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 15th** you can register your group in the online Spreadsheet available at the following link:

https://docs.google.com/spreadsheets/d/1ezLfKgLhlkYkAARjF0SHn-Wl-0ofXOkyGL7aS1JHQXY/edit?usp=sharing

For those of you who will not be able (for any reason) to find the other two people, you will be assigned automatically to other groups that will have not been able to find all the three elements.

To access the final examination the Homework Project must be completed **in all its parts** and submitted no later than **January 10th, 2025**.Students who already attended the course in previous years and got a mark for their project do not need to do the homework project again, but they can if they want to achieve a higher score. However, in case the score of the new Project is lower than the previous one, **it cannot be declined**.

Each group must submit the technical report in .pdf format on "Portale della didattica/ Fundamentals of machine design/Elaborati(Homework)" section. Each group must submit **only one file** with name ***Group_Nr_XX_Surname1_Surname_2_Surname_3.pdf***. No other file other than the technical report will be taken into account (Spreadsheet, matfile, scripts, etc.)

Shortly after the submission deadline you will be notified about the acceptance of the uploaded material. **If the uploaded files are lacking some relevant parts, you will not be able to access the examination until a revised version is uploaded**. The maximum possible score for the new revised version will be 0.

For all accepted works the final mark after the evaluation of the technical report will be in the range C = +3/-3.
Works with score C<0 can be delivered again one week before the next useful written examination date, but the maximum score for the second revision will be C=0.

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.

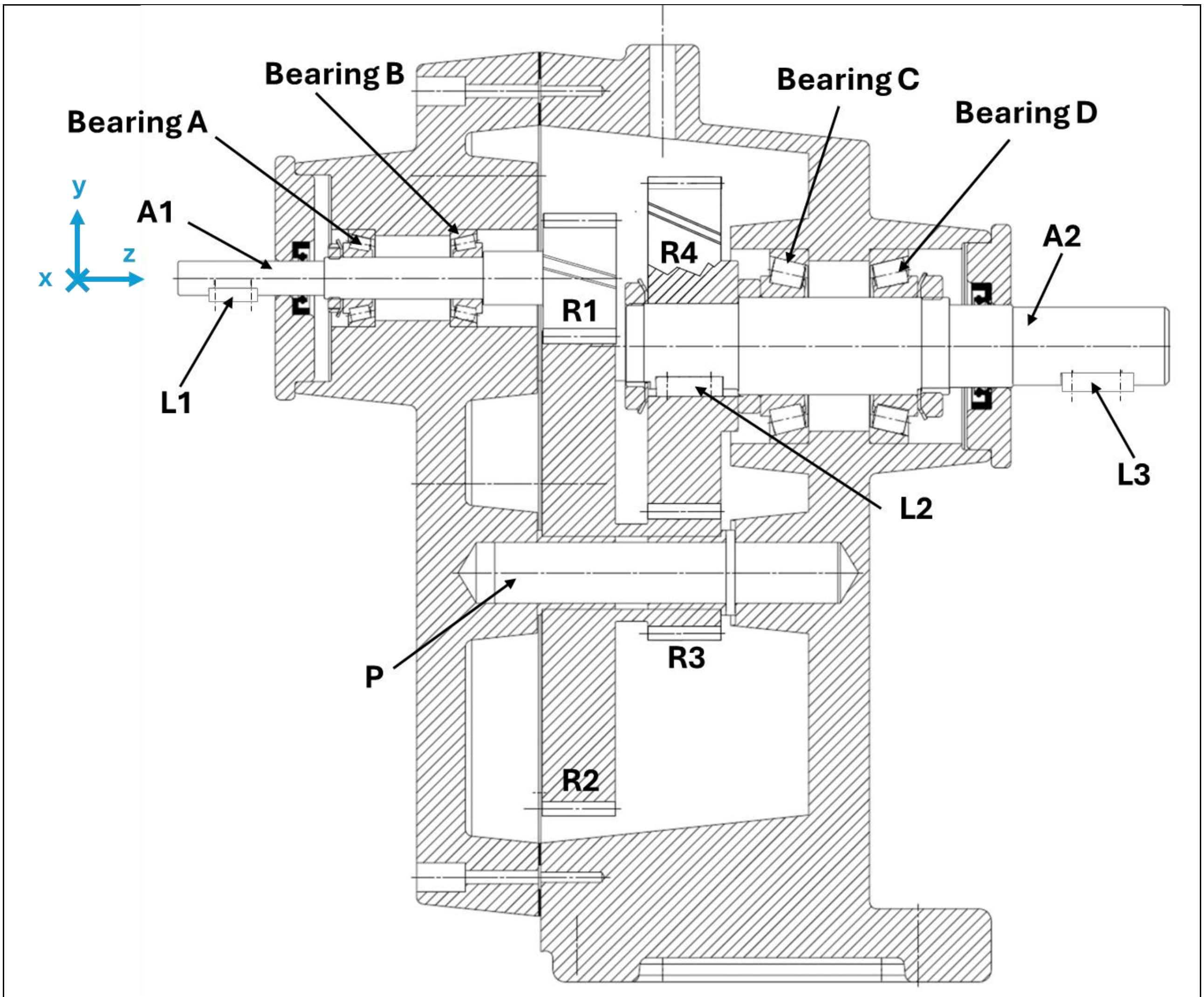


Figure 1. Gearbox assembly drawing (section)

LEGEND:

A1: gearbox input shaft; it receives power from the motor via key **L1**.
R1: helical pinion of the primary reduction stage, it transfers power from shaft **A1** to the driven gear **R2**.
P: intermediate spindle, supporting the single unit including gears **R2** and **R3**.
R2: driven helical gear receiving power from pinion **R1** and transmitting it to pinion **R3**.
R3: helical pinion transmitting power to gear **R4**.
R4: driven helical gear, it transfers power to shaft **A2** via key **L2**.
A2: output shaft, it receives power from gear **R4** and transfers it to the end user via the key **L3**.
L1: key that connects the input shaft **A1** with the **motor**.
L2: key that connects the driven gear **R4** with shaft **A2**.
L3: key that connects the output shaft **A2** with the **mechanical end user.**
Bearings A, B: tapered roller bearings supporting the input shaft **A1.**
Bearings C, D: tapered roller bearings supporting the output shaft **A2.**

Data for static and fatigue analysis on shaft A2 and gears R1 and R3

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 and Figure 2 for the direction of rotation). Power is transmitted to the end user, connected to the output shaft **A2**, through the meshing gears **R1** and **R2** and the meshing gears **R3** and **R4**. The helical gears have the following characteristics:

	R1	R2	R3	R4
normal module, m_n (mm)	2.5	2.5	2.5	2.5
number of teeth, z	18	73	19	51
helix angle, ψ	20°	20°	20°	20°
normal pressure angle, ϕ_n	20°	20°	20°	20°
transmission accuracy level, Q_v	7	7	7	7
tooth face width (mm)	30	30	30	30
Ultimate strength (MPa)	1060	1950	1950	1060
Yield strength (MPa)	950	1400	1400	950
Surface Hardness (HB)	335	560	560	335
Bending fatigue strength σ_{FP} (MPa)	320	450	450	320
Contact fatigue strength σ_{HP} (MPa)	860	1360	1360	860

The output shaft **A2** (Figure 4), which is to be verified both for static loading and fatigue, is made with a 34NiCrMo6 UNI EN10083 hardened and tempered steel (σ_u=1050 MPa, σ_y=950 MPa, σ_{D-1}=520 MPa), and it is supported with two tapered roller bearings (**C** and **D**) in back-to-back configuration.

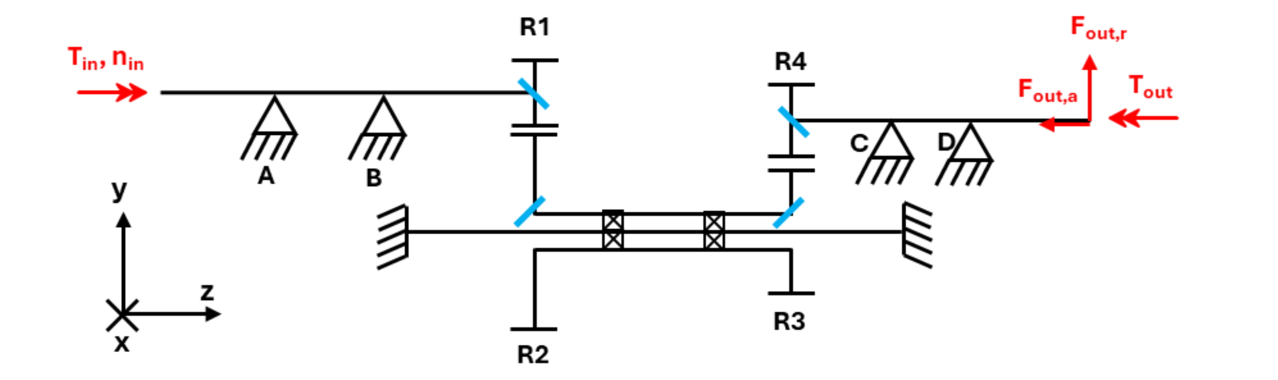


Figure 2. Gearbox functional scheme

Consider:

- The mechanical end user applies a radial force acting along the y-axis and an axial force along the z-axis with directions shown in Figure 2.
- The forces $\mathbf{F_{OUT,r}}$ and $\mathbf{F_{OUT,a}}$ are considered to be applied in the midplane section of the key **L3**.
- The values of the forces must be calculated as a function of the torque on shaft **A2**, as:

$$F_{OUT,r} = \frac{T_{A2}}{0.1}\ N \qquad\qquad F_{OUT,a} = 0.25 \cdot F_{OUT,r}\ N$$

With T_{A2} expressed in Nm
Remember to consider the orientation of the forces with respect to the global reference frame x-y-z shown in Figure 2.

Bearings:

A: SKF 30203 (Explorer).
B: SKF 30203 (Explorer).
C: SKF 30208 (Explorer).
D: SKF 30208 (Explorer).

Pinion **R1** is machined from the same piece of shaft **A1**; helical gears **R2** and **R3** are made in a single piece and are supported by spindle **P** via journal bearings; driven gear **R4** is keyed on shaft **A2** via key **L2**.
The support bearings of shaft **A2** (bearings **A** and **B**) are both tapered roller bearings with back-to-back mounting configuration.
The same configuration is used on shaft **A2**, which is supported by bearings **C** and **D**.

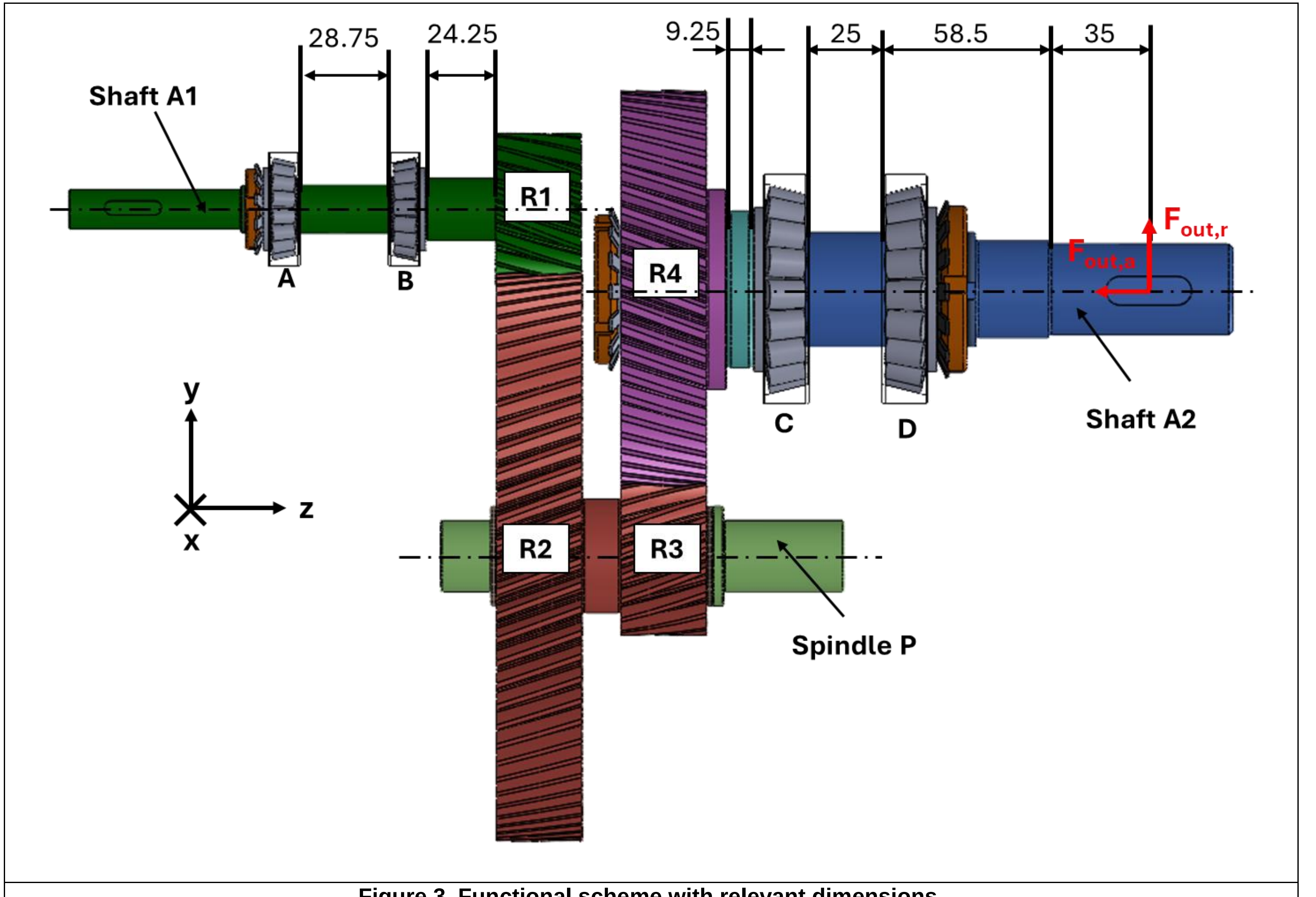


Figure 3. Functional scheme with relevant dimensions.

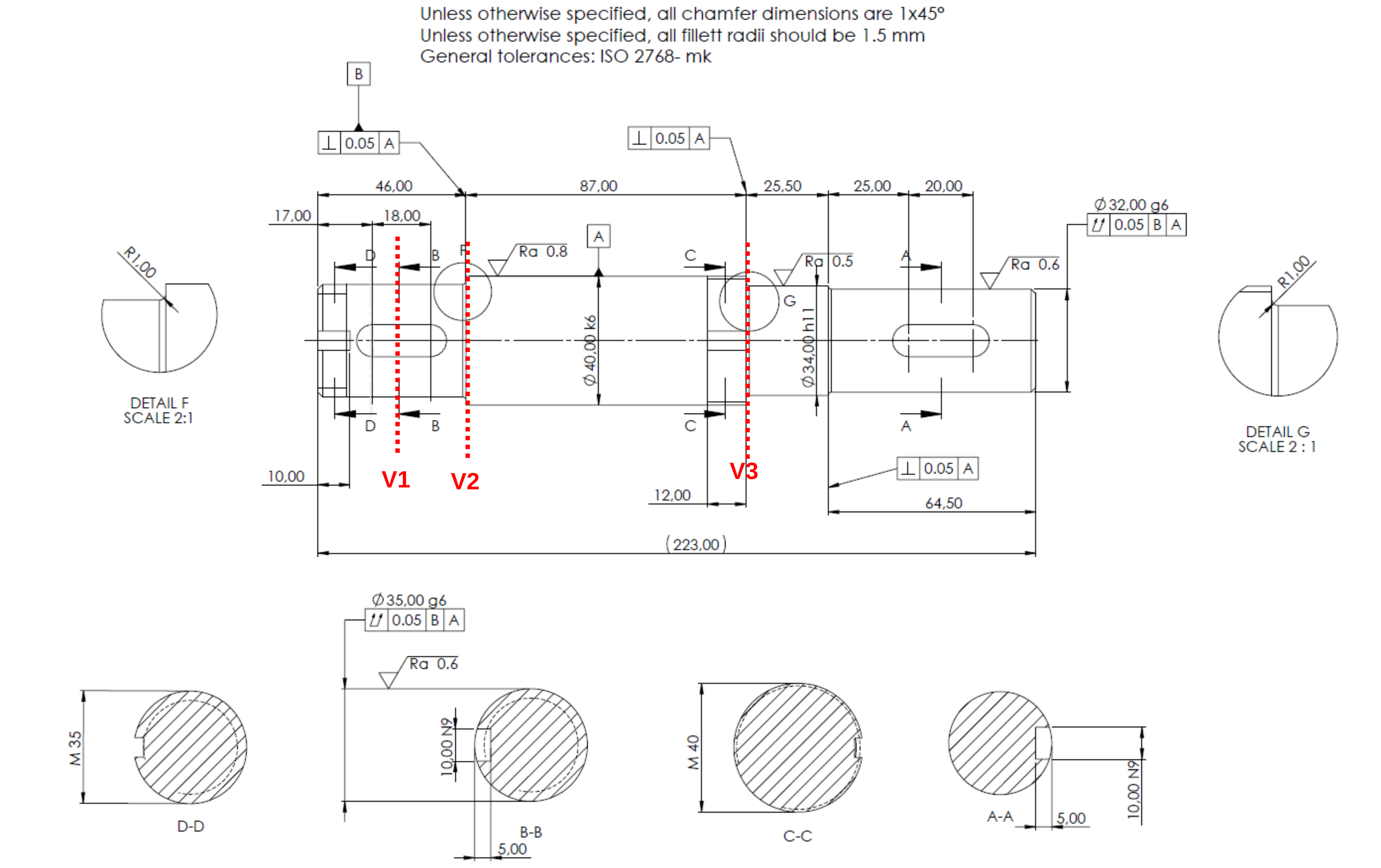


Figure 4. Drawing of shaft A2.

MAIN REQUESTS

Part 1

- Static verification and Fatigue analysis for infinite life for shaft **A2** in the sections shown in Figure 4
  - $\mathbf{V_1}$ in the middle section of the keyseat for key **L2**
  - $\mathbf{V_2}$ in the **shoulder** machined to axially locate gear **R4** on shaft **A2.**
  - $\mathbf{V_3}$ in the **shoulder** on the right side of bearing **D.**

  considering the stress concentration and intensification factors where necessary.
- Bending and contact stress verification for gears **R1** and **R3** for an endurance of 10^8 cycles and a reliability of 99% (based on AGMA D2001-D04). Assume the following conditions:
  - for the commercial enclosed units, a continuous working condition without overloads and uniform power source, with an operating temperature of 60 °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 $Z_N = 1{,}4488 \cdot N^{-0{,}023}$.

Part 2

- Evaluate according to the SKF method the expected life of **all bearings** installed in the gearbox (**A**, **B**, **C**, **D**) both in millions of cycles and operating hours. For the analysis assume:
  - Constant input power and rotational speed
  - Constant operating temperature, T=60°C
  - Slight contamination conditions
  - Reliability: **95%**
  - Oil bath lubrication with **ISO VG 100 oil**
- Evaluate for each bearing the static safety factor and the minimum load.

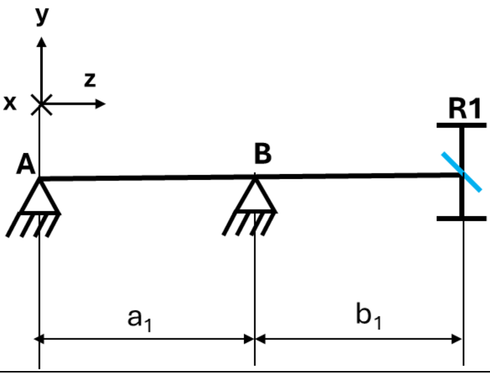
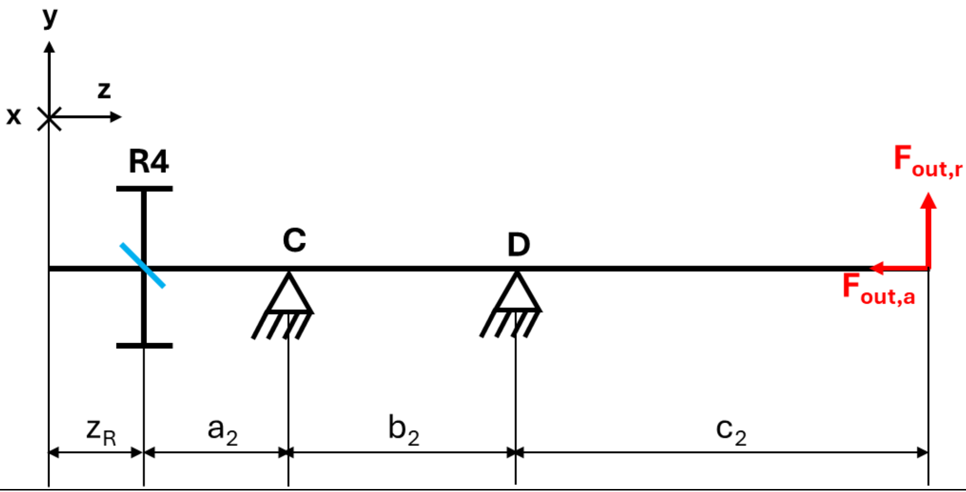
Perform the verifications following the list of calculations reported in the schemes given in the pages below, which represents a complete outline for the solution as well as a suggestion to write the technical report.

NOTE: Use ONLY the direction of rotation proposed in Figure 2 for the input shaft A1, and consequently determine the direction of rotation of all gears.

FINAL OUTCOME

- The results of the calculations for Parts 1 and 2 should be provided and **discussed** in the form of a **technical report** (**text document**).
- The technical report MUST be uploaded as a **.pdf document**, other files with **different extension WILL NOT be considered**.
- 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 are part of the final evaluation, so spend some time to write clear text and produce nice plots and tables!**

SCHEME FOR THE TECHNICAL REPORT: STATIC VERIFICATION

<table>
<tr><th>Shaft A1: External forces</th><th colspan="2">Shaft A2: External forces</th></tr>
<tr><td>$$\begin{cases} F_{t21} = \cdots \text{ N} \\ F_{r21} = \cdots \text{ N} \\ F_{a21} = \cdots \text{ N} \end{cases}$$</td><td colspan="2">$$\begin{cases} F_{t34} = \cdots \text{ N} \\ F_{r34} = \cdots \text{ N} \\ F_{a34} = \cdots \text{ N} \\ F_{OUT,r} = \cdots \text{ N} \\ F_{OUT,a} = \cdots \text{ N} \end{cases}$$</td></tr>
<tr><td colspan="3"><u>Notes</u>: Assume a gear efficiency equal to 1.</td></tr>
<tr><th colspan="3">Shaft A1: Reaction forces</th></tr>
<tr><td colspan="2">
</td><td>$$\begin{cases} a_1 = \cdots \text{ mm} \\ b_1 = \cdots \text{ mm} \end{cases}$$ $$\begin{cases} R_{xA} = \cdots \text{ N} \\ R_{xB} = \cdots \text{ N} \end{cases}$$ $$\begin{cases} R_{yA} = \cdots \text{ N} \\ R_{yB} = \cdots \text{ N} \end{cases}$$</td></tr>
<tr><th colspan="3">Shaft A2: Reaction forces</th></tr>
<tr><td colspan="2">
</td><td>$$\begin{cases} z_R = \cdots \text{ mm} \\ a_2 = \cdots \text{ mm} \\ b_2 = \cdots \text{ mm} \\ c_2 = \cdots \text{ mm} \end{cases}$$ $$\begin{cases} R_{xC} = \cdots \text{ N} \\ R_{xD} = \cdots \text{ N} \end{cases}$$ $$\begin{cases} R_{yC} = \cdots \text{ N} \\ R_{yD} = \cdots \text{ N} \end{cases}$$</td></tr>
<tr><td colspan="3"><u>Notes</u>:

For both shafts, calculate the reaction forces on both planes z-y and z-x.
For both shafts, apply the reaction forces on the <u>pressure centers</u> of the bearings.
For shaft A1, consider the z-axis with the origin in the pressure center of bearing A. Use <u>Figure 3</u> and the bearing data from the catalogue to identify the relevant dimensions (a_1, b_1).
For shaft A2, consider the origin of the z-axis to be placed on the left end (the shaft end which is closer to the gear R3). Use <u>Figure 3</u> and <u>Figure 4</u> to identify the relevant dimensions (z_R, a_2, b_2, c_2).
For shaft A1, consider the forces due to the meshing gears R1 and R2 to be applied in the midplane section of gear R1.
For shaft A2, consider the forces due to the meshing gears R3 and R4 to be applied in the middle section of the keyseat for key L2.
For shaft A2, consider the forces applied by the mechanical end user to be acting in the middle section of the keyseat for key L3.
</td></tr>
</table>

Shaft A1: Axial reaction forces	Shaft A2: Axial reaction forces
$\begin{cases} R_{zA} = \cdots \text{ N} \\ R_{zB} = \cdots \text{ N} \end{cases}$	$\begin{cases} R_{zC} = \cdots \text{ N} \\ R_{zD} = \cdots \text{ N} \end{cases}$

Notes:

- Use the uploaded SKF catalogue to calculate the axial forces produced by each tapered roller bearing.

Shaft A2 Internal loads	
Calculate and PLOT the internal loads and the total bending moment $M_B(z) = \sqrt{M_x^2 + M_y^2}$ in the intermediate shaft A2. Provide the trends of all internal loads with **CLEAR PLOTS**.	$\begin{cases} N(z) = \cdots \text{ N} \\ M_x(z) = \cdots \text{ Nm} \\ M_y(z) = \cdots \text{ Nm} \\ M_B(z) = \cdots \text{ Nm} \\ M_t(z) = \cdots \text{ Nm} \end{cases}$
Stress in the shaft A2: single components	
Calculate and PLOT 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 A2: equivalent stress	
Calculate and PLOT 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 **full yielding**. 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 A2	
Calculate the static safety factors for the three cross sections of the shaft represented in Figure 2. Identify the most critical section and evaluate the minimum safety factor of the shaft $SF_{s,min}$.	$\begin{cases} SF_{V1} = \cdots \\ SF_{V2} = \cdots \\ SF_{V3} = \cdots \\ \\ SF_{s,min} = \cdots \end{cases}$

Notes:

- In the sections with the keyseats, for static verifications, 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 keaysеats.
- For threaded sections, consider a shaft diameter equal to the nominal diameter of the thread, as if there were no threaded parts.

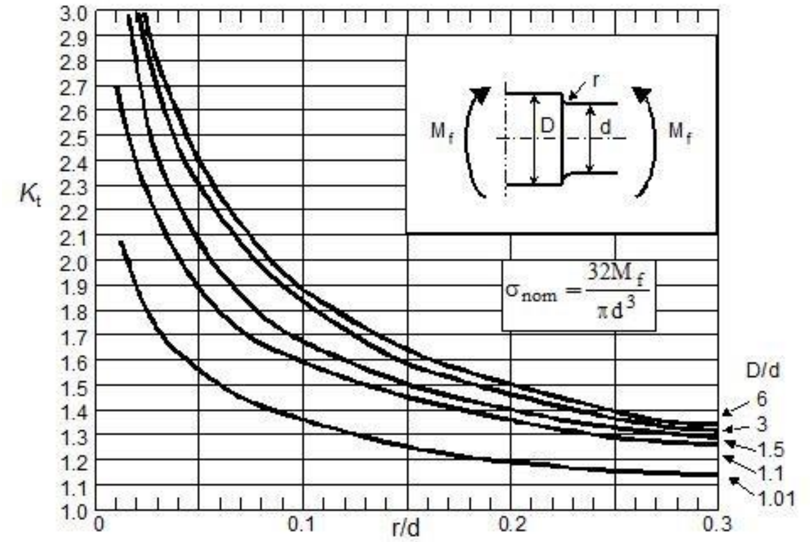
SCHEME FOR THE TECHNICAL REPORT: FATIGUE VERIFICATION

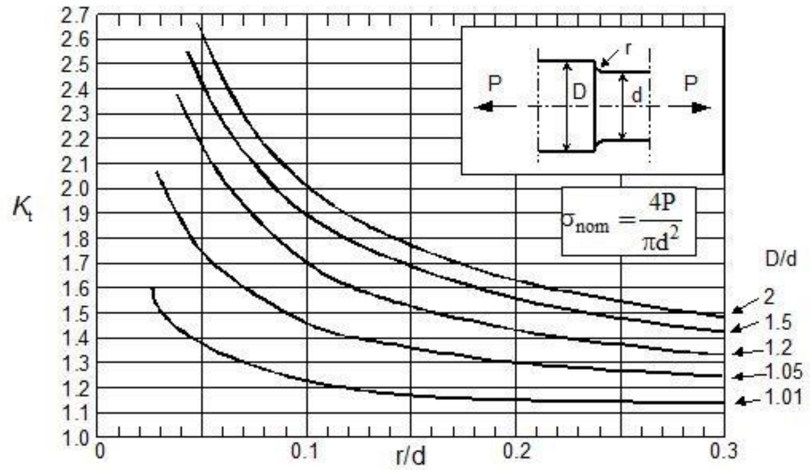
Evaluation of the fatigue stress on Shaft A2	
Evaluate, if present, the mean and alternate stress components due to the normal load, σ_m^N e σ_a^N, mean and alternate stress components due to the bending moment, $\sigma_m^{M_B}$ e $\sigma_a^{M_B}$, and the mean and alternate stress components due to the torsion moment, $\tau_m^{M_t}$ e $\tau_a^{M_t}$ in the sections V1, V2, V3.	For each section V1, V2, V3: $\begin{cases} \sigma_m^N = \cdots \text{ MPa} & \sigma_a^N = \cdots \text{ MPa} \\ \sigma_m^{M_B} = \cdots \text{ MPa} & \sigma_a^{M_B} = \cdots \text{ MPa} \\ \tau_m^{M_t} = \cdots \text{ MPa} & \tau_a^{M_t} = \cdots \text{ MPa} \end{cases}$
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 V1, V2, V3.	$\begin{cases} q = \cdots \\ K_{t,N} = \cdots \\ K_{f,N} = \cdots \\ K_{t,B} = \cdots \\ K_{f,B} = \cdots \\ K_{t,T} = \cdots \\ K_{f,T} = \cdots \end{cases}$
Fatigue limit correction factors	
Considering the working condition and the shaft geometry, see Figure 4, evaluate from the corresponding diagrams the scale effect factor C_S, and the surface finish effect factor C_F for sections V1, V2, V3.	$\begin{cases} C_S = \cdots \\ C_F = \cdots \end{cases}$
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.	$\begin{cases} \sigma_{a,eq} = \cdots \text{ MPa} \\ \sigma_{m,eq} = \cdots \text{ MPa} \end{cases}$ $SF_f = \cdots$

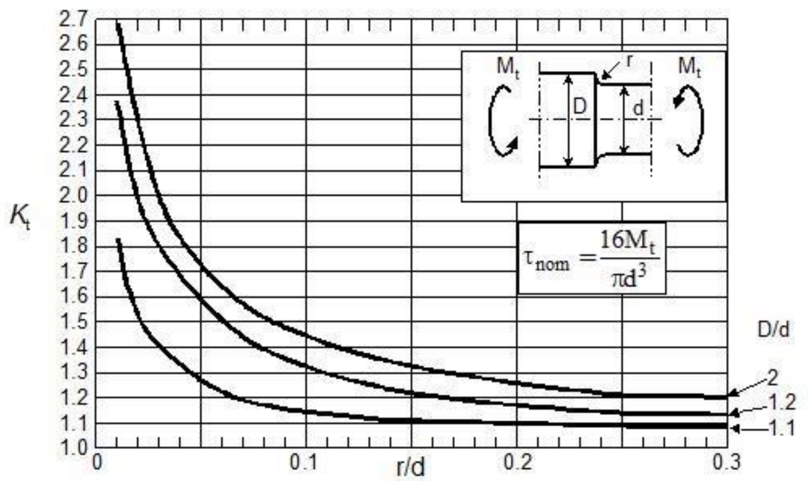
Reminder:

- **Repeat the same procedure for sections V1, V2, V3**.
- 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 V1**, where the keyseat for connection to gear **R4** is milled, calculate the nominal stresses using the full diameter, as if there was no keayseat.
- Treat section **V3** as a simple shoulder with greater diameter equal to the nominal diameter of the thread.
- **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



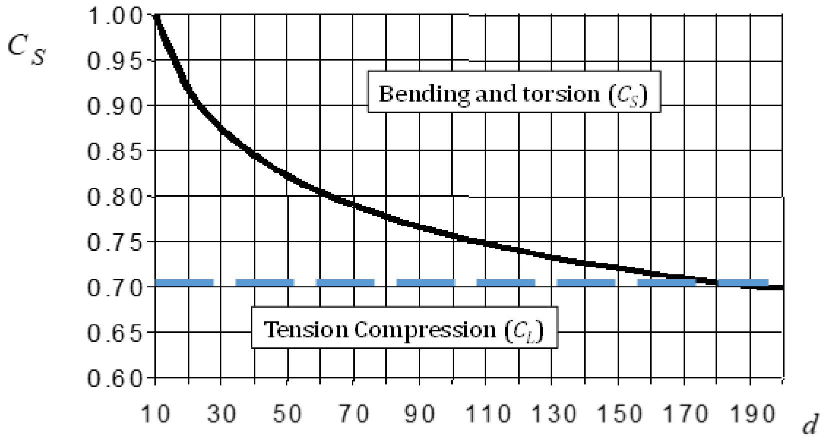




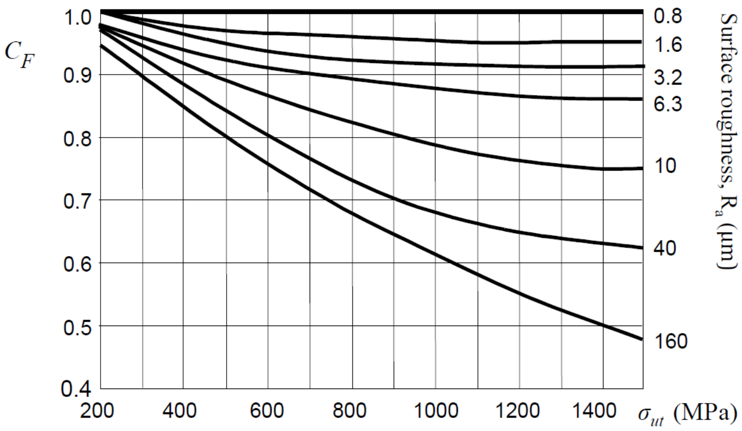
	Bending	Torsion
K_f for keyseat	1.6	2.0

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

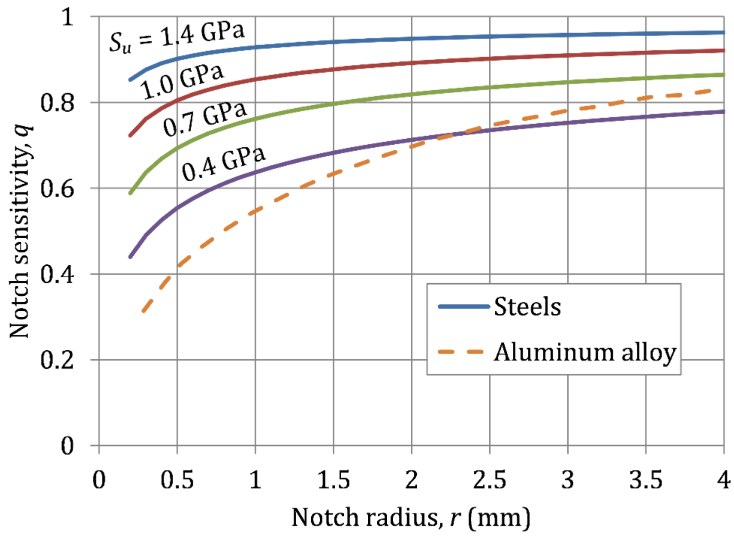
Size Effect Cs



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

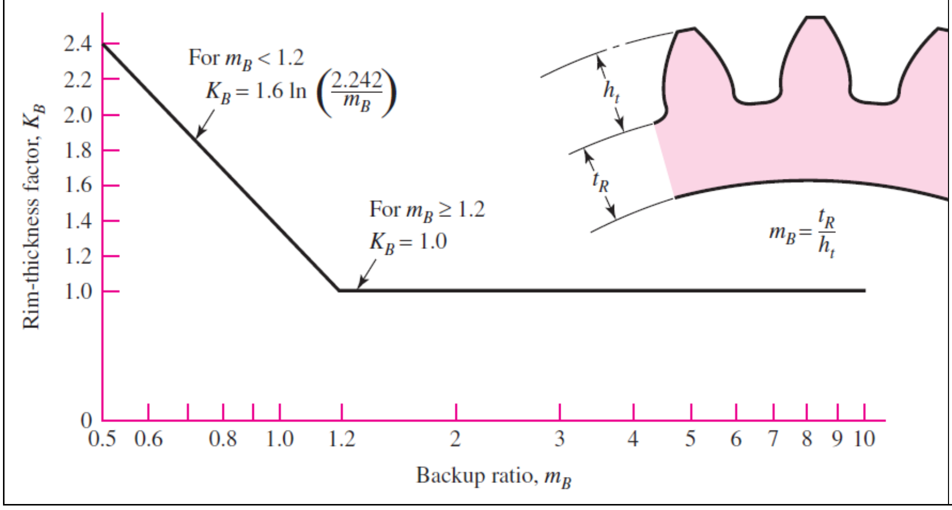
Overload factor K_O

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

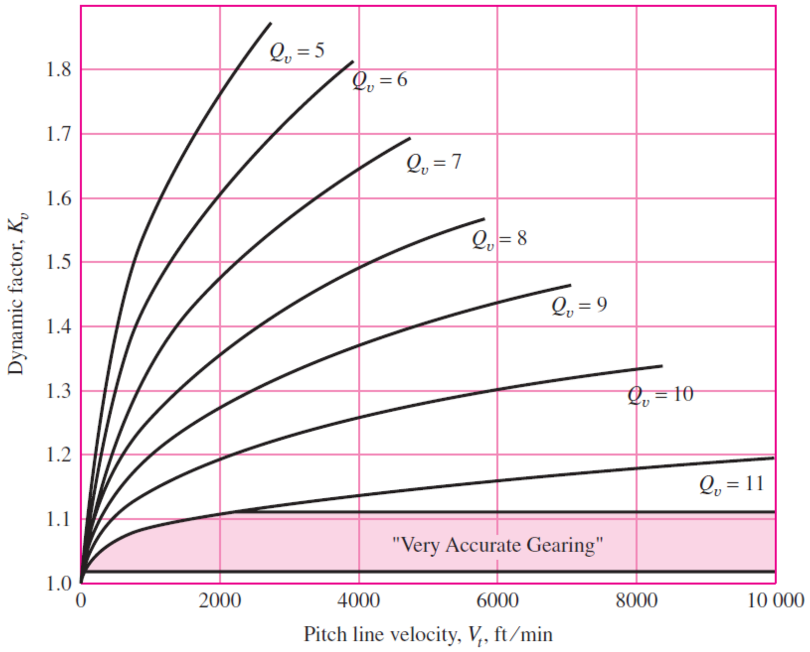
$K_O = \cdots$

Rim-thickness factor K_B



$K_B = \cdots$

<u>NOTE</u>:
For pinion **R3** assume an inner hub diameter of 30 mm.

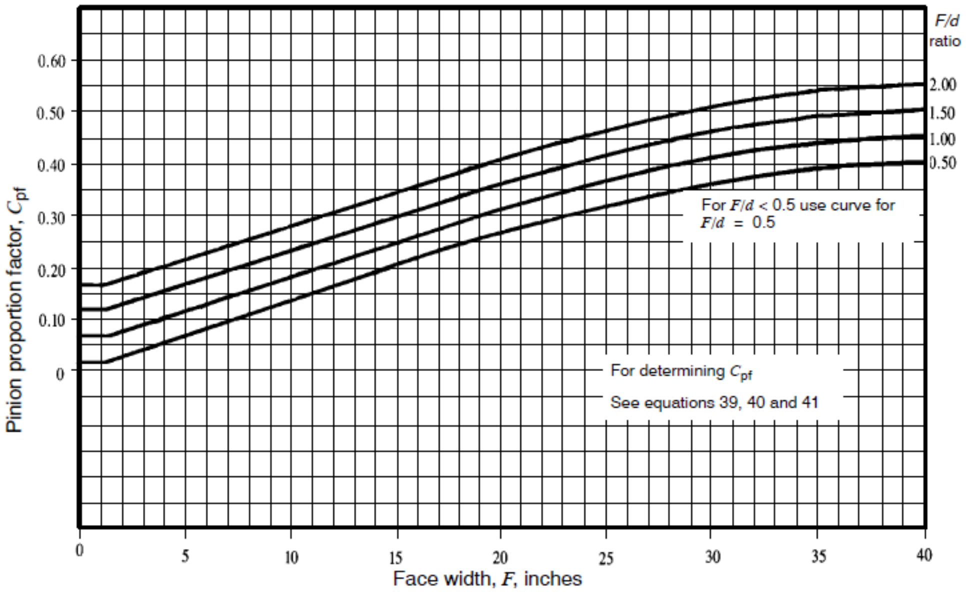
Dynamic factor K_v	
	$K_v = \ldots$ where Q_v is the transmission accuracy-level number and V_t is the peripheral linear speed of the coupling expressed in **ft/min**

Load distribution factor K_H

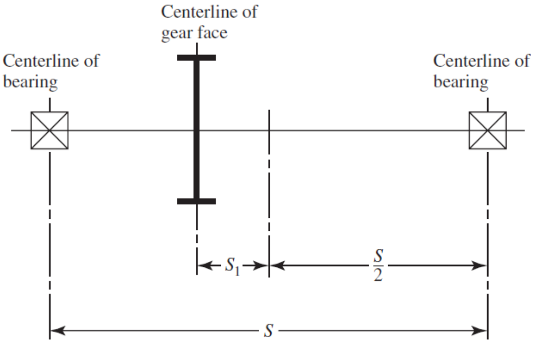
$$K_H = 1 + C_{mc}\left(C_{pf}C_{pm} + C_{ma}C_e\right), \text{ where } C_e = 1.$$

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

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



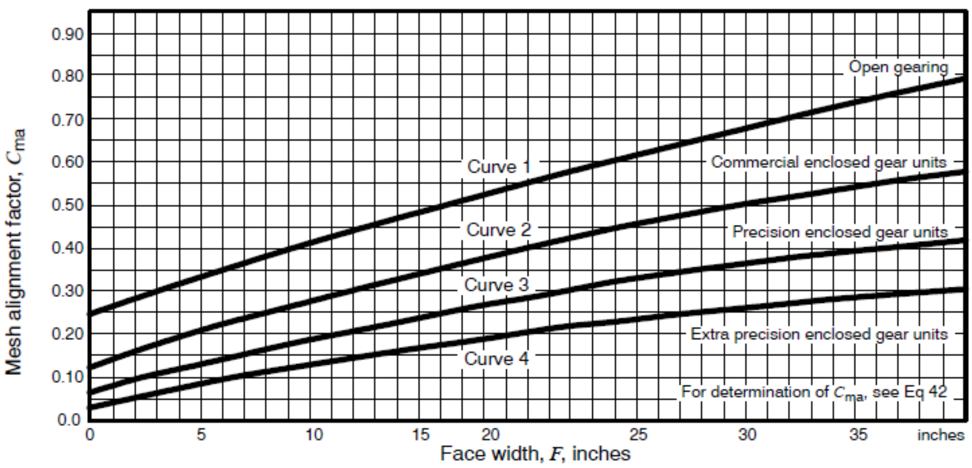
where d is the pitch circle diameter, b is the face width.



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

<u>NOTE:</u>

- For pinion **R3**, which is supported by journal bearings, consider $C_{pm} = 1$.



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

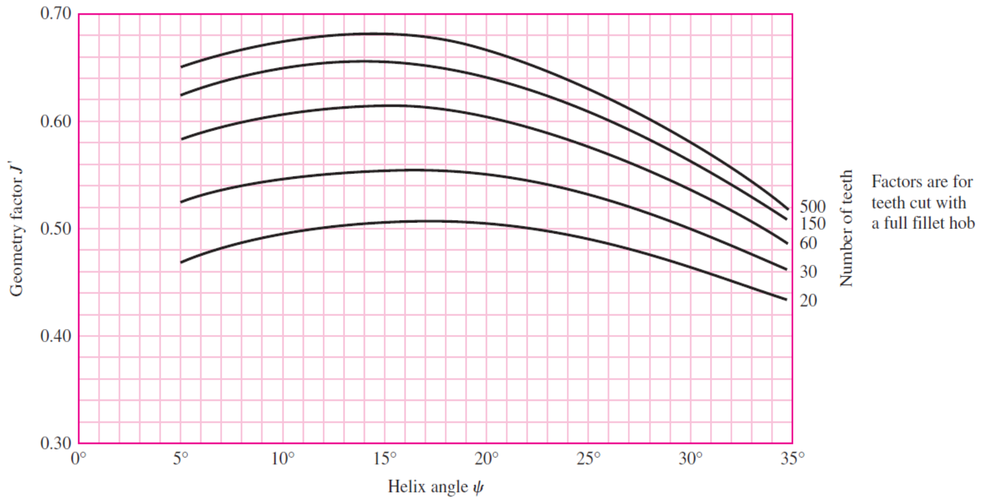
$$K_s = 0{,}843\left(b \cdot m_t\sqrt{Y}\right)^{0{,}0535} = \cdots$$

<u>NOTES</u>:

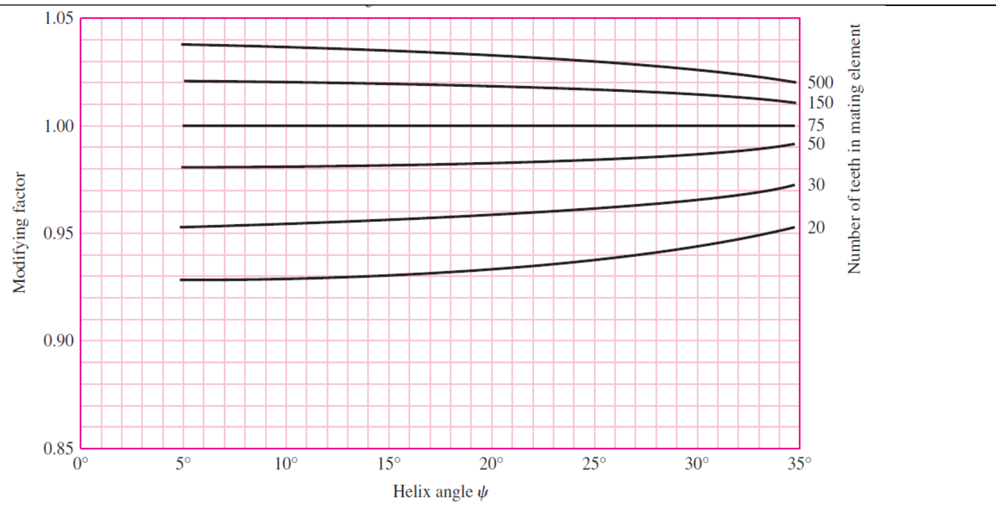
- Express b and m_t in mm in the equation above.
- For helical gears enter the table with the virtual number of teeth:

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

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



$J' = \cdots$

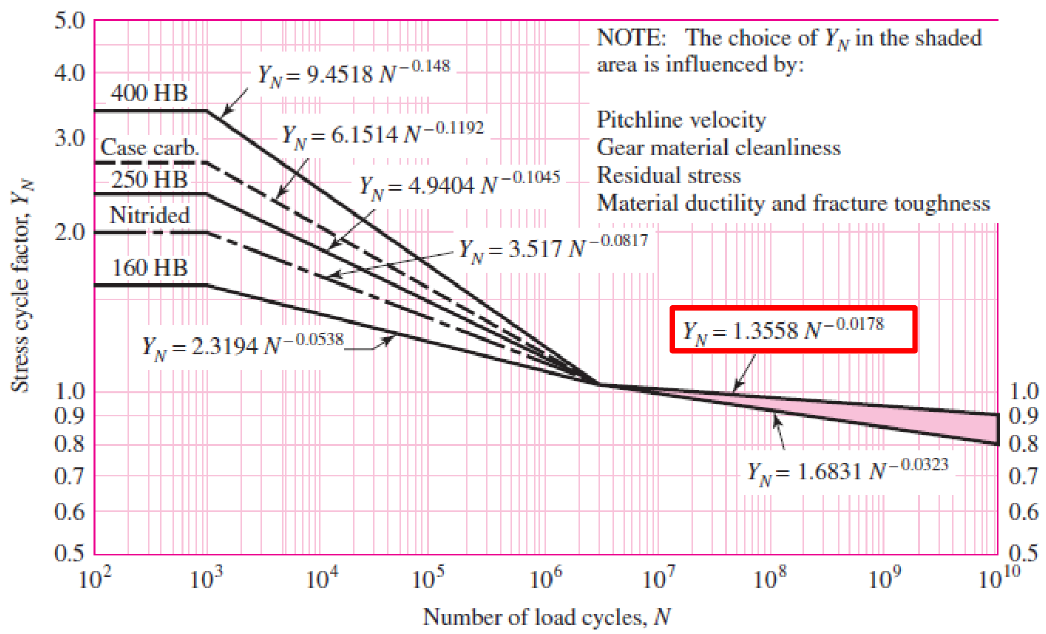


$J'' = \cdots$

Bending safety factor

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

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.

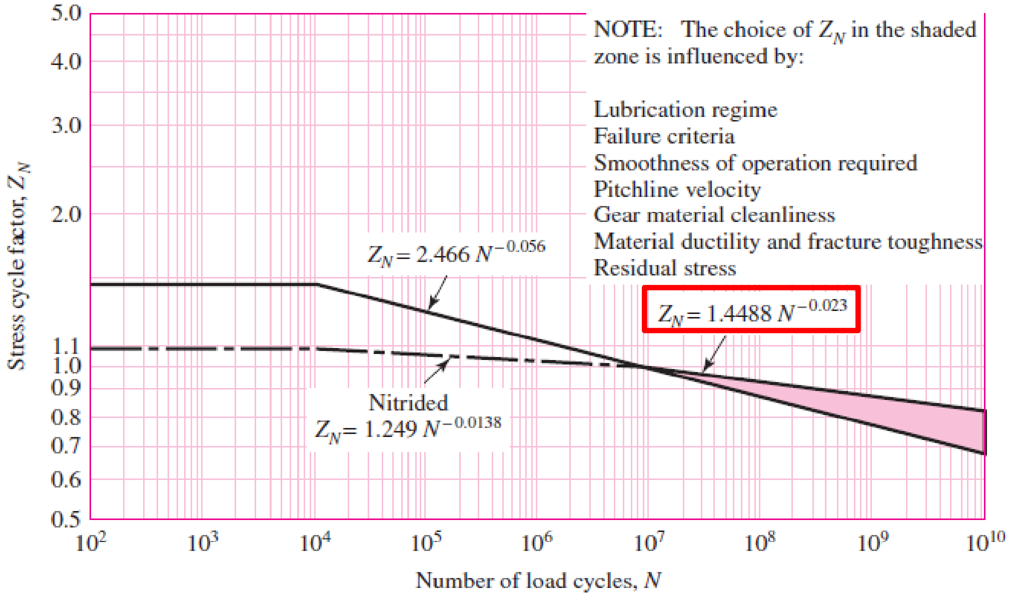
$Z_E = \dots\sqrt{\text{MPa}}$

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. $\begin{cases} Z_A = \min\left[\sqrt{(r_\mathbf{P}+a)^2 - r_{b_\mathbf{P}}^2}, (r_\mathbf{P}+r_\mathbf{G})\mathrm{sen}[\phi_t]\right] \\ Z_B = \min\left[\sqrt{(r_\mathbf{G}+a)^2 - r_{b_\mathbf{G}}^2}, (r_\mathbf{P}+r_\mathbf{G})\mathrm{sen}[\phi_t]\right] \\ Z = Z_A + Z_B - (r_\mathbf{P}+r_\mathbf{G})\mathrm{sen}[\phi_t] \end{cases} \Longrightarrow m_N = \frac{p_n\cos[\phi_n]}{0{,}95\cdot Z},$ 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)

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

Stress cycle life factor Z_N



$Z_N = \cdots$

Hardness-ratio factor Z_W (C_H)

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

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.

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

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 (95%).
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 100 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 all bearings in the gearbox (A, B, C, and D).**

Annex 1: Group input data for gearbox analysis

Group No.	Input Power P_{in} (kW)	Input speed n_{in} (rpm)
1	8,25	3000
2	7,75	2500
3	8,50	2750
4	7,25	2750
5	7,50	2500
6	6,25	2000
7	6,25	2250
8	5,00	2000
9	8,50	3000
10	6,75	2250
11	7,00	3000
12	6,00	2000
13	9,25	3000
14	8,00	3000
15	6,50	2500
16	6,25	2750
17	5,50	2000
18	5,75	2500
19	7,75	2750
20	6,75	2500
21	5,50	2250
22	5,25	2000
23	8,00	2750
24	8,25	2750
25	7,00	2250
26	7,25	3000
27	6,00	2500
28	7,50	3000
29	9,00	3000
30	7,25	2500
31	6,00	2250
32	8,75	3000
33	7,75	3000
34	6,75	2750
35	4,75	2000
36	7,00	2750
37	6,25	2500
38	4,50	2000
39	5,75	2250
40	7,50	2750
41	6,50	2750
42	5,75	2000
43	7,00	2500
44	5,25	2250
45	6,5	2250

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.

	VALUE	UNIT OF MEASUREMENT
SHAFT ANALYSIS		
$R_{x,C}$		N
$R_{y,C}$		N
$R_{x,D}$		N
$R_{y,D}$		N
$R_{z,C}$		N
$R_{z,D}$		N
Mx (V1)		Nm
Mx (V2)		Nm
Mx (V3)		Nm
My (V1)		Nm
My (V2)		Nm
My (V3)		Nm
N (V1)		N
N (V2)		N
N (V3)		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
GEARS – AGMA VERIFICATION		
GEAR R1 - BENDING		
$\left(\frac{F_t}{b \cdot m_t \cdot Y_J}\right)_{\mathrm{R1}}$		MPa
$\sigma_{\max,bending}^{R1}$		MPa
S_F^{R1}		-
GEAR R1 - PITTING		
$\left(Z_E\sqrt{\frac{F_t}{b \cdot d_{\boldsymbol{P}}} \cdot \frac{1}{Z_I}}\right)_{R1}$		MPa
$\sigma_{\max,Pitting}^{R1}$		MPa
S_H^{R1}		-
GEAR R3 - BENDING		
$\left(\frac{F_t}{b \cdot m_t \cdot Y_J}\right)_{\mathrm{R3}}$		MPa
$\sigma_{\max,bending}^{R3}$		MPa
S_F^{R3}		-
GEAR R3 - PITTING		
$\left(Z_E\sqrt{\frac{F_t}{b \cdot d_{\boldsymbol{P}}} \cdot \frac{1}{Z_I}}\right)_{R3}$		MPa
$\sigma_{\max,Pitting}^{R3}$		MPa
S_H^{R3}		-

	VALUE	UNIT OF MEASUREMENT
BEARING ANALYSIS: LOAD CALCULATION		
$F_{r,A}$		kN
$F_{a,A}$		kN
$F_{r,B}$		kN
$F_{a,B}$		kN
P_A		kN
P_B		kN
$F_{r,C}$		kN
$F_{a,C}$		kN
$F_{r,D}$		kN
$F_{a,D}$		kN
P_C		kN
P_D		kN

	VALUE	UNIT OF MEASUREMENT
BEARING ANALYSIS: CORRECTED RATING LIFE		
κ_A		-
κ_B		-
κ_C		-
κ_D		-
$a_{skf,A}$		-
$a_{skf,B}$		-
$a_{skf,C}$		-
$a_{skf,D}$		-
$L_{5mh,A}$		Hours
$L_{5mh,B}$		Hours
$L_{5mh,C}$		Hours
$L_{5mh,D}$		Hours