Homework projet rules

During the course it is **mandatory** to complete a Homework project, that will be assigned to groups of 3 students. Up to **November 21st** you can register your group in the online Spreadsheet available at the following link:

https://docs.google.com/spreadsheets/d/1SLKVPPbPd5pzfBoI3CFYSgv4Pd3MwyhVXmwhsKu9wj0/edit?usp=sharing

The homework project is evaluated at the oral exam, accessed after passing the written test. The score of the oral exam about the homework project activity is in the range C = -2/+2.

Students who did the project in past academic years and received a valid score have two options:

1) The old scores for previous projects can be preserved but they are rescaled by a factor 2/3 (example: actual score: +1.5, new score: +1). In this case, no oral exam is required after passing the written exam, which now includes both exercises and theory questions.
2) Students can choose to do the homework project proposed for the current academic year, but following the new rules. In this case, the previous score is cancelled, and the homework project is evaluated with score in the range +/- 2, during the compulsory oral exam taken after passing the written part, which now includes both exercises and theory questions.

At the oral exam, each student must bring **a printed and bound copy** of the homework project technical report. It is not necessary to upload a pdf version of the homework project on the portal. No other file other than the technical report will be taken into account (Spreadsheet, matfile, scripts, etc.).

Remind that while the homework project is a **teamwork**, the score awarded during the oral examination is individual. **Each member of the group is expected to have a comprehensive understanding of all aspects of the technical report**.

Technical report for the verification of gearbox components

The subject of the technical report is the verification of the major components of the mechanical gearbox shown in Figure 1.

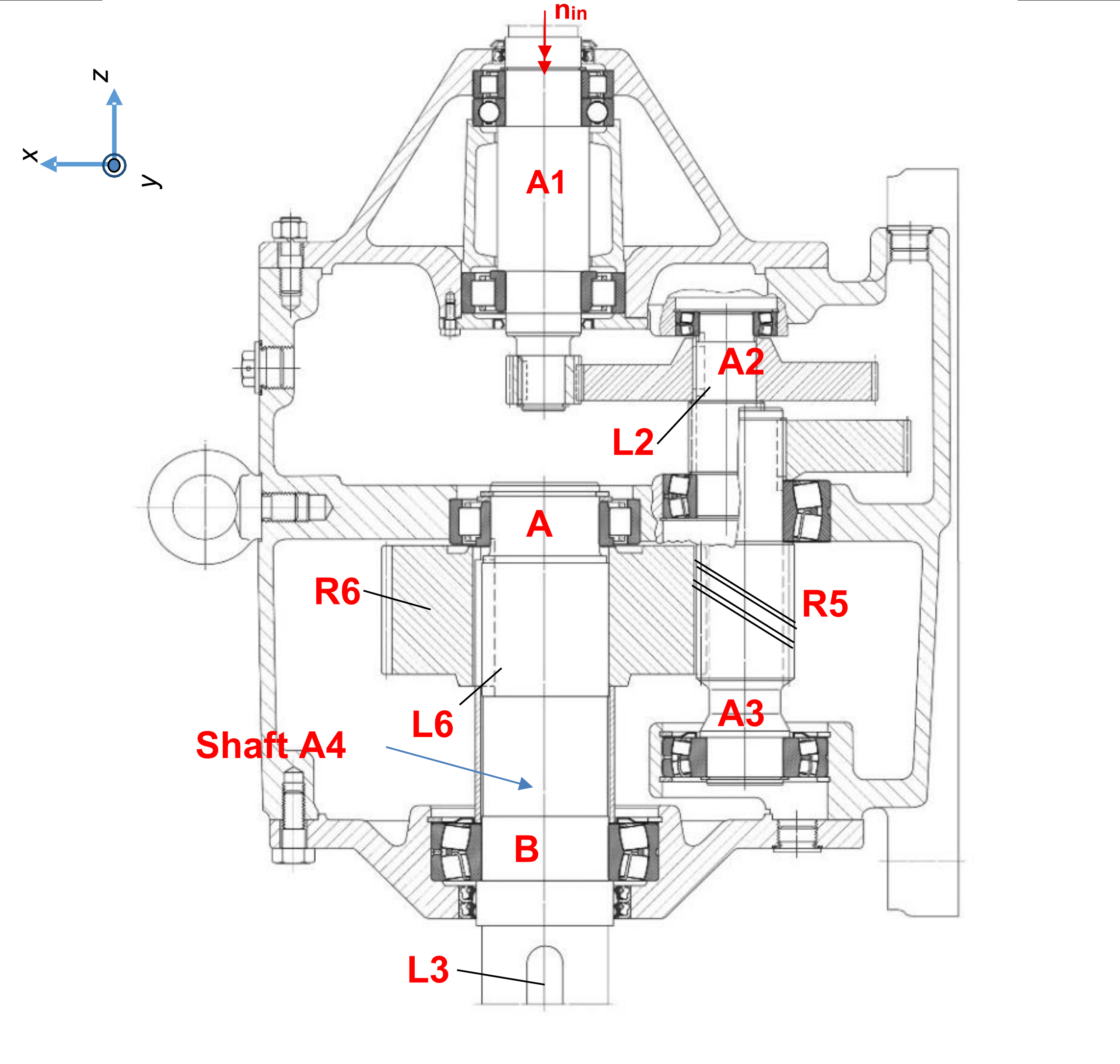


Figure 1. Gearbox assembly drawing (section)

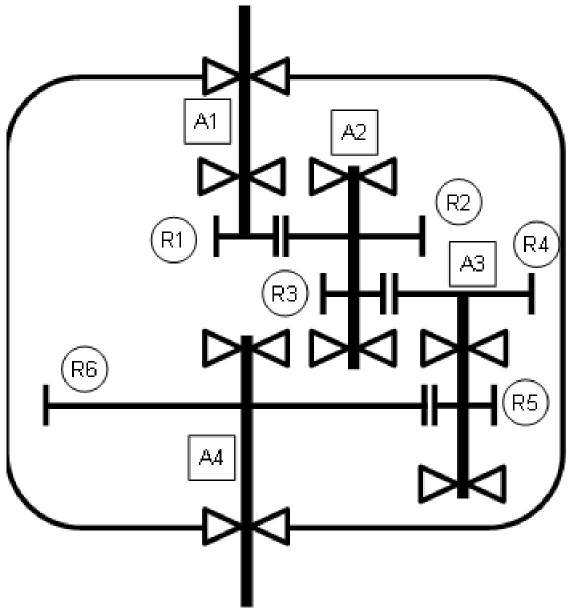


Figure 2. Gearbox functional scheme

LEGEND:

A1: gearbox input shaft; it receives power from the motor and transfers it to **shaft A2** through **pinion R1**.
R1: helical pinion of the first reduction stage, it transfers power from **shaft A1** to the **driven gear R2**.
R2: driven helical gear receiving power from **pinion R1** and transmitting it to **shaft A2**.
A2: secondary shaft, it receives power from **shaft A1** via **gear R2** and transmits it to **shaft A3** through **pinion R3**.
R3: helical pinion receiving power from **shaft A2** and transferring it to **shaft A3**.
R4: driven helical gear, it receives power from **pinion R3** and transfers it to **shaft A3**.
A3: tertiary shaft, receiving power from **shaft A2** via **gear R4** and transmitting it to **shaft A4** via **gear R5**.
R5: helical pinion, it receives power from **shaft A3** and transfers it to **shaft A4.**
R6: driven helical gear receiving power from **pinion R5** and transmitting it to **shaft A4**.
A4: output shaft of the gearbox, it receives power from shaft A3 via gear R6 and transmits it to the mechanical end-user via key **L3**.
L6: key that connects the output shaft **A4** with **gear R6** (UNI 6604-A 28x16x90).
L3: key that connects the output **shaft A4** with the **mechanical end-user** (UNI 6604-A 28x16x90)**.**

Shaft A1 is supported at one end by a roller bearing and at the other end by a pair of ball and roller bearings. Shafts A2 and A3 are supported by double-row spherical roller bearings. Shaft A4 is supported by a cylindrical roller bearing at one end (A) and a double-row spherical roller bearing at the other end (B).

The objective of the technical report is to verify the output shaft A4, gear R6, and bearings A and B that support the output shaft A4.

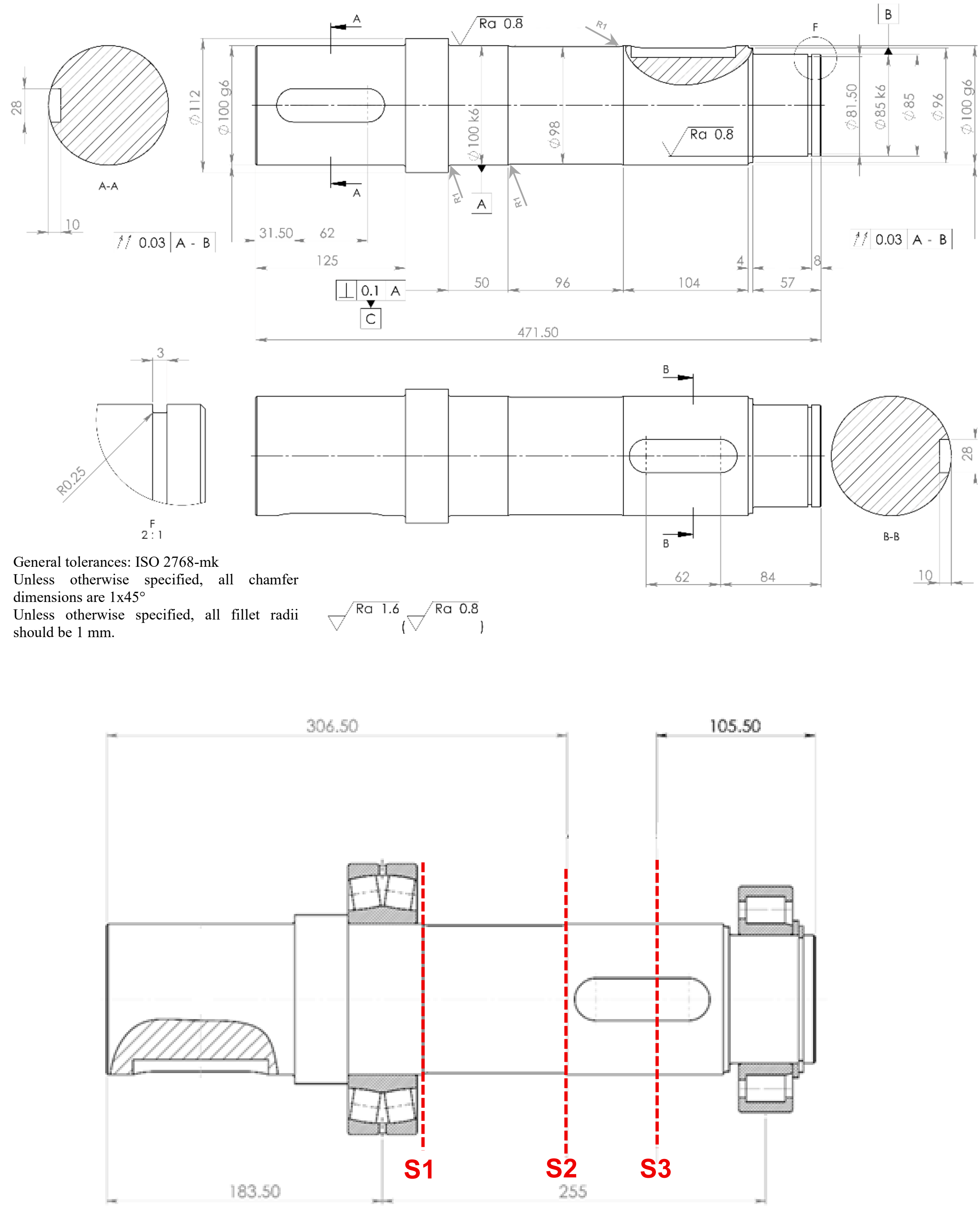


Figure 3. Main sizes of shaft A4 and bearing arrangement

Data for static and fatigue analysis on shaft A4 and gear R6

A motor, rotating at an angular speed n_{in}, provides a power P_{in} to the input shaft **A1** of the gearbox (see Annex 1 for input data). The Gearbox transfers power to the end-user, connected to the output shaft **A4**. The total transmission ratio of the gearbox (speed reducer) is τ_{tot} = 18.32. The gears 5 and 6 are cylindrical helical gears, made of 16MnCr5 carburized and hardened steel, with the following characteristics:

	R5	R6
normal module, m_n (mm)	8	8
number of teeth, z	15	39
helix angle, ψ	15°	15°
normal pressure angle, ϕ_n	20°	20°
transmission accuracy level, Q_v	6	6
tooth face width (mm)	109	109
Ultimate strength (MPa)	1050	1050
Yield strength (MPa)	930	930
Surface Hardness (HB)	525	525

The output shaft **A4** (Figure 3), which is to be verified both for static loading and fatigue, is made with a 34CrNiMo6 normalised steel (σ_u=800 MPa, σ_y=600 MPa, σ_{D-1}=400 MPa), and it is supported with a cylindrical roller bearing (bearing **A**) and a double-row spherical roller bearing (bearing **B**).

Bearings:

A: SKF NJ 217 ECP (Explorer).
B: SKF 24020 CC/W33 (Explorer).

MAIN REQUESTS

Part 1

- Static verification and Fatigue analysis for infinite life for the output shaft **A4** in the sections shown in Figure 3
  - $\mathbf{S_1}$ in diameter variation.
  - $\mathbf{S_2}$ in the **shoulder** machined to axially locate gear **R4** on shaft **A4**.
  - $\mathbf{S_3}$ in the middle of the driven gear **R6.**

  considering the stress concentration and intensification factors where necessary.
- Bending and contact stress verification for the driven gear **R6** for an endurance of 10^7 cycles and a reliability of 99% (based on AGMA D2001-D04). Assume the following conditions:
  - for the commercial enclosed units, a working condition of the driven machine with moderate shocks and uniform power source, with an operating temperature of 50 °C
  - a surface condition factor $Z_R = 1$.
  - for the load distribution factor K_H, the coefficient $C_e = 1$ and uncrowned teeth.
  - for the stress cycle factors Y_N and Z_N , the models $Y_N = 1{,}3558 \cdot N^{-0{,}0178}$ and $Z_N = 1{,}4488 \cdot N^{-0{,}023}$.
  - For the allowable fatigue bending stress, consider $\sigma_{FP} = 860$ MPa.
  - For the allowable fatigue contact stress σ_{HP}, consider a **grade 3** carburized and hardened steel.
  - For all other unknown coefficients, assume values deemed suitable for the design of a speed reducer intended for medium-duty applications.
  - Any additional data should be assumed in accordance with a consistent design approach.

Part 2

- Evaluate according to the SKF method the expected life of the bearings mounted on the output shaft **A4** (**A**, **B**) both in millions of cycles and operating hours. For the analysis assume:
  - Constant input power and rotational speed
  - Constant operating temperature, T=50°C.
  - Typical contamination conditions.
  - Reliability: **90%.**
  - Oil bath lubrication with **ISO VG 220 oil.**
- Evaluate for each bearing the static safety factor and the minimum load.

Perform the verifications following the list of calculations reported in the schemes given in the pages below, which represents a complete outline for the solution as well as a suggestion to write the technical report.

NOTE: Use ONLY the direction of rotation proposed in Figure 1 for the input shaft A1 and consequently determine the direction of rotation of all gears.

FINAL OUTCOME

- The results of the calculations for both Parts 1 and 2 must be provided and **discussed** in the form of a **technical report** (**text document**).
- The technical report must be brought to the oral exam in **printed and bound form**.
- The technical report **MUST be written with a writing program** (Word, LaTeX, Libre Office Writer, etc.), so **it MUST NOT BE a collection of snapshots** of hand-written sheets. **Hand-written documents will not be accepted**.
- **The technical report MUST include a final table like the submission table provided at the end of this document**, to summarize the major numerical results of the required calculations.
- **Remember that CLARITY and GRAPHIC aspects contribute to the final evaluation, so spend some time to write clear text and produce nice plots and tables!**

SCHEME FOR THE TECHNICAL REPORT: STATIC VERIFICATION

Develop the verification calculations for **PART 1** according to the points listed in the attached outline, which not only provide a complete guide for the work to be carried out, but also serve as a suggestion for dividing the technical report into sections and subsections.

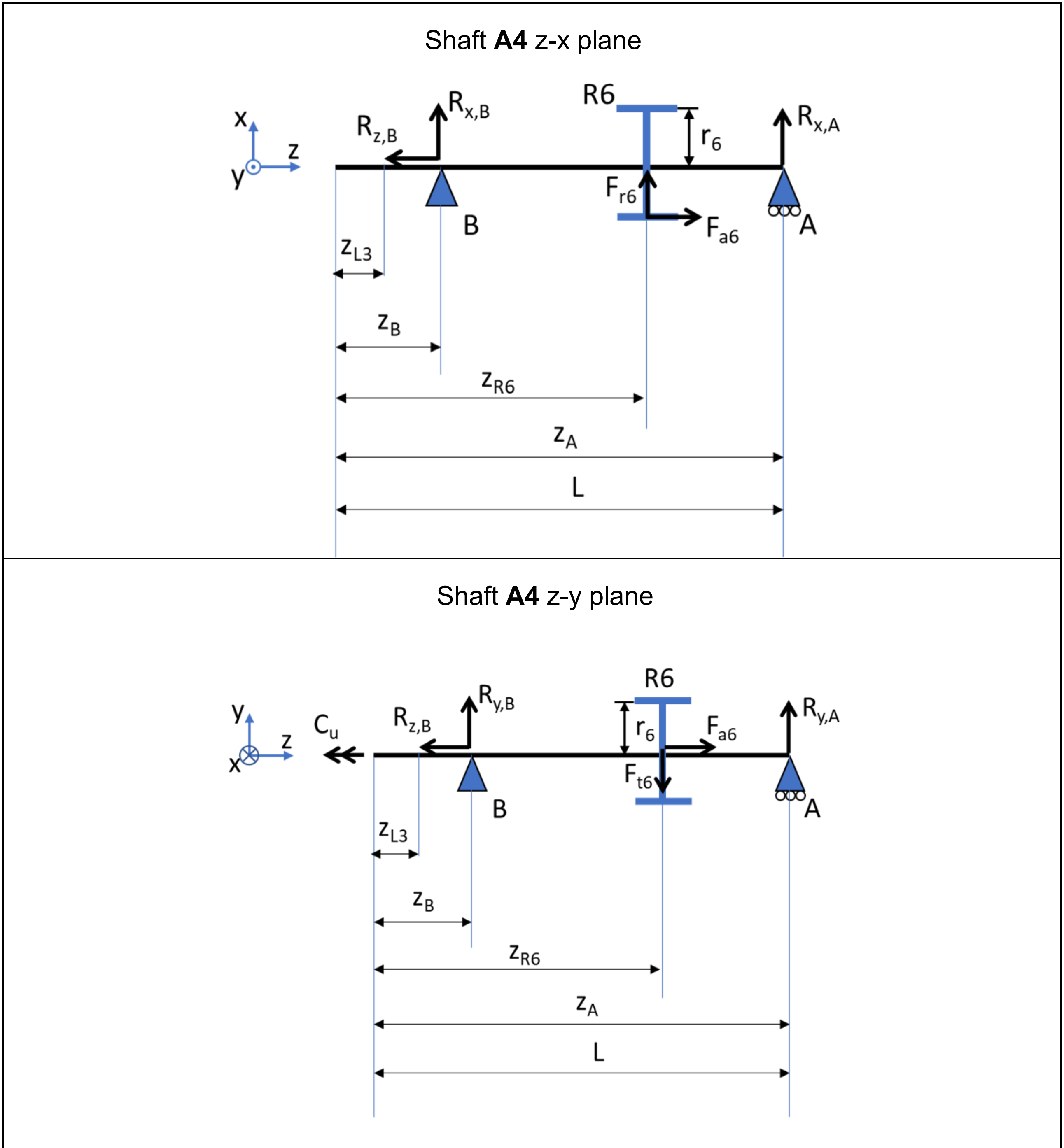
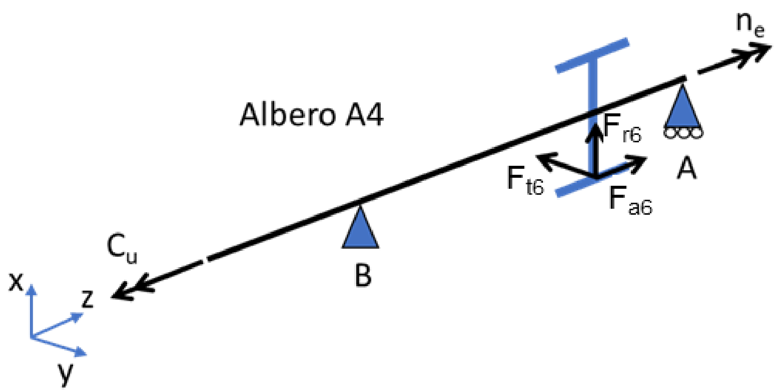


Figure 4. Diagrams for the solution of the static equilibrium of shaft **A4** in the xz and yz planes, consistent with the suggested calculation sequence.

Shaft A4: 3D scheme

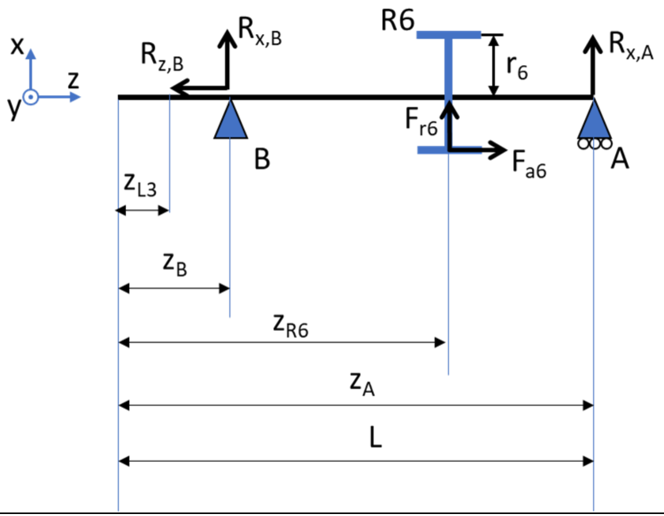


$$\begin{cases} C_u = \cdots \text{Nm} \\ F_{t6} = \cdots \text{N} \\ F_{r6} = \cdots \text{N} \\ F_{a6} = \cdots \text{N} \end{cases}$$

Notes:

- Before proceeding with the calculations, verify that the direction of the forces is consistent with the direction of rotation of the shaft indicated in the diagram and <u>in Figure 1</u>.
- Assume a gear efficiency equal to 1.

Shaft A4: Reaction forces on z-x plane



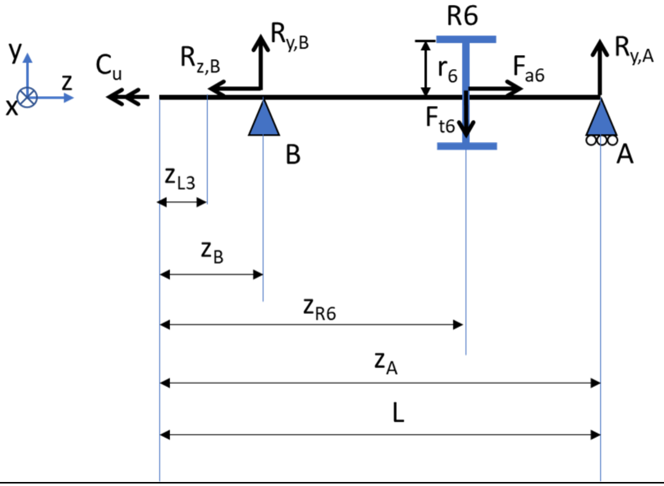
$$\begin{cases} z_{L3} = \cdots \text{mm} \\ z_B = \cdots \text{mm} \\ z_{R6} = \cdots \text{mm} \\ z_A = \cdots \text{mm} \\ L = \cdots \text{mm} \end{cases}$$

$$r_6 = \cdots \text{mm}$$

$$\begin{cases} R_{x,A} = \cdots \text{N} \\ R_{x,B} = \cdots \text{N} \\ R_{z,B} = \cdots \text{N} \end{cases}$$

Notes:

- Considering the actual positions of the points of application along the shaft axis of the bearing reaction forces, determine the required sizes.
- For each gear, consider the forces to be applied in the middle of the face width.

Shaft A4: Reaction forces on z-y plane	
	$\begin{cases} R_{y,A} = \cdots \text{N} \\ R_{y,B} = \cdots \text{N} \end{cases}$

Notes:

- Calculate the reaction forces on both planes z-y and z-x.
- Apply the reaction forces on the pressure centers of the bearings.
- Consider the forces due to the meshing gears **R5** and **R6** to be applied in the middle section of the gear face width.
- Consider the torque required by the mechanical end user to be acting in the middle section of the keyseat for key **L3**.

Shaft A4: Internal loads on z-x plane	
Calculate the internal loads on the z-x plane and **PLOT** the diagrams for axial force $N(z)$ and bending moment $M_y(z)$. Provide the trends of all internal loads with **CLEAR PLOTS**.	$\begin{cases} N(z) = \cdots \text{N} \\ M_y(z) = \cdots \text{Nm} \end{cases}$

Shaft A4: Internal loads on z-y plane	
Calculate the internal loads on the z-y plane and **PLOT** the diagrams for bending moment $M_x(z)$. Provide the trends of all internal loads with **CLEAR PLOTS**.	$M_x(z) = \cdots \text{Nm}$

Shaft A4: Total bending moment	
Calculate and PLOT the trend of the total bending moment $M_{b,tot}(z) = \sqrt{M_x^2(z) + M_y^2(z)}$ along the shaft axis. Provide the trends of all internal loads with **CLEAR PLOTS**.	$M_b(z) = \cdots \text{Nm}$

Shaft A4: Torsional moment	
Calculate and PLOT the trend of the torsional moment along the shaft axis. Provide the trends of all internal loads with **CLEAR PLOTS**.	$M_t(z) = \cdots \text{Nm}$

Shaft A4: Area and section modulus diagrams.	
<u>Calculate and PLOT</u> the trend of the cross-section $A(z)$, of the bending module $W_b(z)$ and of the torsion module $W_t(z)$. Provide the trends of the required quantities with **CLEAR PLOTS**.	$\begin{cases} A(z) = \cdots \text{mm}^2 \\ W_b(z) = \cdots \text{mm}^3 \\ W_t(z) = \cdots \text{mm}^3 \end{cases}$ ***<u>Note</u>***: ○ In the sections with the keyseats, <u>for static verifications</u>, consider a diameter equal to the nominal diameter minus the keyseat depth. For fatigue verifications, calculate the nominal stress using the full diameter, as if there were no keayseats.

Stress in the shaft A4: single components	
<u>Calculate and PLOT</u> the trend of the stresses due to the normal load $\sigma^N(z)$, to the bending moment $\sigma^{M_B}(z)$ and to the torsional moment $\tau^{M_t}(z)$. Provide the trends of all stress components with **CLEAR PLOTS**.	$\begin{cases} \sigma^N(z) = \cdots \text{MPa} \\ \sigma^{M_B}(z) = \cdots \text{MPa} \\ \tau^{M_t}(z) = \cdots \text{MPa} \end{cases}$
Stress on the shaft A4: equivalent stress	
<u>Calculate and PLOT</u> the trend of the resulting normal stress $\sigma^{tot}(z)$ and equivalent stress $\sigma_{id}(z)$. For the evaluation of the equivalent stress, consider the Von Mises criterion and suppose that the material fails for **<u>full yielding</u>**. Provide the trends of these stresses with **CLEAR PLOTS**.	$\begin{cases} \sigma^{tot}(z) = \cdots \text{MPa} \\ \sigma_{id}(z) = \cdots \text{MPa} \end{cases}$
Static safety factor on the shaft A4	
Identify the most critical sections and evaluate the static safety factor against **<u>full yielding</u>**. Identify the minimum safety factor of the shaft $SF_{s,min}$.	$SF_{s,min} = \cdots$
<u>Notes</u>: ○ In the sections with the keyseats, <u>for static verifications</u>, consider a diameter equal to the nominal diameter minus the keyseat depth. For <u>fatigue verifications</u>, calculate the nominal stress using the full diameter, as if there were no keayseats.	

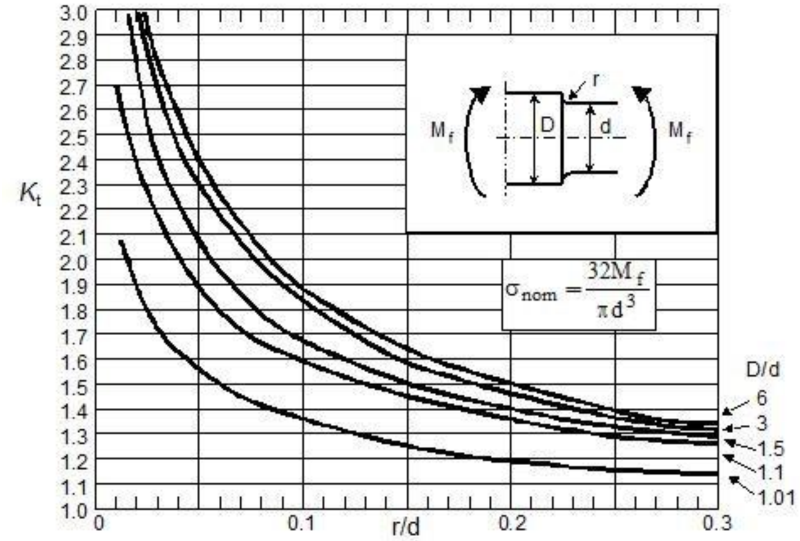
SCHEME FOR THE TECHNICAL REPORT: FATIGUE VERIFICATION

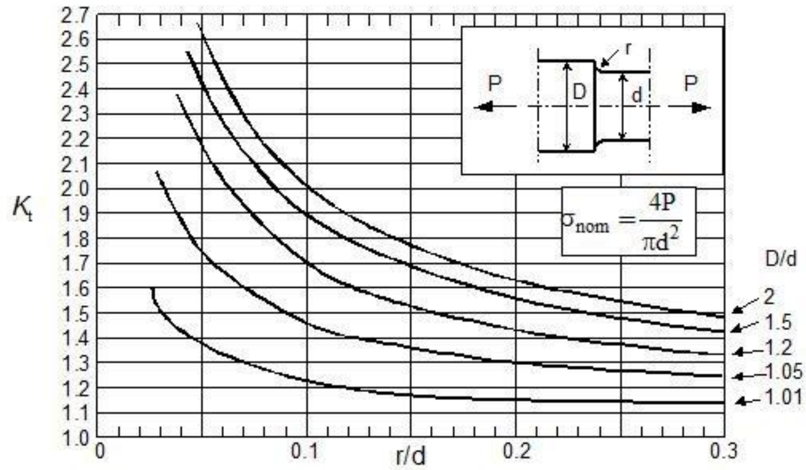
Evaluation of the fatigue stress on Shaft A4	
Evaluate, if present, the mean and alternate stress components due to the normal load, σ_m^N e σ_a^N, mean and alternate stress components due to the bending moment, $\sigma_m^{M_B}$ e $\sigma_a^{M_B}$, and the mean and alternate stress components due to the torsion moment, $\tau_m^{M_t}$ e $\tau_a^{M_t}$ in the sections **S1**, **S2**, **S3**.	For each section S1, S2, S3: $\begin{cases} \sigma_m^N = \cdots \text{MPa} & \sigma_a^N = \cdots \text{MPa} \\ \sigma_m^{M_B} = \cdots \text{MPa} & \sigma_a^{M_B} = \cdots \text{MPa} \\ \tau_m^{M_t} = \cdots \text{MPa} & \tau_a^{M_t} = \cdots \text{MPa} \end{cases}$
Stress concentration factors	
Starting from the notch sensitivity q (evaluated as a function of the notch radius) and the geometric stress raiser notch factor K_t, evaluate the fatigue stress intensification factor K_f for sections **S1**, **S2**, **S3**.	$\begin{cases} q = \cdots \\ K_{t,N} = \cdots \\ K_{f,N} = \cdots \\ K_{t,B} = \cdots \\ K_{f,B} = \cdots \\ K_{t,T} = \cdots \\ K_{f,T} = \cdots \end{cases}$
Fatigue limit correction factors	
Considering the working condition and the shaft geometry, see Figure 3, evaluate from the corresponding diagrams the scale effect factor C_S, and the surface finish effect factor C_F for sections for sections **S1**, **S2**, **S3**..	$\begin{cases} C_S = \cdots \\ C_F = \cdots \end{cases}$
Fatigue limit correction for the component and Haigh diagram	
Considering the working condition and the shaft geometry, evaluate the fatigue limit of the component σ_{D-1}^c and **plot for each section the Haigh diagram** for infinite life with all the relevant information.	$\sigma_{D-1}^c = \cdots \text{MPa}$
Fatigue safety factor	
Considering the fatigue working condition and the shaft geometry, calculate the coordinates of the working point P on the Haigh diagram and evaluate the fatigue safety factor for infinite life, SF_f. Choose the most appropriate definition of the safety factor according to the specific characteristics of the application. **For each section, PLOT the Haigh diagram**.	$\begin{cases} \sigma_{a,eq} = \cdots \text{MPa} \\ \sigma_{m,eq} = \cdots \text{MPa} \end{cases}$ $SF_f = \cdots$

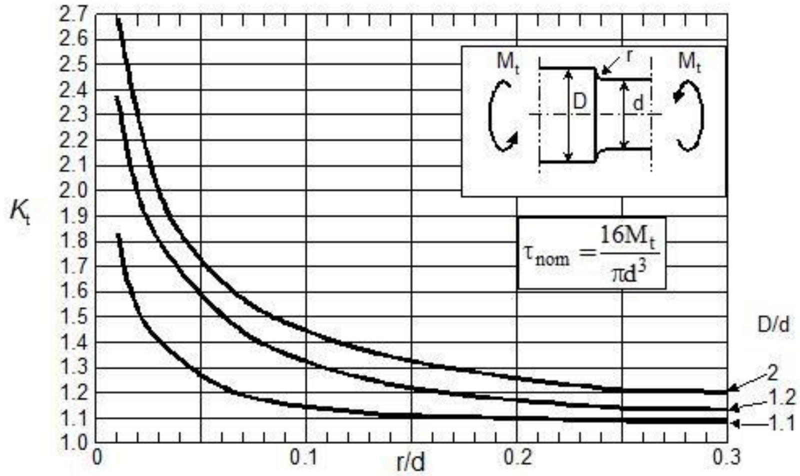
Reminder:

- **Repeat the same procedure for sections S1, S2, S3**.
- Use the diagrams in the following pages for the evaluation of the stress concentration coefficients.
- Use the **Shigley Method** to calculate the alternate and mean equivalent stress components of the working point.
- For **section S3**, where the keyseat for connection to gear **R6** is milled, calculate the nominal stresses using the full diameter, as if there was no keayseat.
- **For each section, it is requested to draw the Haigh diagram and clearly identify the position of the working point!**

Geometrical stress concentration factors Kt



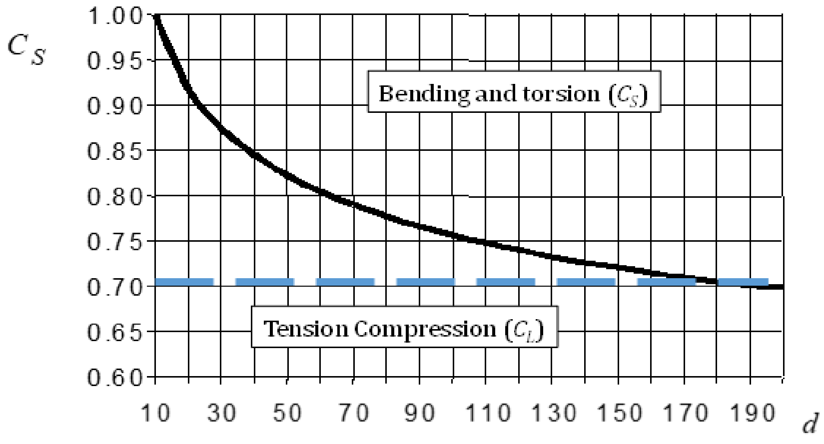




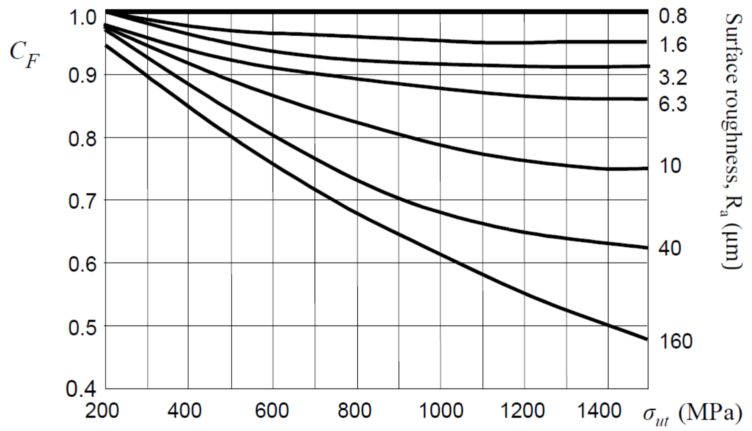
	Bending	Torsion
K_f for keyseat	1.6	2.0

<u>NOTE</u>: For keyseats, consider as the nominal diameter for fatigue stress calculation the full diameter, as if no keyseat was milled on the shaft.

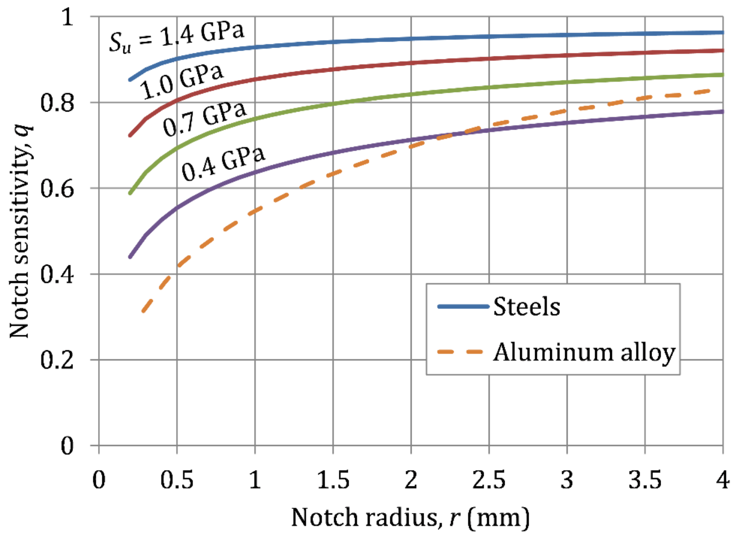
Size Effect Cs



Surface finish effect



Notch sensitivity q



SCHEME FOR THE TECHNICAL REPORT: GEAR TOOTH VERIFICATION

Maximum tooth gear bending stress equation for fatigue

$$\sigma_{\max,bending} = F_t K_O K_B K_v K_H K_s \frac{1}{b \cdot m_t Y_J}$$

Notes:
Express the face width b and the transverse modulus m_t in mm

Face width b

$$b = \min[b_{\mathbf{G}}, b_{\mathbf{P}}]$$

Where G is the gear and P is the pinion of the mating gears

Overload factor K_O

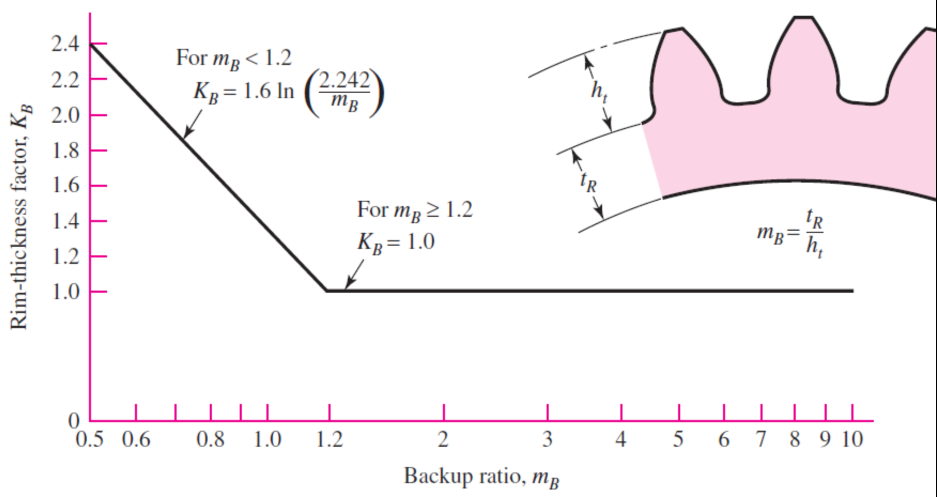
Table of Overload Factors, K_o

	Driven Machine		
Power source	Uniform	Moderate shock	Heavy shock
Uniform	1.00	1.25	1.75
Light shock	1.25	1.50	2.00
Medium shock	1.50	1.75	2.25

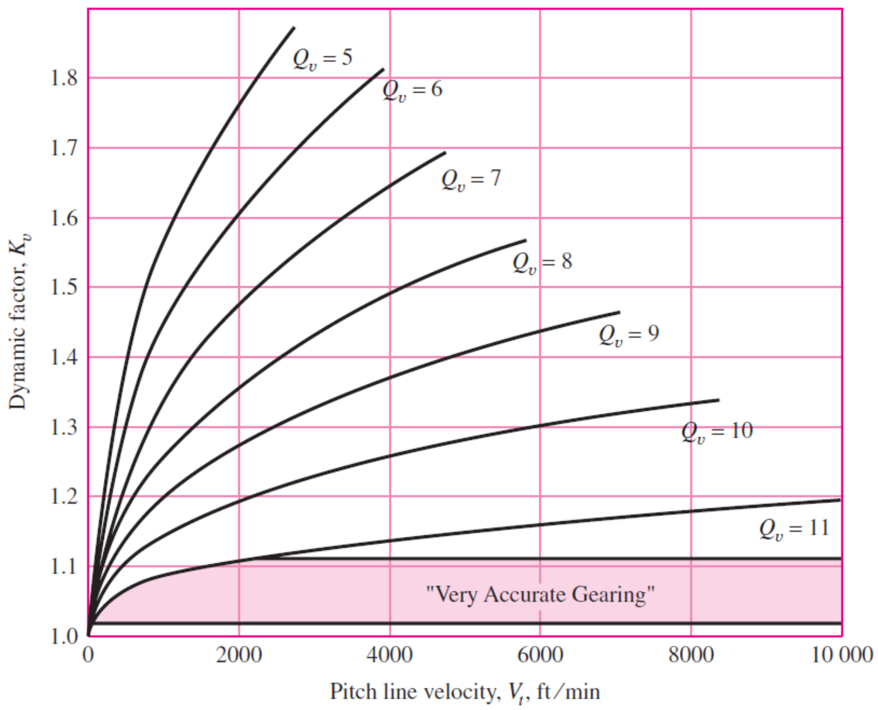
$K_O = \cdots$

NOTE:
Assuming operating conditions different from those indicated in the report, evaluate the influence of the K_0 factor on the fatigue safety factor in bending and pitting.

Rim-thickness factor K_B



$K_B = \cdots$

<table>
<tr><th colspan="2">Dynamic factor K_v</th></tr>
<tr><td>
</td>
<td>$K_v = \dots$

<u>NOTE</u>:
Values in the diagram can be obtained from the following equation, where the peripheral speed should be expressed in ft/min and $Q_v = 6$ is is the transmission accuracy-level number:

$$\begin{cases} V = \cdots \text{ ft/min} \\ B = 0{,}25(12 - Q_v)^{2/3} \\ A = 50 + 56(1 - B) \\ K_v = \left(\dfrac{A + \sqrt{V}}{A}\right)^B \end{cases}$$</td></tr>
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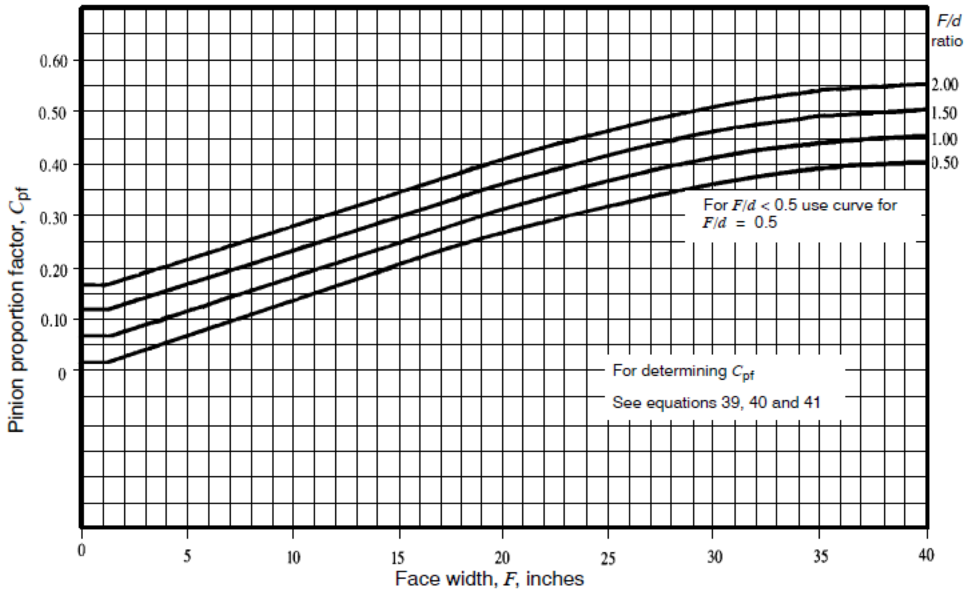
Load distribution factor K_H
$K_H = 1 + C_{mc}(C_{pf} C_{pm} + C_{ma} C_e)$, where $C_e = 1$.

<u>ATTENTION</u>:
Diagrams are provided here with a tooth width b (F in the plots) expressed in inches

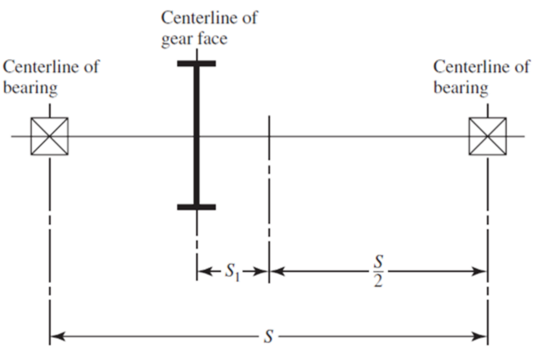
$$C_{mc} = \begin{cases} 1 & \text{for uncrowned teeth} \\ 0{,}8 & \text{for crowned teeth} \end{cases}$$



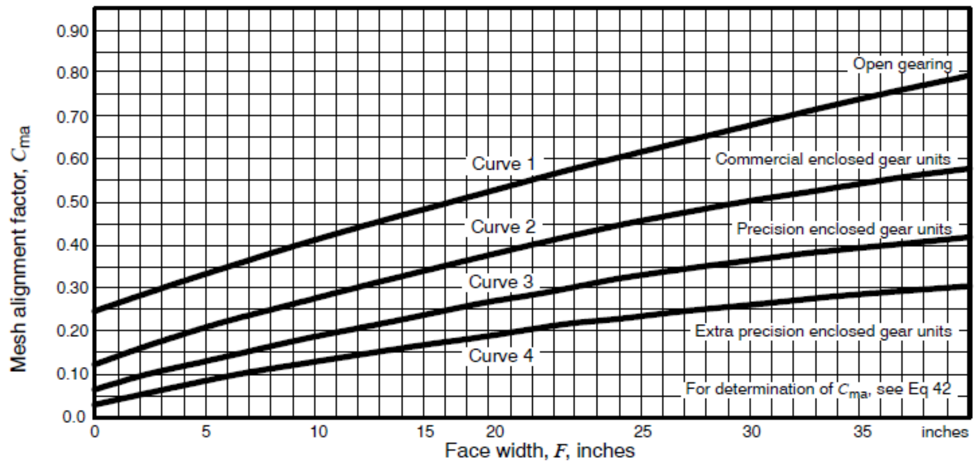
$$C_{pf} = \begin{cases} \dfrac{F}{10d} - 0.025 & F \le 1 \text{ in} \\ \dfrac{F}{10d} - 0.0375 + 0.0125F & 1 < F \le 17 \text{ in} \\ \dfrac{F}{10d} - 0.1109 + 0.0207F - 0.000\,228F^2 & 17 < F \le 40 \text{ in} \end{cases}$$

where d is the pitch circle of the pinion and F is the face width. Both quantities should be expressed in inches in the formulas on the left.

Note that for values of $F/(10d) < 0.05$, $F/(10d) = 0.05$ is used.



$$C_{pm} = \begin{cases} 1 & S_1/S < 0{,}175 \\ 1{,}1 & S_1/S \ge 0{,}175 \text{ or cantilever shaft} \end{cases}$$



$$C_{ma} = A + B \cdot F + C \cdot F^2$$

Condition	A	B	C
Open gearing	0.247	0.0167	$-0.765(10^{-4})$
Commercial, enclosed units	0.127	0.0158	$-0.930(10^{-4})$
Precision, enclosed units	0.0675	0.0128	$-0.926(10^{-4})$
Extraprecision enclosed gear units	0.00360	0.0102	$-0.822(10^{-4})$

Where F is the face width of the gear in inches.

Size factor K_s

Number of Teeth	Y	Number of Teeth	Y
12	0.245	28	0.353
13	0.261	30	0.359
14	0.277	34	0.371
15	0.290	38	0.384
16	0.296	43	0.397
17	0.303	50	0.409
18	0.309	60	0.422
19	0.314	75	0.435
20	0.322	100	0.447
21	0.328	150	0.460
22	0.331	300	0.472
24	0.337	400	0.480
26	0.346	Rack	0.485

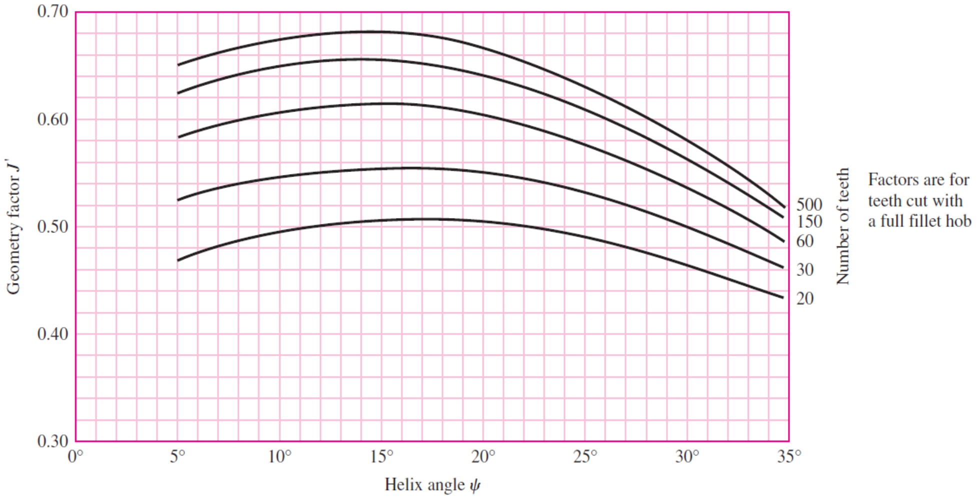
$$K_s = 0{,}843\left(b \cdot m_t \sqrt{Y}\right)^{0{,}0535} = \cdots$$

NOTES:

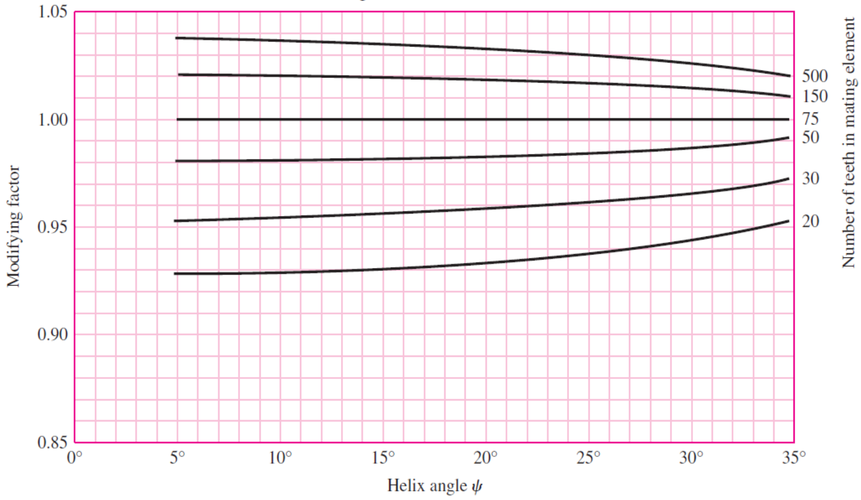
- Express b and m_t in mm in the equation above.
- For helical gears enter the table with the virtual number of teeth:

$$z' = \frac{z}{cos^3\psi}$$

Bending strength geometry factor $Y_J = J' \cdot J''$



$J' = \cdots$



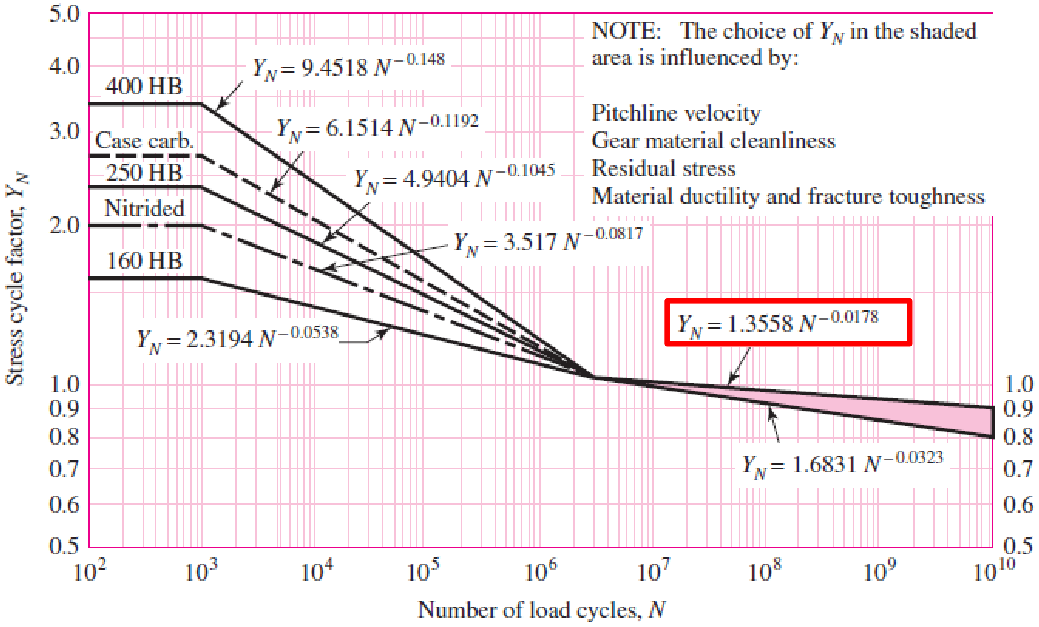
$J'' = \cdots$

Bending safety factor

$$S_F = \frac{\sigma_{FP}}{\sigma_{\max,bending}} \frac{Y_N}{Y_\theta Y_Z}$$

Allowable fatigue bending stress σ_{FP} = 860 MPa for N=10^7 cycles

Stress cycle life factor Y_N



$Y_N = \cdots$

Temperature factor Y_θ

$Y_\theta = 1,$ for temperature up to 120 °C

Reliability factor Y_Z

Reliability	K_R (Y_Z)
0.9999	1.50
0.999	1.25
0.99	1.00
0.90	0.85
0.50	0.70

$Y_Z = \cdots$

Maximum gear contact (pitting resistance) stress equation in fatigue

$$\sigma_{\max,pitting} = Z_E\sqrt{F_t K_O K_v K_s \cdot \frac{K_H}{b \cdot d_P} \cdot \frac{Z_R}{Z_I}}$$

Elastic coefficient Z_E

Table 14–8

Elastic Coefficient C_p (Z_E), $\sqrt{\text{psi}}$ ($\sqrt{\text{MPa}}$)   *Source:* AGMA 218.01

Pinion Material	Pinion Modulus of Elasticity E_p psi (MPa)*	Gear Material and Modulus of Elasticity E_G, lbf/in² (MPa)*: Steel 30×10^6 (2×10^5)	Malleable Iron 25×10^6 (1.7×10^5)	Nodular Iron 24×10^6 (1.7×10^5)	Cast Iron 22×10^6 (1.5×10^5)	Aluminum Bronze 17.5×10^6 (1.2×10^5)	Tin Bronze 16×10^6 (1.1×10^5)
Steel	30×10^6 (2×10^5)	2300 (191)	2180 (181)	2160 (179)	2100 (174)	1950 (162)	1900 (158)
Malleable iron	25×10^6 (1.7×10^5)	2180 (181)	2090 (174)	2070 (172)	2020 (168)	1900 (158)	1850 (154)
Nodular iron	24×10^6 (1.7×10^5)	2160 (179)	2070 (172)	2050 (170)	2000 (166)	1880 (156)	1830 (152)
Cast iron	22×10^6 (1.5×10^5)	2100 (174)	2020 (168)	2000 (166)	1960 (163)	1850 (154)	1800 (149)
Aluminum bronze	17.5×10^6 (1.2×10^5)	1950 (162)	1900 (158)	1880 (156)	1850 (154)	1750 (145)	1700 (141)
Tin bronze	16×10^6 (1.1×10^5)	1900 (158)	1850 (154)	1830 (152)	1800 (149)	1700 (141)	1650 (137)

Poisson's ratio = 0.30.

$$Z_E = \ldots \sqrt{\text{MPa}}$$

Surface strength geometry Z_I
$\begin{cases} r_{b_\mathbf{P}} = r_\mathbf{P}\cos[\phi_t] \\ r_{b_\mathbf{G}} = r_\mathbf{G}\cos[\phi_t] \end{cases}$’ where $r_\mathbf{P}$ e $r_\mathbf{G}$ are the pitch radii of the pinion **P** (the smaller gear) and the mating gear **G** respectively while ${r_b}_\mathbf{P}$ e ${r_b}_\mathbf{G}$ are the base radii of **P** and **G**. ϕ_t is the transverse pressure angle. $$\begin{cases} Z_A = \min\left[\sqrt{(r_\mathbf{P}+a)^2 - r_{b_\mathbf{P}}^2}, (r_\mathbf{P}+r_\mathbf{G})\mathrm{sen}[\phi_t]\right] \\ Z_B = \min\left[\sqrt{(r_\mathbf{G}+a)^2 - r_{b_\mathbf{G}}^2}, (r_\mathbf{P}+r_\mathbf{G})\mathrm{sen}[\phi_t]\right] \\ Z = Z_A + Z_B - (r_\mathbf{P}+r_\mathbf{G})\mathrm{sen}[\phi_t] \end{cases} \Rightarrow m_N = \frac{p_n\cos[\phi_n]}{0{,}95\cdot Z},$$ Where: • m_N is the load sharing ratio, • $p_n = \pi \cdot m_n$ is the normal pitch (with m_n the normal module), • $a = m_n$ is the addendum, • ϕ_n is the normal pressure angle, • ϕ_t is the transverse pressure angle with $\tan[\phi_t] = \frac{\tan[\phi_n]}{\cos[\psi]}$. $$Z_I = \begin{cases} \dfrac{\cos[\phi_t]\mathrm{sen}[\phi_t]}{2m_N}\cdot\dfrac{m_G}{m_G+1} & \text{external meshing} \\ \dfrac{\cos[\phi_t]\mathrm{sen}[\phi_t]}{2m_N}\cdot\dfrac{m_G}{m_G-1} & \text{internal meshing} \end{cases}$$ where $m_G = n_\mathbf{P}/n_\mathbf{G} = d_\mathbf{G}/d_\mathbf{P}$ is the gear ratio.

Wear safety factor (Hertzian contact)

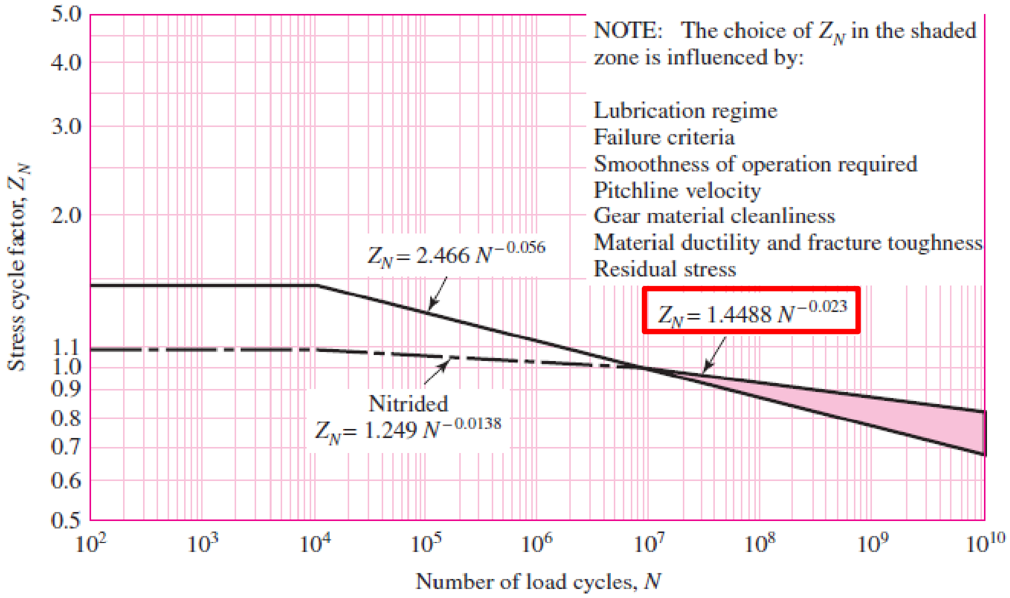
$$S_H = \frac{\sigma_{HP}}{\sigma_{\max,pitting}} \frac{Z_N Z_W}{Y_\theta Y_Z}$$

Values of σ_{HP} (S_C in psi) at 10^7 cycles and reliability 0.99

Material Designation	Heat Treatment	Minimum Surface Hardness[1]	Allowable Contact Stress Number,[2] S_c, psi Grade 1	Grade 2	Grade 3
Steel[3]	Through hardened[4]	See Fig. 14–5	See Fig. 14–5	See Fig. 14–5	—
	Flame[5] or induction hardened[5]	50 HRC	170 000	190 000	—
		54 HRC	175 000	195 000	—
	Carburized and hardened[5]	See Table 9*	180 000	225 000	275 000
	Nitrided[5] (through hardened steels)	83.5 HR15N	150 000	163 000	175 000
		84.5 HR15N	155 000	168 000	180 000
2.5% chrome (no aluminum)	Nitrided[5]	87.5 HR15N	155 000	172 000	189 000
Nitralloy 135M	Nitrided[5]	90.0 HR15N	170 000	183 000	195 000
Nitralloy N	Nitrided[5]	90.0 HR15N	172 000	188 000	205 000
2.5% chrome (no aluminum)	Nitrided[5]	90.0 HR15N	176 000	196 000	216 000

$$S_c = \cdots \text{ psi} \rightarrow \sigma_{HP} = \cdots \text{ MPa}$$

Stress cycle life factor Z_N



$$Z_N = \cdots$$

Temperature coefficient Y_θ

$Y_\theta = 1$ for temperature lower than 120 °C

Reliability factor Y_Z

Reliability	K_R (Y_Z)
0.9999	1.50
0.999	1.25
0.99	1.00
0.90	0.85
0.50	0.70

Hardness-ratio factor Z_W (C_H)

$$A' = \begin{cases} 0 & HB_{\mathbf{P}}/HB_{\mathbf{G}} < 1{,}2 \\ 8{,}98 \cdot 10^{-3}(HB_{\mathbf{P}}/HB_{\mathbf{G}}) - 8{,}29 \cdot 10^{-3} & 1{,}2 \leq HB_{\mathbf{P}}/HB_{\mathbf{G}} \leq 1{,}7 \\ 0{,}00698 & HB_{\mathbf{P}}/HB_{\mathbf{G}} > 1{,}7 \end{cases}$$

where $HB_{\mathbf{P}}$ e $HB_{\mathbf{G}}$ are the Brinell hardness of **P** and **G**.

$$Z_W = 1 + A'(m_G - 1)$$

where $m_G = n_{\mathbf{P}}/n_{\mathbf{G}} = d_{\mathbf{G}}/d_{\mathbf{P}}$ is the gear ratio.

SCHEME FOR THE TECHNICAL REPORT: BEARINGS CALCULATION

Static analysis
• Verify each bearing under investigation for the minimum load. • Provide the Safety Factor (SF) for the static bearing load.

Fatigue analysis: Bearing life estimation
Bearing life analysis (millions of cycles) $L_{nm} = a_1 a_{SKF} \left(\frac{C}{P}\right)^p$ **Bearing life in operating hours** $L_{nmh} = \frac{10^6}{60 \cdot n} L_{nm}$
Life adjustment factor for reliability
Search the uploaded SKF catalogue to obtain the coefficient a_1 for the reliability level of the application (90%).
Evaluation of the contamination
Values for η_c are given for several levels of contamination in the SKF catalogue.
Evaluation of the viscosity ratio
$\kappa = \frac{\nu}{\nu_1}$, with: • ν lubricant viscosity at the given operating temperature • ν_1 minimum required lubricant viscosity for the given working conditions. **Rember that the gearbox is lubricated with oil bath lubrication using the ISO VG 220 oil.**
Evaluation of the minimum required lubricant viscosity
Values for ν_1 are given as a function of d_m and of the shaft rotational speed n. Search the catalogue for the relevant charts.
Evaluation of the lubricant viscosity at a given operating temperature
Values for ν are given as a function of ν_1 and of the operating temperature T. Search the catalogue for the relevant charts.
Evaluation of the coefficient a_{SKF}
Values for a_{SKF} are given as function of $\eta_c \frac{P_u}{P}$ and κ for radial roller bearings where η_c is the contamination factor, P_u is the ultimate fatigue load (catalogue) and k is the viscosity ratio. Search the catalogue for the relevant charts.

Reminder:

- Use the **uploaded SKF catalogue** to find relevant tables and diagrams and for the equivalent load definitions.
- **Repeat the verification for the bearings mounted on the output shaft A4 (A and B).**

Annex 1: Group input data for gearbox analysis

Groups 1-30

Group No.	Input Power P_{in} (kW)	Input speed n_{in} (rpm)
1	34	750
2	34	1100
3	34	1200
4	30	800
5	34	950
6	31	900
7	33	750
8	32	1000
9	34	850
10	31	1250
11	33	1250
12	33	1100
13	33	900
14	31	1150
15	32	750
16	32	1050
17	31	1100
18	30	950
19	34	1000
20	32	1250
21	30	900
22	35	1250
23	34	900
24	33	1150
25	32	1200
26	30	1200
27	35	950
28	31	1050
29	30	1250
30	35	1050

Groups 31-60:

Group No.	Input Power P_{in} (kW)	Input speed n_{in} (rpm)
31	35	1200
32	35	900
33	30	750
34	31	750
35	33	1000
36	32	900
37	30	850
38	35	800
39	33	1200
40	31	850
41	31	950
42	35	1000
43	34	1050
44	31	1200
45	32	800
46	31	1000
47	32	1150
48	33	1050
49	32	1100
50	30	1150
51	30	1050
52	31	800
53	32	850
54	33	950
55	32	950
56	34	800
57	33	850
58	35	750
59	35	1100
60	35	1150

SUBMISSION TABLE FOR KEY CALCULATION RESULTS

In your final report, it is MANDATORY to include tables like the submission tables below filled with the required results.

PART 1

	VALUE	UNIT OF MEASUREMENT
SHAFT ANALYSIS		
$R_{x,A}$		N
$R_{y,A}$		N
$R_{x,B}$		N
$R_{y,B}$		N
$R_{z,A}$		N
$R_{z,B}$		N
Mx (S1)		Nm
Mx (S2)		Nm
Mx (S3)		Nm
My (S1)		Nm
My (S2)		Nm
My (S3)		Nm
N (S1)		N
N (S2)		N
N (S3)		N
$Mb_{tot,max}$		Nm
Mt_{max}		Nm
SF (static)		-
$\sigma_{a,b\ nom}$ (z=m.c.s.)		MPa
$\sigma_{m,b\ nom}$ (z=m.c.s.)		MPa
$\sigma_{a,N\ nom}$ (z=m.c.s.)		MPa
$\sigma_{m,N\ nom}$ (z=m.c.s.)		MPa
$\tau_{a,t\ nom}$ (z=m.c.s.)		MPa
$\tau_{m,t\ nom}$ (z=m.c.s.)		MPa
Kf,b		
$\sigma_{a,b\ eff}$ (z=m.c.s.)		MPa
$\sigma_{m,b\ eff}$ (z=m.c.s.)		MPa
Kf,N		
$\sigma_{a,N\ eff}$ (z=m.c.s.)		MPa
$\sigma_{m,N\ eff}$ (z=m.c.s.)		MPa
Kf,T		
$\tau_{a,t\ eff}$ (z=m.c.s.)		MPa
$\tau_{m,t\ eff}$ (z=m.c.s.)		MPa
$\sigma_{a,eqvP}$ (z=m.c.s.)		MPa
$\sigma_{m,eqvP}$ (z=m.c.s.)		MPa
SF_f (fatigue)		-

- M.c.s. Most critical section

	VALUE	UNIT OF MEASUREMENT
GEAR R6 - BENDING		
$\left(\frac{F_t}{b \cdot m_t \cdot Y_J}\right)_{\mathrm{R6}}$		MPa
$\sigma_{\max,bending}^{R6}$		MPa
S_F^{R6}		-
GEAR R6- PITTING		
$\left(Z_E\sqrt{\frac{F_t}{b \cdot d_{\boldsymbol{P}}} \cdot \frac{1}{Z_I}}\right)_{R6}$		MPa
$\sigma_{\max,Pitting}^{R1}$		MPa
S_H^{R6}		-

PART 2

	VALUE	UNIT OF MEASUREMENT
BEARING ANALYSIS: LOAD CALCULATION		
$\mathbf{F_{r,A}}$		kN
$\mathbf{F_{a,A}}$		kN
$\mathbf{F_{r,B}}$		kN
$\mathbf{F_{a,B}}$		kN
$\mathbf{P_A}$		kN
$\mathbf{P_B}$		kN
$\mathbf{L_{10,A}}$		Mln cycles
$\mathbf{L_{10,B}}$		Mln cycles
BEARING ANALYSIS: CORRECTED RATING LIFE		
$\boldsymbol{\kappa_A}$		-
$\boldsymbol{\kappa_B}$		-
$\mathbf{a_{skf,A}}$		-
$\mathbf{a_{skf,B}}$		-
$L_{10mh,A}$		Hours
$L_{10mh,B}$		Hours