and tempering steels [10.3/3]. b) Figure 10.3/1 a) Schematic time-temperature curve of the normalising of hypoeutectoid steel; b) normalised structure of steel 20MnCr5 (880 C/30 min/air up to 650 C/further cooling in the furnace), V = 500:1 10.3 Heat Treatment 667 According to DIN EN 10052, the A c3 temperature is the one “at which, during heating, the trans- formation of the ferrite into austenite ends”. Normalising temperatures are not specified in DIN EN 10083-2 and DIN EN 10084. The cooling must be slow enough to avoid the formation of bainite or martensite. Thermomechanical treatment with complete recrystallisation of the austenite, the normalising transformation, can also be considered instead of normalising (refer to DIN EN 10052). 10.3.1.2 Pearlitising DIN EN 10052 refers to pearlitising, the isothermal transformation in the pearlite stage, as an- nealing consisting of austenitising, followed by cooling to a temperature in the pearlite stage and holding, so that the austenite completely transforms into ferrite/pearlite or cementite/pearlite . Pearlitising is carried out specifically for low-alloyed case-hardened steels with the aim of pro- ducing a homogeneous structure with good machinability, which is also important for low-distortion case hardening. Figure 10.3/2 Schematic time- temperature curve of the pearlitising of low-alloyed case-hardened steel After heating to the normalising temperature T N (Figure 10.3/2), it is cooled to below 700 C so rapidly that no ferrite is produced, held isothermally until complete transformation and then further cooled as desired (refer to [7.4/11, Volume 2]). 10.3.1.3 Soft Annealing According to DIN EN 10052, soft annealing is referred to in very general terms as “ heat treatment to reduce the hardness of a material to a predetermined value ”. Tempering steels (DIN EN 10083), case-hardened steels (DIN EN 10084) and nitriding steels (DIN EN 10085) are supplied soft annealed. The aim of soft annealing is the transformation of carbide into a spherical form in the soft ferritic matrix in order to improve the machinability and cold formability, and it is also a requirement for a faster and more uniform austenitising upon hardening. According to Figure 10.3/3a, heating is carried out up to just below A c1 with holding for at least two hours. The soft-annealing temperature TG can be calculated from the chemical analysis of the above-mentioned steels using the Ac1 temperature [10.3/3]. According to DIN EN 10052, the A c1 temperature is the one at which, during heating, the formation of austenite begins. Soft annealing temperatures are not specified in DIN EN 10083, DIN EN 10084 and DIN EN 10085. After soft annealing, cooling is carried out slowly in the furnace or cooling pit. Case-hardened steels can often be poorly machined in the soft-annealed condition. 668 10 Manufacturing of Cylindrical Gear Units Figure 10.3/3 a) Schematic time-temperature curve of soft annealing; b) soft-annealed structure of the steel 42CrMo4 (720 C/4 hours/furnace air), V = 500:1 10.3.1.4 Stress-Relief Annealing In principle, stress-relief annealing is possible in all gear materials. According to DIN EN 10052, it is “ heat treatment consisting of heating and holding at a sufficiently high temperature and the subsequent adequate cooling in order to substantially reduce internal stresses without significantly changing the structure ”. The aim is to reduce residual stresses that are higher than the yield stress at the temperature of stress-relief annealing through plastic deformation. The higher the temperature can be selected, the more stresses are relieved. Depending on the material and purpose, stress-relief annealing is carried out between 450 C and 650 C (refer to Figure 10.3/4). Heating must be carried out so slowly that the hot yield stress is achieved uniformly over the entire cross section of the gear. The temperature level (max. 650 C) and holding time depend on the stability of the previous structural condition. Cooling must be carried out in a furnace in order to avoid new stresses through excessively rapid cooling. Figure 10.3/4 Schematic time-temperature curve of stress-relief annealing For quenched and tempered steels, it applies that the temperature of the stress-relief annealing must be at least 30K below the tempering temperature of the quenching and tempering. The desired reduction in residual stress is linked to distortion. An appropriate stock allowance must be provided on workpieces that have already been machined so that finishing in order to correct distortions is possible after the stress-relief annealing. Straightening must not be carried out because it would result in tensile residual stresses. b) 10.3 Heat Treatment 669 10.3.2 Heat Treatment Appropriate for the Strain The aim of heat treatment appropriate for the strain is, in connection with the suitable material, to generate a property level that is higher at any location on the gear than the level of external stress applied there (refer to Section 7.4/1). Quenching and tempering, surface-layer hardening, and the thermochemical processes of case hardening, carbonitriding, and nitriding are mainly used for the heat treatment of gears (refer to Table 7.4/3a, b). Combinations of these methods are possible. The result of the heat treatment appropriate for the strain has a direct influence on the load capacity value of the gearing. Approximate values, which reflect the relationship between material, distinguishing dimension and load capacity for heat treatment appropriate for the strain, are contained in Tables 7.4/14a to c, 7.4/15, 7.4/20, 7.4/23a, b, and 7.4/27. 10.3.2.1 Quenching and Tempering According to DIN EN 10052, “ quenching and “tempering ” is “ hardening and tempering at a higher temperature in order to achieve the desired combination of mechanical properties, particu- larly high toughness and ductility ”. Depending on the case of application, all gear materials that are suitable for thermal surface-layer hardening are suitable for quenching and tempering according to Table 7.4/3b. Tempering steels according to DIN EN 10083, nitriding steels according to DIN EN 10085 and iron casting materials according to DIN EN 10293 are supplied quenched and tempered (condition +QT). The quenching and tempering process of case-hardened steels according to DIN EN 10084 is used to avoid a banded structure from normalising and thus to improve the machinability, for example, during broaching, or to reduce stresses and to make the distortion behaviour of large workpieces more effective and uniform (tip diameter d a > 1 m) according to [10.3/4]. Notes: For materials for quenched and tempered gears, refer to Section 7.4.2.4; for information on heat treatment, refer to 9.3; for heat treatment temperatures, refer to Table 10.3/6a to c at the end of the section . For hardening, the workpieces are heated in one step, depending on the material (unalloyed steels), or a holding step at around 400 C is added (Figure 10.3/5). Holding for austenitising takes place at 30 or 50K above A c3. It is subsequently quenched in water or oil and then tempered below Ac1. The fine-grained tempered structure of the steel 42CrMo4 can be seen in Figure 10.3/5b. Figure 10.3/5 a) Schem atic time-temperature curve of the quenc hing and tempering of low-alloyed steel; b) tempered structure of the steel 42CrMo4 (850 C/20 min/oil + 600 C/2 hours/air, V = 500:1) b) 670 10 Manufacturing of Cylindrical Gear Units The basic principles of the technological process for the quenching and tempering of components are described in detail in DIN 17022-1. In the practice of gear tempering, three technological variants are common: 1. The gear blank is supplied quenched and tempered with a greater machining allowance. 2. The soft-annealed or normalised blank is pre-machined and quenched and tempered with stock allowance without gearing. 3. The component toothed before quenching and tempering is quenched and tempered (distortion!). In the first and second variant, it must be noted that with R m > 1000 N/mm2 the machinability can become worse. Due to the growing application of gears with hardened surface layers, quenching and tempering loses importance as a final state appropriate for the strain. Exceptions are, for example, very large gears. However, quenching and tempering is important for the production of a high tooth core strength for subsequent surface-layer hardening or nitriding. The structure formation achieved in the quenching process is decisive for the result of quenching and tempering. Since the checking of the structure is very complex after hardening, the hardness is measured in the quenched state and set in ratio to the maximum hardness of the given steel, which is dependent only on the carbon content (refer to degree of hardening R in Section 7.4.2.4). In the heat-treatment shop, the hardness must therefore be measured in the quenched condition in order to control the subsequent tempering process. The hardness after tempering (tempered hard-ness) must be calculated at a given tempering temperature depending on the degree of hardening R with a known chemical analysis [7.4/12]. Figure 10.3/6 depicts the tempering graph for the steel 42CrMo4 of an average chemical analysis. The required hardness at a given tempering temperature by reference to the degree of hardening is independent of the distinguishing dimension (refer to [7.4.1.2]). By choosing a correspondingly low tempering temperature, the drawing requirement of the tempered hardness and strength can often still be fulfilled at a degree of hardening of R < 0.6. The negative effects of increasing the ferrite/pearlite content in the mixed structure should not be compensated for. Despite the achievement of the predetermined hardness, reductions in the toughness and fatigue strength are the result. Figure 10.3/6 Tempering graph of the steel 42CrMo4 10.3 Heat Treatment 671 The properties of the quenched and tempered gears are significantly influenced by • the hardenability of the respective material (refer to Section 7.4.1.2 and 7.4.2.4) and • the quenching and tempering parameters described. All further strength values for determining the gear endurance limit are derived from the tempered hardness, as explained in Section 7.4.2.4. In certain cases, a high toughness at room temperature or ambient temperature (down to –50 C) is required of gears made of tempering steels. The toughness is independent of the type of steel when the hardenability and grain size of the steel after hardening guarantees fine-grained martensite. Figure 10.3/7 shows the structural dependence of the toughness on tempering steels ([10.3/3, p. 801]). Notes on the use of gears made of tempering steel at low ambient temperatures can be found in [7.4/10]. The FATT (Fracture Arrest Transition Temperature) is the temperature for 50% ductile fracture surface of the notched specimen. Structure proportions after quenching and tempering: F - ferrite KG - grain size P - pearlite ASTM 3 - coarse grain with grain size 3 B - bainite ASTM 7 - fine grain with grain size 7 M - martensite ([10.3/3], p. 801) Figure 10.3/7 Transition temperature for 50% ductile fracture as a function of yield stress and structure The strength and hardness after the quenching and tempering of case-hardened steels according to DIN EN 10084 as a “special treatment” cannot be predicted according to the documents for tempering steels. The main reason is the lower carbon content compared with tempering steels. The strength and hardness of quenched and tempered case-hardened steels with the usual tempering temperatures for tempering steel is significantly below the values of the core hardness after case hardening. Depending on the tempering temperature, 20 to 30% can be given as a rough approximate value. The time-temperature curve is the same as illustrated schematically in Figure 10.3/5. 672 10 Manufacturing of Cylindrical Gear Units After the hardening from the core hardening temperature, tempering is carried out between 500 C and 700 C (refer to Table 10.3/2b). The result of a tempering experiment with 20MnCrS5 is shown in Figure 10.3/8. Figure 10.3/8 Result of the tempering experiment with 20MnCrS5 at different tempering temperatures 10.3.2.2 Austempering According to DIN EN 10052, austempering, isothermal transformation in the bainite stage , is a “heat treatment consisting of austenitising and the subsequent staged quenching to a temperature above M s (the beginning of the transformation from austenite to martensite) at such a rate that any formation of ferrite or pearlite is avoided, and holding at that temperature to partially or completely transform the austenite into bainite ”. (The term is also formulated analogously in ISO 4885, Austempering). Notes: For materials, refer to Section 7.4.2.4; for information on heat treatment, refer to 9.2. Metallurgical basic principles of the austempering of steels are to be taken from continuous and isothermal time-t emperature transformation graphs [7.4/11, Volume 1, refer to “Tempering steels” in the Krupp steel catalogue “High quality steels”, 2000 edition]. These apply only to a specific chemical composition and certain austenitising conditions (temperature and holding time). In addition to alloyed tempering steels, carburised case-hardening steels, carbonitrided case-hardened and tempering steels and spheroidal graphite cast iron are suitable for the bainitic transformation after continuous quenching. A prerequisite for the austempering of tempering steels is a temporally broad austenitic region, the stability of which increases with an increasing amount of hardenability-enhancing alloying elements. It is apparent from such a graph (refer to “Tempering steels” in the Krupp steel cata-logue “High quality steels”) that, for example for steel 34CrNiMo6, the bainite region is between approximately 330 C (martensite start temperature) and approximately 500 C. After austeni-tising at 830 C, the minimum time up to the start of the austenite-bainite transformation at ap-proximately 380 C is 40 seconds. During this time, the component must be quenched and held at this temperature for approximately 25 minutes (shortest time interval). Subsequent cooling is optional. This results in bainite through to the core of the workpiece without ferrite or pearlite. 10.3 Heat Treatment 673 Figure 10.3/9 shows the schematic temperature-time curve for the steel 34CrNiMo6. After a possible holding at 400 C, the steel is heated to the hardening temperature (at least 830 C/20 min) under inert gas or in a neutral salt bath and then abruptly quenched in a salt bath (possibly with the addition of water), the temperature of which is in the range of the formation of bainite, and then held until complete transformation at 380 C/25 min. The struc-ture (hardness H 380 HV) consists of cementite plates with embedded ferrite. Further cooling is optional. Significant advantages of austempering over quen- ching and tempering are as follows: • It provides high static, dynamic, and cyclical strength. • There is minimal distortion. • The dimensional change is reproducible, so components are only austempered after teeth manufacturing. • Tempering is omitted, so there is no risk of temper brittleness. After quenching in a salt bath, thorough washing with additional wastewater treatment is necessary. The treatment is carried out either in a shaft furnace under an inert gas with transfor-mation in a salt bath or in a chamber furnace with integrated salt bath [10.3/5]. The disadvantage of salt-bath quenching can be circumvented through so-called “dry” austempering [10.3/6] using high-pressure gas quenching, which is still in the experimental stage. The transformation takes place in furnaces pre-heated to the bainite temperature. 10.3.2.3 Surface Hardening According to DIN EN 10052, surface hardening is to be understood as “ hardening with austeni- tising limited to the surface layer ”. It is useful to characterise the term by the type of heating, for example, flame hardening, induc- tion hardening, electron-beam hardening, or laser-beam hardening . Electron or laser beams are not yet used for standard gear hardening because the generation of the conformal hardness layer is only conditionally possible. Other high-energy methods, such as impulse hardening, are not suitable for gears. Notes : For materials for surface-hardened gears, refer to Section 7.4.3.1; for heat treatment data, refer to Section 9.3; for heat-treatment temperatures, refer to Table 10.3/6 at the end of the section. The purpose of the method is to give the workpiece a high surface hardness dependent primarily on the carbon content through partial surface-layer heating followed by external or self-quen-ching, combined with a hardness transition to a predominantly quenched and tempered core material that is as gradual as possible. The interaction of the increase in hardness and transfor-mation-related residual stress results in the increased fatigue strength, and the high degree of hardness results in the improved resistance to abrasive wear. The technological sequence of the methods of surface-layer hardening is described in detail in DIN 17022-5. Figure 10.3/9 Schematic time-temperature curve of austempering of the steel 34CrNiMo6; F - ferrite, P - pearlite, B - bainite, M - martensite 674 10 Manufacturing of Cylindrical Gear Units Flame and induction hardening require external quenching through a liquid medium. The beam methods, which until now have rarely been used for gears, work with self-quenching due to the low heated volume and even higher temperature gradients (refer to Figure 10.3/10). Very fine martensite occurs as a result. The hardening temperature of flame or induction hardening T H-FI results from the selected method variant. A high hardening temperature is advantageous for austenitising over a short time period (e.g., tooth space hardening), whereas a lower hardening temperature is advantageous for austenitising over a longer time period (e.g., spin hardening). It should be noted that the austenitising temperature increases with the heating rate [7.4/11, Volume 3]. Ultimately, the precise temperatures measured with non-contact measurement devices that are to be selected are an operational experience. After the flame or induction hardening, annealing is carried out immediately at a low tempera-ture for 1 hour (150 C to 180 C, rarely up to 220 C). It applies to electron- and laser-beam hardening that the hardening temperature approaches the melting temperature with increasing alloy content. The limit hardening temperature of tempering steels should be at least 250K below the melting temperature of pure iron. Holding does not occur at this temperature. Tempering is not common after electron- or laser-beam hardening. The surface-hardening methods with flame or induction heating are carried out with equipment similar to machine tools. Here, the machine realises the movements, and the tool with which the heat is generated and the quenching is carried out, the burner or the inductor, has a variety of forms with an integrated cooling sprinkler. Figure 10.3/10 Schematic time-temperature curve of the surface layer hardening: a) flame or induction hardening with external quenching and tempering; b) electron- or laser-beam hardening with self-quenching; TS - melting temperature of pure iron, 1536 C; TH-FI - austenitising temperature of flame or induction hardening in C; TH-EL - austenitising temperature of electron- or laser-beam hardening in C; Ttemp - tempering temperature C; tH - holding time in hours or seconds The advantages are • the usually finer martensitic hardening structure compared to volume hardening, • the good reproducibility of the coating properties with the appropriate expertise, • the low or no surface oxidation, • the targeted partial hardening, 10.3 Heat Treatment 675 • the method-dependent hardening depth, which can be controlled over a wide range, • the lower distortion with a positive impact on the grinding costs compared to volume hardening, • the fast operational readiness of the previously established system, • the possibility of computerised process control and monitoring, • the possible integration into manufacturing lines, • the low energy consumption, especially in the case of small piece numbers and large workpiece dimensions or weights, and • the cleanliness and environmental friendliness. Disadvantages are • the lower load capacity of surface-hardened gears compared to case-hardened gears (refer to Figure 7.4/22), • the emergence of more abrupt transitions of the distribution of hardness compared to thermo- chemical methods, • the limited application for complex workpiece geometries and small lot sizes, and • the usually much higher plant costs compared to furnaces. In many cases, flame and induction hardening are equal in terms of the treatment result and the achievable load capacity, so clear differences and advantages of the methods are only visible in the specific component and the resulting treatment. The shot peening of the tooth root surface in order to improve the strength at the root is advan- tageous in fillets that have not been hardened. Flame hardening: The heating for the austenitising of the surface layer is carried out with natural gas-oxygen or acetylene-oxygen mixtures. Special burners, which are adapted to the tooth profile, or ring burners are required for the heating. After reaching the austenitising temperature monitored with a radiation pyrometer, either the tooth space, the individual tooth or the entire gear is quenched after the method-dependent holding time. The corresponding working methods typical for gears, which are also suitable for induction hardening, are • tooth space hardening including and excluding tooth root hardening in the feeding method, • two-flank hardening excluding tooth root hardening in the feeding method, and • spin hardening including and excluding the root, which can take place as a stand-circulation method (ring burner stands, gear spins) or as a feed-circulation method (ring burner moves from top to bottom, gear spins). In the case of tooth space hardening, two-flank hardening, and the feeding-circulation method, the cooling sprinkler is downstream of the burner or inductor. Water or an aqueous polymer solution serves as a quenching liquid. The gear is quenched in a bath with water, an aqueous polymer solution or oil only in the stand-circulation method. The aim of flame hardening is a conformal formation of the hardness layer, which, however, can be approximately realised only with gears with large modules through tooth space hardening including the root. All other variants differ from this. This results in the different tooth root endurance strength [7.4/73.1], [7.4/73.2]. The possible “hardness contours” and application limits of the technological variants with their effect on the gear endurance limit are shown and described in Table 10.3/1, according to [10.3/4, p. 661]. 676 10 Manufacturing of Cylindrical Gear Units The surface hardening depth is determined by the depth of the surface layer heated at the hardening temperature, the hardenability of the material and the effect of the quenching medium. Thus, for example, through tooth space hardening with a given burner output, the surface hardening depth is controlled by the feed rate of the burner through the tooth space, depending on the module. The macro etching figure (hardness contour) and the microstructure from the edge and from the transition to the core of a flame-hardened tooth ( m n = 8 mm, steel 42CrMo, quenched and tempered) with the corresponding hardness-depth curves of the flank and root can be seen in Figure 10.3/11. According to the macro etching image, almost four times the surface hardening depth can be found in the middle of the flank compared to the tooth root, which is confirmed by the hardness-depth curves. When the core hardness is high (> 300 HBW) or the heating time is long, a low hardness can occur at the transition zone of the hardness-depth curve compared to the core hardness, colloquially known as a hardness sack, which can be seen in Figure 10.3/11 (refer to DIN 17022-5). This effect also applies to induction hardening. The gears must be tempered at a low temperature (150 C to 220 C/1 hour) immediately after the surface-layer hardening. A drop in hardness of 5 to 7 HRC is possible compared to the quenching hardness. Steel-dependent surface hardening for the initial quenched and tempered state of the material is shown in Table 7.4/8a, b. Figure 7.4/15 provides information on the determination of the surface-hardening depth. Module-dependent surface-hardening depths are shown in Table 7.4/17. Figure 10.3/11 Macro etching figure (V = 1:2), microstructure (V = 500:1) and hardness depth curves of a flame- hardened tooth made from steel 42CrMo4 10.3 Heat Treatment 677 Table 10.3/1 Scope of Application of Flame and Induction Hardening [10.3/4, p. 661] Variant and hardness contour Effect Scope of application Flame hardening HF induction hardeningMF induction hardening MF/HF induction hardening a) Single-shot gear hardening excluding the root Increase of σHlim compared to quenching and tempering; σHlim 20% lower than after case hardening; σ FE max. such as after quenching and tempe- ring if the distance from the contour end to the root circle is min. 0.2 m n up to 450 mm (larger with special machines) m n 6 (12) mm 10 to 150 mm depending on generator power and gearing width, m n 2 mm 35 to 500 mm depending on generator power and gearing width, m n = 2.5 to 6 mm Not applicable b) Single-shot gear hardening including the root Increase in σ Hlim com- pared to quenching and tempering; σHlim 20% lower than after case hardening; σ FE approx. 30% higher than after quenching and tem-pering, 30% lower than after case hardening up to 450 mm m n 6 (10) mm 10 to 150 mm depending on generator power and gearing width, m n 5 mm 35 to 500 mm depending on generator power and gearing width, m n = 2.5 to 6 mm Not applicable c) Conformal single- shot gear hardening Equal σHlim such as after single-shot gear hardening without fil- let; σ FE about 15% lower than after case hardening; requirement: tempered structure Not applicable Not applicable Not applicable 250 mm m n = 2.5 to 5 mm d) Tooth-by-tooth gear hardening, two-flank hardening refer to a) Unlimited diameter mn 6 mm Diameter dependent on machine mn 2 mm Diameter dependent on machine mn 5 mm Not applicable e) Tooth-by-tooth gear hardening, tooth space hardening refer to b) Unlimited diameter mn 10 mm Diameter dependent on machine mn 4 mm Diameter dependent on machine mn = 5 to 30 mm Not applicable Gear design and variations of the surface-layer hardening influence the distortion. Special design advice can be found, for example, in [10.3/7], [10.3/8]. As a general conclusion for a given gear design, it is considered that the greater the hardness case volume ratio to the tooth or gear volume, the greater the distortion. 678 10 Manufacturing of Cylindrical Gear Units Advantages of flame hardening are • the achievement of very large surface-layer hardening depths (refer to Table 7.4/17) and • the low equipment costs compared to other surface-hardening methods. Disadvantages are • the costly manufacturing of specific burners adapted to the tooth or gear form and • the mostly non-conformal case and the reduced strength at the root compared to nitrided and especially case-hardened gears. In some cases, the combination of thermochemical and thermal surface-layer hardening is used. Examples are carburised gears, which are subsequently advantageously treated in spin hardening [7.4/75], and flame-hardened gears, the tooth root of which was not hardened and in which the tooth root endurance strength can be significantly increased with minimal distortion through subsequent nitriding [10.3/9]. It should be noted that the pitting endurance limit of the flame-hardened teeth is reduced by the effect of tempering during nitriding. Induction hardening : Induction hardening has replaced flame hardening in many cases. Detailed information on the induction hardening of gears is available, among others, in [7.4/73] and [10.3/10]. Heating up to the austenitising of the surface layer occurs through eddy currents induced by an electromagnetic alternating field in the workpiece (tooth). Due to the rapid heating and the very short holding time compared to volume heating , the term short-term austenitising is also used. Information about the achievable heating rate can be found in the continuous time-temperature-transformation diagrams [7.4/11, Volume 3]. Requirements are the toothing geometry (tooth space, tooth form, tip diameter) and very accu-rately adjusted inductors , the manufacture of which requires particular experience (refer to, e.g., [10.3/10, p. 194 et seq.]). The coupling distance , the exact distance between the inductor and the tooth surface to be hardened, is very important for the achievable hardness contour . External and internal gearing is hardened in the same way. The penetration depth of the induced current limited to the workpiece edge is frequency depen-dent. The higher the frequency, the lower is the penetration depth. This law (skin effect) is also used for gear heating by applying a high frequency (HF with 102 to 103 kHz) and a medium frequency (MF with 2.5 to 10 kHz). Mains frequency (50 Hz) is rarely suitable for gear heating. The surface-hardening depth can be estimated as a function of the power density and the fre-quency at a given temperature and holding time [10.3/10, pp. 93–95]. The hardness variants applied for gears are individual tooth space hardening with tooth root har- dening, individual tooth space hardening (two-flank hardening) without tooth root hardening and single-shot gear hardening with and without tooth root hardening. The achievable hardness contours are shown in Table 10.3/1 with their advantages and disadvantages with respect to the tooth root and tooth flank endurance limit. In the case of gears with large dimensions and large module, individual tooth space hardening as two-flank hardening and individual tooth space hardening with tooth root hardening are mainly used. Single-shot gear hardening is particularly suitable for smaller dimensions and moduli. This hardness variant, which has previously been reported [10.3/11], has become established parti-cularly in series manufacturing due to more advanced plant techniques [10.3/12]. Compared to the previously known through-hardening of the tooth (Table 10.3/1, Figure b), it is now possible to harden gears so that they have an accurate contour or at least near accurate contour (Table 10.3/1, Figure c). A distinction is made between the single-frequency and the dual-frequency method with different technological variants, which are described in detail in [10.3/13] with schematic temperature-time curves. The methods are suitable for moduli m n = 2.5 to 5 mm at tip 10.3 Heat Treatment 679 diameters da 250 mm, depending on the generator output. Technological data, such as genera- tor output, austenitising temperature (controlled by a radiation pyrometer or thermal-vision camera), heating time and quenching time, depend on the hardness task and must be determined experimentally. Research is being carried out on the optimisation of the process parameters of induction hardening using numerical simulation [10.3/14]. The achieved surface-hardening depth is dependent on the hardenability of the steel, its initial structure (necessarily quenched and tempered) and the effect of the quenching medium, such as water, aqueous polymer solution or oil [10.3/15]. More alloying elements are necessary on the steel side for the module-dependent increasing surface-hardening depths and increasing dimensions. Values of steel-dependent surface hardness are shown in Table 7.4/8a, b. The hardened structure in the surface layer and the curve of the hardness-depth curve are roughly comparable to those of flame hardening (refer to DIN 17022-5). Differences in the hardness profile between the tooth flanks, where the hardness is higher than in the root radius, are unavoidable. The hardness at the hardened tooth tip is at its greatest. Results concerning the load capacity of gears hardened along the tooth contour are not known. It can be assumed that the tooth flank endurance limit is the same as in gears with single (tooth) space hardening including the tooth root. The tooth root endurance limit will be approximately 15% lower than the case-hardened gears. The advantages and disadvantages of surface hardening initially mentioned are also applicable for induction hardening. There is widespread use of inductive gear hardening in the series manufacturing of automotive engineering, wherein, among other things, the superior efficiency of the induction hardening of gears along the tooth contour is highlighted in comparison with case hardening [10.3/16]. Inductive surface-layer hardening is possible with gears with m n 3 mm, made from sintered steels with high density. The carbon content of these steels should be 0.35 to 0.50%. A polymer solution, which simultaneously provides protection against corrosion, is recommended as a quenching medium. Notes: For the determination of surface-hardening depths, see Figure 7.4/15. For module-dependent surface- hardening depths, see Table 7.4/17. Laser-beam hardening The application of laser-beam hardening for gears was already studied in the 1980s [10.3/17]. At that time, it was considered questionable whether laser-beam hardening for gears was gaining in importance [10.3/18]. The reasons were that, in the case of larger modules, the necessary surface-hardening depth is not reached, the conformal hardness layer cannot be produced economically, and there are differences in the hardness and surface-hardening depth due to the trace overlaps. In some cases, however, laser-beam hardening for gears continues to be studied [7.4/78]. Electron-beam hardening (EB hardening) The hardness layer adapted to the tooth form has not yet been realised economically for all gear forms. Examples are flat rings with face gearing, special gearing with large module (face widths up to over 100 mm with tooth heights up to 15 mm) and special gearing with flat flanks (chain wheels) [7.4/28, pp. 59–60]. Because of the procedurally limited surface-hardening depth ( 2 mm), this surface-layer hardening is especially suitable for increasing the wear resistance [7.4/77]. The improvement of the pitting and tooth flank endurance limit is thus given only in individual cases. Both beam methods can achieve a hardness that is 100 to 200 HV greater than in the case of flame or induction hardening. 680 10 Manufacturing of Cylindrical Gear Units 10.3.2.4 Case Hardening According to DIN EN 10052, “ case hardening ” is defined as “ carburisation or carbonitriding with subsequent treatment leading to hardening ”. Together with the subsequent tempering, case hardening is a combined process. The aim of case hardening is to achieve a hard surface layer through transformation hardening (formation of martensite) with a decrease in hardness through to the ductile core material that is as gradual as possible. The composite material withstands the strain affecting gears through volume fatigue, contact fatigue, boundary-layer fatigue, abrasion and static overload (refer to Section 7.4.1). Case hardening has undergone such an extensive development, especially in the last 20 years, that it cannot be described in detail. For this reason, reference is made to two German-language monographs [10.3/19], [10.3/20]. Notes: For materials for case-hardened gears, refer to Section 7.4.3.2; for information on heat treatment, refer to Section 9.2; for carbonitriding, refer to Section 10.3.2.5; for heat-treatment temperatures, refer to Table 10.3/6a to c at the end of the section. Carburisation: According to DIN EN 10052, “ carburisation” is the “ thermochemical treatment of a workpiece in the austenitic state for the enrichment of the surface layer with carbon, which is then present in a solid solution in the austenite ”. “The carburised workpiece is then hardened (directly or after reheating). It is recommended to indicate the medium in which the carburisation will be carried out, for example, carburisation in gas: gas carburisation, carburisation in powder: powder carburisation, carburisation in plasma: plasma carburisation.” Primarily gases/gas mixtures and molten salts, and rarely powders and pastes, are used as carburisers. The most common and best technically controlled method is gas carburising at atmospheric pressure . It is carried out in the temperature range between 880 C and 980 C, mainly at 930 C. The upper limit is determined by the resistance of the metallic retort of the furnace to tempe-rature changes. Depending on the method, propane, natural gas and methane, for example, are used as carbon-emitting gases, usually as mixtures with air or oxygen [10.3/19, pp. 16–17]. A prerequisite for carbon diffusion is that there is always sufficient carbon present in the carburisation gas and the surface for the absorption of atomic carbon is reactive, that is, metallic bright and clean. Oily, rusty or oxidised surfaces reduce or prevent carbon adsorption. The aim of carburisation is a steel-dependent surface carbon content of C R 0.8% with a gradual transition to the core carbon content CK: this is called a carbon profile. The limitation of the surface carbon level is an essential prerequisite for the high-quality carburisation of gears in order to avoid carbide precipitation and to limit the surface oxidation as well as the structural residual austenite in the surface layer. Through these prerequisites, grinding cracks and overheating can be avoided. According to ISO 6336-5 (and DIN 3990-5), they are not permissible for the quality MQ. Grinding cracks are visible to the naked eye. Grinding burn is detected by etching. This problem is described in great detail in [10.3/21, p. 688 et seq.]. An increase in surface carbon content over 0.6% C can cause a drop in the tooth root endurance limit. The tooth flank endurance limit can increase up to 0.9% C at the surface [10.3/22]. A good compromise should be the surface carbon content in accordance with Table 10.3/2. Recommended surface carbon contents for surface layer structures low in residual austenite are given in Table 10.3/2 for the selected case-hardened steels according to Table 7.4/20. As can be seen from Table 10.3/2, the sum of the alloying elements compared to the pure iron predomi-nantly results in a higher carbon absorption, which is taken into account by the alloying factor k L 10.3 Heat Treatment 681 [10.3/19]. Simply formulated, it indicates that the required carbon level CP of the carburisation atmosphere must be divided by the alloying factor kL so that the desired surface carbon content is present in the steel. Example: 18CrNiMo7-6, kL = 1.13; necessary carbon level of the carburisation atmosphere: CR = 0.7%, CP = 0.62% (refer to Table 10.3/2). Table 10.3/2 Surface Carbon Content, Average Alloying Factors and Required C Potential for Surface Layer Structure Low in Residual Austenite in Accordance with DIN 17022-3 Material Surface carbon content CR [Mass %] Alloying factor kL [-] C level of the atmosphere CP [%] Pure iron, carburised 0.80 1.00 0.80 16MnCr5 0.72 1.11 0.66 20MnCr5 0.70 1.13 0.62 17CrNi6-6 0.65 1.11 0.59 15NiCr13 0.60 0.98 0.61 20NiCrMoS6-4 0.75 1.04 0.72 18CrNiMo7-6 0.70 1.13 0.62 14NiCrMo13-4 0.75 1.00 0.75 20NiCrMo13-4 0.75 1.01 0.74 Multi-stage carburisation with different carbon levels (for term, refer to DIN EN 10052) is state of the art and practised at case-hardening depths CHD 1 mm. In the first stage, the carbon level Cp is set so high that hardly any soot forms in the furnace retort or hardly any carbides form in the surface layer [10.3/19]. In the second stage, the carbon level Cp and usually the temperature are lowered so that the desired surface carbon content is achieved. In order to improve the edge hardenability, especially for large gears, up to 5% ammonia can be added to the gaseous atmosphere at the end of the carburisation process (in the last 30 minutes). The basic profile of the carbon curve (carbon profile) is shown schematically in Figure 10.3/12 for the two-stage carburisation widely used in practice (from DIN 17022-3). Process computers control the carburisation process on the basis of the measurement of the oxygen partial pressure with the oxygen probe [10.3/19, p. 63 et seq.]. The indirect measurement of the characteristic values of gas, such as CO 2 content using infrared absorption and water vapour content using dew-point measurement, which are thermodyna-mically related to the carbon level [10.3/19], is still used today in older systems. The carbon curve can also be varied through two-stage carburisation in a salt bath [10.3/23] when molten salts with a different activator content corresponding to 1.1% and 0.8% of the surface carbon content are used. Powder and paste carburisation cannot be controlled and no longer have practical significance. In addition to the achieved surface carbon content, the result of carburisation is documented as the carburisation depth. According to DIN EN 10052, this is the “ vertical distance from the surface up to a limit that indicates the thickness of the layer enriched with carbon. This limit must be specified exactly ”. For example, At 0.35 is “a practical introduction of a limit” [7.4/39]. The depth denoted by At0.35 can yield the case-hardening depth after hardening when the steel has sufficient case hardenability with the selected quenching intensity in the existing gear dimensions [10.3/24]. The carburisation depths for gears are between 0.2 and 10 mm. Great carburisation depths can only be realised economically through the two-stage process using high-temperature carburisation (over 950 C to 1050 C) [10.3/19] and with process times of up to 682 10 Manufacturing of Cylindrical Gear Units 100 hours [10.3/25]. For this reason, high demands are simultaneously placed on the fine-grained steel, that is, uniformly fine grained after a long holding at a high temperature [7.4/32] to [7.4/35], and on the thermal durability of the carburisation furnace. Carburisation of the entire gear is economical. This is not always possible due to reasons of machining and distortion. Areas that must remain “soft” can be most easily protected by masking pastes. The subsequent removal of carburised surfaces in the soft annealed condition is also possible. Figure 10.3/12 Schematic carbon profile curves with two-stage carburisation, from DIN 17022-3: C S - saturation value of the austenite for carbon C R - surface carbon content CK - core carbon content CG - carbon content limit in order to determine the carburisation depth At Gas carburisation in a vacuum, developed after 1970 (technical usage: low-pressure carburisation, vacuum carburisation), allows for temperatures up to 1000 C at gas pressures between 1 and 30 mbar, due to the furnace materials used (e.g., SiC, graphite), and thus much shorter treatment times. It is now always used in conjunction with high-pressure gas quenching (LP/HP method). There is a wide range of plant technology with usable dimensions up to 600 mm wide, 750 mm high and 1000 mm in length available for this purpose [10.3/26], so even relatively large gears can be treated. The process run in the furnace chamber is simplified in Figure 10.3/13: • Evacuation at room temperature up to approximately 1.5 mbar • heating in nitrogen up to the desired carburisation temperature between 960 C and 1050 C, possibly with a short holding at 700 C to 800 C • evacuation to the required process pressure • introduction of the carburisation gas with the specification of the temperature-dependent carbon flow (mainly acetylene because of the much higher carbon supply compared to propane) in the pulse-pause ratio, that is, carburisation is very fast at C contents up to the soot limit, which is about 1.3% C at 1000 C for the steel 18CrNiMo7-6 • diffusion phase: The short-term pulse-pause sequence is repeated several times with a longer diffusion phase at the end until the target values of the surface carbon content and carburisation depth are reached. In the final, longer diffusion phase, the reduction to the hardening temperature takes place. In order to reduce distortion, after diffusion it can be cooled in the pearlite stage without lowe-ring the temperature, held isothermally, reheated to the hardening temperature and then quen-ched with different gases (N 2, He) and gas pressures of 10 to 20 bar with a high gas flow rate (refer to Figure 10.3/14d). 10.3 Heat Treatment 683 Figure 10.3/13 Schematic time-temperature curve of gas carburisation in a vacuum, according to [10.3/26] If low-pressure carburisation is carried out wit h the aid of plasma, a sputtering step under hydrogen is added to activate the surface during heating. Methane is the carbon dispenser. The electrical parameters with a dominant influence are the current density, voltage and frequency. This is in addition to the pulse-pause ratio. In contrast to regulated gas carburisation, the performance of the carbon diffusion of the gas carburisation in a vacuum is controlled. The carbon profile results from the carburisation-diffusion phases, the duration and number of which are determined in advance by carburisation calculations. Hardening after carburisation: According to DIN EN 10052, the aim of hardening is to “ austenitise and cool under such conditions that an increase in hardness occurs through a more or less complete transformation of austenite into martensite and, where appropriate, into bainite ”. Essentially, there are four different temperature-time sequences suitable for gears, which are represented in Figure 10.3/14a to d. Gears can be hardened at the core hardening temperature in order to achieve the optimum fatigue strength. A greater distortion can occur in comparison to surface-layer hardening. Table 10.3/3 contains the method description with notes on advantages and disadvantages, which apply to classic carburisation and carburisation in a vacuum. The selected quenching medium influences the hardening behaviour of the gears, expressed mainly in terms of surface hardness, hardness-depth curve, tooth core hardness and distortion. By choosing the gas quenching system (an external chamber with the dimensions of the furnace chamber), the quenching speed can be varied and, thus, influence can be exerted on the heat transfer coefficient [10.3/27]. Quenching with hydrogen, also as a mixture with nitrogen, is offered, and quenching with air is being tested. 684 10 Manufacturing of Cylindrical Gear Units Figure 10.3/14 Schematic time-temperature sequences of the case hardening of gears: a) direct hardening: gas carburisation; b) direct hardening: low-pressure carburisation (see continuation in c, d); A c3-core - austenitising temperature for the non-carburised core material Ac3-surface - austenitising temperature for the carburised surface The advantages of this relatively new method compared to gas carburisation and hardening are as follows: • The process is shortened through the selection of carburisation temperatures up to 1050 C. • It has low gas and energy consumption. • The quenching effect can be varied by the gas type, gas pressure and gas velocity. • Gears are blank after hardening, surface oxidation is avoided, and washing is eliminated. • Carburisation is uniform, with only small CHD differences between flank and root. • Distortion is reduced and more uniform. • Partial hardening is possible with plasma carburisation without the use of masking pastes. • The lower cost of hard finishing can compensate for higher steel prices. • Vacuum and plasma carburisation plants work without prior conditioning. Disadvantages compared to quenching in oil are as follows: • The quenching effect is lower (H 0.25, refer to Figure 7.4/10). • At the same characteristic dimensions, higher alloyed, more expensive steels must be used in order to achieve the required core hardness HC 34 HRC for the quality MQ. • The steel 23MnCrMo5, the alloying technology of which is cheaper, was used [10.3/27]. • “Normal” case-hardened steels exhibit grain growth at a high temperature and over a long period of time. • The fine-grain steels supplied are more expensive. • Surface-layer damage occurs due to element evaporation. • New investments to replace the proven method are expensive. • Gas recovery with the use of helium is mandatory. • Dimensions for large gears, which are usually case hardened in a shaft furnace, are limited. 10.3 Heat Treatment 685 Table 10.3/3 Advantages and Disadvantages of the Hardening Methods Suitable for Gears after Carburisation Disadvantages • Surface oxidation or Mn evaporation • Expensive fine-grain steels with the use of high-temperature carburisation • Steels of higher hardenability with the use of gas quenching • Washing after oil quenching • Increased energy and time consumption compared to direct hardening • Inert gas is necessary to prevent decarburisation during hardening or intermediate annealing • Cracks in the Cr-Ni steel during slow cooling • High plant, energy and time consumption • Inert gas is necessary during isothermal transformation Advantages • Saving of treatment time and energy • Saving of plant capacity • Variation of the quenching effect • Low-distortion hardening without force • No washing after gas quenching • Almost no limitation of the case-hardening depth • Correction of distortions is possible though removal of the carburised layer • Advantageous initial structure for hardening • Fine-needle martensite after hardening • Low distortion • Advantageous initial structure for hardening • Fine-needle martensite after hardening • Prevention of hardening cracks with the use of stainless steel • No fine-grain steel required Method description Direct hardening; Immediately after the end of the one- or two-stage carburisation, quenching is carried out in fluid media or with gases directly from this temperature or usually from the core hardening temperature after prior reduction of the temperature and holding, depending on the type and geometry of the steel (Figure 10.3/22a b). Suitable for gears of almost all dimensions. Single hardening: After carburisation, cooling is carried out slowly depending on the steel and geometry. Reheating to the hardening temperature should be carried out for gears above Ac3 of the core material. Then, depending on the type of steel and geometry, quenching is carried out in fluid media or with gases (Figure 10.3/22c). Suitable for gears for which mechanical machining is necessary after carburisation, or for gears that are not manufactured from fine-grain steel. If, due to the steel or dimensions, the hardness is too great for mechanical machining after slow cooling, it is soft annealed (intermediately annealed) (Figure 10.3/22c). Hardening after isothermal transformation: After carburisation, the gear is cooled to a temperature of 620 C to 680 C, held isothermally at this tempe- rature, reheated to the core hardening temperature and then quenched in liquid media or gases. Suitable for gears that are sensitive to distortion and cracking and that are not manufactured from fine-grain steel (Figure 10.3/22d). 686 10 Manufacturing of Cylindrical Gear Units Figure 10.3/14 Schematic time-temperature sequence of the case hardening of gears: c) single hardening with intermediate annealing Figure 10.3/14 Schematic temperature-time sequence of the case hardening of gears: d) hardening after isothermal transformation The decisive factors are the type of medium, its temperature, concentration and flow velocity, as well as the time sequence of the three cooling phases (film boiling, nucleate boiling and con-vection) determined by this. In general, the distortion is less after mild quenching. The gears should be washed in oil after quenching. 10.3 Heat Treatment 687 After hardening, usually only the surface hardness is measured, which should be between 62 and 67 HRC. Other material and component properties are determined only after the annealing. Gas quenching with nitrogen, helium or helium-nitrogen mixtures has a lower quenching effect compared to the media used during quenching after gas carburisation, such as salt baths, oil or aqueous solutions. This shows a comparison according to [10.3/27]. While surface layer damage through surface oxidation is unavoidable during gas carburisation [10.3/28], vacuum carburisation results in, for example, the evaporation of manganese and thus the same negative effect of the reduction in hardenability in the surface layer [10.3/28]. This can be remedied by the mechanical removal of the damaged layer or by shot peening [10.3/30]. Tempering It is state of the art that gears are annealed at 170 C to 220 C for between 2 and 5 hours, independent of the case-hardened steel used and the intended use. This results in advantages and disadvantages for the properties achieved. It should be noted that, for economical reasons and with respect to the modified properties of the gears, tempering can be dispensed with [10.3/31]. In this case, it is necessary to consider what is to be achieved by annealing, as is shown by the following points: • Decrease in the surface hardness (Figure 10.3/15), [10.3/32] • Decrease in the residual compression stresses (Figure 10.3/16), [10.3/32] • Increase in the static and dynamic toughness [10.3/31] • Reduction in the risk of cracking • Improvement in the grindability A drop in the load capacity of the gear is associated with the reduction in surface hardness, case- hardening depth and residual compression stresses [10.3/33]. The predetermined surface hardness for gears, for example, for moderate requirements of 58 to 63 HRC (refer to Section 7.4.3.2), is realised for the steel 20MoCr5 by annealing between 275 C/2 hours and 175 C/2 hours, according to Figure 10.3/15 (prerequisite: 67 HRC quenching hardness). Figure 10.3/16 Residual compression stress relief in case-hardened teeth through tempering [10.3/32] Figure 10.3/15 Surface hardness of the steel 20MoCr5 as a function of tempering temperature and time [10.3/32] 688 10 Manufacturing of Cylindrical Gear Units The tempering effect acts on the entire hardness-depth curve (for an evaluation of the hardness- depth curve, refer to Figure 7.4/17). Annealing reduces the surface hardness of the steel 20MoCr5, for example, from 790 HV0.5 to 690 HV0.5 after 275 C/2 hours, and the case-hardening depth drops from 0.94 mm in the hardened condition to 0.87 mm (175 C/2 hours) or to 0.76 mm (275 C/2 hours). As a result of refrigeration after hardening, an increase in hardness is possible due to the transformation of residual austenite (refer to Table 10.3/2). Through the use of fine-grain steel, the fatigue strength can be significantly increased with corre- sponding case hardening compared to coarse-grain steel [10.3/34]. An increase in toughness can be identified with decreasing grain size. Grinding burn and grinding cracks are known to markedly reduce the load capacity of the gear [10.3/35]. A coarse grain, residual austenite, and reticular and coarse carbides can have a negative impact on the grindability. The simulation of the case hardening, including the estimation of the endurance limit, has been studied in great detail [10.3/36], [10.3/37]. The “target” structure of the case-hardened surface layer and of the hardened core material is described in detail in ISO 6336-5. Table 10.3/4: Selection of Typical Structural and Hardness Requirements for Case-hardened Gears (Quality MQ) Surface structure Core structure • fine-needle martensite • no surface decarburisation • no reticular or coarse carbides • residual austenite content 30% • surface oxidation depth 0.02 mm • no grinding burn Example: Steel 20MoCr5: martensite and 25% residual austenite Surface hardness: 58 to 63 HRC • martensite and bainite • no ferrite at the grain boundaries Example: Steel 20MoCr5: martensite and bainite Core hardness: 34 HRC The influence of an “undesirable” structure on the load capacity of the gear is shown in Table 10.3/5a, b. 10.3 Heat Treatment 689 Table 10.3/5a Influence of Undesirable Structures on the Load Capacity of the Gear Remedy Vacuum or plasma carburisation; grinding of the damaged layer; shot peening under defined condit- ions cancels the damaging effect Regulated carburisation with sur- face carbon content adapted to the steel (refer to Table 103/6); lowering of the hardening tempe- rature during direct hardening; single hardening from the opti-mised core hardening temperature Hardening and intermediate annealing after carburisation with a regulated C level; removal of the decarburised layer; shot peening under defined conditions cancels the damaging effect Effect Greatly reduced edge harden-ability, lower edge hardness, residual stress reversal of pres- sure in tension; reduction in the load capacity of the gear. Reduction in tooth load capacity, drop in hardness to the edge, maximum hardness below the surface; improvement in ductility; increase in the risk of grinding cracks, shift in the maximum residual compression stress in the depth; reduction in the load capacity of the tooth root. Partial to complete formation of ferrite; large drop in hardness and compressive stress to the edge; reduction in tooth load capacity. Structure, structural defects Surface oxidation Internal oxidation of the alloying elements, especially Mn and Si, through a carburisation medium containing oxygen; up to max. 0.040 mm below the surface Residual austenite Excessively high C content in the austenite; incomplete martensite transformation when martensite start point is below room temperature. Top figure: 20MoCr5 with 1.14% surface carbon content, approx. 55% residual austenite (dark martensite needles in light austenite), V = 500:1 Surface decarburisation Diffusion of carbon from the surface layer during annealing in a medium containing oxygen; up to 0.5 mm below the surface. Bottom figure: 20MoCr5, decarburised at a depth of 0.5 mm, V = 100:1 690 10 Manufacturing of Cylindrical Gear Units Table 10.3/5b Influence of Undesirable Structures on the Load Capacity of the Gear Remedy Regulated carburisation with surface carbon content adapted to the steel (refer to Table 10.3/6); in the case of insufficient surface carbon content, either accelerated cooling after carburisation or after soft annealing Sufficiently long carburisation adapted to the type of steel; steel with sufficient edge hardenability; more abrupt quenching is necessary Steel with sufficient core hardenability; hardening from the optimised core hardening temperature; improved quenching effect Effect Carbides in globular form improve the load capacity of the tooth flanks; reticular carbides have an embrittling effect: increase the risk of grinding cracks, reduce the load capacity of the tooth flanks Hardness is too low due to soft ferrite; drop in residual compression stresses to the edge; reduction in the load capacity of the gear Core hardness is too low due to soft ferrite; reduction in the load capacity of the gear with ferrite in the tooth core Structure, structural defects Carbide precipitation Excessively high C content in the austenite, precipitated at grain boundaries as a carbide in addition to pearlite during slow cooling (refer to Single hardening). Top figure: 16MnCr5, coarse and reticular carbides (light) in fine-grain martensite, Ferrite in the surface layer Insufficient carburisation or excessively mild quenching or steel with insufficient edge hardenability Ferrite in the core Steel with insufficient core hardenability; excessively low hardening tempe- rature or excessively mild quen- ching Bottom figure: 20MoCr5, V = 500:1 10.3 Heat Treatment 691 Distortion : The distortion related to the system properties as the sum of all the dimensional and form changes of a gear, due to the process-related thermal, mechanical and chemical effects, is caused primarily by phase changes and elastic and plastic deformations. The determination of distortion parameters after the case hardening of gears has great economic importance, particularly for the subsequent finishing (after hardening) [10.3/21, pp. 380–389]. In the case of measurements of distortions on gears, it can be assumed that all influences through primary shaping, forming and manufacturing steps before the heat treatment are included in the result. The following are considered the main causes of gear distortion [10.3/38]: • the gear design (50 to 60%), • the steel and its hardenability (20 to 30%) and • the manufacturing process, including heat treatment (5 to 10%). The dominant influence of the gear design on the distortion is described by the term heat-treatment- related design (refer to Section 7.4.1). The general design rules, such as symmetry, uniform mass distribution, flowing cross-sectional transitions, and no ribbing in the web area, also apply to gears [7.4/6]. The following distortion trends are known for different gear forms made from case-hardened steel: • During hardening, gears grow in the direction of their largest expansion [10.3/4]. • Gear shafts become longer; consequently, in the case of helical gearing, the distortion acts in the sense of reducing the helix angle [10.3/39]. • In the case of gear shafts, the shrinkage of the gearing can be reduced by a longitudinal bore [10.3/4]. • Wheel discs or rims increase in the radial or tangential direction and thereby become thinner [10.3/39]. • With the growth of the wheel discs, the base tangent length bec omes greater [10.3/4]. • In the case of wide gearing, the diameter can be increased more than in the tooth centre by the more abrupt cooling at the end faces [10.3/4]. Experience has shown that gears as solid disks up to 1.2 m in diameter are brought to the required dimensions through finishing (after hardening). In the case of larger wheels (up to 4 m in diameter, usually as a welded construction) and thus a greater increase in diameter, the pinion may need to be adjusted. The conclusion is that a quality-oriented finishing is only possible with the help of systematic distortion measurements. The principles of deformation in connection with measurement results are described in great detail in [10.3/40]. 10.3.2.5 Carbonitriding According to DIN EN 10052, carbonitriding is the “ thermochemical treatment of a workpiece above Ac1 (723 C in the iron-carbon system) in order to enrich the surface layer with carbon and nitrogen. Both elements are then in the austenite in a solid solution ”. It is recommended to indicate the medium in which the carbonitriding will be carried out, for example, carbonitriding in gas: gas carbonitriding, carbonitriding in a salt bath: salt bath carbonitriding. Only carbonitriding above the core hardening temperature of case-hardened steels (refer to DIN EN 10084) and the hardening temperature of tempering steels (refer to DIN EN 10083-3) is suitable for gears with high load capacity. In terms of the method, the upper temperature limit is 930 C. 692 10 Manufacturing of Cylindrical Gear Units The carbonitriding process is similar to that of carburisation with direct hardening and it has the same goal (refer to Section 10.3.2.4). Due to the additional nitrogen diffusion, edge hardenability, hardness deepening and tempering resistance are improved in comparison to case hardening. Notes: For materials for carbonitrided gears, refer to Section 7.4.3.3; for information on heat treatment, refer to Section 9.2; for heat-treatment temperatures, refer to Table 10.3/6a, b at the end of the section. Carbonitriding is possible in gas at atmospheric pressure and in a vacuum, in plasma and in molten salts. Gas carbonitriding at atmospheric pressure using natural gas (split propane), with the addition of propane and ammonia, is the method most often used. Other C-emitting gas mixtures can also be used. The aim is a total content of carbon and nitrogen of not more than 1%, for example, 0.7% C and 0.3% N. About 2 to 4 volume percent of ammonia at temperatures above 850 C to 930 C is sufficient to keep the N content < 0.3%. As with carburisation, the C level is measured and regulated, and the constant addition of ammonia (up to 10%) is set by the flow meter. Hardening takes place immediately after the diffusion process (refer to Figure 10.3/14a). In the diffusion phase, a higher ammonia content is set, which is then reduced after lowering to the hardening temperature. The optimal treatment parameters (temperature, time, sum of C + N) for a given steel are still the subject of investigation. Temperatures 930 C are aimed for in order to achieve greater hardening depths for use on gears with large modules. Quenching is carried out in oil. After washing, tempering is carried out at 180 C to 220 C for 2 to 5 hours. Carbonitriding in a vacuum differs only slightly from carburisation. The carbon and nitrogen diffusion does not take place at the same time, contrary to the atmospheric pressure method. Carbon is diffused in the first step, followed by only nitrogen. After lowering the hardening temperature, it is quenched with nitrogen (Figure 10.3/17). The nitrogen content is regulated by the volume of NH 3 and duration. As with gas carbonitriding, a ratio of about 0.7% carbon to between 0.2% and 0.4% nitrogen should be obtained [10.3/41], [10.3/42]. Figure 10.3/17 Schematic time-temperature curve of vacuum carbonitriding, according to [10.3/42] 10.3 Heat Treatment 693 With a C + N total content of 1%, there is a high residual austenite content (up to 50%). It reduces the maximum achievable surface hardness from ≈ 67 HRC to ≈ 63 HRC. The residual austenite content can be influenced by the choice of higher tempering temperatures. With an increasing carbonitriding temperature and time, as well as an increasing addition of ammonia, pores occur in the surface layer, which reduce the endurance limit (refer to Fi-gure 10.3/18). The inevitable porous zone should be thinner than 0.02 mm for the material quality MQ. Where appropriate, the carbon-nitride-hardened gears are not tempered if they are mountable without finishing (after hardening). If tempering is required, temperatures that are higher than after case hardening must be chosen, as nitrogen improves the tempering resistance. Tempering up to 300 C is possible. The case-hardening depth reached is determined by the diffusion temperature and time. Greater case-hardening depths are possible if carbonitriding is carried out at temperatures > 860 C. The de-termination of the case-hardening depth is carried out according to DIN EN ISO 2639. The choice of a lower diffusion temperature compared to carburisation leads to a lower heat-treatment distortion after the direct hardening of gears with m n 6 mm. This advantage is used whenever there are measurements of deformations, in order to perform the finishing of the gearing (shaving or grinding) before carbonitriding. Until now, the carbon-nitride-hardening of gears has been widespread mainly in the automotive industry. 10.3.2.6 Nitriding/Nitrocarburising According to DIN EN 10052, the methods of nitriding and nitrocarburising are particularly used for gears. “Nitriding ” is the “ thermochemical treatment for the enrichment of the surface layer of a workpiece with nitrogen ”. “Nitrocarburising ” is the “ thermochemical treatment for the enrichment of the surface layer of a workpiece with nitrogen and carbon to form a link layer”. In DIN EN 10052, it is recommended to specify the medium in which treatment will be carried out for this method, for example, nitriding in gas: gas nitriding, plasma nitriding: plasma nitriding (without oxygen), likewise, nitrocarburising (sulfo-nitrocarburising) in gas, in plasma, in a salt bath. For “ nitriding ”, a broad temperature range between 380 C to 650 C is specified, which is used depending on the method variant [10.3/43]. The contrast to nitrogen case hardening is clear. This nitriding method has been further developed on a large scale. It is advantageous to denote nitrogen as a basic element and other elements, such as carbon, oxygen and sulphur, as addi-tional elements and to use “nitriding” as a generic term for the large group of methods for pro-ducing surface layers containing nitrate. As a result of “ nitriding/nitrocarburising ”, a split layer primarily emerges, which consists of the link layer and the diffusion layer. This surface layer is hard and relatively brittle. The hardness transition to the core hardness, which should be as high as possible through tempering (> 300 HV), is gradual to steep, depending on the material. As a result of the increase in hardness Figure 10.3/18 Pores in the unetched carbonitrided sur- face layer of the steel 34Cr4, V = 500:1; pore depth approx. 0.05 mm 694 10 Manufacturing of Cylindrical Gear Units and residual compression stresses in the surface layer, the composite layer withstands the volume and contact fatigue. The link layer shows the high resistance to boundary-layer fatigue, abrasion, adhesion and corrosion (tribo-oxidation). Special nitride precipitation in the diffusion layer increases the heating hardness and resistance to thermal fatigue. Notes: For materials for nitrided gears, refer to Section 7.4.3.5; for information on heat treatment, refer to Section 9.3; for heat-treatment temperatures, refer to Table 10.3/6a to c at the end of the section. On the basis of the gear endurance limit values according to Figure 7.4/22a in comparison with Figure 7.4/22b, it could be determined that much higher values are achieved with gas- and plasma-nitrided quenched and tempered Al-free nitriding steels with σHlim = 1200 N/mm2, σFlim = 850 N/mm2, than with tempering and case-hardened steels for nitrocarburised gears with σHlim = 800 N/mm2, σFlim = 640 N/mm2. One reason for this is that, in the case of nitrocarburis- ing, it mainly depends on the ε link layer, and a thickness of approximately 20 μm is sufficient. The nitriding hardness depth is thereby limited. Table 7.4/25 underlines this conclusion and displays a high resistance to abrasive and adhesive wear combined with a high resistance to corrosion for nitrocarburised gears, in addition to the lower pitting and tooth root endurance limit. Therefore, nitriding in gas and gas mixtures without an additional gas containing carbon is mostly used for gears. Nitriding in plasma is becoming increasingly widespread. Gas and plasma nitrocarburising are possible. Molten salt for nitrocarburising is still valid. Nitrocarburising in powder is hardly used for gears. An essential prerequisite for reproducible treatment results in gaseous atmospheres is the cleaning of the gears before nitriding/nitrocarburising in order to eliminate residues of a mostly unknown type created in the manufacturing process by washing or radiation/blasting/shot peening and rarely by pickling. The widespread washing can be performed with hot aqueous solutions, solvent cleaners or chlorinated hydrocarbons [7.4/52, p. 116 et seq.]. The potential sputtering during plasma nitriding is used for fine cleaning. The surface passivation has an inhibitory effect on the nitrogen diffusion. This barrier layer is the result of mechanical machining with a wide variety of auxiliary substances. The passivation increases with an increasing content of special nitride formers (especially > 1.5% Cr). This can be remedied by fine blasting or preoxidation (refer to Figure 10.3/19). Nitrocarburising in a salt bath is unproblematic regarding passivated surfaces. Various additional gases are used for the methods used in industry for producing boundary layers containing nitride on the basis of ammonia, for example, NH 3 + N 2, NH 3 + H 2 or NH 3 + CO, NH 3 + CO 2 or NH 3 + air [7.4/52, p. 129]. The common advantage of all the method variants is, in contrast to martensitic hardening, the precipitation and particle strengthening without structural transformation as a major reason for the low distortion after nitriding. A requirement is to avoid the temperature-dependent α ĺ γ transformation in the structure close to the surface layer. The nitriding temperature is theoretically limited to 590 C (A 1 point of the iron-nitrogen system). With the increasing content of alloying elements that form aluminium and special nitrides, the temperature limit increases for transformation-free nitriding. According to [10.3/44], this increase can be calculated over 590 C for low-alloyed steels. Compared to case hardening, the process design of nitriding is relatively simple. After mostly slow heating, holding is carried out at about 300 C for “preoxidation” under air/water vapour (Fe 3O4 in the nanometer range) for depassivation. Washing is then performed with N 2 until the O 2 content is below 1%. At the beginning of nitration, the link layer is usually formed on the surface, consisting of the nitrides ε-Fe 2-3N and γƍ-Fe 4N), the existence range of which depends on the NH 3 content and the nitriding temperature and is described by the Lehrer diagram [7.4/52, pp. 130–132 and DIN 17022-4]. The growth of the link layer and diffusion layer is controlled by temperature and 10.3 Heat Treatment 695 time. For a given fresh gas composition and quantity, the process is regulated and kept constant by the nitriding index K N and the oxidation index KO using sensors (e.g., O 2 probe) [10.3/45], [10.3/46]. For gears, nitriding between 500 C and 600 C is conventional. The nitriding atmosphere consists of NH 3, usually in a mixture with N 2, H 2 and air. The nitriding time can be up to 200 hours (“deep nitriding”, refer to [7.4/108]). When the thickness of the link layer is to be limited through generation of the γƍ phase, it is advantageous to nitrate at a constant temperature at first briefly with a high nitrogen supply and then with a low supply, which is referred to as initial nitriding. Cooling after nitriding should be carried out up to 150 C under N 2, and only then is the charge removed. In the case of the gene- ration of a ε link layer with high resistance to corrosion, “post-oxidation” can be performed as a holding stage at about 500 C under air or water. The schematic process run of nitriding is shown in Figure 10.3/19. Figure 10.3/19 Schematic time-temperature curve of gas nitriding With respect to the variable layer formation on gears, plasma nitriding [7.4./52, pp. 171–189], [10.3/47] is comparable to gas nitriding. In addition to temperature and time, the process parameters are the gas composition (N 2 and H 2 in different ratios depending on the type of layer), gas pressure (vacuum approximately 10 mbar) and electric voltage, and pulse duration and pulse repetition during pulse plasma nitriding. The components serve as the cathode, the container wall as the anode. The result is a visible purple glow in the transition zone, in which the reactive nitrogen is present. The initial stage is therefore somewhat shorter than in gas nitriding. The required nitriding time is the same as in gas nitriding. Figure 10.3/20 shows the simplified schematic process run of plasma nitriding for the component [7.4/52, p. 179]. A further development of plasma nitriding is the use of an “active grid” (ASPN: active screen plasma nitriding). In this case, the glow discharge is moved from the charge surface to a flexible metal grid in the furnace chamber [10.3/48]. The much lower energy consumption (glow transition zone to the grid no longer on all components) and very uniform nitration also of passivated surfaces without prior sputtering are cited as advantages compared to the current state of plasma nitriding [10.3/49]. 696 10 Manufacturing of Cylindrical Gear Units The aim of nitrocarburising in gas, in plasma and in a salt bath is to generate compound layers consisting of ε carbonitride. The process flows are similar to those in gas and plasma. During nitro- carburising, many carbon carriers are available as additional gases, which influence the phase composition of the link layer [7.4/52, p. 129]. Plasma nitrocarburising works with CH 4 as a carbon- emitting gas. In both of these process flows, as indicated in Figure 10.3/19, post-oxidation can be carried out at the end of the actual diffusion process when a high corrosion resistance is required for gears. Salt bath nitrocarburising exhibits some special features. The gears must be completely dry. This is achieved by preheating to 400 C. For treatment between 540 C and 630 C, different salts containing cyanide are available with possible porosity of the link layer, depending on the temperature [7.4/52, pp. 191–230]. After the diffusion phase, it is transferred to a second salt bath with a temperature of about 400 C for quenching. This quenching variant has the advantage of a lower risk of distortion, and the heat has a simultaneous oxidising effect (enhanced resistance to corrosion). Salt bath treatment is always associated with a costly washing and salt treatment process. In addition to temperature and time, the determination of the chemical composition (concentration of cyanate, cyanide and iron through titration) and the amount of air introduced are used for process control. The salt bath method requires strict regard for the environment. Figure 10.3/20 Schematic time-temperature curv e of plasma nitriding, according to [7.4/52] For reasons of distortion, nitrided gears should have as few machining residual stresses as possi-ble before nitriding. They are ready for mounting. All nitriding methods are capable of forming a surface layer consisting of a link layer and precipitation layer. According to Table 7.4/25, three types of layers are possible for gears: 1. The composite layer consisting of a ε(+ γƍ) link layer with thicknesses of 20 m and a diffusion layer (precipitation layer) with nitriding hardness depths of 1 mm. With a variable nitriding hardness depth in gaseous media, the growth of the link layer can be influenced by the nitriding index, gas flow, temperature (500 C to 600 C) and treatment time (up to 200 hours). Gas or salt bath nitrocarburising can be used for nitriding hardness depths of 0.5 mm. Nitriding hardness depths of > 0.5 mm are advantageously generated in gaseous atmospheres or in plasma. 10.3 Heat Treatment 697 2. The composite layer consisting of a γƍ link layer with thicknesses of 10 m and a diffusion layer with nitriding hardness depths of 1 mm. These layers, which are mostly used for gears with high load capacity, require a limited nitriding capability of the medium. This is ensured during sensor-controlled gas nitriding through the lowering of the nitriding index (e.g., K N < 0.8) and during plasma nitriding, among others, due to the reduction in the nitrogen content in the N 2-H2 mixture. The required nitriding hardness depth is adjustable by temperature and time (K N denotes the nitriding intensity) [10.3/45], [10.3/46]. 3. The nitrided layer without a link layer (diffusion layer only) with nitriding hardness depths of 0.8 mm. The nitriding capability of the medium must be lowered to such an extent (e.g., in gas, KN < 0.3; in plasma, H 2/N2 < 8) that the nitride covering does not occur. The nitrogen diffusion into the depth is slowed down. Hardness depths of > 0.5 mm can be produced economically only at temperatures above 570 C and over a long time (> 50 hours) [10.3/50]. As can be seen from the partial images a) and b), at the same temperature and time, the link layer is formed at different thicknesses due to the different nitriding capability of the medium. The nitriding hardness depth is almost the same when the same surface state is present. a) V = 500:1 h(VS) = 0.031 b) V = 500:1 h(VS) = 0.009 c) V = 500:1 h(VS) = 0 d) V = 500:1 h(VS) = 0.011 a) ε (+ γƍ) link layer, 31 μm thick, nitriding hardness depth = 0.60 mm, gas oxinitrided 570 C/32 hours/ KN = 3, V = 500:1; b) γƍ link layer, 9 μm thick, nitriding hardness depth 0.55 mm, gas oxinitrided 570 C/ 32 hours/ KN = 0.8, V = 500:1; c) no link layer, nitriding hardness depth 0.42 mm, gas oxinitrided 570 C/32 hours/ KN = 0.2, V = 500:1; d) ε(+γƍ) link layer with chipped porous zone, V = 1000:1; e) hardness-depth curve Figure 10.3/21 Surface layer structure and hardness-depth curves of the steel 30CrMoV9 after different nitration 698 10 Manufacturing of Cylindrical Gear Units The link layers have a dark outer porous zone. The pores are formed as a result of the recombination of diffusing nitrogen atoms to molecular nitrogen [10.3/51]. Their formation is unavoidable during gas nitriding. The formation of pores may be restricted through plasma nitriding. Since the porous zone is softer than the underlying non-porous nitride layer, it can improve the running-in characteristics when the pressure is not too high. It flakes off at high pressure (Figure 10.3/21d). The unwanted, ductility-reducing grain boundary cementite (Figure 10.3/21a) can be suppressed by the choice of material (steel with low carbon content) and technology (controlled process: limited thickness of the link layer, targeted decarburisation), but not completely avoided. In contrast to case hardening, where the hardening profile is, largely independently of the steel composition, determined mainly by the carbon level and carburisation temperature and time, nitrided steels have characteristic hardening profiles. The hardening profiles are primarily influ-enced by the initial structure, nitriding temperature, nitriding time and gas composition. As an example, Figure 10.3/22 shows the hardness-depth curves of the steels X6CrMo4, 17CrMoV10, 31CrMoV9 and X40CrMoV5-1, which have been quenched and tempered under the same conditions after gas nitriding at 500 C/96 hours. The different content of carbon and alloying elements causes very different hardening profiles, linked to the corresponding residual stress profiles (refer to Figure 10.3/23). Figure 10.3/22 Hardness-depth curves of the quenched and tempered steels X6CrMo4, 17CrMoV10, 31CrMoV9 and X40CrMoV5-1, Ttemp = 620 C/2 hours, nitrided at 500 C/96 hours, KN = 9, from [10.3/52] Distortion The low systemic distortion (sum of dimensional and form changes), even with complex compo- nents such as gears, is considered a major advantage of nitriding/nitrocarburising over surface hardening, case hardening and carbonitriding. Straightening in order to correct distortions of nitrided gear shafts, which have a link layer, should be avoided, since these are not plastically deformable and can tear. Positive residual compression stresses are relieved. Nitrided gears without link layers can be straightened if a re-sidual compression stress relief can be tolerated. In order to exclude the release of machining stresses, as well as structural transformations of the previously tempered steel during nitriding, which may lead to distortion, the temperature and time of tempering during quenching and tempering, of stress relief annealing and of nitriding must be matched using the Hollomon-Jaffe parameter [10.3/3]. In principle, the (fine) finishing of the teeth should be performed before nitriding in order to achieve the positive effect of the link layer. 10.3 Heat Treatment 699 A volume and, thus, dimension increase occurs as a result of the nitrogen and carbon diffusion. This is dependent on the material and the method parameters and, in the case of close tolerances, can be accounted for by the minus allowance during (fine) finishing. The positive dimension increase is associated with a specific mass gain. The dimension increase is particularly critical on the gear edges, where an overshoot occurs as a result of bilateral diffusion, which must be limited by cham-fering the edges. Residual stresses The positive effect of residual stress profiles on the endurance limit with residual compression stresses at the surface and with a transition to residual tensile stresses has long been known through nitriding. For a given dimension, the residual stress formation is dependent on the type of steel, its structure formation and the surface layer structure attained by nitriding. The long-term nitriding at a low temperature is advantageous here. In particular, the heating hardness plays a major role. The higher it is with the nitriding temperature, the higher the maximum of the residual stress [10.3/52]. The residual stress profiles for the four steels shown in Figure 10.3/22 are compared in Figure 10.3/23. As well as the different hardness profiles, different residual stress profiles with an almost similar trend also result. The difference from the residual stress development after case hardening, which is negatively affected by tempering (refer to Figure 10.3/16), is obvious and is an argument for nitriding. Figure 10.3/23 Residual stress profiles of the quenched and tempered steels X6CrMo4, 17CrMoV10, 31CrMoV9 and X40CrMoV5-1, Ttemp = 620 C/2 hours, nitrided at 500 C/96 hours, KN = 9, from [10.3/52] 10.3.2.7 Boriding According to DIN EN 10052, boriding is the “ thermochemical treatment for the enrichment of the surface layer of a workpiece with boron, with the aim of forming a boride layer ޵ . “It is recommended to indicate the medium in which boriding will be carried out, for example, boriding in powder: powder boriding, boriding in paste: paste boriding.” Notes: For materials for borided gears, refer to Section 7.4.3.6; for information on heat treatment, refer to Section 9.3. The aim of boriding is the generation of a very hard and wear-resistant layer. Boride layers are characterised to a large extent by resistance to scaling, resistance to corrosion and heating hard-ness [10.3/1, pp. 176–178], [10.3/3, pp. 887–896]. 700 10 Manufacturing of Cylindrical Gear Units All ferrous materials customarily used for gears may be used for boriding. However, for gears made of case-hardened, tempering or tool steel, boriding can be used for wear protection under extreme strain (inadequate lubrication, dry run) only in exceptions. Boriding is possible in powder (mainly used) [10.3/53], [10.3/54], [10.3/55], in gas [10.3/56], in pulsed plasma [7.4/56], [10.3/57], in pastes [10.3/58] and in a fluid bed [7.5/54]. Boriding in a salt bath has not prevailed. The boriding temperatures are between 850 C and 1050 C, usually around 900 C. As a result, after boriding, direct hardening must be performed, or a subsequent hardening (O 2-free inert gas or vacuum necessary) is required in a polymer solution or oil, in order to ensure the support effects for the hard boride layer. Where appropriate, this is followed by tempering at low to medium tempera-tures (up to 500 C). The temperature-time curve during powder boriding is similar to that of quenching and tempering. Depending on the boriding temperature and the thickness of the boride layer adjusted to the in-tended use, the boriding time is between 1 hour and 20 hours [7.4/55], [10.3/3, p. 888]. The aim should be the monophasic Fe 2B layer with layer thicknesses up to approximately 40 μm. The layer hardness reaches up to 2000 HV0.05. The layer exhibits interlocking to the core material (refer to DIN EN ISO 1463). The boride layer grows on the surface without significant boron diffusion inside the material. This creates a dimension increase, which can be up to 30% of the layer thickness. This can be offset through the minus allowance during machining by chip removal. As a result of quenching by the relatively high treatment temperature, distortion can be expected after the hardening. In order to reduce the distortion, the reduction of machining stresses should be carried out through stress-relief annealing with subsequent dimensional correction. The mechanical post-machining to reduce the occurring roughness [10.3/53] is possible by grinding, lapping or polishing. Straightening after boriding leads to tearing of the layer. On the material side, the result of boriding is evaluated with the layer hardness (microhardness measurement HV0.05 to HV0.2), the layer thickness (DIN EN ISO 1463:2004-08) and the layer formation (phases, transition to the basic material). 10.3.2.8 Coating According to DIN 8580, the main group No. 5 is designated by the term “ coating ” (refer to DIN EN 10052 and ISO4885). “Coating from the gaseous or vaporous state (vacuum coating) ” and “coating from the ionised state ” belong to the main group No. 5. In these two methods used for gears, a layer is deposited on the surface without further diffusion into the interior (refer to Table 7.4/2). Coating is always carried out in the final state of the gear after the heat treatment appropriate for the strain. Two types of coating are primarily used for gears: • layers of very high hardness, called hard material layers , which are produced mainly by PVD or PACVD treatment (with temperature-time curves, which can also be considered as heat treatment) and • layers with low to moderate hardness, which are caused by galvanised or chemical coating at temperatures below 100 C and which do not belong to heat treatment . Notes: For materials for coated gears, refer to Section 7.4.3.7; for information on heat treatment, refer to Section 9.3. 10.3 Heat Treatment 701 Hard material layers The cost-intensive generation of multiple hard-material layers improves, in particular, the resistance to micropitting and (in smaller gears) pitting, increases the scuffing load capacity (reduction in friction) and the resistance to corrosion and facilitates the running-in characteristics. Depending on the strain, the follow ing are suitable as coating ma terials: carbides, nitrides, oxides, borides of different metals in various forms, as well as their combinations, and diamond-like carbon. The layers are very thin (mostly 10 μ m), and the possible layer hardening is between 1000 and 3000 HV0.05. Hard material layers can be produced in a vacuum with different physico-chemical principles at different temperatures. A distinction is made between the following methods: • Chemical vapour deposition (CVD) in a temperature range of around 700 C to 1050 C. It involves the chemical separation of solids. This is based on reactive gases that flow around the component and react with it to form firmly adherent layers. Due to the high coating temperature, this method is not suitable for the coating of gears from case-hardened or tempering steels. • Plasma-assisted chemical vapour deposition (PACVD) in the temperature range from about 150 C to 700 C. The glow discharge in a vacuum with a pulsed direct current enables the layer formation at a low temperature. As in plasma nitriding, the components are connected as a cathode, and the recipient is the anode. The conformal glow transition zone realises the coating on all sides. The reaction media required for the coating are fed in gaseous form and must be swirled around. The residual gas is pumped off. • Physical vapour deposition (PVD) in a temperature range of about 200 C to 500 C. It is based on the high-energy vacuum evaporation of a dispenser material (target) with subsequent precipitation on the component, where the components have to be moved to the conformal coating. The steel to be coated should have a high hardness in its surface layer in order to support the hard material layer [7.4/58], [7.4/59]. Ni trided tempering steels, nitrided steels of high core strength and case-hardened steels are particularly suitable for hard material layers on gears. WC/C layers [7.4/61], [7.4/62] and hard amorphous carbon layers with diamond-like properties (diamond-like carbon: DLC), which are obtained by PACVD at deposition temperatures of 150 C (below the tempering temperature range of case-hardened steels) [10.3/59], are favoured for the coating of gearing. A DLC coating (Figure 10.3/23) is very smooth and largely unstructured. The wheel and pinion should be coated. The surface roughness is in the range of several micrometers. A DLC coating is particularly suitable for the prevention of scuffing wear in the case of inadequate lubrication or dry running. Its e ffect during dry running is temporary because the thin layers are removed at high strains. Figure 10.3/24a DLC layer, approx. 10 μm thick [10.3/60] Figure 10.3/24b: Gear with DLC layer [10.3/61] 702 10 Manufacturing of Cylindrical Gear Units References [10.3/62] and [10.3/63] offer understanding on the many possibilities of coating with hard material. More information on layer generation using PVD and CVD methods can be found in the VDI Guideline 3198. The VDI Guideline 3824 Part 1 provides information on layer prop-erties. Detailed agreements should be made with a company that performs the coating, taking into account the tempering resistance of the basic material and a possibly existing intermediate layer (e.g., nitrided layer). Layers with low to moderate hardness These are used to prevent scuffing during the run-in period, as well as to temporarily prevent corrosion, and have no negative effect on the load capacity of the tooth root and tooth flank. Copper plating occurs in acidic electrolytes at 40 C to 60 C, and a layer thickness of 4 to 10 μm is desirable (refer to drawing specification in Section 9.3). For more information, refer to [10.3/64]. Phosphating takes place in an aqueous solution of zinc or manganese phosphate at 30 C to 95 C and, with treatment times between 2 and 60 minutes, results in layer weights of up to 60 g/m 2 (refer to VDI/VDE 2420, Part 6, and drawing specification) in an increase in the hot scuffing load capacity by phosphating or copper plating the flanks. Resins have been developed as layers for reducing friction, improving the running-in charac-teristics and the resistance to corrosion, which are embedded in the MoS 2, graphite or PTFE. These layers (anti-friction coatings) are to prevent direct contact of the tooth flanks. Following prior fine blasting and phosphating, the coating can be carried out by immersion or spraying. Partial coating is possible. The layer thicknesses are between 10 and 20 μm [10.3/66]. In Tables 10.3/6a, b and c, heat-treatment temperatures for tempering, case-hardened and nitrided steels are indicated over the standards. 10.3 Heat Treatment 703 Table 10.3/6a Heat Treatment Temperatures of Gear Materials Nitriding, nitrocarburising Gas, plasma, salt bath nitro- carburising6) C 550 to 590 1) The temperatures in the lower range are generally used for hardening in water, those in the upper range are for hardening in o il. 2) Minimum austenitising time of 30 minutes (guide value). 3) When choosing the quenching medium, the influence of other parameters, such as design, size and hardening temperature, on the properties and susceptibility to cracking should be considered. Others, for example, a synthetic quenching medium, may also be used. W - water, SyA - synthetic quenching medium. 4) Minimum tempering time of 60 minutes (guide value) 5) Quench large forged pieces predominantly in oil or in aqueous solutions (synthetic quenchi ng medium), there is a risk of crack ing in water. 6) The wide temperature range covers all method variants of nitriding. 7) Dashes in the table mean that the treatment is not co mmon or not practical. Gas and plasma nitriding6) C 500 to 590 (380 to 650) Carbonitriding over A c3 Tem pering4) C 170 to 210 — Direct hardening in oil C 830 to 860 840 to 870 830 to 860 840 to 870 830 to 860 840 to 870 830 to 860 — Surface layer hardening Tem- pering4) C 150 to 180 (220) Flame and induction hardening3) C 870 to 900/W 890 to 920/W 870 to 900/W 890 to 920/W 870 to 900/W 880 to 910/W — 7) 880 to 920/oil or SyA — 880 to 920/oil or SyA Quenching and tempering Tem- pering4) C 550 to 660 540 to 680 540 to 660 550 to 650 550 to 720 Quenching medium3) Water or oil Oil or water Water or oil Oil (water) Air or oil Hardening1) 2) C 820 to 860 830 to 870 820 to 860 830 to 870 820 to 860 830 to 890 820 to 880 840 to 870 820 to 850 830 to 860 830 to 860 865 to 885 920 to 970 Tempering steels, DIN EN 10083 Open-die Forgings DIN EN 10250 C45E, C45R 38Cr2 46Cr2 32Cr4 41Cr4 34CrMo4 42CrMo4 30CrMoV9 5) 36CrNiMo45) 34CrNiMo65) 30CrNiMo85) 36NiCrMo165) 40CrMoV13-95) 704 10 Manufacturing of Cylindrical Gear Units Table 10.3/6b Heat Treatment Temperatures of Gear Materials Nitriding, nitrocarburising Gas, plasma, salt bath nitrocarbu- rising C 550 to 590 1) Quenching and tempering to improve hardenability and reduce distortion. 2) The carburisation temperature is dependent on the chemical composition of the steel, the mass of the gears and the carburiser. During the direct hardening of steels, a temperature of 950 C is generally not exceeded. For specific methods, e.g. under v acuum, higher carburisation temperatures (e.g. 1020 C to 1050 C) are not uncommon. 3) Core hardness temperatures are indicated. To reduce the risk of distortion, the lower temperature limit is selected. Austenitising time 30 to 35 minutes (guide value) 4) The tempering temperature is varied depending on the achieved quenching hardness and the desired final hardness, as is the te mpering time between 2 and 5 hours. Gas and plasma nitriding C 500 to 590 Carbonitriding over A c3 Tem- pering4) C 150 to 240 Direct har- dening in oil C 830 to 940 (1000) Case hardening Tempe- ring4) C 170 to 220 Hardening in oil3) C 860 to 900 830 to 870 840 to 880 830 to 670 825 to 880 Intermediate annealing (IA), isothermal trans- formation (IT) C 630 to 700/IA 620 to 680/IT Carburi- sation2) C 880 to 1050 (1100) Quenching and tempering1) Tempering C 500 to 700 Hardening in oil3) C 860 to 900 830 to 870 840 to 880 830 to 870 825 to 880 Case-hardened steels, DIN EN 10084 20MnCr5 17CrNi6-6 20NiCrMoS6-4 15NiCr13 14NiCrMo13-4 18NiCrMo7-6 20NiCrMo13-4 10.3 Heat Treatment 705 Table 10.3/6c Heat Treatment Temperatures of Gear Materials Nitriding, nitrocarburising Gas, plasma, salt bath nitrocarburising C 550 to 590 1) The temperatures in the lower range are generally used for hardening in water, those in the upper range are for hardening in o il. 2) Minimum austenitising time of 30 minutes (guide value). 3) When choosing the quenching medium, the influence of other parameters, such as design, size and hardening temperature, on the properties and susceptibility to cracking should be considered. Others, for example, a synthetic quenching medium, may also be used. 4) The tempering temperature, duration time = 2h, depends on the temperature of following nitriding. It should be 50 K lower than the nitriding temperature. The Hollomon-Jaffe-Parameter has to be checked, see 10.3.2.7 5) Accelerated quenching should be used to avoid tempering brittleness. 6) Stress relief annealing should be done at 20 to 50 K below nitriding temperature. The Hollomon-Jaffe-Parameter has to be check ed. 7) Steel 30MnVS6 is used in precipitation hardening state. 8) The wide temperature range is valid for all technology versions of nitriding. Gas and plasma nitriding8) C 500 to 590 (380 to 650) Stress relief annealing before nitriding6) C 480 to 570 Quenching and tempering Quenching5) air — air air air air air air air air pressure air or oil water or oil Tempering4) C 500 to 590 — 600 to 700 500 to 700 550 to 700 580 to 630 600 to 660 600 to 700 570 to 700 550 to 720 550 to 650 570 to 660 Quenching medium3) oil — oil (water) oil oil (water) oil oil oil air or oil oil or warm bath oil (water) Hardening1) 2) C 890 to 930 — 940 to 980 900 to 950 900 to 925 850 to 880 910 to 950 900 to 925 870 to 910 920 to 970 1020 to 1060 850 to 900 Nitriding steels, DIN EN 10085 20MoCr4 30MnVS6 7) 15CrMoV5-9 17CrMoV10 Ovako 225A 24CrMo13-6 31CrMoV9 32CrMoV5 32CrMoV13-10 33CrMoV12-9 40CrMoV13-9 X40CrMoV5-1 32CrAlMo7-10 706 10 Manufacturing of Cylindrical Gear Units 10.4 Symbols and Symbol Explanations of Section 10.3 A - structure austenite Ac1 C start temperature of austenite creating during heating (DIN EN 10052) Ac3 C start temperature of conversion of ferrite to austenite during heating (DIN EN 10052) Ac3-core C austenitising temperature of not- carburised core material Ac3-surface C austenitising temperature of car- burised surface material At mm carburising depth At0.35 mm carburising depth for carbon content limit of 0.35% B - structure bainite CG ma-% carbon content limit for definition of carburising depth CK ma-% core carbon content CL ma-% carbon level of alloyed steel CP ma-% carbon level CR ma-% border carbon content CS ma-% saturation value of austenite for carbon CHD mm case hardening depth for hardness limit of 550 HV1 F - structure ferrite FATT C fracture arrest transition temperature HC HV1 core hardness HV0.5 KG - grain size according to ASTM KN - nitriding reference number KO - oxidising reference number kL - alloying factor M - structure martensite Ms - martensite start point mn mm normal module N - number of load cycles Nht mm nitriding hardness depth according to DIN 50190-3 R - hardening degree Rm N/mm2 tensile strength Rp0.2 N/mm2 0.2% offset yield strength s mm distinguishing dimension, nominal thickness (DIN EN 10025-2) TA C austenitising temperature Ttemp C tempering temperature TG C soft annealing temperature TH-Fi C austenitising temperature of flame or induction hardening TH-EL C austenitising temperature of electron or laser beam hardening TS C melting temperature of pure iron tbath h duration of bath quenching ttemp h duration of tempering V - zoom in ratio İ - iron nitrite Fe 2-3N Ȗƍ - iron nitrite Fe 4N For additional symbols see Section 7.4. H RTETECHNIK Hardening and quality management from small parts to large gearboxes for the automotive industry, wind turbines, mechanical engineering, and general metal processingWe set standards. Quality is our passion.“Ahead through knowledge and expertise” A company of the HQM Group.www.hqm-gmbh.de Dr. Ing. S. Kr ger, company founder 709 Table of Appendices No. Title Page 1 Tooth Root Geometry ....................................................................................................... ............................. 710 1.1 Algorithm for Calculating the Tooth Root Geometry at a Cross Section with Given Tangent Angle ș ........ 710 1.2 Relative Tooth Root Curvature Radius for Basic Rack ISO 53 (DIN 867) ................................................... 712 1.3 Relative Tooth Root Thickness............................................................................................ .......................... 713 1.4 Notch Parameter ......................................................................................................... ................................... 714 2 Details for Calculating the Stress Correction Factor YS and the Relative Sensitivity Factor YįrelT According to ISO 6336-3 (DIN 3990-3) (1987 Issue) 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Root Geometry at a Cross Section with Given Tangent Angle ș (see Figures 2.3/9 to 2.3/11), Equations • Input data for the calculation: Normal pressure angle Į0 Helix angle ȕ Normal module mn Number of teeth of the gear to be manufactured z1 Profile shift factor of the gear to be manufactured x1 Number of teeth of the tool (shaper cutter) z0; on hobs approximately z0 = 100000 Profile shift factor of the tool x0; for hobs x0 = 0 Tool profile addendum haP0 Tool tip rounding radius ࣁaP0 Remaining protuberance (machined part) spr Tangent angle ș • Calculation algorithm for the coordinates of the tangent point, FnN and s Fn Number of teeth z, z0 (number of teeth of equivalent gear) b2: cosȕcosȕz z= ; b0 2 0: cosȕ cosȕz z= Operating pressure angle Įw0 of the generating pairing (two iteration steps are usually sufficient) 01 00 01ȟ2t a n Į invĮxx zz+=++ Initial value: 3w0Į 3ȟ = Iteration: w0 w0 w0 2 w0ȟinvĮ Į :Į tanĮ− = + Generating centre distance a0 Gear ratio u0 00 1uz z= Generating pitch circle radii rw 0 1 01waru=+, w0 w 1 0rr u= Tool base circle radius rb0 b0n 0 00.5 cos Į =rm z Radius r M on which the centre point M of the tool tip rounding is located ap0 0a P0 Mn 0 nn 2h zrm xmm=+ + − N Pressure angle ĮM for radius rM b0 M MĮ arc cosr r= Thickness half angle ǻĮ at point M pr 00 z 0M 0b 0aP0 0.5ʌ2t a nĮĮ invĮ invĮs x zr− +Δ= − + −N Appendix 1 Tooth Root Geometry 711 Appendix 1.1 Algorithm for Calculating the Tooth Root Geometry at a Cross Section with Given Tangent Angle ș (see Figures 2.3/9 to 2.3/11), Equations (continued) Auxiliary angle ȥ (two iteration steps are usually sufficient) 1) 01ȥʌ ș z=+ Initial value: 0ȥȥ= Iteration: w0 MȜ cosȥr r= 0 0ȥȥȥarc cosȜĮuy−=− + Δ + w0 2 0M1s i n ȥ1 1Ȝryur′=+ − ⋅ − ȥ:ȥy y=−′ termination at 610y y− ′< Distance K of the point M from the contact point of the pitch circles 1 Mw 0 01ʌsinȥĮ ȥ ș sinȥzKr rzz=+ Δ + − − − • Parameters for the root geometry Coordinates of the tangent point (with sinȥ Kh=Δ ;KT M= ; refer to Figure 2.3/9) w1 aP0sin(ȥș )( ) c o s ș Xr K=− − + N w1 aP0cos(ȥș )( ) s i n ș Yr K=− − + N (equivalent spur gearing!) Tooth root thickness sFn Fn2=s X Tooth root fillet radius ࣁFn 2 Fn aP0 w0 w1 w0 w1sinȥ+K Krr rr=+ +NN • Minimal tooth root fillet radius at the transition of the tooth root fillet to the root circle Minimum distance Kmin of the centre point M of the tool tip rounding from the contact point of the pitch circle T min M w0=−rr K Minimal tooth root fillet radius ࣁFmin minFmin aP02 min w0 w1 w0 w1sinȥ+K Krr rr=+ +NN 1 )Note: The equation to determine ȥ follows from 0 = ȥ – Q1 – ș with (refer to Figure 2.3/9) Q1 = Ȧ1 + ʌ/z1; Ȧ1 = (z0/z)1 Ȧ 0, Ȧ0 = į – ȥ – ǻĮ ; į = arccos( rw0/rM cosȥ) for the basic equation used here: 0 = ȥ (1 + z0/z1) – (z 0/z1) arccos( rw0/rM cosȥ) + (z0/z1) ǻĮ – ʌ/z1 – ș. The results are identical to those of the iteration according to the equation for ȥ in Figure 2.3/9 and Section 2.3. 712 Appendix 1 Tooth Root Geometry Appendix 1.2 Relative Tooth Root Curvature Radius for Basic Rack ISO 53 (DIN 867) c) with protuberance NaP0/mn = 0.4 haP0/mn = 1.4 spr/mn = 0.02 b) NaP0/mn = 0.38 haP0/mn = 1.25 no protuberance a) ࣁaP0/mn = 0.25 haP0/mn = 1.25 no protuberance Appendix 1 Tooth Root Geometry 713 Appendix 1.3 Relative Tooth Root Thickness (See Appendix 1.1) c) with protuberance NaP0/mn = 0.4 haP0/mn = 1.4 spr/mn = 0.02 b) NaP0/mn = 0.38 haP0/mn = 1.25 no protuberance a) NaP0/mn = 0.25 haP0/mn = 1.25 no protuberance 714 Appendix 1 Tooth Root Geometry Appendix 1.4 Notch Parameter (See Appendix 1.1) c) with protuberance NaP0/mn = 0.4 haP0/mn = 1.4 spr/mn = 0.02 b) NaP0/mn = 0.38 haP0/mn = 1.25 no protuberance a) NaP0/mn = 0.25 haP0/mn = 1.25 no protuberance Appendix 2 Details for calculating Y S and YįrelT according to ISO 6336-3 715 Appendix 2 Details for Calculating the Stress Correction Factor YS and the Relative Sensitivity Factor YįrelT According to ISO 6336-3 (DIN 3990-3) (1987 Issue) Due to the authors’ own experiences and findings, other relationships with the factors given below were specified in the previous sections. • The stress correction factor Y S in accordance with DIN 3990 (1987 issue) Ss1 1.21 2.3(1.2 0.13 )+=+LYL q (A2/1) Fn Fe Ls h= ()Fn F s 2 qs=N For s Fn, hFe, FN= FnNrefer to Figure 6.5/8, Equation (6.5/31) and Appendix 1.1 Note: YS was designated as the stress concentration factor in the authors’ own studies depicted • The dynamic sensitivity factor Y įrelT in accordance with ISO 6336-3 (DIN 3990, 1987 issue) for the calculation of the endurance limit The dynamic sensitivity factor YįrelT is the ratio of the sensitivity factor of the toothing to be calculated ( Yį) to the sensitivity factor of the toothing of the test gear (Y įT). Then YįrelT is calculated from the following equation: į įrelT įTYYY=\* T1Ȥ 1Ȥ∗′ += ′ +N N (A2/2) The slip-layer thickness can be taken from the following Table A2.1 as a function of the material. Table A2.1 Values of the Slip-layer Thickness ′N[mm] No. Basic material in accordance with ISO 6336-3, DIN 3990 (1987) ′N DIN 3990 (1987 issue) contains the following note: The equation for YįrelT applies to m = 5 mm. The size effect is included with YX. Explanation of abbreviations: NT nitrided wrought steels, nitriding steels NV through hardened wrought steels, nitrided, nitrocarburised CHD case-hardening depth IF induction- or flame-hardened St steel V through hardened GG grey cast iron GGG nodular cast iron GS cast steel ı B = Rm, ıS = Re, ı0.2 = Rp0.2 1 2 3 4 5 6 7 8 9 10 11 GG; ı B = 150 N/mm2 GG, GGG (ferr.); ıB = 300 N/mm2 GGG with increasing perlitic structure: approximation to steel NT, NV St, GS; ıS = 300 N/mm2 St, GS; ıS = 400 N/mm2 V, GS, GGG (perlitic, bainitic); ıS = 500 N/mm2 V, GS, GGG (perlitic, bainitic); ıS = 600 N/mm2 V, GS, GGG (perlitic, bainitic); ı0.2 = 800 N/mm2 V, GS, GGG (perlitic, bainitic); ı0.2 = 1000 N/mm2 case hardened, IF (in root fillet) 0.3124 0.3095 0.1005 0.0833 0.0445 0.0281 0.0194 0.0064 0.0014 0.0030 Related stress gradient: ()\* p s ȤȤ 12 q =+ (A2/3) where \* pȤ= 0.2 and \*Ȥ=\* TȤ for the standard reference test gear where qsT = 2.5. 716 Appendix 3 Flank **Backlash** and Flank Modifications Appendix 3.1 Recommendations for Selecting the Flank **Backlash** and the Base Tangent Length or Tooth Thickness Allowances The normal **backlash** (meshing **backlash**) j n is the shortest distance between the non-working flank of the teeth of a gear pair when their working flanks have contact. The following relationship exists between the normal **backlash** jbn and the circumferential **backlash** j wt (in the form of an arc on the pitch circle): () wt n w t bbcosĮcosȕ =jj The angle of the pinion ĳj1 is the angle by which the pinion can be rotated between the flank contacts when the wheel is not rotating: ()j1 bn 1 n nĳ 2/ c o s Į =jz m The size of the **backlash** is essentially determined by the tooth thickness allowances Esn. These can be substituted by the span allowances EWn (allowances in the direction of the flank normals). Wn b sncosȕ =EE If no special requirements or experiences exist, the sum of the amounts of the upper allowances can be selected from Figure 3.1b. For a uniform distribution, we have We1 We2 We1 We2 0.5 == − + EE E E The lower allowances are derived using the tolerance Tw for each wheel (1 and 2) as follows: Wi 1 , 2 W e 1 , 2 W 1 ,2 =− EE T In general terms, the tolerance TW1,2 lies in the range W1,2 Wn1,2 Wn1,2 ;0 . 0 3 t o 0 . 0 8 TT T == and is usually selected using TWn1,2 = 0.05 mm. Ignoring other influences (e.g., difference in thermal expansion, centre distance deviation), the normal **backlash** results from We1 We2 nm in(0)=+bjE E As a result of several influences, the actual **backlash** differs from jbn min(0) , both when mounting and in operation . Probably the smallest normal **backlash** jn min in operation is We1 We2 bn bnm in (0)=+− Δ jE E j 222 2 i1 i2 2 T GGȕ į bn aEl tw b'' '' rElsinĮcosȕ ȕ cosȕ()22 2 2 t a n Į 2t a nĮ2c o s ȕ Δ++ + + ⋅+ + + Δ≈ ⋅ ⋅ aFF f f b bbAf fLLj where Fi" is the double-flank total composite deviation; Aa centre distance allowance; fȈį inclination deviation; fȈȕ axial alignment deviation; b/LG face width over bearing distance; fT thermal expansion difference between the gears and the housing; faEl the closest approximation of the gears as a result of elastic deformations in the direction of the centre distance; ǻȕrEl angle difference of the gear pair shafts as a result of the elastic deformation perpendicular to the line connecting the gear centres (tangential). For a very precise analysis, it is necessary to take into account the bearing clearance and the housing deformation. For values, refer to ISO 1328, ISO/TR 10064, DIN 3963, and DIN 3964. Figure 3.1a **Backlash**, tolerances and deviations of base tangent length Figure 3.1b Sum of deviations and normal **backlash** versus centre distance For a few gears, for example, those with fast torque reversal and control gears , the ope- rating flank **backlash** should be as close to zero as possible. For high-speed gear units , however, the gear losses are partly reduced with a corres- ponding increase in flank **backlash** and tip clearance. For the minimal **backlash** used in other cases, refer to the diagram in Figure 3.1b. Appendix 3 Flank **Backlash** and Flank Modifications 717 Appendix 3.2 Flank Modification Flank modification is an intentional variation of the existing tooth flank as it is theoretically expressed in the basic parameters of the gearing. Its intention is • to minimise strain and/or • to minimise rotation angle variation, excitation of vibrations as the cause of noise in the gear unit. Deformations in the gearing and the surroundings of the gearing (shafts, bearings, housing) due to the torque transmitted and other external loads on the shaft can serve as a guide to the order of magnitude of the modifications. An exact recalculation demands a 3D view and the use of numerical methods (e.g., LVR [6.4/33], RIKOR [6.4/34]). An approximation method will first be given as follows, separately for the tooth traces (axial direction) and for the profile (addendum direction). The starting point is usually the mean values (expected values) and the fluctuations (a multiple of the standard deviation), delineated by the tolerance limits; refer to Section 6.4, Eq. (6.4/ 74). Axial direction: Expected values ȕi,f Fluctuations 2 ȕifΣ (A3.2/1) Addendum direction: Expected values Įi,f Fluctuations 2 ĮifΣ (A3.2/2) Modifications to the Tooth Trace The expected values are determined from the deformations or lowerings in the shafts, bearings, and housing due to the expected values of the load and bearing clearances. They then result in the expected values of the line of contact deviation of the gear pairing ( fi1, fi21). It is common practice to balance the deformations of the shafts and bearings resulting from the mean values (expected values) of the load, as well as the tilting that results from bearing clearances or bearing clearance differences, by changing the helix angle of one of the gears of the pairing [ tooth trace angle modification C Hȕ (Table A3.2/1, No. 1)]. In cases where major angle modifications become necessary, it is clearer and more accurate for the functionality, manufacture and gear testing to specify the modified helix angle instead of a tooth trace angle modification CHȕ, which in actuality is not really an angle change but rather a position change. Overloading of the tooth ends, such as by the protruding mating gear, can be minimised by tooth trace end relief CȕI,II (Table A3.2/1, No. 2). Fluctuations in the relative position of the tooth trace angle can be compensated for by tooth trace crowning Cȕ (Table A3.2/1, No. 3). Combinations of end re lief and crowning are to be approximated by a tooth trace modification, expressed by a parabola of higher order. Depending on the order of the parabola, different specifications can be given for ǻ Cȕ through the face width bF. As far as the efficacy of the gearing modifications goes, the effective sum of the modifications of pinion and wheel in the field of action are decisive; the modifications can only be effected by applying them to one gear or distributing them on the pairing. The tooth trace crowning Cȕ results from the statistically superimposed tooth trace alignment tolerances, the total position tolerances, and the fluctuations in the deformations and lowerings due to bearing clearance tolerances [refer also to Eq. (6.4/72) and (6.4/75)]. It is favourable to choose the crowning Cȕ for instance in ()ȕ2 ȕi0.4... 0.5 f C≈ (A3.2/3) The exact size, taking tolerancing into account, must be determined through numerical analysis. Often it is favourable for the course of the modification to choose a parabola of higher order, such as of third order. Modifications to the Profile For the profile, modifications CHĮ to the profile angle ( transverse profile angle modification , Table A3.2/1, No. 4) make sense when deformations of the gear rim (e.g., in the case of thin, elastic gear rims) lead to a general skewing of the entire tooth. More common are tip and root relief , which, however, should in no way merge angularly into the neighbouring contour (Table A3.2/1, No. 5). A parabolic gradient – also of third or higher order – is usually propitious. With regard to having an influence on transmission errors (fluctuations in stiffness), transverse profile crowning (height crowning) is often very effective (Table A3.2/1, No. 6). The size of the profile modification ( CHĮa, CHĮf) in the sense of relief in parts of the length of path of contact should correspond to the expected value of the tooth deformation fz0 chosen at a maximum. Often, for zones which are in particular danger (e.g., concerning pitting, scuffing), a larger relief is undertaken. 718 Appendix 3 Flank **Backlash** and Flank Modifications Appendix 3.2 Flank Modification (continued) In the case of spur gearing, when determining the length of the profile correction LĮa, LĮf, one differentiates between short and long profile correction (Table A3.2/1, No. 5, bottom). With respect to the rotation angle change, a short profile relief is more propitious than long relief, particularly in the partial load area. With respect to strain, both often produce no significant differences if one disregards superimposed additional dynamic loads. In the case of helical gearing, it is often useful to assume short profile relief with a parabolic gradient (e.g., a parabola of third order) and to determine the most favourable size of relief and length with the help of numerical methods. Spatial Modifications Spatial modifications are mainly useful for helical gearing. To be considered are diagonal relief (Table A3.2/1, No. 8), bias (Table A3.2/1, No. 9) and topographical modification (Table A3.2/1, No. 7). They can be utilised on the condition that precise numerical calculation methods and appropriate manufacturing options are used. Basic Remarks Too large of crowning both in the profile and in the face width reduces the resistance to damage. It is also unpro- pitious for the load-bearing capacity to stipulate an elliptical contact pattern at full load. Angular transitions in the profile and in the width are unfavourable because they cause local overloading of the tooth flanks. Since manufacturing is not possible without deviations, it also makes sense to allocate on the drawing a (+/–) tolerance to the nominal dimensions of the reliefs. Attention must be paid to the respective smallest dimension to be realised in terms of production. Control of the measurements with templates is usually too global and admits unfavourable formings. The consequences the actual variation (limits of size) will have for the load-bearing capacity should be determined mathematically. Table A3.2/1 Modifications and Their Indication on Drawings Modifications Possibilities for indication on drawings 1. Tooth trace angle modification CHȕ 2. Tooth trace end relief CȕI, CȕII Note: The gradient of the end relief just drawn is here merely the simplified depiction of the parabolic flank end modification, not the actual form strived for! Appendix 3 Flank **Backlash** and Flank Modifications 719 Appendix 3.2 Flank Modification (continued) Modifications Possibilities for indication on drawings 3. Tooth trace crowning Cȕ 4. Profile slope modification CHĮ 5. Tip and root relief CĮa, CĮf Short relief Long relief 720 Appendix 3 Flank **Backlash** and Flank Modifications Appendix 3.2 Flank Modification (continued) Modifications Possibilities for indication on drawings 6. Profile crowning CĮ 7. Topological modification 8. Diagonal relief CEa, CEf 9. Bias SĮ, Sȕ Indication of the size of the relief Cij on the points (i, j) Appendix 4 Manufacturing Profile Shift Coefficient xE (Definition, Calculation) 721 Appendix 4 Manufacturing Profile Shift Coefficient xE (Definition, Calculation) The sum of the profile shift factors determined in Section 2.1.2 and 2.1.3 – and thus their distribution on the gears of the pairing – apply to the zero-**backlash** mating. Since a zero-**backlash** operation is not possible, due, among other things, to the tooth system deviation and the thermal expansion, it is necessary to produce a smaller tooth thickness compared to the zero-**backlash** nominal value for production purposes. Figure A4/1 depicts a gear with nominal values and Figure A4/2 a protuberance tool. The tool with a straight-flanked, gear-rack-shaped basic profile in protuberance design represents the most common design in the manufacture of large gears. Figure A4/1 Allowances on a toothing Figure A4/2 Protuberance tool (depicted without undercut) (gear-rack-shaped basic profile) The following parameters are either defined or determined in the construction or are based on the manufacturing processes: • The greatest ( E si) and the smallest **backlash** allowance ( Ese). • The machining allowance p s in relation to the tooth thickness. • The change in the root circle ǻdf as a result of the radial infeed of the tool corresponding to the **backlash** allowances ( Es), the machining allowance ( ps) and the tooth thickness alteration ( ǻs0). Here, x1 and x 2 are the profile shift factors determined for the zero-**backlash** tooth pairing. If the minus allowances for tooth thicknesses are achieved by a radial infeed of the gear-rack-shaped tool, then other profile shift factors are crucial for the tooth thicknesses and tooth root geometry. These are referred to as generating addendum modification coefficients and denoted by xE1 or xE2 for the pinion or the wheel: 0 E nn n n nns 2t a nĮ sinĮ 2t a nĮEs qxxmm mΔ=+ + +⋅⋅ ⋅ (A4/1) where Es is the **backlash** allowance ( Es < 0), for which the tooth geometry should be determined q is the machining allowance perpendicular to the tooth flank ǻs0 is an increase in the tooth thickness of the tool (for example, to limit the radial infeed in case of quantitatively large allowances Es) Note: For precise analyses of the tooth root stress, it is necessary to determine, among other things, the combined tooth form factors YFS1,2 or form factors YF1,2 with xE1,2. For quantitatively large allowances and small modules, including for high-speed gear units, the influence of the generating addendum modification xE1,2 can be significant, particularly on the tooth root stress. 722 Appendix 5 Vibration Analysis of Complex Drive Systems with Multi-body Simulation Appendix 5 Vibration Analysis of Complete Drive Systems with Multi-body Simulation 1 Vibration analysis using multi-body simulation (MBS) A knowledge of stresses is of key importance in component dimensioning. In addition to the vibrations the load is subjected to, additional dynamic stresses caused by vibrations and un-favourable load distributions as a result of deformations also represent a problem. A detailed modelling of the components using the finite element method (FEM) or the boundary element method (BEM) allows the analysis of the deformations and stresses for the quasi-static load to be performed with a high degree of precision. These methods can also be used over several steps of a meshing period, such as to determine the quasi-static traces of the transmission error. A direct analysis of the vibrations involved in complex models, however, results in an unjustifiably high level of computing time. The use of a more simple vibration model with masses and spring-damper elements is better suited for the calculation of dynamic stresses. The vibration analysis can be very efficient carried out in the frequency range with analytical solutions for linear systems. As soon as larger-scale non-linearities take effect in the model, such as clearance in bearings and gearings, it is only possible to achieve analytical solutions using extremely high computing time and reduced precision. The calculation using time-domain integration lends itself to such cases. In this process, the displacement resulting in extremely short time intervals for an existing load initiation is calculated in advance, and the resulting load distribution is subsequently determined for the next stage of the integration process. The computing time is directly dependent on the time interval selected. Modern integration processes allow an internal adjustment of the interval as a function of the displacement profile calculated. Despite this adjustment, calculations, in which the stimulation from the tooth meshing is to be taken into account, result in the requirement to use an extremely short time interval, as the tooth stiffness is very high and would result in a divergent build-up of the tooth forces calculated in the event of too-long time intervals. In spite of this requirement involving a large number of computation steps, the time-domain calculation offers many advantages if the multi-body simulation (MBS) is used. This method foregoes the necessity of complying with a particular specified model structure required for the analytical treatment of the mathematical model in the calculation soft-ware. Typical MBS applications are in the investigation of the driving properties of vehicles on various types of road unevenness and when negotiating bends for the purpose of improving the chassis suspension and simulating the behaviour of the vehicle in emergency situations. Multi-body simulation is also used in the dimensioning of wind turbines. The increasing hardware power enables an ever finer detailing of the elements originally regarded as rigid bodies, and this also applies to tooth meshing. 2 Modelling multi-body systems As in the case of the classical spring-mass vibration model, it is necessary to divide up the mechanical system to be investigated into individual bodies as a basis for defining the inter-actions between them. Although in the simplest approach these bodies are rigid, for higher-quality analyses a linear elastic behaviour – determinable in separate calculations – can also be assigned to them. The interactions are described using massless coupling elements that are able to map the viscoelastic and elastoplastic behaviour of elastic elements. The best approximations to the real behaviour of the system are achieved by modelling using elastic bodies. Appendix 5 Vibration Analysis of Complex Drive Systems with Multi-body Simulation 723 Appendix 5 Vibration Analysis of Complete Drive Systems with Multi-body Simulation (continued) The elastic component deformations can be very easily combined with the non-linear kinematics of the body frequently resulting from connections with inherent play and non-linear coupling elements. The modelling can ensue completely on a three-dimensional basis in translation and rotation, whereby large shifts and distortions are also permissible. This modelling freedom is a crucial advantage compared to modelling with classical vibration models, which were often restrictted to rotary vibrations in the area of drive technology or additionally only contained individual translational degrees of freedom. Furthermore, the graphical support provided by modern calculation programs in mapping the structure of the model and assembling the vibration model itself has proved extremely beneficial for our modelling understanding. The extensive use of substructure techniques, with which various levels of the model assembly can rapidly and clearly be made visible, has also contributed to this understanding. At the same time, substructures in the model can be used on a repeated basis while also facilitating the assembly of the model if they are made available as basic detailed elements by the analytical software. The substructures themselves are MBS models, which can be readily replaced in the overall model for the purpose of simplification or more precise detailing. As far as product development is con-cerned, MBS still possesses the decisive advantage that the modelling does not have to remain restricted to the mechanical map of the system because it is possible to incorporate not only hydraulic controls but also the effect of electrical control and regulation elements. One example of this is to be seen in the special “hardware in the loop” MBS variant that makes use of complex MBS models to simulate vehicle behaviour in order to test the effectiveness of control software. In this early development process, this is already possible well before the existence of the real hardware in the form of the vehicle itself. Alongside the ADAMS/Simulink basic software [A5/1], [A5/2] for MBS calculations, a large number of program packets can be used with extensive model catalogues, e.g., [A5/3] and [A5/4]. 3 Mapping the tooth meshing in the multi-body model Various levels of detailing can be used to record the meshing in the MBS model, ranging from a simple approach via a spring-damper element with constant stiffness through to the mapping of the effective stiffness via multi-dimensional characteristic diagrams as a function of centre dis-tance, centre tilting and load. The constant gearing stiffness is then sufficient when the excitation resulting from the meshing has scarcely any influence on the calculation results, such as with excitation in the rotational frequency of the input shaft. As soon as the tooth force vibrations are of interest over a mesh period or it is a question of investigating the effects of the meshing on the operating noise, then a more precise modelling of the meshing is required. The calculation of the effective gearing stiffness using a detailed contact model for each time step results in a clearly too-high computing time. Simple approaches for the definition of the stiffness vibrations are often implemented in the MBS programs. The nominal value of the stiffness of a tooth pair is de-termined in accordance with ISO 6336 (refer to Section 6.2.3). Correction factors are used for considering the effects of tip relief and the changing length of the line of contact at the beginning and end of contact on helical gears. The effective gearing stiffness ultimately results from the superpositioning of the stiffnesses of all the tooth pairs in contact. 724 Appendix 5 Vibration Analysis of Complex Drive Systems with Multi-body Simulation Appendix 5 Vibration Analysis of Complete Drive Systems with Multi-body Simulation (continued) The use of a Fourier series as the stiffness function over a mesh period (refer to Section 6.3.4.4) is particularly suitable if the results of separate tooth contact analyses are to be used for the stiff-ness variations. In the simplest case, the influences of shifts and tilts of the wheels on the contact conditions are neglected, and the gearing stiffness to be used is determined as a function of the mesh position only. In order to minimise the excitation of vibrations and improve the load dis-tribution, flank modifications can be used, which affect the actual tooth deformation under load exactly like the mesh misalignments, and thus influence the stiffness effective in a given meshing position. As long as the calculation for a particular load condition ensues with no major tooth load vibrations, the use of the stiffness trace for this nominal load is sufficiently precise. If the progression of the tooth deformation calculated in a tooth contact analysis is divided into an elastic deformation component and a component from modifications and mesh misalignments, it is possible to take account of the load sensibility in the immediate proximity of the nominal load used with more accuracy. In order to record the resulting tooth deformation over a larger load range as precisely as possible in the calculation, it is necessary to calculate the tooth deformation curves in advance and interpolate the tooth stiffness to be used as a function of load and mesh position in the MBS model. In order to take into account the influences of centre distance alte-rations resulting from shifts and the effective axial skew due to the wheels in the tooth contact, several levels of detailing are again available. The influences of tilt on the load distribution along the face width can be very well recorded using a toothing model by means of several tooth slices. If the effective tooth stiffness for each time step is calculated from the centre distance currently present at the tooth slice in question (by means of approximations), then a rough estimation of the influences of the change in the local transverse contact ratio on the stiffness is thereby achieved. In order to record the influences of flank modifications on the load distribution and the effective gearing stiffness thereby resulting as accurately as possible, separate tooth contact analyses are required in association with variations of the axis misalignment and the centre distance for each load condition. The stiffness applicable for a given time step must be deter-mined by means of multi-dimensional interpolation from the matrix of stiffness traces as a function of the axial bearings. 4 Sample calculations The point of inflection of the resonance curve for a spur gearing (refer to Section 6.3.4.6), at which a flank separation occurs under load as a result of the high fluctuation of the tooth stiff-ness, can be determined with an MBS model with no increased calculation effort. Other calcu-lation methods are either completely unable to record such non-linearities – or if they are, then only with an extremely high effort. Figure A5/1 depicts the calculation results for such a simu-lation [A5/1], which also shows the differences between the increasing and decreasing speed curves. Figure A5/2 depicts the 3D calculation model for a wind turbine. Of particular interest here are the actual strains on the gear as a function of the vibrations in the system. The noise excitations resulting from the meshing under real operating conditions are also an objective of the MBS calculation. For this purpose, the gear toothing is modelled in detail. With such com-plex models, it is possible to relatively quickly obtain statements relating to the design of the gears while retaining clarity – such statements hardly being possible on the basis of measure-ments to a comparable degree. Appendix 5 Vibration Analysis of Complex Drive Systems with Multi-body Simulation 725 Appendix 5 Vibration Analysis of Complete Drive Systems with Multi-body Simulation (continued) Figure A5/1 Variation in the angle of rotation for a gear with inherent **backlash** at increasing and decreasing speeds with the point of inflexion in the resonance curve in the region of the flank 726 Appendix 5 Vibration Analysis of Complex Drive Systems with Multi-body Simulation Appendix 5 Vibration Analysis of Complete Drive Systems with Multi-body Simulation (continued) Figure A5/2 Example of a wind turbine with detailed modelling of the gear [A5/5] 5 Special literature on Appendix 5 [A5/1] ADAMS Advanced Multibody Simulation, MSC Software Corporation, Santa Ana, CA [A5/2] SIMULINK Simulation and Model-Based Design, The Math Works, Inc., Natick, MA [A5/3] SIMULATIONX MKS-Simulationsumgebung der ITI GmbH (ITI Company MBS simulation environ- ment), Dresden, Germany [A5/4] SIMPACK MKS-Simulationsumgebung der INTEC GmbH (INTEC Company MBS simulation environment), Wessling, Germany [A5/5] SIMPACK GEARWHEEL Toolbox und Interne Dokumentation zur Simulationsumgebung SIMPACK der INTEC GmbH (Toolbox and internal documentation on the SIMPACK simulation environment), Wessling, Germany Eickhoff Antriebstechnik GmbH I Am Eickhoffpark 1 I D-44789 Bochum I Tel. +49 234-975-0 I Fax +49 234-975-2579 I www.eickhoff-bochum.deANTRIEBSTECHNIK GEARBOXES FOR WIND TURBINESLARGE FORCES EFFICIENTLYCONTROLLING Appendix 6 Supplement for Internal Gears 729 Appendix 6 Detailed Calculation of the Stress Concentration Factors at the Tooth Root Fillet of Internal Gears A: Solid toothed rim: Tip factor Y FS Y FS = (YFa + YFad)YSab (A6/1) YFa according to Equation (6.5/34c); Fan n Fad nF nsin cosĮmYs =− ⋅ α; 0.3388 Fn Sab Fn21.418 Ys− = N (A6/2) Remark: YFa 0, Y Fad 0 B: Elastically designed toothed rim: valid for: sR/sFn 0.7, N Fn/mn = 0.15 to 0.3, 0.04 2N Fn/sFn 0.25 Stress concentration factors at tangent angle R = 60 on the root fillet: Due to Ft: ()2.561 0.3388 0.055 ımax t RF n R SFt Fn Fn Fn180ș 20.324 1.418 cos 1.2 60ʌ−− ⋅ =+ ⋅ ⋅⋅ − DF ssYsssN (A6/3) () ()0.54 Fn Fn ımax t 1.6 Fn nRF n2 0.562ș 7.662 4.590 =− + ⋅ Fsm ssNN[rad] (A6/4) for ımaxș∞ 35ʌ 180D D, ımaxș∞=35ʌ 180D D has to be used Due to MbK (bending moment in the rim): SMbY =()1.102 0.3045 ımax bK RF n Fn Fn180ș 20.201 1.595 cos 1.4 60ʌ−− ⋅ −+ ⋅ ⋅ ⋅ − DM s ssN (A6/5) ()Fn Fn ımax bk Fn n2ș 4.014 3.351 1.239 =− + + MsmNN [rad] (A6/6) for ımaxș∞ 35ʌ 180D D, ımaxș∞=35ʌ 180D D has to be used Due to compressive load: ımax sad sab180școs 1.3 60ʌYY∞ ⋅ =⋅ − ; 0.3388 Fn Sab Fn21.418− =⋅ YsN (A6/7) 0.54 Fn Fn ımax Fn n2ș 7.662 4.590∞ =− + smNN [rad]; (A6/8) for ımaxș∞ 35ʌ 180D D, ımaxș∞=35ʌ 180D D has to be used 730 Appendix 7 Supplement for Load Capacity Based on Hertzian Pressure Appendix 7 Supplement for Load Capacity Based on Hertzian Pressure Curves of Strength and Stress Normal to the Flank in the Depth Direction for Case-hardened Gears A more precise analysis of the flank load capacity of the surface area of hard edge layers requires the comparison of the strain and strength depth profiles for the equivalent continuous load and for the maximum load occurring (impact load) . The following text provides calcu- lations on this issue (partly as approximations) for the case-hardened layer. These statements also apply on an analogous basis to other edge layers, such as nitride layers. 1 General information The experimental determination of the sustainable Hertzian contact stress generally ensues on relatively small test beds with low numbers of teeth ( a = 90 to 160 mm, z = 16 to25). Here the maximum strain is to be found near the surface. Furthermore, the flash temperature in contact, the bending at the meshing region subsequently overrolled, the shear load due to friction and the decrease in strength (drop in hard-ness), usually present with the residual compression stress relief as a result of surface oxidation or surface decarburisation and tempering effects when finishing, cause a displacement of the maximum strain nearer to the surface. The local friction coefficients, which are greater in size than the mean friction coefficients, also result in a shift of the maximum strain to the outer edge zones . Because of this, the pits can also start from cracks forming on the surface. For large gears, the maximum strain lies deeper under the surface. Another, smaller strength (hardness and residual stress) then occurs, which often results in a lower load capacity that is not explicable with the normal calculation. The case-depth curve is of fundamental impor-tance. It has been depicted in a generalised manner for two typical materials from several measurements via related values (Figures A7.1, A7.2 ). Figure A7.1 Related case depth curve, 16MnCr5, case-hardened and tempered. CHD550 is the case hardness depth at hardness 550 HV. Appendix 7 Supplement for Load Capacity Based on Hertzian Pressure 731 Appendix 7 Supplement for Load Capacity Based on Hertzian Pressure Curves of Strength and Stress Normal to the Flank in the Depth Direction for Case-hardened Gears (continued) Figure A7.2 Related case depth curve, 18CrNiMo7-6, case-hardened and tempered 2 Fatigue strength The calculation of the strain along the line of contact ensues as per Section 6.5.1. It is recommended that the depth zıvGmax of the maximum strain of the equivalent stress under the surface be determined. For this purpose, Equations (6.5/2b) to (6.5/3a) – or in prepared form with Equations (7.4/15) and (7.4/16) – are available. The flank strength ı Hlim is dependent on the ratio of this depth zıvGmax to the case-hardening depth CHD . For ( zıvGmax/CHD > 1), a significant fall-off of ı Hlim compared to the value of ıHlim specified in Section 7.4, Table 7.4/20, is to be expected. Even in the ratio lying in the boundary condition ( zıvGmax/CHD ĺ 1), a fall-off of the endurance flank strength is also to be expected compared to the value determined on small gears, commonly specified in tables due to the drop in hardness and the changed residual stress, pointed out by the hardness characteristics depicted in Figures A7.1 and A7.2 as well as the drop in the residual compression stress. Test results on the fall-off of ı Hlim, which is roughly linear with the hardness, are available ( Niemann and Glaubitz , VDI journal, 21 February 1951, Volume 93). It may therefore be concluded that the position of maximum strain should lie as far as possible under the limit of the case- hardening depth (recommended: zıvGmax≤0.7CHD ) or, alternatively, that the fall-off in strength be taken into account: ıHlim(z > 0)/ıHlim(z = 0) ≈HHV (z > 0)/ HHV (z = 0) (A7/1a) 3 Exceeding the local yield stress: Plastic deformation The hardness of the hardened case merges into the core strength at approximately a depth of (z = 2 CHD) . Previous experience has shown that it is sufficient to set the equivalent stress ıvG in this depth (z = 2 CHD) against the yield stress Rp0.2. The safety S is given by S = Rp0.2/ıvG(z = 2 CHD ) ≥ 1 (A7/1b) When there are no special values available, the yield stress R p0.2 of the core material can be assumed as follows: 16MnCr5, where Hcore ≈350 HV, Rm ≈1200 N/mm2: Rp0.2 ≈750 N/mm2 18CrNiMo7-6, for Hcore ≈400 HV, Rm ≈1300 N/mm2: Rp0.2 ≈850 N/mm2 732 Appendix 7 Supplement for Load Capacity Based on Hertzian Pressure Appendix 7 Supplement for Load Capacity Based on Hertzian Pressure Curves of Strength and Stress Normal to the Flank in the Depth Direction for Case-hardened Gears (continued) For the calculation of the equivalent stress depth curve to check the occurrence of plastic deformations , as a result of which pitting damage and fractures by the tooth above the tooth root can start ( flank fractures ), the maximum load should be used as the basis, or, alternatively, an application factor K Ast should be assumed for impacts. The equivalent stress is derived from the three stress components. These components, based on the Hertzian contact stress, are depicted in Figure A7.3 or taken from Equations (A7/2) to (A7/5). The stress ız for the depth coordinate z = 0 corresponds to the Hertzian contact stress [ ız(z =0) = ıH]. In the region of transition to the core hardness, the von Mises’ maximum distortion energy theory can be considered as valid (ductility) and the residual stress neglected (as an approximation) . z / b H Figure A7.3 The three stress components related to TH at Hertzian contact and the equivalent stress VGı according to von Mises’ maximum distortion energy theory; for the depth curve refer to Equations (A7/2) to (A7/5), [7.4/86] 2 xHHHı 2Ȟı 1zz bb =− ⋅ + − (A7/2) 2 H yH2H H12 ıı 2 1z b z bz b + =− − + (A7/3) H z2 Hıı 1z b=− + (A7/4) () () ()22 2 VG x y y z z x1ıı ı +ıı +ıı2 =⋅ − − − (A7/5) The following approach is expedient for the practical calculation : - Calculate the Hertzian contact stress ıH for the maximum load. - Calculate the half-width of the Hertzian contact deformation bH as per Equation (6.5/3) or (6.5/3a) and the stress components ıx,y,z for the depth z = 2 CHD [Equations (A7/2) to (A7/5)]. - Calculate TVG as per Equations (A7/2) to (A7/5) or ( TVG/TH) from Figure A7.3 for z/bH where z = 2CHD . We then obtain TVG = (TVG/TH) TH. - Proof of avoiding plastic deformation below the surface [Equation (A7/1b)] and possible consequences, such as flank fracture and spalling. Tx cTH Ty cTH Tz cTH TVG cTH Appendix 8 Load-dependent Transverse Contact Ratio 733 Appendix 8 Load-dependent Transverse Contact Ratio (refer to Figure 3.2/2) AgΔPre-meshing EgΔ Post-meshing (0)İİ ǻ İ (A8 /1) =+ () e0İpAE= (A8/1a) eE Aǻİpg gΔ + Δ= (A8/1b) )Įcos(ʌe ⋅ ⋅ = m p ()()1E 2A 2E 1Ab1 b2 1E 2A 1E 2A t 0 e11 11İİrrzf zf FKp ⋅⋅ + ⋅⋅++≈+NN NNNN (A8/2) valid for gΔ< <N, 05K.≈ ȈfA Relative shift of the tooth pair coming into contact at A (A ƍ) as a result of the elastic deformation (and of the positional deviation) of the other tooth pairs already in contact ȈfE Relative shift of the tooth leaving the contact at E (E ƍ) due to the elastic deformation (and positional deviation) of the other tooth pairs remaining in contact Calculation : ȈfA,E from the sum of the deformations and positional deviation at A and E, respectively. For deviation-free conditions, ȈfA,E = fz0 , fz0 from Equation (6.4/43). Note: A precise calculation is also possible with the program LVR (refer to Section 6.4). 734 Appendix 9 Optimisation of Tooth Root Fillet Appendix 9 Optimisation of Tooth Root Fillet 1 Influence of tooth root fillet on the tooth root stress The maximum local tooth root stress is essentially influenced by the load initiation (tooth force, load application angle, bending moment arm) and the tooth root geometry (tooth root thickness, fillet radius). This can be considered as the product of a nominal stress and a stress correction factor, comprising the notch effect. If the fillet radius can be designed as large as possible, then only a low notch effect will occur on the tooth root. Rolling processes, in which the tools are equipped with a circular arc tip rounding, are often used for the manufacture of gears. Rolling results in a trochoid, which exhibits a vari-able curvature along the tooth root fillet. For forming processes, the root fillet is a direct image of the tool geometry. Both types of processes result in the maximum possible circular tip rounding of the tool based on the specification that the centre point of the tip rounding lies on the centre line of tool profile and the tooth root fillet produced ends at the root form circle required. Although the tooth root fillet thus generated exhibits a low notch effect, a clear increase in stress occurs at the critical root cross section in the region of the 30 tangent. It is not possible to achieve an almost uniform distribution of stress along the tooth root fillet with a circular-arc tool tip rounding. Figure A9/1 depicts tooth root stress distributions along a tooth generated with tools having different kinds of tip rounding. The full tip rounding allows a reduction in stress of 12%, while an optimised root fillet provides for a further 17% stress reduction. The target of a low stress, which should be as constant as possible, is achieved with this optimised curve. Figure A9/1 The potential of the optimised tooth root fillet in relation to stress reduction 2 The potential of the optimised tooth root fillet with regard to stress reduction Figure A9/1 depicts an example of an optimised root fillet. In contrast to root fillets generated with tools having a circular-arc tip rounding, the flank curvature is extremely low in the region of the 30 tangent (large radius of curvature). Although there is a sizable curvature present in the region of the centre gap, this does not result in a critical stress maximum in the tooth space centre. Various strategies exist to find an “optimal” root geometry. Most of these processes are based on an iterative modification of the root geometry as well as a calculation of the resulting local stress using either the boundary element method or the finite element method. The involute of the tooth and the tip edge path of the mating gear are assumed in the elliptical-arc root fillet presented below. An iterative stress calculation is not required. The root fillet determined leads to a 10% to 20% reduction of the maximum tooth root stress compared to the Appendix 9 Optimisation of Tooth Root Fillet 735 Appendix 9 Optimisation of Tooth Root Fillet (continued) manufacture using hobs with full tip rounding. Although this reduction is not necessarily the maximum improvement that can be achieved, the remaining potential for any further reduction nevertheless is low. Even the definition to define the condition for the optimum is questionable because it is possible to determine an optimal root fillet for each load application point, which is no longer the optimum for other load application points. An optimum is found when the local stress remains practically constant over a large region of the tooth root fillet. If a root fillet is optimised for the load application on the tooth tip, the stress maxima for load applications with a lower lever arm are to be found in the vicinity of the tooth space centre. This displacement of the stress maximum in this region may have additional negative consequences since this is where the compressive stress leads to an increase in the reverse load when the adjacent tooth is subject to load. 3 Optimised “elliptical” root fillet An optimised root fillet can, for example, be generated by a combination of an ellipse and a circular arc. The ellipse describes the root fillet in the region of the tooth space centre, while the circular arc is used to connect the ellipse to the involute. Figure A9/2 depicts the first step in the design of the root fillet. The starting point is the involute and the path of the tip edge of the mating gear, obtained with a tip circle radius increased in size by Δr a2 for the optimisation. An increase in the tip circle is recommended in order to prevent interferences of the mating gear tooth tip with the optimised root fillet. For the design of the ellipse, a point P T is determined along the path of the tooth tip edge, which exhibits a specific tangent angle ξ as a function of the force application specified for the optimisation. Both the distance xT from the tooth space centre line and the tangent angle ξ are used to determine the ellipse (refer to Figures A9/3, A9/4). A point P1 is defined on the ellipse as the starting point for the circular arc, which describes the root fillet up to the starting point of the involute P 0. While the gradient of the root fillet is continuous at the transition from the ellipse to the circular arc, it is not necessarily continuous at the transition from the circular arc to the involute. This does not lead to any unintended increase in the local stress because a bulge – and not a local recess in terms of a notch – arises at the point where the involute begins. The definitive root fillet is depicted in Figure A9/5. The design of an optimal root fillet presented is clearly deter- mined by the gear and mating gear parameters. The only free parameter is an additional increase in the tip diameter of the mating gear used in the curve calculation to avoid tooth-tip interferences. The increase of the tip diameter selected should not be too large in order to keep the associated increase in the Figure A9/3 Ellipse through the point PT Figure A9/4 Definition of the circular arc by the points P0 and P 1 Figure A9/2 Tangent point PT on the tip edge path of the mating gear 736 Appendix 9 Optimisation of Tooth Root Fillet Appendix 9 Optimisation of Tooth Root Fillet (continued) tooth root stress (due to the greater lever arm) as small as pos- sible. The form of the root fillet is influenced by the tangent angle ξ specified. A smaller tangent angle is suitable for a force application close to the tip circle, while for low-lying force applications (external point of single tooth contact associated with a greater transverse contact ratio) a tangent angle greater than 40 is sensible. The most favourable angle can be iteratively derived from calculated distributions of tooth root stress. Figure A9/5 Optimised root fillet 4 Tooth root stresses on hobbed and optimised root fillets Some sample calculations should indicate the reduction of the maximum local tooth root stress achievable with the optimised root fillet. The stress gradient was calculated using a boundary element method [A9/1, A9/2]. The hob used possesses a full tip rounding, enabling the root fillet generated to represent an optimum for a circular tool tip rounding. A further reduction in stress can still be achieved using the opti-mised ellipse-shaped root fillets. Standard gears with z = 44 and x = 0 (or z = 21 and x = 0.5) were used in Examples 1 and 2 (Figure A9/6 and A9/7). The force application was offset by 0.4 times the mesh position of the load application on the tooth tip, corresponding to a force application at the external point of single tooth contact for a transverse contact ratio of İ Į = 1.4. In Example 3 (Figure A9/8), an extra- depth gear with the number of teeth z = 35 and x = 0.5 was used, while in Example 4 (Figure A9/9) an internal gear with z = –60 and x = –0.5 was used. For external gears, the reduction in stress calculated lies in the range of 10% to 15%. For internal gears, a corresponding reduction of 28% is achieved since the shaper cutter possesses a relatively small tip rounding radius. Optimisation of Tooth Root Fillet Figure A9/6 Distribution of stress on a standard gear with the number of teeth z = 44, x = 0; left side: with traditional hob; right side: optimised ( 12.5% stress reduction ) FFσmax = 146.6 MPa at θ = 29.6 F σmax = 167.6 MPa at θ = 35.8 Appendix 9 Optimisation of Tooth Root Fillet 737 Appendix 9 Optimisation of Tooth Root Fillet, continued Figure A9/7 Distribution of stress on a standard gear with the number of teeth z = 21, x = 0.5; left side: with traditional hob; right side: optimised ( 10% stress reduction) Figure A9/8 Distribution of stress on an extra-depth gear with the number of teeth z = 35, x = 0.5; left side: with traditional hob; right side: optimised ( 14.7% stress reduction) Figure A9/9 Distribution of stress on an internal gear with the number of teeth z = –60, x = –0.5; left side: with traditional shaper cutter; right side: optimised ( 28% stress reduction) ( z0 = 30, ࣁa0 = 0.2 mn) Fσmax = 124.5 MPa at θ = 60.2 F σmax = 173.5 MPa at θ = 49.3 Fσmax = 180.7 MPa at θ = 24.5 Fσmax = 211.8 MPa at θ = 29.8 Fσmax = 130.4 MPa at θ = 51.7 F σmax = 144.9 MPa at θ = 37.2 738 Appendix 9 Optimisation of Tooth Root Fillet Appendix 9 Optimisation of Tooth Root Fillet, continued 5 Tools for optimised tooth root fillets The optimised root fillet can be manufactured in the rolling process, as depicted in Figure A9/10. The tool contour corresponds to the conjugate gear for the optimised tooth contour. Tool contours for forming processes can be similarly derived directly from the contour of the opti-mised root fillet. In the event that only the flanks of the optimised gear are finished in a hardened state, it is necessary in the finishing to extremely precisely tune the contour of the tool to the optimised pre-toothing root fillet in order to avoid unintentional processing approaches in the finishing. Figure A9/10 Manufacture of gearing unit as per Figure A9/6 using a hob; left side: meshing position on manufacturing; right side: profile of the hob 6 Literature [A9/1] Brechling, J. Zur Berechnung der Spannungsverteilung im festen K rper im Fall des ebenen Spannungs- oder Verfor- mungszustandes infolge u erer Belastung (On calculating the stress distribution in a rigid body in the case of uniform stress of deformation conditions due to external load). Maschinenbautechnik 18 (1969) 12 [A9/2] PC Program PCSpann “Spannungsverteilung an evolventischen Verzahnungen” (Stress distribution on involute gears); Version 1.3 2/2000, TU Dresden, Institute for Machine Elements and Machine Design -6-40246 -6 -4 -2 0246 x [mm]y [mm]1 x Appendix 10 Special Involute Gears 739 Appendix 10.1 Influences of an Enlarged Pressure Angle and Reduced Tooth Height, Example: Gearing data chosen for low gearing losses! General (constant) data: a = 116.5 mm, ȕ = 10 , mn = 3 mm, b = 70 mm, ࣁaP0/mn = 0.4, İȕ = 1.29, z1 = 25, z2 = 49, pro = 0.1 mm, Įpro = 10 , = 0.08 Figure A10.1/1 Gearing according to ISO 53 (DIN 867) with haP0/mn = 1.4, x1 = 0.5, x2 = 0.902; İĮ = 1.339, İȕ = 1.29, İȖ = 2.629 Figure A10.1/2 30 gearing with reduced tooth height with haP0/mn= 1.0, x1 = 0.5, x2 = 0.8205; İĮ = 0.727, İȕ = 1.29, İȖ = 2.017 Table A10.1/1 Ratios of Strains and Losses TH30 /TH20 TF1\_30 /TF1\_20 TF2\_30 /TF2\_20 K30 /K20 kk 30 20ff 30 30 20 201Ș 1ȘV V −=− 1.13 to 1.22 1.008 to 1.004 0.833 to 0.85 0.98 to 0.809 2.25 to 0.96 0.5 to 0.63 Where: K flash temperatures; f deformation variation along a pitch; V gear losses; 30 (Index): Gearing with Į = 30 ; 20 (Index): Gearing with Į = 20 Notes: In the calculations with LVR [6.5/105], a load was chosen for ıH 200 to 1000 N/mm2. The ranges of strain given for ıH30 /ıH20 , and so on, are due to the load dependency of the transverse contact ratio, which LVR takes into account. Here, the optimum was not sought fo r the Hertzian stress factor and bending stress. A significant reduction of losses is possible with this and a similar design, but it is usually associated with an increase in the Hertzian stress factor and possibly other stresses . 740 Appendix 10 Special Involute Gears Appendix 10.2 Symmetric and Asymmetric Gearing (Calculation of the Stresses by D. Kr ger [4/6]) 1 Symmetric gearing a) z = 20 b) z = 50 Figure A10.2/1 Stress distribution, profile: ISO 53 (DIN 867), İĮ = 1.2, x = 0, ȕ = 0 , haP0 =1.25 mn, ࣁaP0 = 0.38 mn a) z = 20 b) z = 50 Figure A10.2/2 Stress distribution, Į = 28 , İĮ = 1.2, x = 0, ȕ = 0 , haP0 =1.40 mn, ࣁaP0 = 0.1 mn, spr = 0.1 mm (protuberance), mn = 4 mm Figure A10.2/3 Stress gradient depending on the rotating angle Figure A10.2/4 Smith chart with modified strength at the root, caused by the spread of stress from the neighbouring tooth Appendix 10 Special Involute Gears 741 Appendix 10.2 Symmetric and Asymmetric Gearing (continued) (Calculation of the Stresses by D. Kr ger [4/6]) In the case of symmetric gearing according to ISO 53 (DIN 867) and the full gear body, the tooth root breakage generally starts from the tension side. The absolute value of the stress at the compression side is greater. Since the strength is also greater with a decreasing mean stress (direction) compression range, the tension side remains an area where cracks begin. In the analysis of the load capacity of the tooth root, running tests are carried out in order to ob- tain the strength value. Thus, the influence of the stress from the neighbouring tooth is globally recognised. In contrast, also due to this influence, the strength values determined in the pulsator test differ (slightly) from the results of the running tests due to the lack of stress superposition by the neighbouring tooth. 2 Asymmetric gearing In the case of asymmetric gearing, the stress on the compression side – with greater differences in the profile angle of the load and non-working flank – is generally not insignificant and is often a decisive influence. The double amplitude of the stress and the modified strength at the root with the safety case should then be taken as a basis. a) z = 20 b) z = 50 Figure A10.2/5 ĮI = 28 , ĮII = 18 , İ Į = 1.2, x = 0, ȕ = 0 , haP0 = 1.4 m n, ࣁaP0 = 0.31 mn, spr = 0.1 mm (protuberance), mn = 4 mm a) z = 20 b) z = 50 Figure A10.2/6 ĮI = 35 , ĮII = 18 , İ Į = 1.2, x = 0, ȕ = 0 , haP0 = 1.4 m n, ࣁaP0 = 0.14 mn, spr = 0.1 mm (protuberance), mn = 4 mm 742 Appendix 10 Special Involute Gears Appendix 10.2 Symmetric and Asymmetric Gearing (continued) (Calculation of the Stresses by D. Kr ger [4/6]) a) z = 20 b) z = 50 Figure A10.2/7 Stress distribution: ĮI = 45 , ĮII = 20 , İĮ = 1.2, x = 0, ȕ = 0 , haP0 = 1.15 mn, ࣁaP0 ≈0, spr = 0 (no protuberance) On closer analysis, the strains in the root fillets must be examined point by point. The double amplitudes are to be formed first. On the tension side, the compressive stresses are to be superimposed, which arise due to the load of the neighbouring tooth and spread to the tension side. For the compression side, it applies analogously that the tensional stresses, which taper off only in the neighbouring root fillet during the loading of the neighbouring tooth, are to be superimposed. However, the maximum of the double amplitude does not yet represent the location of the minimal safety for the respective boundary point. It must be determined only by the strength value applicable to the respective location. This is dependent on the mean stress ıFm. The inclinations of the limiting lines for the upper and lower stress of the Smith chart determine its influence. Whether the smallest safety is calculated on the tension side or on the compression side depends not only on the tooth geometry, but essentially on the ratio of the pure fatigue limit ıFEw to the pulsating fatigue strength ıFE. Considering initially only the tension side, for which the tooth root load capacity analysis is usually performed according to ISO 6336-3 in the case of symmetric teeth (ISO 53), the safety SF results in a simplified form as follows: ıFE FE FFı ıFfSf= (A10.2/4) Here, fıFE represents the increase in the allowable double amplitude compared to the pure pulsating fatigue strength ıFE, and fF represents the increase in the pure pulsating stress ıF as a result of the spread of the stress of the neighbouring tooth. Up to a profile angle ĮI 28 of the load flank and at a ratio of ıFEw/ıFE, fıFE/fF is 1 and, as for the symmetric gearing according to ISO 6335-3, the safety case can be calculated at the tension side with the stress ıF, which is present there, and with the strength value ıFE specified in ISO 6336-5. Literature: [4/5] to [4/10] - Standard safety factor: ()()FE F0 F1Sσ=σĭ (A10.2/1) - Safety factor from the span of the stress amplitude: FE F(1) FF 21() ()S=+σ σσĭĭ (A10.2/2) - Safety factor, such as Equation (A10.2/2), but with consideration of the increasing strength with decreasing mean stress: ()FE mod F2 FF 21() ()S =+σ σσĭĭ (A10.2/3) Figure A10.2/6 Safety relations (qualitative representation) Appendix 11 Hardness and Tensile Strength for Unalloyed and Low-Alloyed Steels 743 Appendix 11 Hardness and Tensile Strength for Unalloyed and Low-Alloyed Steels Tensile strength Vickers hardness Brinell hardness Rockwell hardness Tensile strength Vickers hardness Brinell hardness Rockwell hardness <N/mm > [HV10] [HB] [HRC] <N/mm > [HV10] [HB] [HRC] 255 80 76 1155 360 342 36.6 270 85 80.7 1190 370 352 37.7 285 90 85.5 1220 380 361 38.8 305 95 90.2 1255 390 371 39.8 320 100 95 1290 400 380 40.8 335 105 99.8 1320 410 390 41.8 350 110 105 1350 420 399 42.7 370 115 109 1385 430 409 43.6 385 120 114 1420 440 418 44.5 400 125 119 1455 450 428 45.3 415 130 124 1485 460 437 46.1 430 135 128 1520 470 447 46.9 450 140 133 1555 480 456 47.7 465 145 138 1595 490 466 48.4 480 150 143 1630 500 475 49.1 495 155 147 1665 510 485 49.8 510 160 152 1700 520 494 50.5 530 165 156 1740 530 504 51.1 545 170 162 1775 540 513 51.7 560 175 166 1810 550 523 52.3 575 180 171 1845 560 532 53 595 185 176 1880 570 542 53.6 610 190 181 1920 580 551 54.1 625 195 185 1955 590 561 54.7 640 200 190 1995 600 570 55.2 660 205 195 2030 610 580 55.7 675 210 199 2070 620 589 56.3 690 215 204 2105 630 599 56.8 705 220 209 2145 640 608 57.3 720 225 214 2180 650 618 57.8 740 230 219 660 58.3 755 235 223 670 58.8 770 240 228 20.3 680 59.2 785 245 233 21.3 690 59.7 800 250 238 22.2 700 60.1 820 255 242 23.1 720 61 835 260 247 24 740 61.8 850 265 252 24.8 760 62.5 865 270 257 25.6 780 63.3 880 275 261 26.4 800 64 900 280 266 27.1 820 64.7 915 285 271 27.8 840 65.3 930 290 276 28.5 860 65.9 950 295 280 29.2 880 66.4 965 300 285 29.8 900 67 995 310 295 31 920 67.5 1030 320 304 32.2 940 68 1060 330 314 33.3 1095 340 323 34.4 From DIN EN ISO 18265 1125 350 333 35.5 744 Appendix 12 Strength Values for Gears of Structural Steel, Cast Iron, Cast and Tempered Steel Appendix 12 Approximate Values for Pitting Endurance Limit ıHlim and Tooth Root Strength ıFE, for Gears Made of Structura l Steel, Cast Iron, Cast Steel and Unalloyed Tempering Steel without Surface Layer Hardening, Material Quality MQ in Accordance with ISO 6336-5 (DIN 3990-5) NFlim 9) 3 106 3 106 3 106 3 106 3 106 3 106 3 106 1) The areas of hardness on the tooth from ıHlim and ıFE are taken from the Figures 1a to 3c from ISO 6336-5 (DIN 3990-5). 2) Normalising rolling, rolled thermomechanically. 3) Bainitic cast iron in accordance with DIN EN 1564 is not yet registered in ISO 6336-5 (DIN 3990- 5). 4) At H < 180 HBW the structure contains a large proportion of ferrite; not favourable for gears. 5) For calculating ıFE: H = 160 to 250 HBW. 6) For calculating ıFE: H = 200 to 300 HBW. Spheroidal graphite cast iron in accordance with DIN EN 1563 can have up to 360 HB. 7) For calculating ıFE: H = 115 to 210 HV10. 8) Steels with > 0.32% carbon content; wh en making a selection the hardenability must be taken into consideration. 9) At N < N Hlim or N < N Flim a further drop in endurance limit is to be expected. According to results and experience until now, in cases of load cycle numb ers of N > N lim and increases of load cycle numbers by a factor of 10 the further drop in endurance strength is each 5% (refer to factors YNT, ZNT). Note: N Hlim and NFlim are load cycle numbers which mark the transition from limited life strength to endurance strength (note remark 9)). σFE [N/mm ] 1.0 H+150 0.444 H+33.34 0.667 H+213.3 0.7 H+240 0.632 H+207.4 0.625 H+355 0.8 H+272 σFE [N/mm ] 1) 260 to 360 100 to 140 320 to 380 380 to 450 280 to 340 480 to 580 380 to 440 NHlim 9) 5 107 5 107 5 107 5 107 5 107 5 107 5 107 σHlim [N/mm ] 1.0 H+200 1.0 H+140 1.364 H+159 1.417 H+215 0.933 H+294 1.375 H+265 0.933 H+364 σHlim [N/mm ] 1) 310 to 410 290 to 380 350 to 500 470 to 640 420 to 485 540 to 760 490 to 560 Hardness on tooth H 1) 110 to 210 HBW 150 to 240 HBW 4) 140 to 250 HBW 5) (160 to 250) 180 to 300 HBW (200 to 300) 6) 135 to 210 HV10 (115 to 210) 7) 200 to 360 HV10 135 to 210 HV10 Heat treatment, condition at time of delivery Normalised 2) Predominantly pearlitic in cast state Not decarburising annealed in cast state Predominantly pearlitic in cast state Normalised and quenched and tempered, unalloyed Quenched and tempered, alloyed Normalised, quenched and tempered Material group Structural steels, DIN EN 10025 Lamellar-graphite cast iron, DIN EN 1561 Black heart malleable cast iron, DIN EN 1562 Spheroidal graphite cast iron DIN EN 1563 3) Cast steel for general use, DIN EN 10293 Tempering steels, unalloyed, DIN EN 10083-2 8) Appendix 13 Test Procedures for Scuffing Load Capacity 745 Appendix 13 Test Procedures for Scuffing Load Capacity (see Section 6.5.5.7) The two further FZG test methods A10/16.6R/120 and A/2.8/50 developed in addition to the A/8.3/90 standard test have found their place in ISO 14635 as Part 2 and Part 3 and will appear as DIN ISO 14635-2 and DIN ISO 14635-3 [6.5/79]. The draft of DIN ISO14635-2 [6.5/93] includes the FZG test method A10/16.6R/120 for determining the relative scuffing load capacity of highly EP-alloyed lubricants, as they are used in many vehicles and some stationary systems. Compared to the standard test, the following areas have been tightened: • Pinion width has been halved to b = 10 mm • Circumferential speed in the pitch circle has been doubled to / = 16.6 m/s • Test rig runs in reverse operation (gear drives pinion) • Start temperature in oil sump raised to 120 C from load stage 4 onwards Aside from the effective face width, the gearing data is identical to that listed in Table 6.5/17 (refer to Section 6.5.5.7) for the standard test. The conditions of the test can be found in Table A13/1. Table A13/1 (For Section 6.5.5.7) Test Conditions for FZG Test Method A10/16.6R/120 [6.5/93] Duration per load stage 21700 revolutions of the motor (roughly 7.5 min) Motor speed 2910 min–1 3% Direction of rotation of the motor Counterclockwise Oil temperature at the beginning of load stage 4 and in each subsequent load stage (120 3) C The damage load stage refers to the load stage in which the total area of the damages to all 16 active tooth flanks of the pinion exceeds 100 mm2. Here excessive wear must be ruled out by limi- ting the weight loss on the gear to a maximum value of 20 mg. The test criteria are summarised in Table A13/2. Table A13/2 (For Section 6.5.5.7) Test Conditions for FZG Test Method A10/16.6/90 [6.5/93] Damage area on pinion A <mm2> Weight loss on gear ǻm <mg> Result 100 20 Passed 100 > 20 Invalid 1) > 100 Not required Failed 1) Determination of scuffing load capacity not possible. Using the splash lubrication method, the test gear pair runs in the lubricant to be tested at a constant speed for a predetermined number of revolutions. The load on the tooth flanks is raised in stages, according to Table A13/3. Beginning with load stage 5, the pinion tooth flanks are exa-mined for surface damage at the end of each load stage. The test is finished when the damage criteria are satisfied or load stage 10 is run through without the occurrence of the damage criteria. 746 Appendix 13 Test Procedures for Scuffing Load Capacity Appendix 13 Test Procedures for Scuffing Load Capacity (see Section 6.5.5.7), continued Table A13/3 (For Section 6.5.5.7) Load Stages for FZG Test Method A10/16.6/90 [6.5/93] Load stage Torque on pinion M1T [Nm] Normal tooth force Fn [N] Hertzian contact stress at the pitch point ıH0 [N/mm2] 1 2 3 4 5 6 7 8 9 10 3.3 13.7 35.3 60.8 94.1 135.5 183.4 239.3 302.0 372.6 99 407 1 044 1 799 2 786 4 007 5 435 7 080 8 949 11 029 206 417 670 878 1 093 1 311 1 527 1 742 1 960 2 176 The draft of DIN ISO 14635-3 [6.5/94] describes the FZG test method A/2.8/50 for determining relative scuffing load capacity and wear characteristics of low-viscosity greases for gears for en-closed transmissions. Here the greases must have a sufficient degree of flowability in the FZG test gear box. The gearing data (A) correspond to those of the standard test in accordance with Table 6.5/17; the circumferential speed at the pitch circle is 2.8 m/s. The rest of the test conditions are summarised in Table A13/4. Table A13/4 (For Section 6.5.5.7) Test Conditions for FZG Test Method A/2.8/50 [6.5/94] Duration per load stage 21700 revolutions of the motor (roughly 45 min) Motor speed 500 min–1 3% Direction of rotation of the motor Clockwise Fill level of test lubricant (1.25 0.05) litres Lubricant temperature at the outset in load stage 1 Room temperature Lubricant temperature at the outset of load stage 4 and each subsequent load stage (50 3) C (must be set using the temperature controller) The damage load stage refers to the load stage in which the sum of the damages (width of all scratches and scuffing marks) on the active tooth flanks of the 16 pinion teeth exceeds one face width or 20 mm. The load on the tooth flanks is raised in stages until the damage load stage is reached in accordance with Table A6.5/19 (refer to Section 6.5.5.7). If the test is finished without satisfying the damage criteria, then the result “damage load stage greater than 12” and the weight loss of the pinion and wheel are to be specified. As a supplement to the stage test A10/16.6R/90 in accordance with DIN ISO 14635-2, in which the load on the tooth flanks is raised in stages, the S-A10/16.6R/90 direct test without running-in was developed in the context of research by the Research Association for Drive Technology (FVA) [6.5/80]. In the direct test without running-in, a load equivalent to the anticipated damage load stage is applied directly, and damages or passed specimens are then determined. By omitting running-in, the test method is tightened considerably so that lubricants with higher load-bearing capacity can be tested. More tests were carried out to determine the influence of the oil temperature on the scuffing load capacity [6.5/95]. Appendix 14 Supplement for Power Losses and Efficiency, Transmission Heating, Lubrication 747 Appendix 14.1 Supplement for Power Losses and Efficiency (see Section 6.6.1.2) An exact analysis of the gear losses necessitates calculation in the entire field of action. This can be achieved with the PC program LVR [6.4/33]. During tests by Wimmer , the easy-to-calculate tooth loss coefficient HV in accordance with Ohlendorf [6.6/1] was expanded. Moreover, it was through his efforts and the local point of view that it became possible to calculate the helical gearing and take the load distribution into account. Equation (6.6/13), which has been utilised more than any other to date, references the nominal tooth geometry of spur gearing, without taking manufacturing deviations and gearing flexibilities into account. The tooth loss coefficient determined this way results from the gearing parameters for spur gearing, with the position of the pitch point in the single meshing region. For general gearing, in analogy to the LVR program, Wimmer [6.6/43] introduced the local- geometric gear power loss factor H VL, which is included in Equation (A14.1/1): () ()g N VL et bt t b 0A, , 1ddbE yxxy fx yHx ypF===⋅ / / (A14.1/1) Here the following apply: Pet Normal base pitch on the base circle, in mm y Coordinates in the gearing width direction, in mm x Coordinates on the line of action (distance from pitch point), in mm fN Local line load per face width, in N/mm /b Circumferential speed on the base circle, in m/s The factor HVL is not as easy to calculate as the loss factor HV according to Ohlendorf. That is why it was also integrated into the “RIKOR” program, where the load distribution is taken into account for the affected gearing and the losses are also taken into account for helical gearing better than ever before, as is already the case in the program LVR [6.4/33]. The agreement between the loss factors H V and HVL is relatively good for normal spur gearing, whereas in the case of corrected extra-depth gearing, helical gearing and special forms of gearing large deviations can occur. In particular, flank corrections that are not taken into account in the loss factor H V are registered by the local geometric tooth loss coefficient HVL. However, the “classic” loss factor H V is still very valuable for modern preliminary designs since it makes it easy to identify the significance of balanced sliding speed. 748 Appendix 14 Supplement for Power Losses and Efficiency, Transmission Heating, Lubrication Appendix 14.2 Supplement to Bearing Losses (see Section 6.6.1.3) The roller bearing coefficients f0, f1 and f 2 listed in the draft of ISO/TR 14179-2 [6.6/53] are reproduced in the following Tables A14.2/1, A14.2/2 and A14.2/3. Table A14.2/1 Roller Bearing Coefficient f0 1) in Accordance with ISO/TR 14179-2 [6.6/53] Bearing type Method of lubrication Grease Oil mist Oil bath Oil injection, oil bath with vertical shaft Deep-groove ball bearings Single row Paired 0.75 to 2 2) 3 1 2 2 4 4 8 Self-aligning ball bearings 1.5 to 2 2)0.7 to 1 2) 1.5 to 2 2) 3 to 4 2) Angular-contact ball bearings Single row Paired 2 4 1.7 3.4 3.3 6.5 6.6 13 Four-point bearings 6 2 6 9 Cylinder rolling bearing (with cage) Bearing series 10, 2, 3, 4 Bearing series 22 Bearing series 23 0.6 0.8 1 1.5 2.1 2.8 2.2 3 4 2.2 3) 3 3) 4 3) Cylinder rolling bearing (full complement) Single row Paired 5 4) 10 4) - - 5 10 - - Needle bearings 12 6 12 24 Spherical roller bearings Bearing series 213 Bearing series 222 Bearing series 223, 230, 239 Bearing series 231 Bearing series 232 Bearing series 240 Bearing series 241 3.5 4 4.5 5.5 6 6.5 7 1.75 2 2.25 2.75 3 3.25 3.5 3.5 4 4.5 5.5 6 6.5 7 7 8 9 11 12 13 14 Tapered roller bearings Single row Paired 6 12 3 6 6 12 8 to 10 2, 3) 16 to 20 2, 3) Axial deep-groove ball bearings 5.5 0.8 1.5 3 Axial cylinder rolling bearings 9 - 3.5 7 Axial needle bearings 14 - 5 11 Axial spherical roller bearings Bearing series 292 E Bearing series 292 Bearing series 293 E Bearing series 293 Bearing series 294 E Bearing series 294 - - - - - - - - - - - - 2.5 3.7 3 4.5 3.3 5 5 7.4 6 9 6.6 10 1) The values given are valid for uniform conditions; (2 to 4) f0 is to be used for the calculation in cases of freshly greased bearings. 2) The lower values are valid for light bearings and the higher values for heavy bearings within a group of holes. 3) Valid for injection lubrication. The values given are to be doubled in cases of oil-bath lubrication with a vertical shaft. 4) Valid for low rotational speeds up to 20% of the reference speed (refer to bearing tables). At higher rotational speeds f0 is to be doubled. Appendix 14 Supplement for Power Losses and Efficiency, Transmission Heating, Lubrication 749 Appendix 14.2 Supplement to Bearing Losses (see Section 6.6.1.3), continued Table A14.2/2 Roller Bearing Coefficient f1 and Equivalent Bearing Load P1 in Accordance with ISO/TR 14179-2 [6.6/53] Bearing type f1 P1 1) Deep-groove ball bearings (0.0006 to 0.0009) ( P0/C0)0.5 2)3 Fa - 0.1 Fr Self-aligning ball bearings 0.0003 ( P0/C0)0.4 1.4 Y2 Fa - 0.1 Fr Angular-contact ball bearings Single row Paired 0.001 (P 0/C0)0.33 0.001 (P 0/C0)0.33 Fa - 0.1 Fr 1.4 Fa - 0.1 Fr Four-point bearings 0.001 ( P0/C0)0.33 1.5 Fa + 3.6 Fr Cylinder rolling bearing (with cage) Bearing series 10 Bearing series 2 Bearing series 3 Bearing series 4, 22, 23 0.00020 0.00030 0.00035 0.00040 F r 3) Fr 3) Fr 3) Fr 3) Cylinder rolling bearing (full complement) 0.00035 Fr 3) Needle bearings 0.00200 Fr Spherical roller bearings Bearing series 213 Bearing series 222 Bearing series 223 Bearing series 230, 241 Bearing series 231 Bearing series 232 Bearing series 239 Bearing series 240 0.00022 0.00015 0.00065 0.00100 0.00035 0.00045 0.00025 0.00080 The following applies for all bearing series: 1.35 Y 2 Fa, if Fr/Fa < Y2 Fr [ 1 + 0.35 ( Y2 Fa/Fr)3], if Fr/Fa Y2 Tapered roller bearings Single row Paired 0.00040 0.00040 2 Y F a 1.2 Y2 Fa Axial deep-groove ball bearings 0.0008 ( Fa / C0)0.33 Fa Axial cylinder rolling bearings, axial needle bearings 0.00150 Fa Axial spherical roller bearings Bearing series 292 E Bearing series 292 Bearing series 293 E Bearing series 293 Bearing series 294 E Bearing series 294 0.00023 0.00030 0.00030 0.00040 0.00033 0.00050 The following applies for all bearing series: F a (Fmax 0.55 Fa) P0 Equivalent static bearing load 1) If P1 < Fr is true, then P1 is to be set the same as Fr. 2) The lower values are valid for light bearings and the higher values for heavy bearings within a group of holes. 3) For cylinder rolling bearings with additional axial load, refer to Equation (6.6/27). The load-dependent friction torque M 1 according to Equation (A14.2/1) is expanded in ISO/TR 14179-2 with the introduction of the exponents a and b. The following applies: 3 11 1 m 10abMf P d−= (A14.2/1) The exponents a and b can be found in Table A14.2/3. 750 Appendix 14 Supplement for Power Losses and Efficiency, Transmission Heating, Lubrication Appendix 14.2 Supplement to Bearing Losses (see Section 6.6.1.3), continued Table A14.2/3 Exponents a and b in Equation (6.6/26) Self-aligning Bearings in Accordance with ISO/TR 14179-2 [6.6/53] Table A14.2/4 Roller Bearing Coefficient f2 for Cylin- der Rolling Bearing in Accordance with ISO/TR 14179-2 [6.6/53] Bearing type a b Bearing type Grease Oil 213 1.35 0.2 Bearing with cage 222 1.35 0.3 EC design 0.003 0.002 223 1.35 0.1 All others 0.009 0.006 230 1.5 -0.3 Full-complement bearings 231, 232, 239 1.5 -0.1 Single row 0.006 0.003 240, 241 1.5 -0.2 Double row 0.015 0.009 For all other bearings a = b = 1. Among other authors the roller bearing coefficients f0, f1 and f 2 differ to some extent from those given in ISO/TR 14179-2 [6.6/53]. Thus, for example, the roller bearing coefficient f2 can be determined based on the axial load and on the lubrication condition in accordance with [6.6/12] from the diagram in Figure A14.2/1. Figure A14.2/1 Roller bearing coefficient f2 for cylinder rolling bearings with axial load [6.6/12], dependent on the axial load and on the lubrication condition The variables necessary in Figure A14.2/1 are to be used as follows: f\* 0.0048 for bearings with cage 0.0061 for full-complement bearings dm Mean bearing diameter in mm Ȟb Service viscosity of the oil or the basic grease-oil in mm2/s (e.g., according to Figure 6.6/23) n Speed of the inner ring in min–1 Fa Axial load in N D Outer diameter of the roller bearing in mm d Nominal diameter of the roller bearing in mm (hole diameter) Appendix 14 Supplement for Power Losses and Efficiency, Transmission Heating, Lubrication 751 Appendix 14.2 Supplement to Bearing Losses (see Section 6.6.1.3), continued In the current issue of the SKF roller bearing catalogue [6.6/11], a new method for calculating fric- tion torque is introduced. The following relationship applies: lsea drag rr slMM M M M=++ + (A14.2/2) Here the following apply: M Total friction torque Mrr Roll friction torque Msl Sliding friction torque Mseal Friction torque of contact seals Mdrag Friction torque caused by flow, splash or spray losses This method is intended for determining approximate values for the friction torque of individual bearings under the following operating conditions: • Grease lubrication or normal oil lubrication methods (oil bath, oil air lubrication or oil injection methods) • Loads equivalent to or larger than the minimum load • Loads unchanging in direction and size • Working **backlash** Determining the individual parts of the total friction torque is a relatively complex task and time consuming. That is why an appropriate calculation programme should be used, which is available in the “Interactive SKF Bearings Catalogue”. Refer to [6.6/11] for more details. 752 Appendix 14 Supplement for Power Losses and Efficiency, Transmission Heating, Lubrication Appendix 14.3 Supplemental Information Regarding Thermal Balance (Appendix to Section 6.6.3.4) A new version of ISO/TR 14179-2 [6.6/53] is available [6.6/8], [6.6/9]. Notable differences to Section 6.6.3.4 shall be shown here. Table A14.3/3 provides information on a few new symbols from ISO/TR 14179-2. The emission ratios İ listed in Table A14.3/1 are specified as a supplement to Equations (6.6/56) and (6.6/57). Table A14.3/1 Emission ratio İ [6.6/53] Material Treatment İ Cast iron Hammer blow 0.60 to 0.80 Mechanically processed 0.35 to 0.45 Steel Rolled 0.80 to 0.90 Mechanically processed 0.15 Processed and treated with oil 0.35 Sand blasted 0.35 Sand blasted and treated with oil 0.50 to 0.60 Aluminium Oxidised surface 0.15 Mechanically processed 0.05 to 0.10 All materials painted External oil or dust layer 0.90 to 0.95 1 Heat dissipation via the gear housing First of all, the following equations are valid: (6.6/43), (6.6/46), (6.6/51) and (6.6/57). The heat dissipation via the gear housing is composed of a convective portion and a radiation portion. The convective portion can be produced by free or forced convection. For this, further relationships shall be given. Surface heat transfer coefficient through convection: air air con K,free K,forced ca ca1AA AA∗ α= α − + α ⋅ ⋅ η (A14.3/1) with wall air wall∗ ∞−η=−KK KK For a housing without heat fins one can say: - For free convection ( /Luft < 1.5 m/s) the following applies: 0.3 0.1 wall K,free 18273H− ∞ ∞ ϑ− ϑα= ϑ+ (A14.3/2 ) - For forced convection (/Luft > 1.5 m/s) the following applies: ()0.64 K,forced x0.0086 Re l′ ⋅α= (A14.3/3) Appendix 14 Supplement for Power Losses and Efficiency, Transmission Heating, Lubrication 753 Appendix 14.3 Supplemental Information Regarding Thermal Balance (Appendix to Section 6.6.3.4), continued with 2ee2.5GrRR′=+ air airxlReυ⋅=ν () ()3 GR 2 Ra ir 273gHGrϑ− ϑ= ϑ+ νD Here the following apply: Įcon Surface heat transfer coefficient through convection in W/(m2K) ĮK,free Surface heat transfer coefficient through free convection in W/(m2K) ĮK,forced Surface heat transfer coefficient through forced convection in W/(m2K) Aair Fan-ventilated housing area in m2 Aca Total external housing area in m2 H Total height of the gear housing in m KG Housing wall temperature in C Kair Fan air temperature in C KR Ambient temperature in C lx Ventilated length (path of the current thread along the housing wall) in m Re Reynolds number Gr Grashof number /air Ventilation speed in m/s ȃair Kinematic viscosity of the air in m2/s g 9.81 m/s2 For a housing with heat fins in accordance with Figure A14.3/1 it follows: - For purely convection (Aair = 0): ()pro fin fin ca K,free rad fin K,free rad ca fin ca1A AA AA A α= α + α η + − ⋅ α + α (A14.3/4) with the fin thermal efficiency () ()fin fin fintanh ml mlη= pro ca rad fin fin fin2A Amsα+ α ⋅ =⋅λ Here the following apply: Įca External or air-side surface heat transfer coefficient in W/(m2K) Įrad Heat transfer coefficient due to radiation in W/(m2K) Afin Total fin area (housing, external) in m2 Apro Projected fin area (housing, external) in m2 Șfin Thermal efficiency of fin m Fin factor sfin Thickness of a fin in m Ȝfin Thermal conductivity coefficient of the fin in W/(mK); refer to Table A14.3/2 for values 754 Appendix 14 Supplement for Power Losses and Efficiency, Transmission Heating, Lubrication Appendix 14.3 Supplemental Information Regarding Thermal Balance (Appendix to Section 6.6.3.4), continued 1 Environment 2 Oil film 3 Oil sump Figure A14.3/1 Housing with heat fins - For free convection and fan-ventilated fin area (Afin = A air): ()pro air air ca K,forced rad fin K,free rad ca air caĮ 1A AA AA A∗ α= η + α η + − ⋅ α + α (A14.3/5) - For free and forced convection (Aair > A fin): () ()air air fin ca K,free rad K,forced rad ca ca pro fin K,forced rad rip ca fin1AA A AA A A AA∗ ∗ −α = − ⋅ α + α+ ⋅ α η + α+ +⋅ α η + α ⋅ η (A14.3/6) ∗η Refer to Equation (A14.3/1) finη Refer to Equation (A14.3/4) Appendix 14 Supplement for Power Losses and Efficiency, Transmission Heating, Lubrication 755 Appendix 14.3 Supplemental Information Regarding Thermal Balance (Appendix to Section 6.6.3.4), continued 2 Heat dissipation via the foundation The calculation of the conductivity of the foundation assumes the division of the gear unit foundation into multiple separate fins and uses the fin equation known from thermodynamics. The partial heat flows (areas A qi) are added together to equal the total foundation conductivity (refer to Figure 6.6/12): () ()fun ii n fun i fun fun fun qi i fun i1 ii fun itanh 1t a n hmLmQf A m mLm∗ ∗ ∗ ∗ = ∗α+λ=λ Δ ϑ ⋅α+λ (A14.3/7) with 0.16 foot1.46 botAfA = ()fun n R 0.62 Δϑ = ϑ − ϑ Further, with heat dissipation on the foundation, for the reciprocal length coefficient mi\* the follo- wing applies: Upwards and downwards: Only upwards: fun qi i fun qiUmA∗α=λ fun qi i fun qi0.75UmA∗ α=λ Here the following apply: funQ Heat flow due to conductivity via the gear unit foundation in W Ȝfun Thermal conductivity coefficient of the foundation in W/(mK) (refer to Table A14.3/2 for values) Aq Area of the section of a partial fin of the foundation in m2 (refer to Figure 6.6/12) Įfun Surface heat transfer coefficient on the gear unit foundation in W/(m2K) L Ventilated length in m (refer to Figure 6.6/10) Aroot Root contact area of the gear unit in m2 Abot Gear unit bottom area in m2 Uq Circumference of the section of a partial fin of the foundation in m (refer to Figure 6.6/12) Table A14.3/2 Thermal Conductivity Coefficient Ȝ [6.6/53] Material Ȝ W/(mK) Aluminium 180 Steel 50 Cast iron 42 Concrete 1 Air 0.027 756 Appendix 14 Supplement for Power Losses and Efficiency, Transmission Heating, Lubrication 3 Heat dissipation via shafts and couplings The content of ISO/TR 14179-2 agrees completely with Equations (6.6/68) to (6.6/78). Table A14.3/3 Comparison of Some of the Equation Symbols Used in Appendix 14.3 (and Section 6.6) with ISO/TR 14179-2 Appendix 14.3 and Section 6.6 ISO/TR 14179-2 Description according to ISO/TR 14179-2 Appendix 14.3 and Section 6.6 ISO/TR 14179-2 Description according to ISO/TR 14179-2 LuftA airA Ventilated housing area K,erzwα K,forcedα Heat transfer coefficient due to forced convection Fu A botA Gear unit bottom area K,freiα K,freeα Heat transfer coefficient due to free convection KA caA Overall housing area (external) Kα conα Heat transfer coefficient due to convection ripA finA Total fin area (housing external) Sα radα Heat transfer coefficient due to radiation BODA botA Gear unit bottom area Fϑ+ funTΔ 0.62( Toil -T∞) qiU iU Circumference of the foundation, part i ripη fη Fin efficiency ripl finl Depth of one fin Gϑ (Unit C) wallT (Unit K)Temperature of housing wall Srip finį Thickness of one fin Rϑ (Unit C) T∞ (Unit K) Ambient temperature FQ funQ Heat flow across foundation ϑ (Unit C) oilϑ (Unit K) Oil temperature /luft /air Impingement velocity Fλ funλ Thermal conductivity of foundation aα caα Air-side heat transfer coefficient at housing ripλ finλ Thermal conductivity of fin Appendix 14 Supplement for Power Losses and Efficiency, Transmission Heating, Lubrication 757 Appendix 14.4 Alternative Lubrication (see Section 6.6.4.1) A few results on lubrication alternatives will be presented as suggestions. Tests were carried out on oil-free operation of power-transmitting gear units [6.6/48] to [6.6/51]. The goal of the work here was to achieve a more environmentally friendly operation without oil and still not permit any negative effects on the load capacity of the gears. Of all the lubricants tested for this application in collaboration with the Leibniz Institute of Polymer Research Dresden (IPF), PTFE micropowder proved to be the most suitable alternative gear transmission lubricant. These powders possess very good tribological properties and also do not lead to environmental damage. Initially, coasting tests were carried out with various products in a back-to-back gear test rig as a comparison with lu bricating oil, in order to demonstrate the general suitability as transmission lubricant. Table 6.6/12 lists the main gearing data. The test rig was loaded with a test load, corresponding to a nominal contact stress of ı H0 = 600 N/mm2, and subsequently accelerated to a trigger speed of nan = 1000 min–1. After that, the drive motor was disconnected from the test rig by a switchable coupling and the coasting time measured. The result showed that the PTFE powder lubrication generally had longer coasting times than flanks lubricated with oil. Table A14.4/1 Main Data of the Test Gearing Name Formula symbol Dimension Pinion Gear Number of teeth z - 17 23 Normal module mn mm 6 Centre distance a mm 125 Pressure angle Įn 20 Helix angle ȕ 0 Face width b mm 16 Profile shift factor x - 0.4888 0.4600 Tooth loss coefficient HV - 0.173 Circumferential speed /t m/s 5.56 Max. sliding speed /g m/s 2.17 Figure A14.4/1 Modified gear housing with guide and swirl mechanisms [6.6/49] Guide plate Lashes 758 Appendix 14 Supplement for Power Losses and Efficiency, Transmission Heating, Lubrication Appendix 14.4 Alternative Lubrication (see Section 6.6.4.1), continued In the coasting tests the swirl behaviour was also observed, which is strongly dependent on the production process and on previous irradiation of the powder. Sufficient swirling of the powder, which translates into the regeneration of the lubricating film, can be achieved using additional agents. In previous running tests, elastic lashes were fastened to the gear shafts and on the flanges of the gears and guide plates placed in the gear housing (Figure A14.4/1). Furthermore, stepped tests were carried out for a selection of powders to assess the lubrication behaviour under conditions similar to long-term operation. Here the stepped tests were carried out at a constant input rotational speed of nan = 1000 min–1 in eight load steps, whose nominal contact stress corresponded to a range of ıH0 = (300) to 1200 N/mm2. The test duration was 60 minutes at load step 1, and 90 minutes at each subsequent load step. In addition to load torque, also loss torque and the resulting mass temperature were registered. Due to the lack of heat transmission with the powder lubrication, the pinion mass temperatures measured at the end of each load step were considerably higher than for oil lubrication. Whereas with oil lubrication a temperature of roughly 90 C resulted after the highest load step, the mass temperatures for powder lay in the 150 C range, depending on the powder product. If such tempera-tures are allowed to act on case-hardened gearing for a longer period of time, it can lead to tempe-ring effects (loss of hardness). Despite the high thermal load, the PTFE powder and the gearing suffered no detectable negative effects during the relatively short running time, however. In the case of some powders, the formation of permanently bonding PTFE coatings on the tooth flank could be observed. After the long-term test of more than 10 million load cycles with a select powder at a load of ı H0 = 1200 N/mm2 and an input rotational speed of nan = 1000 min–1, no detectable damage to the gearing was apparent. As such, it was possible to convincingly verify the functional capability of the lubrication. With that, an environmentally friendly and physiologically unobjectionable lubricant is available, at the very least for a certain area of application. As further variation of oil-free gear units, different coatings were tested on the tooth flank surfaces. However, dry running for an unlimited time under typical flank loads was not achievable this way. According to findings to date, the advantage of the coatings can only be clearly seen in the areas of the scuffing load capacity and the micropitting load capacity. As reason for the somewhat improved pitting load capacity and root load capacity which were occasionally noticed because of the coatings, primarily the unfavourable formations of the outer edge layer in the previous production stages, such as after case hardening and grinding, can be considered. The following among others should be observed for practical applications of the alternative powder lubrication: • Ensure the formation of regenerating coatings through sufficient swirling • Use for circumferential speeds that are not too high • Gear materials or treatments with sufficient temperature stability (nitrided, among others) • Alternative cooling (e.g., compressed air) • Lower shaft-to-bearing surface-heat transfer, e.g., through grooves (possibly helical) at the bearing seat • Temperature-stabilised bearings, ceramic bearings, or encapsulated bearings with a lubricant resistant to raised temperatures Appendix 15 Manufacturing of Cylindrical Gears, Tools 759 Appendix 15 Manufacturing of Cylindrical Gears, Tools Tool Shapes Figure A15.1 Top to bottom: disk-type shaper cutter; bell-type shaper cutter; shank shaper cutter for helical gears Figure A15.4 View into the working space of a ring honing machine (advertising picture of Reishauer AG) Figure A15.2 Full-body roaming needle with roughing zone, finishing bush, end part and work pieces Figure A15.3 Spur and helical internal gears with finishing bushes for broaching the toothing Figure A15.5 Hone ring made of fused aluminium oxide and work piece (advertising picture of Reishauer AG) 760 Appendix 15 Manufacturing of Cylindrical Gears, Tools Appendix 15 Manufacturing of Cylindrical Gears, Tools (continued) Figure A15.11 Dressing of a ceramically bound profile grindin g disk(front view ) Figure A15.10 Dressing of a ceramically bound profile grindin g disk (side view ) Figure A 15.9 Profile grinding with ceramically bound grinding disk Figure A15.8 Profile grindin g with galvanically coated CBN grinding disk Figure A15.7 Combination of continuous grinding worm (back) for roughing and profile grinding disk (front) for finishin gFigure A15.6 Continuous generating grinding with CBN-grinding tool, front: finishi ng worm, back: roughing wormrom the best advice to the best productionver 1,000,000 versions, but only one original )/(1'(5 KHOLFDO JHDU XQLWV RIIHU WKH ULJKW VROXWLRQ IRU \RXU DSSOLFDWLRQ Appendix 16 Gear Transmissions, Selected Examples 763 Appendix 16.1 Bevel Gear-Cylindrical Gear Transmission, Siemens AG (Flender Industriegetriebe GmbH, Penig, Germany) 764 Appendix 16 Gear Transmissions, Selected Examples Appendix 16.2a, b Planetary Transmissions a) Planetary unit in combination with two cylindrical gear stages in a wind turbine transmission; Siemens AG (Winergy AG, Voerde/Friedrichsfeld, Germany) b) Planetary unit with stepped planets in a marine gas turbine drive (Entwicklungsbau Pirna, Germany) Appendix 16 Gear Transmissions, Selected Examples 765 Appendix 16.2c, d Planetary Transmissions (continued) c) Planetary transmission (ABUS Transmission Plant, Dessau, Germany) d) Planetary transmission for wind power units, overriding drive with hydrodynamic torque converter (Voith, Crailsheim, Germany) 766 Appendix 16 Gear Transmissions, Selected Examples Appendix 16.2e Planetary Transmissions (continued) e) Diagram of a large transmission of a bucket-wheel excavator (ABUS Transmission Plant, Dessau, Germany) Excellent availability and high reliability have priority in power station technology. Voith Turbo offers a wide range of variable speed drives based on the hydrodynamic principle. Low investment costs and energy savings in power plants are achieved with Voith Turbo geared variable-speed couplings in power plant applications like boiler feed pumps, boiler fans and coal mills. In combined power stations, variable speed couplings control the speed of boiler feed pumps. As a result, the pump operates with optimum efﬁciency allowing considerable energy cost savings. Additionally Voith Turbo torque converters for gas turbine starting devices offer an unrivalled availability in this application. Voith Turbo GmbH & Co. KG Variable Speed DrivesVoithstr. 174564 Crailsheim, GermanyTel. +49 7951 32-261vs.drives@voith.comwww.voith.com/vsdThe Most Reliable Way to Highest Performance. Appendix 16 Gear Transmissions, Selected Examples 769 Appendix 16.2f Planetary Transmissions (continued) f) Large transmission of a bucket-wheel excavator, design detail (ABUS Transmission Plant, Dessau, Germany) 770 Appendix 16 Gear Transmissions, Selected Examples Appendix 16.2g, h Planetary Transmissions (continued) g) Turbo planetary transmission (BHS Getriebe GmbH, Sonthofen, Germany) h) Turbo planetary transmission, fixed planet carrier (BHS Getriebe GmbH, Sonthofen, Germany) Appendix 16 Gear Transmissions, Selected Examples 771 Appendix 16.3 Fluid Transmission for Rail Drives, Str mungsmaschinen, Pirna, Germany 772 Appendix 16 Gear Transmissions, Selected Examples Appendix 16.4 Transmissions: Multi-stage Compressor Drive (Pumpen- und Gebl sewerk PGW, Leipzig, Germany) Appendix 16 Gear Transmissions, Selected Examples 773 Appendix 16.5a Simple Manual Passenger Car Transmission 774 Appendix 16 Gear Transmissions, Selected Examples Appendix 16.5b Eight-speed Automatic Car Transmission, ZF Friedrichshafen AG THE FUTURE OF MOBILITY IS IN MOTION. Since its foundation in 1915, ZF has evolved into a global leader in driveline and chassis technology as well as active and passive safety technology with134,000 employees. From now on, we will offer all relevant technologies forthe megatrends of the future from one source. To learn more about autonomousdriving, safety and efficiency, visit us at zf.com/technology-trends Appendix 16 Gear Transmissions, Selected Examples 777 Appendix 16.6 Large Cylindrical Gear Transmission (Power ca. 100 MW) of a Gas Turbine Power Unit; Renk, Augsburg, Germany 778 Appendix 16 Gear Transmissions, Selected Examples Appendix 16.7 Multi-Stage Compressor Transmission, BHS Sonthofen, Germany Appendix 16 Gear Transmissions, Selected Examples 779 Appendix 16.8 Gas Turbine Drive – Fan Drive Gear Unit of PW1100G-JM Engine, Pratt & Whitney 1 Fan 2 Planetary Gear Set Integrated Planetary Gear Set (Five Planets, Helical Gearing) 780 Appendix 17 Standards (Selection) Appendix 17.1 Gear Standards (Selection) ISO ISO 53 Cylindrical gears for general and heavy engineering - Standard basic rack tooth profile (DIN 867) ISO 54 Series of modules for cylindrical gears (DIN 780-1) ISO 1328 Cylindrical gears Part 1: ISO system of flank tolerance classification: Definitions and allowable values of deviations relevant to flanks of gear teeth Part 2: ISO system of accuracy: Definitions and allowable values of deviations relevant to radial composite deviations and runout information ISO 6336 Calculation of load capacity of spur and helical gears Part 1: Basic principles, introduction and general influence factors Part 2: Calculation of surface durability (pitting) Part 3: Calculation of tooth bending strength Part 5: Strength and quality of materials Part 6: Calculation of service life under variable load ISO 7902 Hydrodynamic plain journal bearings under steady-state conditions - Circular cylindrical bearings Part 1: Calculation procedure Part 2: Functions used in the calculation procedure Part 3: Permissible operational parameters ISO/ TR 10064 Cylindrical gears; code of inspection practice Part 1: Inspection of corresponding flanks of gear teeth Part 2: Inspection related to radial composite deviations, runout, tooth thickness and **backlash** Part 3: Recommendations relative to gear blanks, shaft centre distance and parallelism of axes ISO/TR 13989 Calculation of scuffing load capacity of cylindrical, bevel and hypoid gears Part 1: Flash temperature method Part 2: Integral temperature method ISO/TR 14179 Gears - Thermal capacity Part 1: Rating gear drives with thermal equilibrium at 95 C sump temperature Part 2: Thermal load-carrying capacity ISO/TR 15144 Calculation of micropitting load capacity of cylindrical spur and helical gears DIN EN ISO DIN EN ISO 128 Technical drawings - General principles of presentation Part 24: Lines on mechanical engineering drawings (ISO 128-24:1999) DIN EN ISO 1101 Geometrical product specifications (GPS) - Geometrical tolerancing - Tolerances of form, orientation, location and run-out (ISO 1101: 2012 + Cor. 1:2013) DIN EN ISO 1302 Geometrical product specifications (GPS) - Indication of surface texture in technical product documentation - Amendment 2: Indication of bearing ratio requirements (ISO 1302:2002/DAM 2:2010) DIN EN ISO 9717 Metallic and other inorganic coatings - Phosphate conversion coating of metals (ISO 9717:2010) DIN EN ISO 13691 Petroleum and natural gas industries - High-speed special-purpose gear units (ISO 13691:2001) DIN ISO / DIN EN DIN ISO 128 Part 24: Technical drawings - General principles of presentation - Part 24: Lines on mechanical engineering drawings (ISO 128-24:1999) DIN ISO 2768 Part 2: General tolerances; geometrical tolerances for features without individual tolerances indications; identical to ISO 2768-2:1989 DIN ISO 13715 Technical drawings - Edges of undefined shape - Vocabulary and indications on drawings (ISO 13715:2000) Appendix 17 Standards (Selection) 781 Appendix 17.1 Gear Standards (Selection), continued DIN ISO / DIN EN DIN ISO 14635 Gears - FZG test procedures Part 1: FZG test method A/8.3/90 for relative scuffing load-carrying capacity of oils (ISO 14635- 1:2000) Part 2: FZG step load test A10/16.6R/120 for relative scuffing load-carrying capacity of high EP oils (ISO 14635-2:2004) Part 3: FZG test method A/2.8/50 for relative scuffing load-carrying capacity and wear characteristics of semifluid gear greases (ISO 14635-3:2005) DIN ISO 18653 Gears - Evaluation of instruments for the measurement of individual gears (ISO 18653:2003) DIN ISO 21771 Gears - Cylindrical involute gears and gear pairs - Concepts and geometry (ISO 21771:2007) DIN ISO 21772 Gears - Cylindrical involute gears and gear pairs - Definition of deviations DIN ISO 21773 Gears - Cylindrical involute gears and gear pairs - Inspection dimensions of tooth thickness DIN EN 10025 Hot-rolled products of structural steels - Conditions; German version EN 10025-1:2004 DIN EN 10083 Hot rolled products of structural steels - EN 10025-1:2004 DIN EN 23741 Acoustics; determination of sound power levels of noise sources; precision methods for broad-band sources in reverberation rooms (ISO 3741:1988); replaced by: DIN EN ISO 3741: 2001-01, DIN EN ISO 3741:2009-11, DIN EN ISO 3741:2011-01 DIN DIN 189 Driving elements; Sole plates; Main dimensions DIN 780 Part 1: Series of modules for gears; modules for spur gears DIN 867 Basic rack tooth profiles for involute teeth of cylindrical gears for general engineering and heavy engineering DIN 868 General definitions and specification factors for gears, gear pairs and gear trains DIN 1825 (Shaper cutter) Pinion-type cutters for cylindrical gears; disc-gear cutters for spur gears DIN 1825 DIN 1826 DIN 1828 (Shaper cutter) Pinion-type cutters for cylindrical gears; disc-gear cutters for spur gears (Shaper cutter) Pinion-type cutters for cylindrical gears; extended hub gear cutters for spur gears (Shaper cutter) Pinion-type cutters for cylindrical gears; shank-gear cutters for spur gears DIN 3960 Definitions, parameters and equations for involute cylindrical gears and gear pairs, withdrawn; replaced by: DIN 21773:2014-08, DIN 21773:2014-08, DIN ISO 21771:2014-08, DIN 21772:2012-07 DIN 3960 Supplement 1: Definitions on involute cylindrical gears and gear pairs; classification of the equations DIN 3961 Tolerances for cylindrical gear teeth; bases DIN 3962 Tolerances for deviations of individual parameters Part 1: Tolerances for cylindrical gear teeth Part 2: Tolerances for cylindrical gear teeth; tolerances for tooth trace deviations Part 3: Tolerances for dylindrical gear teeth; tolerances for pitch-span deviations DIN 3962 Part 1: Tolerances for cylindrical gear teeth; tolerances for deviations of individual parameters DIN 3963 Tolerances for cylindrical gear teeth; tolerances for working deviations DIN 3964 Deviations of shaft centre distances and shaft position tolerances of casings for cylindrical gears; withdrawn DIN 3966 Part 1: Information on gear teeth in drawings; information on involute teeth for cylindrical gears DIN 3967 System of gear fits; **backlash**, deviation and tolerances of teeth thickness, general bases DIN 3969 Part 1: Surface roughness of tooth flanks; roughness parameters, surface grades DIN 3970 Master gears for checking cylindrical gears - Gear blanks and gearing, 2010-04 (replacement for DIN 3970-1, DIN 3970-2) 782 Appendix 17 Standards (Selection) Appendix 17.1 Gear Standards (Selection), continued DIN DIN 3972 Reference profiles of gear-cutting tools for involute tooth systems according to DIN 867 (ISO 53) DIN 3977 Measuring element diameters for the radial or diametral dimension for testing tooth thickness of cylindrical gears DIN 3979 Tooth damage on gear trains; designation, characteristics, causes DIN 3992 Addendum modification of external spur and helical gears DIN 3993 Geometrical design of cylindrical internal involute gear pairs Part 1: Basic rules Part 2: Diagrams for geometrical limits of internal gear-pinion matings Part 3: Diagrams for the determination of addendum modification coefficients Part 4: Diagrams for limits of internal gear - Pinion-type cutter matings DIN 3998 Denominations on gears and gear pairs Part 1: General definitions Part 2: Cylindrical gears and gear pairs DIN 3998 Part 1: Denominations on gears and gear pairs Supplement 1: Denominations on gears and gear pairs; alphabetical index of equivalent terms DIN 8000 Design dimensions and errors of hobs for involute spur gears; fundamental terms DIN 8580 Manufacturing processes - Terms and definitions, division DIN 8589 Manufacturing processes, chip removal - Part 12: Belt grinding (sanding); classification, subdivision, terms and definitions DIN 13320 Acoustics - Spectra and frequency curves, concepts, representation DIN 21683 Acoustics - Preferred reference quantities for acoustic levels (ISO 1683:1983); withdrawn; replaced by DIN EN ISO 1683:2008-11 DIN 21772 Gears - Cylindrical involute gears and gear pairs - Definition of deviations (replacement of DIN 3960) DIN 21773 Gears - Cylindrical involute gears and gear pairs - Inspection dimensions of tooth thickness DIN 31652 Plain bearings; hydrodynamic plain journal bearings designed for operation under steady-state conditions circular; withdrawn; commended: ISO 7902-1,-2,-3:1998-07 DIN 45635 Measurement of noise emitted by machines, airborne noise emission Part 1: Enveloping surface method; basic method, divided into three grades of accuracy Part 2: Reverberation room method; basic measurement method (precision method); withdrawn; replaced by: DIN EN ISO 3741:2001-01, :2009-11 :2011-01 Part 11: Enveloping surface method; internal combustion engines Part 23: Gear transmission Part 38: Enveloping surface method, reverberation room method and in-duct method; fans DIN 45645 Determination of rating levels from measurement data Part 1: Noise immission in the neighbourhood Part 2: Determination of the noise rating level for occupational activities at the workplace for the level range underneath the given risk of hearing damage DIN 45667 Classification methods for evaluation of random vibrations DIN 45681 Acoustics - Determination of tonal components of noise and determination of a tone adjustment for the assessment of noise immissions DIN 50320 Wear; terms, systematic analysis of wear processes, classification of wear phenomena; withdrawn DIN 51501 Lubricants; lubricating oils L-AN; minimum requirements; withdrawn DIN 51509 Part 1: Selection of lubricants for gears; gear lubricating oils Part 2: Selection of lubricants for gears; semi-fluid lubricants DIN 51517 Lubricants - Lubricating oils Part 1: Lubricating oils C; minimum requirements Part 2: Lubricating oils CL; minimum requirements Part 3: Lubricating oils CLP; minimum requirements DIN 58400 Basic rack profile for spur gears with involute teeth for fine mechanics Appendix 17 Standards (Selection) 783 Appendix 17.1 Gear Standards (Selection), continued (ANSI) AGMA (selection) AGMA 901-A92 A rational procedure for the preliminary design of minimum volume gears AGMA 908-B89 (Revision of AGMA 226.01) Geometry factors for determining the pitting resistance and bending strength of spur, helical and herringbone gear teeth AGMA 911-A94 Design guidelines for aerospace gearing AGMA 12-A04 Mechanisms of gear tooth failures AGMA 913-A98 Method for specifying the geometry of spur and helical gears AGMA 914-B04 (Revision of AGMA 299.01) Gear sound manual Part I: Fundamentals of sound as related to gears Part II: Sources, specifications and levels of gear sound Part III: Gear noise control AGMA 918-A93 A summary of numerical examples demonstrating the procedures for calculating geometry factors for spur and helical gears AGMA 923-B05 Metallurgical specifications for steel gearing AGMA 925-A03 Effect of lubrication on gear surface distress AGMA 927-A01 Load distribution factors - Analytical methods for cylindrical gears AGMA 933-B03 Basic gear geometry AGMA 938-A05 Shot peening of gears AGMA 940-A09 Double helical epicyclic gear units ANSI AGMA 1010-E95 Appearance of gear teeth - Terminology of wear and failure ANSI AGMA 1012-G05 Gear nomenclature, definition of terms with symbols ANSI AGMA 2101-D04 Fundamental rating factors and calculation methods for involute spur and helical teeth (metric edition) ANSI AGMA 6032-A94 Standard for marine gear unit: Rating AGMA ISO 14179-1 Gear reducers - Thermal capacity, based on ISO/TR 14179-1 784 Appendix 17 Standards (Selection) Appendix 17.2 Material Standards (Selection) DIN EN ISO 642 Steel - Hardenability test by end quenching (Jominy test) DIN EN ISO 2639 Steels - Determination and verification of the depth of carburized and hardened cases DIN EN ISO 18265 Metallic materials - Conversion of hardness values DIN ISO 15787 Technical product documentation - Heat-treated ferrous parts - Presentation and indications (ISO 15787:2001) DIN EN 1561 Founding - Grey cast irons DIN EN 1562 Founding - Malleable cast irons DIN EN 1563 Founding - Spheroidal graphite cast irons DIN EN 1564 Founding - Ausferritic spheroidal graphite cast irons DIN EN 2832 Aerospace series; hydrogen embrittlement of steels; notched specimen test (will be replaced by DIN EN ISO 6872) DIN EN 10002-5 Tensile testing of metallic materials; method of testing at elevated temperature; withdrawn DIN EN 10020 Definition and classification of grades of steel DIN EN 10025-2 Hot-rolled products of structural steels - Part 2: Technical delivery conditions for non-alloy structural steels DIN EN 10027-1 Designation systems for steels - Part 1: Steel names DIN EN 10027-2 Designation systems for steels - Part 2: Numerical system DIN EN 10052 Vocabulary of heat-treatment terms for ferrous products, replaced by ISO 4885 (Ferrous products, heat treatment vocabulary) DIN EN 10083-1 Steels for quenching and tempering - Part 1: General technical delivery conditions DIN EN 10083-2 Steels for quenching and tempering - Part 2: Technical delivery conditions for non-alloy steels DIN EN 10083-3 Steels for quenching and tempering - Part 3: Technical delivery conditions for alloy steels DIN EN 10084 Case hardening steels - Technical delivery conditions DIN EN 10085 Nitriding steels - Technical delivery conditions DIN EN 10250-1 Open die steel forgings for general engineering purposes - Part 1: General requirements DIN EN 10250-2 Open die steel forgings for general engineering purposes - Part 2: Non-alloy quality and special steels DIN EN 10250-3 Open die steel forgings for general engineering purposes - Part 3: Low-alloy special steels DIN EN 10293 Steel castings for general engineering uses; withdrawn DIN EN 10328 Iron and steel - Determination of the conventional depth of hardening after surface heating DIN EN 10267 Ferritc-pearlitic steels for precipitation hardening from hot-working temperatures DIN EN 12540 Corrosion protection of metals - Electrodeposited coatings of nickel, nickel plus chromium, copper plus nickel and copper plus nickel plus chromium DIN 17021-1 Heat treatment of ferrous metals; material selection; steel selection according to hardenability DIN 17022-1 Heat treatment of ferrous materials - Methods of heat treatment - Part 1: Hardening, austempering, annealing, quenching, tempering of components DIN 17022-3 Heat treatment of ferrous materials; heat treatment methods; case hardening DIN 17022-4 Heat treatment of ferrous materials - Methods of heat treatment - Part 4: Nitriding and nitrocarburizing DIN 17022-5 Heat treatment of ferrous materials - Methods of heat treatment - Part 5: Surface hardening Appendix 17 Standards (Selection) 785 Appendix 17.2 Material Standards (Selection), continued DIN 17023 Heat treatment of ferrous metals - (WBA) Forms - Heat-treatment order DIN 30902 Light-microscopical determination of the depth and porosity of the compound layer of nitrided and nitro-carburized ferrous parts DIN 50190-3 Hardness depth of heat-treated parts; determination of the effective depth of hardening after nitriding DIN 50938 Black oxide coatings on iron or steel - Requirements and test methods DIN 50960-1 Electroplated coatings - Designation in technical documents DIN 50960-2 Electroplated coatings - Part 2: Indications on technical drawings VDI 2736 Thermo-plastic toothed gears VDI 3198 Coating of tools for cold massive forming VDI 3824 sheet 1 Quality assurance for PVD and CVD hard layer coatings, characteristics and fields of application of hard layer coatings VDI 3824 sheet 2 Quality assurance for PVD and CVD hard layer coatings, requirements for tools and work pieces intended for coating VDI 3824 sheet 3 Quality assurance for PVD and CVD hard layer coatings, manufacturing processes and activities VDI 3824 sheet 4 Quality assurance for PVD and CVD hard layer coatings, test planning for hard layer coatings VDI/VDE 2420 Metal surface treatment in precision engineering, non-metallic coatings I; chemical and electro- chemical methods, sheet 6 SEP 1520 Microscopic testing of carbide structure in steels with picture series SEP 1664 Determination of formulas by multiple regression for calculating the hardenability in the end quenching test based on the chemical composition of steels SEP 1664 Supplement Determination of formulas by multiple regression for calculating the hardenability in the end quenching test based on the chemical composition of steels SEW 550 Steels for large forgings; quality specifications SEW 685 Cast steel tough at sub-zero temperature; technical delivery conditions SEW 835 Cast steel for flame and induction hardening 786 Appendix 17 Standards (Selection) Appendix 17.3 International Material Denomination (Selection) EN standard (Europe) Mate- rial number Formerly nationally standardised steel types ASTM SAE J (USA) JIS (Japan) GOST (Russia) DIN France BS/GB Unalloyed tempering steels (DIN) EN 10083-2 C35E C35R 1.1181 1.1180 (Ck 35) Cm 35 [XC 38 H1] [XC 38 H1u] (080M36) - SAE 1035 SAE 1038 S35C S35CM S38C 35 C45E C45R 1.1191 1.1201 (Ck 45) Cm 45 XC45 [XC 48 H1] [XC 48 H1u] (080M46) - SAE 1045 S45C S45CM S48C 45 C60E C60R 1.1221 1.1223 Ck 60 Cm 60 - - (070M60) 060A62 SAE 1060 S58C S60C-CSP S60CM S65C-CSP S65CM 60 Alloyed tempering steels (DIN) EN 10083-3 34Cr4 34CrS4 1.7033 1.7037 34Cr4 34CrS4 (32 C 4) (32 C 4 u) (530M32) - SAE 5132 SAE 5135 SCr430 \*) SCr430H SCr435 SCr435M SCr2H 35Ch - 34CrMo4 34CrMoS4 1.7220 1.7226 34CrMo4 334CrMoS4 (34 CD 4) (34 CD 4 u) (708M32) - SAE 4135H SAE 4137 SAE 4140 SCM435 \*) SCM435H SCM435M SCM435TK 35ChM AS38ChGM - 34CrNiMo6 1.6582 (34CrNiMo6) - (817M40) SAE 4340 SNCM439 38Ch2N2MA 42CrMo4 42CrMoS4 1.7225 1.7227 42CrMo4 42CrMoS4 42 CD 4 42 CD 4 u 709M40 708M40 - SAE 4140 SAE 4140H SCM440RCH SNB7 Class 2SCM440H SCM4H SCM440 38ChM - 36NiCrMo16 1.6773 - 35 NCD 16 - - 51CrV4 1.8159 50CrV4 (50CV 4) [735A50] SAE 6150 SAE 6150H SUP10 SUP10M 50ChFA 50ChGFA Nitriding steels EN 10085 31CrMo12 1.8515 31CrMo12 31CrMoV9 1.8519 31CrMoV9 33CrMoV12-9 1.8522 - Unalloyed case hard. steels C15 1.0401 C15 080A15 080M15 SAE 1015 SAE 1017 - S15C S15CK 15 C22 1.0402 C22 SAE 1020 SAE 1022 S20C S22C 20 Alloyed case hardening steels EN 10084 16MnCr5 16MnCrS5 1.7131 1.7139 16MnCr5 16MnCrS5 16MC5 - 590M17 527M17 - SAE 5120 SAE 4118H - - 18ChG - 20MnCr5 20MnCrS5 1.7147 1.7149 20MnCr5 20MnCrS5 20MC5 - - SAE 5120 - SMnC420 \*) SMnC420H SMnC21H 18ChG - 18CrNiMo7-6 1.6587 17CrNiMo6 18NCD6 - SAE 4320 SAE 4320H SAE 8822H - 20NiCrMo13-4 1.6660 - - - 6493 (AMS) 6492 (AMS) AFP steels EN 10267 30MnVS6 1.1302 27MnSiVS6 - - 15V24 46MnVS6 1.1304 44MnSiVS6 - - 46MnVS3 1.1305 42MnSiVS3-3 - - Denomination in parentheses means that the chemical composition differs only slightly from EN 10083-2, -3. Square brackets are used in case of larger deviations from EN 10083-2, -3. \*) All without additional sulphur content. Appendix 18 Factors and Equations used in American Standards 787 Appendix 18 Factors and Equations used in American Standards Special Factors and Equations, USA (see also ANSI/AGMA 1012-G05) Symbol To find Having Equation dP DP Diametral pitch Module m (mm) d25.4Pm= (in.) dP DP Diametral pitch Circular pitch p dʌPp= dP DP Diametral pitch Number of teeth z and pitch diameter d d=zPd p CP Circular pitch Module m (mm) ʌ 25.4mp= (in.) p CP Circular pitch Diametral pitchdP ʌ dpP= p CP Circular pitch Number of teeth z and pitch diameter d ʌ=dpz z Number of teeth Pitch diameter d and circular pitch p ʌ=dzp a Center distance Number of pinion teeth z 1, number of gear teeth z 2 and diametral pitch Pd 12 d2+=zzaP (in case of Įt = Įtw) a Center distance Number of pinion teeth z 1, number of gear teeth z 2 and circular pitch p 12 d2+=zzaP (in case of Įt = Įtw) a Center distance Gear an pinion pitch diameter d 2, d1 12 2+=dda (in case of Įt = Įtw) px Axial pitch Circular pitch p and helix angle ȕ xtan pp= db Base diameter Pitch diameter d and transverse pressure angle Įt db=d cosĮt Millimeter per in = 25.4; m,z, Įt, Įtw, px, d, d b, p, according to ANSI/AGMA 1012-G05 The Circular pitch is the distance on the pitch circle (or pitch line) between similar sides of adjacent teeth. The Diametral pitch is the ratio of the number of teeth to the pitch diameter. VLHPHQV FRP JHDUXQLWV What keeps movingibliography General literature (primarily of a historical nature) [1] Fronius, S.: Konstruktionsleh re – Antriebselemente [ Theory of Design – Drive Elements ]. Berlin: Verlag Technik 1979 [2] Niemann ,G.; Winter, H.: Maschinenelemente [ Machine Elements ]. Band II: Getriebe allgemein, Zahnradgetriebe – Grundlagen, Stirnradgetriebe [ Volume II: Gears, General Aspects, Gear Drives – Basic Principles, Cylindrical Gear Units ]. Band III: Schraubrad-, Kegelrad-, Schnecken-, Ketten-, Riemen-, Reibradgetriebe, Kupplungen, Bremsen, Freilauf [ Volume III: Crossed Helical, Bevel, Worm, Chain, Belt, Friction Gears, Clutches, Brakes, Freewheel ]. Berlin: Springer 1983 [3] Keck, K.F.: Die Zahnradpraxis [ Gear Practice ]. Band I: Geradzahn-Stirnr der [ Volume I: Spur Gears ]. 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