گزارش پاراگراف‌های حاوی کلمه Backlash

تعداد 55 پاراگراف حاوی کلمه 'backlash' پیدا شد:

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## پاراگراف صفحه 28، شماره 28.2:

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) Plücker, 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) Saalschütz, Luis (German university lecturer on mathe- matics and mechanics in Königsberg) • Published the first German-language specialist work “On the Theory of Involute Teeth” Königsberg (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) Fölmer, 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)

### جملات مرتبط:

- 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.

## پاراگراف صفحه 71، شماره 71.2:

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.

### جملات مرتبط:

- 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.

## پاراگراف صفحه 89، شماره 89.2:

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)

### جملات مرتبط:

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## پاراگراف صفحه 89، شماره 89.4:

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).

### جملات مرتبط:

- 1) To avoid clamping, backlash is necessary.

- This is subsequently realised by a backlash allowance for the tooth thickness or respectively span.

## پاراگراف صفحه 90، شماره 90.2:

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 .

### جملات مرتبط:

- 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.

## پاراگراف صفحه 94، شماره 94.2:

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 ß 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

### جملات مرتبط:

- 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.

## پاراگراف صفحه 106، شماره 106.2:

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§· ¨¸ ©¹ (2.2/22a) The root diameter d f is naP fn = 2 + hdd m x cm∗§·−−¨¸©¹ (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.

### جملات مرتبط:

- Other things that have an influence on the root diameter are the backlash allowances because they are usually considered by tool advance or retraction.

## پاراگراف صفحه 133، شماره 133.2:

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.

### جملات مرتبط:

- 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.

## پاراگراف صفحه 225، شماره 225.2:

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).

### جملات مرتبط:

- 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.

## پاراگراف صفحه 230، شماره 230.2:

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:

### جملات مرتبط:

- As such, usually only linearised models (without backlash of the tooth) can be calculated.

## پاراگراف صفحه 231، شماره 231.2:

• 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].

### جملات مرتبط:

- 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.

- 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.

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- 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]).

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## پاراگراف صفحه 232، شماره 232.2:

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.

### جملات مرتبط:

- 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.

## پاراگراف صفحه 374، شماره 374.2:

CW H 11 h HT CT WT1.4ȗ ı ıȗWc N§·=⋅ ⋅ ⋅¨¸ ©¹N 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 Voßiek 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)

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## پاراگراف صفحه 384، شماره 384.2:

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).

### جملات مرتبط:

- 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.

## پاراگراف صفحه 571، شماره 571.2:

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].

### جملات مرتبط:

- 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 .

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## پاراگراف صفحه 572، شماره 572.2:

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

### جملات مرتبط:

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- Figure 8/1 Tooth thickness limit deviations and backlash

## پاراگراف صفحه 592، شماره 592.1:

8.4 Backlash Allowance Parameters 575

### جملات مرتبط:

- 8.4 Backlash Allowance Parameters 575

## پاراگراف صفحه 592، شماره 592.2:

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

### جملات مرتبط:

- 8.4 Backlash Allowance Parameters 8.4.1 Backlash Figure 8/50 Runout testing

## پاراگراف صفحه 592، شماره 592.3:

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

### جملات مرتبط:

- 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).

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## پاراگراف صفحه 593، شماره 593.2:

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

### جملات مرتبط:

- The thickness of the deformed wire shows the backlash.

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## پاراگراف صفحه 593، شماره 593.3:

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:

### جملات مرتبط:

- 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.

## پاراگراف صفحه 593، شماره 593.4:

(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

### جملات مرتبط:

- (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.

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- 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.

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## پاراگراف صفحه 594، شماره 594.1:

8.4 Backlash Allowance Parameters 577

### جملات مرتبط:

- 8.4 Backlash Allowance Parameters 577

## پاراگراف صفحه 594، شماره 594.2:

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

### جملات مرتبط:

- 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.

## پاراگراف صفحه 596، شماره 596.1:

8.4 Backlash Allowance Parameters 579

### جملات مرتبط:

- 8.4 Backlash Allowance Parameters 579

## پاراگراف صفحه 598، شماره 598.1:

8.4 Backlash Allowance Parameters 581

### جملات مرتبط:

- 8.4 Backlash Allowance Parameters 581

## پاراگراف صفحه 598، شماره 598.3:

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

### جملات مرتبط:

- 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.

## پاراگراف صفحه 601، شماره 601.2:

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

### جملات مرتبط:

- 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.

## پاراگراف صفحه 604، شماره 604.2:

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

### جملات مرتبط:

- 8.9 Use of Spline Gauges The determination of the described quality and backlash allowance parameters is insufficient particularly in the evaluation of splines.

## پاراگراف صفحه 605، شماره 605.2:

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

### جملات مرتبط:

- 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

## پاراگراف صفحه 605، شماره 605.3:

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

### جملات مرتبط:

- 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

## پاراگراف صفحه 662، شماره 662.2:

- 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.

### جملات مرتبط:

- To realise the infeed of the hone, the backlash of the guiding pair of discs can be exploited.

## پاراگراف صفحه 722، شماره 722.2:

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- 715 3 Flank Backlash and Flank Modifications ....................................................................................

- 716 3.1 Recommendations for Selecting the Flank Backlash and the Base Tangent Length or Tooth Thickness Allowances ...................................................................................................

## پاراگراف صفحه 729، شماره 729.1:

716 Appendix 3 Flank Backlash and Flank Modifications

### جملات مرتبط:

- 716 Appendix 3 Flank Backlash and Flank Modifications

## پاراگراف صفحه 729، شماره 729.2:

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++ + + ⋅+ + +­ ½ ªº§·§· §· §· ° ° «»Δ≈ ⋅ ⋅ ¨¸¨¸® ¾ ¨¸ ¨¸ ¨¸¨¸«» ©¹ ©¹ ° ° ©¹©¹«»¬¼¯ ¿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.

### جملات مرتبط:

- 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.

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## پاراگراف صفحه 729، شماره 729.3:

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.

### جملات مرتبط:

- 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.

## پاراگراف صفحه 730، شماره 730.1:

Appendix 3 Flank Backlash and Flank Modifications 717

### جملات مرتبط:

- Appendix 3 Flank Backlash and Flank Modifications 717

## پاراگراف صفحه 731، شماره 731.1:

718 Appendix 3 Flank Backlash and Flank Modifications

### جملات مرتبط:

- 718 Appendix 3 Flank Backlash and Flank Modifications

## پاراگراف صفحه 732، شماره 732.1:

Appendix 3 Flank Backlash and Flank Modifications 719

### جملات مرتبط:

- Appendix 3 Flank Backlash and Flank Modifications 719

## پاراگراف صفحه 733، شماره 733.1:

720 Appendix 3 Flank Backlash and Flank Modifications

### جملات مرتبط:

- 720 Appendix 3 Flank Backlash and Flank Modifications

## پاراگراف صفحه 734، شماره 734.2:

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.

### جملات مرتبط:

- 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.

## پاراگراف صفحه 734، شماره 734.3:

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.

### جملات مرتبط:

- 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 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.

- 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.

## پاراگراف صفحه 738، شماره 738.3:

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

### جملات مرتبط:

- 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

## پاراگراف صفحه 763، شماره 763.3:

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.

### جملات مرتبط:

- 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.

## پاراگراف صفحه 790، شماره 790.2:

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)

### جملات مرتبط:

- 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.

## پاراگراف صفحه 791، شماره 791.2:

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)

### جملات مرتبط:

- 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)

## پاراگراف صفحه 829، شماره 829.3:

Literature on Section 8 (Quality Assurance) [8/1] DIN 3960:1987-03 Begriffe und Bestimmungsgrößen für Stirnräder (Zylinderräder) und Stirnradpaare (Zylinderradpaare) mit Evolventenverzahnung [ DIN 3960:1987-03 Definitions, Parameters and Equations for Involute Cylindrical Gears and Gear Pairs ] [8/2] DIN 3961:1978-08 Toleranzen für Stirnradverzahnungen; Grundlagen [ DIN 3961:1978-08 Tolerances for Cylindrical Gear Teeth: Fundamentals ] [8/3] DIN 3962-1:1978-08 Toleranzen für Stirnradverzahnungen; Toleranzen für Abweichungen einzelner Bestimmungsgrößen [ DIN 3962-1:1978-08 Tolerances for Cylindrical Gear Teeth; Tolerances for Deviations of Individual Parameters ] [8/4] DIN 3962-2:1978-08 Toleranzen für Stirnradv erzahnungen; Toleranzen für Flankenlinienabweichun- gen [ DIN 3962-2:1978-08 Tolerances for Cylindrical Gear Teeth; Tolerances for Tooth Trace Deviations ] [8/5] DIN 3962-3:1978-08 Toleranzen für Stirnradverzahnungen; Toleranzen für Teilungs-Spannen-abweichungen [ DIN 3962-3:1978-08 Tolerances for Cylindrical Gear Teeth; Tolerances for Sector Pitch Deviations ] [8/6] DIN 3963:1978-08 Toleranzen für Stirnradverzahnungen; Toleranzen für Wälzabweichungen [DIN 3963:1978-08 Tolerances for Cylindrical Gear Teeth; Tolerances for Composite Deviations ] [8/7] DIN 3964:1980-11 Achsabstandsabmaße und Achslagetoleranzen von Gehäusen für Stirnradgetriebe [DIN 3964:1980-11 Deviations of Shaft Centre Distances and Shaft Position Tolerances of Casings for Cylindrical Gears ] [8/8] DIN 3967:1978-08 Getriebe-Passsystem; Flankenspiel, Zahndickenabmaße, Zahndickentoleranzen, Grundlagen [ DIN 3967:1978-08 System of Gear Fits; Backlash, Tooth Thickness Allowances, Tooth Thickness Tolerances; Principles ] [8/9] DIN 3970:2010-04 Lehrzahnräder zum Prüfen von Zylinderrädern – Radkörper und Verzahnung [DIN 3970:2010-04 Master Gears for Checking Cylindrical Gears – Gear Blanks and Gearing ] [8/10] ISO 21771:2007-09 Gears – Cylindrical Involute Gears and Gear Pairs – Concepts and Geometry [8/11] DIN ISO 21771:2014-08 Zahnräder – Zylinderräder und Zylinderradpaare mit Evolventenverzahnung – Bestimmungsgrößen und Geometrie [ DIN ISO 21771:2014-08 Gears – Cylindrical Involute Gears and Gear Pairs - Concepts and Geometry ] [8/12] DIN 21772:2012-07 Zahnräder – Zylinderräder und Zylinderradpaare mit Evolventenverzahnung – Definition der Abweichungen [ DIN 21772:2012-07 Gears – Cylindrical Involute Gears and Gear Pairs - Definition of Deviations] [8/13] DIN EN ISO 286-1:2010-11 Geometrische Produktspezifikation (GPS) – ISO-Toleranzsystem für Längenmaße – Teil 1: Grundlagen für Toleranzen, Abmaße und Passungen [ DIN EN ISO 286- 1:2010-11 Geometrical Product Specification (GPS) – ISO Code System for Tolerances on Linear Sizes – Part 1: Basis of Tolerances, Deviations and Fits ] [8/14] DIN EN ISO 286-2:2010-11 Geometrische Produktspezifikation (GPS) – ISO-Toleranzsystem für Längenmaße – Teil 2: Tabellen der Grundtoleranzgrade und Grenzabmaße für Bohrungen und Wellen [ DIN EN ISO 286-2:2010-11 Geometrical Product Specifications (GPS) – ISO Code System for Tolerances on Linear Sizes – Part 2: Tables of Standard Tolerance Classes and Limit Deviations for Holes and Shafts ] [8/15] DIN EN ISO 3274:1998-04 Geometrische Produktspezifikationen (GPS) – Oberflächenbeschaffenheit: Tastschnittverfahren – Nenneigenschaften von Tastschnittgeräten [ DIN EN ISO 3274:1998-04 Geometrical Product Specifications (GPS) – Surface Texture: Stylus Method – Nominal Characteristics of Contact (Stylus) Instruments ] [8/16] DIN 3966-1:1978-08 Angaben für Verzahnungen in Zeichnungen; Angaben für Stirnrad- (Zylinderrad) Evolventenverzahnungen [ DIN 3966-1:1978-08 Informa tion on Gear Teeth in Drawings; Information on Involute Teeth for Cylindrical Gears ]

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- Literature on Section 8 (Quality Assurance) [8/1] DIN 3960:1987-03 Begriffe und Bestimmungsgrößen für Stirnräder (Zylinderräder) und Stirnradpaare (Zylinderradpaare) mit Evolventenverzahnung [ DIN 3960:1987-03 Definitions, Parameters and Equations for Involute Cylindrical Gears and Gear Pairs ] [8/2] DIN 3961:1978-08 Toleranzen für Stirnradverzahnungen; Grundlagen [ DIN 3961:1978-08 Tolerances for Cylindrical Gear Teeth: Fundamentals ] [8/3] DIN 3962-1:1978-08 Toleranzen für Stirnradverzahnungen; Toleranzen für Abweichungen einzelner Bestimmungsgrößen [ DIN 3962-1:1978-08 Tolerances for Cylindrical Gear Teeth; Tolerances for Deviations of Individual Parameters ] [8/4] DIN 3962-2:1978-08 Toleranzen für Stirnradv erzahnungen; Toleranzen für Flankenlinienabweichun- gen [ DIN 3962-2:1978-08 Tolerances for Cylindrical Gear Teeth; Tolerances for Tooth Trace Deviations ] [8/5] DIN 3962-3:1978-08 Toleranzen für Stirnradverzahnungen; Toleranzen für Teilungs-Spannen-abweichungen [ DIN 3962-3:1978-08 Tolerances for Cylindrical Gear Teeth; Tolerances for Sector Pitch Deviations ] [8/6] DIN 3963:1978-08 Toleranzen für Stirnradverzahnungen; Toleranzen für Wälzabweichungen [DIN 3963:1978-08 Tolerances for Cylindrical Gear Teeth; Tolerances for Composite Deviations ] [8/7] DIN 3964:1980-11 Achsabstandsabmaße und Achslagetoleranzen von Gehäusen für Stirnradgetriebe [DIN 3964:1980-11 Deviations of Shaft Centre Distances and Shaft Position Tolerances of Casings for Cylindrical Gears ] [8/8] DIN 3967:1978-08 Getriebe-Passsystem; Flankenspiel, Zahndickenabmaße, Zahndickentoleranzen, Grundlagen [ DIN 3967:1978-08 System of Gear Fits; Backlash, Tooth Thickness Allowances, Tooth Thickness Tolerances; Principles ] [8/9] DIN 3970:2010-04 Lehrzahnräder zum Prüfen von Zylinderrädern – Radkörper und Verzahnung [DIN 3970:2010-04 Master Gears for Checking Cylindrical Gears – Gear Blanks and Gearing ] [8/10] ISO 21771:2007-09 Gears – Cylindrical Involute Gears and Gear Pairs – Concepts and Geometry [8/11] DIN ISO 21771:2014-08 Zahnräder – Zylinderräder und Zylinderradpaare mit Evolventenverzahnung – Bestimmungsgrößen und Geometrie [ DIN ISO 21771:2014-08 Gears – Cylindrical Involute Gears and Gear Pairs - Concepts and Geometry ] [8/12] DIN 21772:2012-07 Zahnräder – Zylinderräder und Zylinderradpaare mit Evolventenverzahnung – Definition der Abweichungen [ DIN 21772:2012-07 Gears – Cylindrical Involute Gears and Gear Pairs - Definition of Deviations] [8/13] DIN EN ISO 286-1:2010-11 Geometrische Produktspezifikation (GPS) – ISO-Toleranzsystem für Längenmaße – Teil 1: Grundlagen für Toleranzen, Abmaße und Passungen [ DIN EN ISO 286- 1:2010-11 Geometrical Product Specification (GPS) – ISO Code System for Tolerances on Linear Sizes – Part 1: Basis of Tolerances, Deviations and Fits ] [8/14] DIN EN ISO 286-2:2010-11 Geometrische Produktspezifikation (GPS) – ISO-Toleranzsystem für Längenmaße – Teil 2: Tabellen der Grundtoleranzgrade und Grenzabmaße für Bohrungen und Wellen [ DIN EN ISO 286-2:2010-11 Geometrical Product Specifications (GPS) – ISO Code System for Tolerances on Linear Sizes – Part 2: Tables of Standard Tolerance Classes and Limit Deviations for Holes and Shafts ] [8/15] DIN EN ISO 3274:1998-04 Geometrische Produktspezifikationen (GPS) – Oberflächenbeschaffenheit: Tastschnittverfahren – Nenneigenschaften von Tastschnittgeräten [ DIN EN ISO 3274:1998-04 Geometrical Product Specifications (GPS) – Surface Texture: Stylus Method – Nominal Characteristics of Contact (Stylus) Instruments ] [8/16] DIN 3966-1:1978-08 Angaben für Verzahnungen in Zeichnungen; Angaben für Stirnrad- (Zylinderrad) Evolventenverzahnungen [ DIN 3966-1:1978-08 Informa tion on Gear Teeth in Drawings; Information on Involute Teeth for Cylindrical Gears ]

## پاراگراف صفحه 830، شماره 830.2:

[8/17] DIN EN ISO 4288:1998-04 Geometrische Produktspezifikation (GPS) – Oberflächenbeschaffenheit: Tastschnittverfahren – Regeln und Verfahren für die Beurteilung der Oberflächenbeschaffenheit [DIN EN ISO 4288:1998-04 Geo metrical Product Specifications (GPS ) – Surface Texture: Stylus Method - Rules and Procedures for the Assessment of Surface Texture ] [8/18] ISO 1328-1:2013-09 Cylindrical Gears – ISO System of Flank Tolerance Classification – Part 1: Definitions and Allowable Values of Deviations Relevant to Flanks of Gear Teeth [8/19] ISO 1328-2:1997-08 Stirnräder – ISO-Toleranzsystem – Teil 2: Definitionen und zulässige Werte für Zweiflanken-Wälzabweichungen und Rundlaufabweichungen [ ISO 1328-2:1997-08 Cylindrical Gears – ISO System of Accuracy – Part 2: Definitions and Allowable Values of Deviations relevant to Radial Composite Deviations and Runout Information ] [8/20] ISO/TR 10064-1:1992-02 Stirnräder; Richtlinien für die Prüfung; Teil 1: Prüfung gleichnamiger Zahnflanken [ ISO/TR 10064-1:1992-02 Cylindrical gears; Code of Inspection Practice; Part 1: Inspection of Corresponding Flanks of Gear Teeth ] [8/21] ISO/TR 10064-2:1996-03 Stirnräder – Anleitung für die Durchführung der Prüfu ng – Teil 2: Prüfung von Zweiflankenwälzabweichungen, Rundlauf, Zahndicke und Flankenspiel [ ISO/TR 10064-2:1996- 03 Cylindrical Gears – Code of Inspection Practice – Part 2: Inspection related to Radial Composite Deviations, Runout, Tooth Thickness and Backlash ] [8/22] ISO/TR 10064-3:1996-08 Stirnräder – Anleitung für die Durchführung der Prüfung – Teil 3: Empfeh- lungen zu Radkörper, Achsabstand und Achsparallelität [ ISO/TR 10064-3:1996-08 Cylindrical Gears – Code of Inspection Practice – Part 3: Recommendations relative to Gear Blanks, Shaft Centre Distance and Parallelism of Axes ] [8/23] ISO/TR 10064-4:1998-10 Stirnräder (Zylinderräder) – Richtlinien für die Prüfung – Teil 4: Empfehlungen bezogen auf die jeweilige Oberflächenbeschaffenheit und zur Tragbildprüfung [ISO/TR 10064-4:1998-10 Cylindrical Gears – Code of Inspection Practice – Part 4: Recommendations relative to Surface Texture and Tooth Contact Pattern Checking ] [8/24] ISO/TR 10064-5:2005-04 Zylinderräder – Richtlinien für die Prüfung – Teil 5: Empfehlung zur Be- urteilung der Messunsicherheit von Zahnradmessgeräten [ ISO/TR 10064-5:2005-04 Cylindrical Gears – Code of Inspection Practice – Part 5: Recommendations relative to Evaluation of Gear Measuring Instruments ] [8/25] DIN ISO 18653:2009-12 Zahnräder – Bewertung von Messgeräten für die Messung von Zahnrädern [DIN ISO 18653:2009-12 Gears – Evaluation of Instruments for the Measurement of Individual Gears ] [8/26] VDI/VDE 2607:2000-02 Rechnerunterstützte Auswertung vo n Profil- und Flankenlinienmessungen an Zylinderrädern mit Evolventenprofil [ VDI/VDE 2607:2000-02 Computer-aid ed Evaluation of Profile and Helix Measurements on Cylindrical Gears with Involute Profile ] [8/27] VDI/VDE 2608:2001-03 Einflanken- und Zweiflanken-Wälzprüfung an Zylinderrädern, Kegelrädern, Schnecken und Schneckenrädern [ VDI/VDE 2608:2001-03 Tangential Composite and Radial Composite Inspection of Cylindrical Gears, Bevel Gears, Worms and Worm Wheels ] [8/28] VDI/VDE 2609:2000-10 Ermittlung von Tragbildern an Verzahnungen [ VDI/VDE 2609:2000-10 Determination of Tooth Contact Patterns for Gearings ] [8/29] VDI/VDE 2610:2014-05 Format für den Austausch von Verzahnungsdaten [ VDI/VDE 2610:2014-05 Gear Data Exchange-Format (GDE Format) – Definition ] [8/30] VDI/VDE 2612:2000-05 Profil- und Flankenlinienprüfung an Zylinderrädern mit Evolentenprofil [VDI/VDE 2612:2000-05 Profile and Helix Checking of Involute Cylindrical Gears ] [8/31] VDI/VDE 2613:2003-12 Teilungs- und Rundlaufprüfun g an Verzahnungen – Zylinderräder, Schneckenräder, Kegelräder [ VDI/VDE 2613:2003-12 Pitch an d Runout Testing on Gearings - Cylindrical Gears, Wormwheels, Bevel Gears ] [8/32] VDI/VDE 2615:2006-08 Rauheitsprüfung an Zylinder- und Kegelrädern mit Tastschnittgeräten [VDI/VDE 2615:2006-08 Surface Roughness Measurement of Cylindrical Gears and Bevel Gears by means on Stylus-type Instruments ] [8/33] ANSI/AGMA 2015-1-A01 Accuracy Classification System – Tangential Measurements for Cylindrical Gears [8/34] ANSI/AGMA 2015-2-A06 Accuracy Classification System – Radial Measurements for Cylindrical Gears [8/35] AGMA 915-1-A02 Inspection Practices – Part 1: Cylindrical Gears – Tangential Measurements [8/36] AGMA 915-2-A05 Inspection Practices – Part 2: Cylindrical Gears – Radial Measurements [8/37] Bösser, F.; Trenk, M.; Wisweh, L.; Wengler, S.: Gewissheit über die Unsicherheit, Kalibrierung von realen Werkstücken [ Certainty on the Uncertainty, the Calibration of Real Workpieces ]. Qualität und Zuverlässigkeit 45(2000) 5, S. 612–616

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