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 24th** you can register your group in the online Spreadsheet available at the following link:

https://docs.google.com/spreadsheets/d/1IVxn8lTE-JhheZVp9LNvYxbZp1I8Q8ycVeg_4xaj7FI/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 **19th of January 2024**.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, etc.)

Shortly after the submission deadline you will be notified about the acceptance of the uploaded material. **If the uploaded files are missing of 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 components of the mechanical gearbox represented in Figure 1.

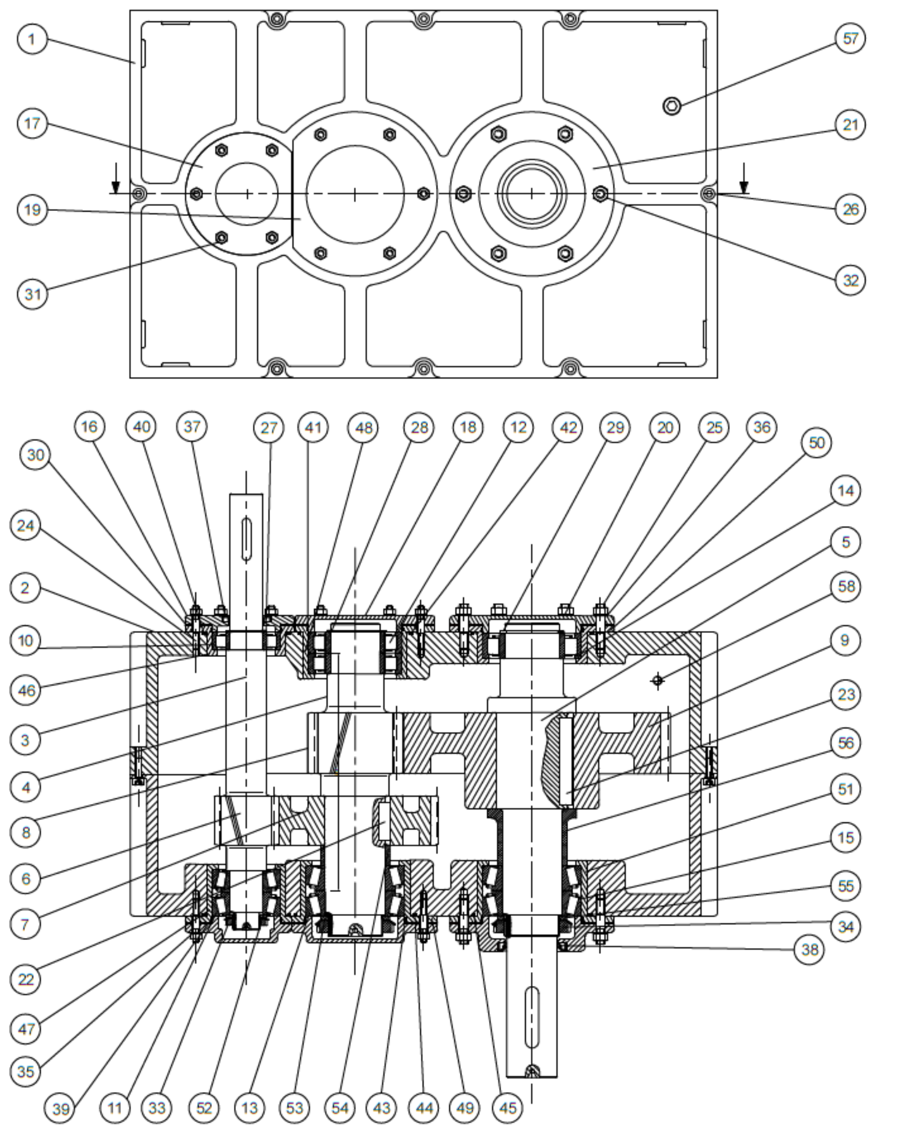

Figure 1. Gearbox assembly drawing

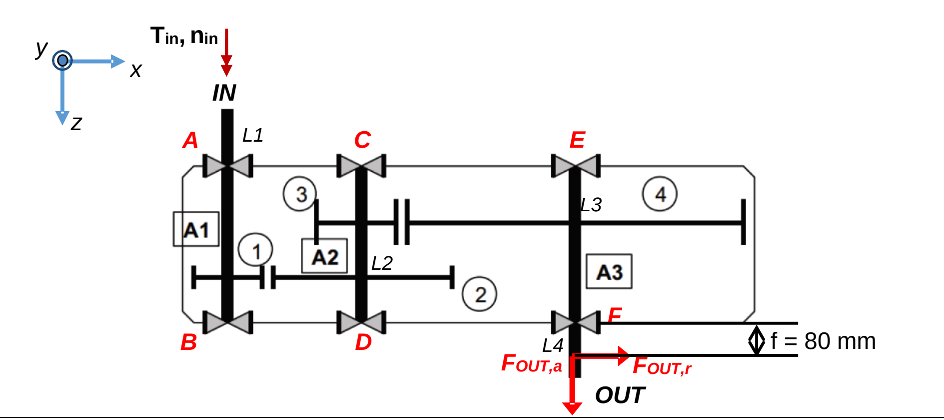

Figure 2. Gearbox functional scheme

A1: gearbox input shaft; it receives power from the motor with the key **L1**.
A2: intermediate shaft; it receives power from shaft **A1** by means of gear **G2** and transmits power to shaft **A3** with gear **G3**.
A3: output shaft of the gearbox; it receives power from shaft **A2** by means of gear **G4** and transmits power to the user with the key **L4**.
G1-G4: helical gears (Helix orientations are given in Figure 1).
L1: key on the input shaft **A1**; it receives the power from the motor.
L2: key that connects shaft **A2** with gear **G2.**
L3: key that connects shaft **A3** with gear **G4**.
L4: key on the output shaft **A3**; it transmits the power to the end user.
$F_{OUT,r}$ - $F_{OUT,a}$: external forces applied to the shaft **A2** by the end user.

Consider:

- The forces **$F_{OUT,r}$** and **$F_{OUT,a}$** are applied at distance f = 80 mm from the midplane of the matched bearing unit in support **F**
- The values of the forces must be calculated as a function of the torque on shaft A3, as:

$$F_{OUT,r} = \frac{T_{A3}}{0.25}\ N \qquad\qquad F_{OUT,a} = 0.25 \cdot F_{OUT,r}\ N$$

With T_{A3} 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: NU 206 ECP; B: 30206 DF ~~30206 X/DF~~ matched unit in face-to-face arrangement
C: 2x NU 209 ECP; D: 32011 X/DF matched unit in face-to-face arrangement
E: NU 2210 ECP; F: 32011 X/DF matched unit in face-to-face arrangement

Commented [MM1]: Consider the 30206 DF bearing from the uploaded SKF catalogue

The gear **G1** is machined from the same piece of shaft **A1** as well as gear **G3** on shaft **A2**.
Gear **G4** is keyed with **L3** on shaft **A3**. Shafts **A1** and **A3** are supported by means of a cylindrical roller bearing on one side and a matched unit of tapered roller bearings on the other. Shaft **A2** is supported by means of a pair of single-row cylindrical roller bearings on one side and a matched unit of tapered roller bearings on the other.

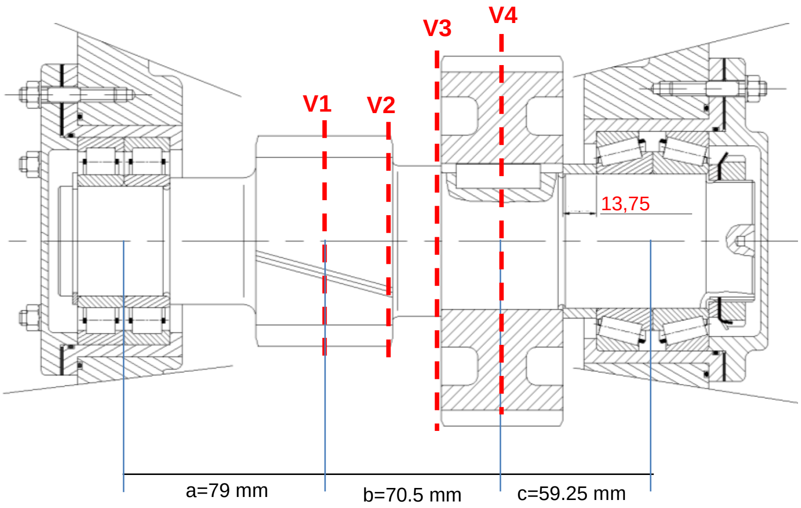

Figure 3. Shaft **A2** and related components

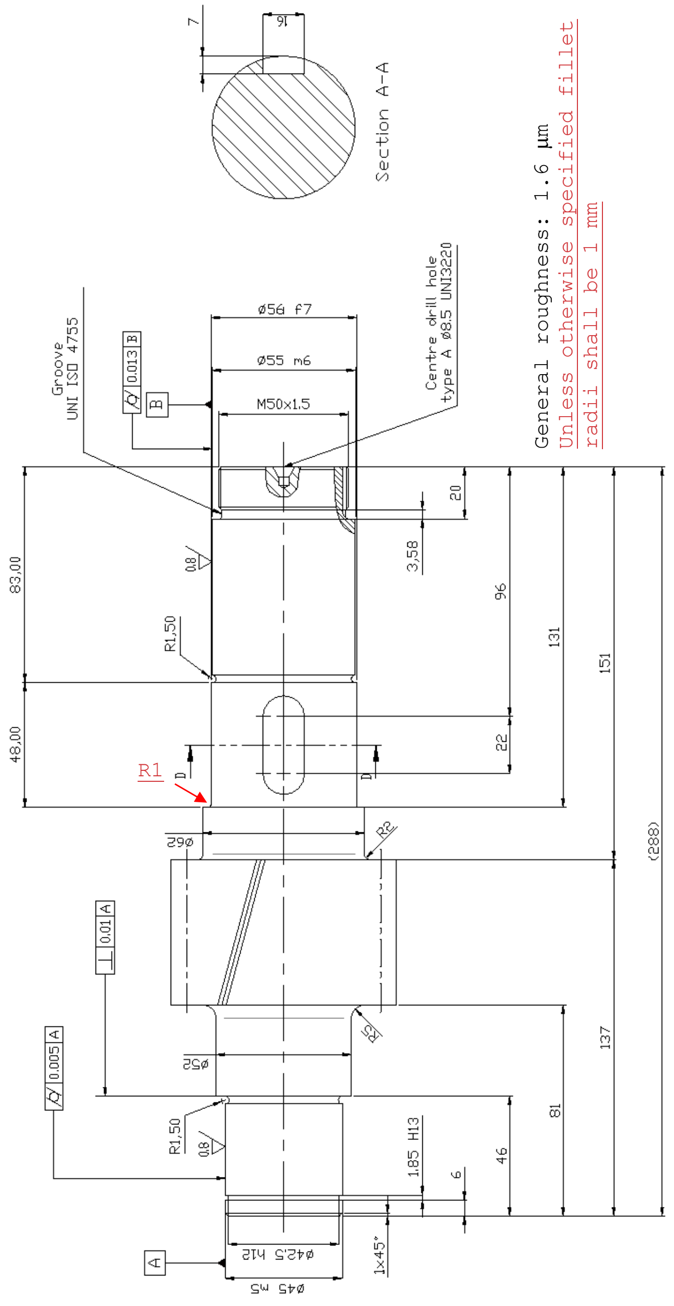

Figure 4. Shaft **A2** technical drawing

Data for static and fatigue analysis on shaft A2 and gear G1

A motor**,** rotating at an angular speed of n_{in} rpm, provides a power P_{in} kW to the input shaft **A1** of the gearbox (see Annex 1 for input data and Figure 2 for the direction of rotation). The power is transmitted to shaft **A2** through the meshing helical gears **G1** and **G2**, and to the output shaft **A3** by the meshing helical gears **G3** and **G4**.
The intermediate shaft **A2** (Figure 3) is made with a 16MnCr5 hardened and tempered steel (σ_u=1060 MPa, σ_y=930 MPa, σ_{D-1}=700 MPa), is supported on one side by a pair of cylindrical roller bearings of the type **NU 209 ECP** (from SKF) and a matched unit of tapered roller bearings of the type **32011 X/DF** (from SKF) in a face to face arrangement.
The helical gears have the following characteristics:

	G1	G2	G3	G4
normal module, m_n (mm)	3	3	4	4
number of teeth, z	16	48	21	63
helix angle, ψ	15°	15°	15°	15°
normal pressure angle, ϕ_n	20°	20°	20°	20°
transmission accuracy level, Q_v	8	8	8	8
tooth face width (mm)	45	45	56	56
Ultimate strength (MPa)	1060	1060	1060	1060
Yield strength (MPa)	930	930	930	930
Surface Hardness (HB)	700	700	700	700

Part 1

- Static verification and Fatigue analysis for infinite life for shaft **A2** in the sections shown in Figure 3
  - $\mathbf{V_1}$ in the mid-section of gear **G3**;
  - $\mathbf{V_2}$ in the **shoulder** given by **G3** of shaft **A2**;
  - $\mathbf{V_3}$ in the **shoulder** used to axially constrain the gear **G2**;
  - $\mathbf{V_4}$ in the mid-section of gear **G2** whereby the shaft features a **keyseat**.

  considering the stress concentration and intensification factors where necessary.
- Bending and contact stress verification for gear **G3** for an endurance of $\underline{10^8}$ 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, E and F) both in millions of cycles and operating hours. For the analysis let's assume:
  - Constant input power and rotational speed
  - Constant operating temperature, T=60°C
  - Normal cleanliness level
  - Reliability 90%
  - **Choose the lubricant viscosity grade** to ensure a value of the viscosity ratio κ as close as possible to 1 on the bearings of the output shaft A3. Then use the **selected lubricant** to perform the calculations on the bearings mounted on the other two shafts (A1 and A2).
- Evaluate for each bearing the static safety factor and the minimum load.

Perform the verifications following the list of calculations reported in the schemes below, which represents a complete outline for the solution as well as a suggestion to write the technical report.
Remember to provide the numerical values of all your calculations adding to your report a finale table like the submission table provided at the end of this document.

SCHEME FOR THE TECHNICAL REPORT: STATIC VERIFICATION

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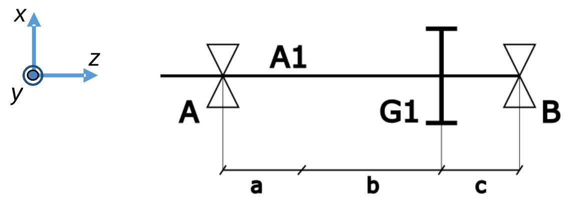
Shaft A1: External forces	Shaft A2: External forces	Shaft A3: External forces
$\begin{cases} F_{t21} = \cdots \mathrm{N} \\ F_{r21} = \cdots \mathrm{N} \\ F_{a21} = \cdots \mathrm{N} \end{cases}$	$\begin{cases} F_{t12} = \cdots \mathrm{N} \\ F_{r12} = \cdots \mathrm{N} \\ F_{a12} = \cdots \mathrm{N} \\ F_{t43} = \cdots \mathrm{N} \\ F_{r43} = \cdots \mathrm{N} \\ F_{a43} = \cdots \mathrm{N} \end{cases}$	$\begin{cases} F_{t34} = \cdots \mathrm{N} \\ F_{r34} = \cdots \mathrm{N} \\ F_{a34} = \cdots \mathrm{N} \\ F_{OUT,r} = \cdots \mathrm{N} \\ F_{OUT,a} = \cdots \mathrm{N} \end{cases}$

Notes:

Assume a gear efficiency equal to 1.

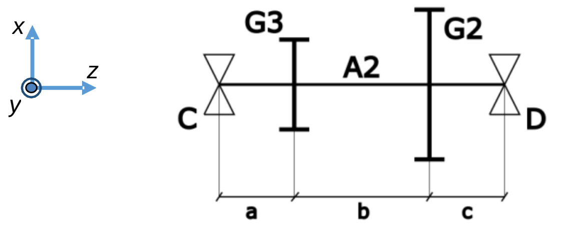
Shaft A1: Reaction forces



$$\begin{cases} R_{xA} = \cdots \mathrm{N} \\ R_{xB} = \cdots \mathrm{N} \end{cases}$$

$$\begin{cases} R_{yA} = \cdots \mathrm{N} \\ R_{yB} = \cdots \mathrm{N} \end{cases}$$

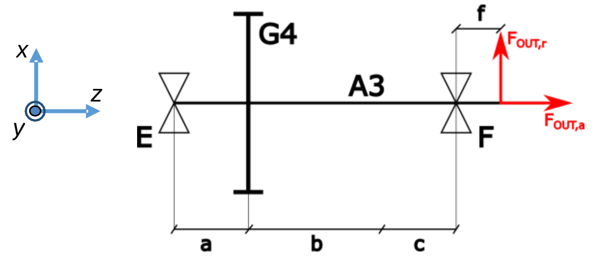
Shaft A2: Reaction forces



$$\begin{cases} R_{xC} = \cdots \mathrm{N} \\ R_{xD} = \cdots \mathrm{N} \end{cases}$$

$$\begin{cases} R_{yC} = \cdots \mathrm{N} \\ R_{yD} = \cdots \mathrm{N} \end{cases}$$

Shaft A3: Reaction forces



$$\begin{cases} R_{xE} = \cdots \mathrm{N} \\ R_{xF} = \cdots \mathrm{N} \end{cases}$$

$$\begin{cases} R_{yE} = \cdots \mathrm{N} \\ R_{yF} = \cdots \mathrm{N} \end{cases}$$

Notes:

- For the pair of roller bearings on shaft A2 (support C), consider a single application point in the midplane section of the unit. For the matched units of tapered roller bearings in face-to-face configuration, consider a single application point in the midplane section of the unit.
- Forces from the gears of the system can be considered applied on the midplane section of gears.

Shaft A1: Axial reaction forces	Shaft A2: Axial reaction forces	Shaft A3: Axial reaction forces
$\begin{cases} R_{zA} = \cdots \text{ N} \\ R_{zB} = \cdots \text{ N} \end{cases}$	$\begin{cases} R_{zC} = \cdots \text{ N} \\ R_{zD} = \cdots \text{ N} \end{cases}$	$\begin{cases} R_{zC} = \cdots \text{ N} \\ R_{zD} = \cdots \text{ N} \end{cases}$

Notes:

- For each shaft, identify the bearing acting as the roller support and the bearing behaving as the pinned support.

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 and plot their trends.	$\begin{cases} N(z) = \cdots \text{ N} \\ M_x(z) = \cdots \text{ Nm} \\ M_y(z) = \cdots \text{ Nm} \\ M_B(z) = \cdots \text{ Nm} \\ M_t(z) = \cdots \text{ Nm} \end{cases}$
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 $\tau^{M_t}(z)$.	$\begin{cases} \sigma^N(z) = \cdots \text{ MPa} \\ \sigma^{M_B}(z) = \cdots \text{ MPa} \\ \tau^{M_t}(z) = \cdots \text{ MPa} \end{cases}$
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**.	$\begin{cases} \sigma^{tot}(z) = \cdots \text{ MPa} \\ \sigma_{id}(z) = \cdots \text{ MPa} \end{cases}$
Static safety factor on the shaft A2	
Calculate the static safety factors for the four cross sections of the shaft represented in Figure 2. Identify the most critical section and evaluate the minimum safety factor of the shaft $SF_{s,min}$.	$\begin{cases} SF_{V1} = \cdots \\ SF_{V2} = \cdots \\ SF_{V3} = \cdots \\ SF_{V4} = \cdots \\ SF_{s,min} = \cdots \end{cases}$

Notes:

- For **pinion G3**, which is directly machined from the same piece of the shaft, consider a shaft diameter equal to the diameter of the **gear dedendum circle**.

SCHEME FOR THE TECHNICAL REPORT: FATIGUE VERIFICATION

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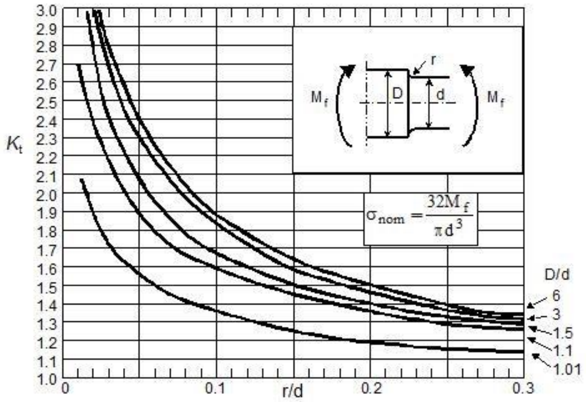
Evaluation of the fatigue stress on Shaft A2	
Evaluate, if present, the mean and alternate stress components due to the normal load, σ_m^N e σ_a^N, mean and alternate stress components due to the bending moment, $\sigma_m^{M_B}$ e $\sigma_a^{M_B}$, and the mean and alternate stress components due to the torsion moment, $\tau_m^{M_t}$ e $\tau_a^{M_t}$ in the sections V1, V2, V3 and V4.	For each section V1, V2, V3 and V4: $\begin{cases} \sigma_m^N = \cdots \text{MPa} & \sigma_a^N = \cdots \text{MPa} \\ \sigma_m^{M_B} = \cdots \text{MPa} & \sigma_a^{M_B} = \cdots \text{MPa} \\ \tau_m^{M_t} = \cdots \text{MPa} & \tau_a^{M_t} = \cdots \text{MPa} \end{cases}$
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 and V4.	$\begin{cases} q = \cdots \\ K_{t,N} = \cdots \\ K_{f,N} = \cdots \\ K_{t,B} = \cdots \\ K_{f,B} = \cdots \\ K_{t,T} = \cdots \\ K_{f,T} = \cdots \end{cases}$
Fatigue limit correction factors	
Considering the working condition and the shaft geometry, evaluate from the corresponding diagrams the scale effect factor C_S, and the surface finish effect factor C_F for sections V1, V2, V3 and V4.	$\begin{cases} C_S = \cdots \\ C_F = \cdots \end{cases}$
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.	$\begin{cases} \sigma_{a,eq} = \cdots \text{MPa} \\ \sigma_{m,eq} = \cdots \text{MPa} \end{cases}$ $SF_f = \cdots$

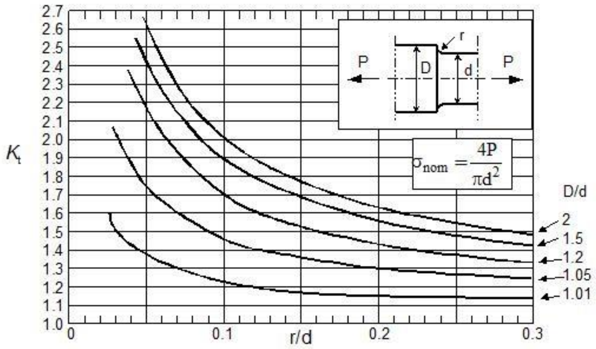
Commented [MM2]: For V3 consider fillet radius of 1 mm.

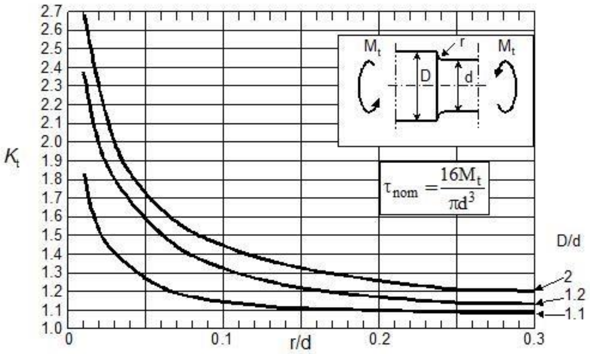
Reminder:

- **Repeat the same procedure for sections V1, V2, V3 and V4**. For V3 consider fillet radius of 1 mm.
- Use the diagrams in the following pages for the evaluation of the coefficients.
- Use the **Shigley Method** to calculate the alternate and average equivalent stress components of the working point.
- For **pinion G3**, which is directly machined from the same piece of the shaft, consider a shaft diameter equal to the diameter of the gear **dedendum circle**.
- **For each section, it is requested to draw the Haigh diagram and identify the position of the working point!**

Geometrical stress concentration factors Kt

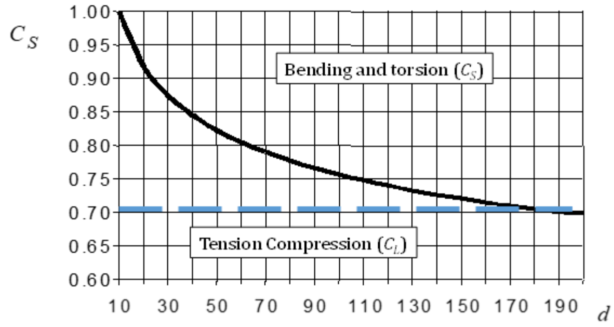




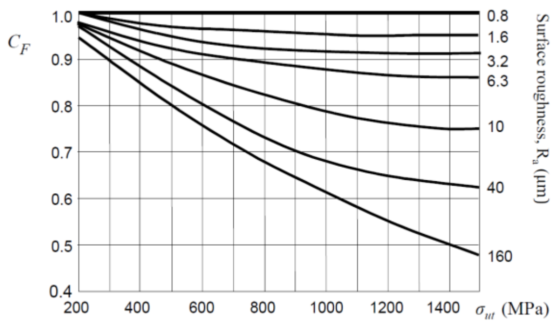


	Bending	**Torsion**
K_f for the keyseat	1.6	2.0
K_f for machined pinion on shaft	1.2	1.5

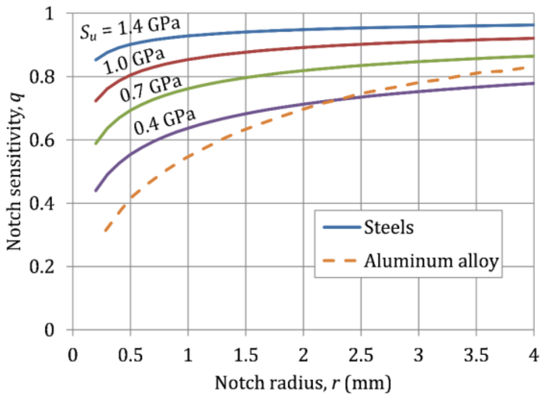
Size Effect Cs



Surface finish effect



Notch sensitivity q



SCHEME FOR THE TECHNICAL REPORT: GEAR TOOTH VERIFICATION

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Maximum tooth gear bending stress equation for fatigue

$$\sigma_{\max,Fatigue} = F_t K_0 K_B K_v K_H K_s \frac{1}{b \cdot m_t Y_J}$$

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 couple

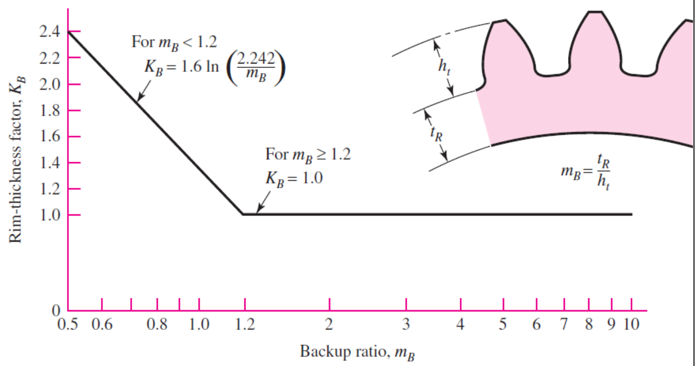
Overload factor K_0

Table of Overload Factors, K_o

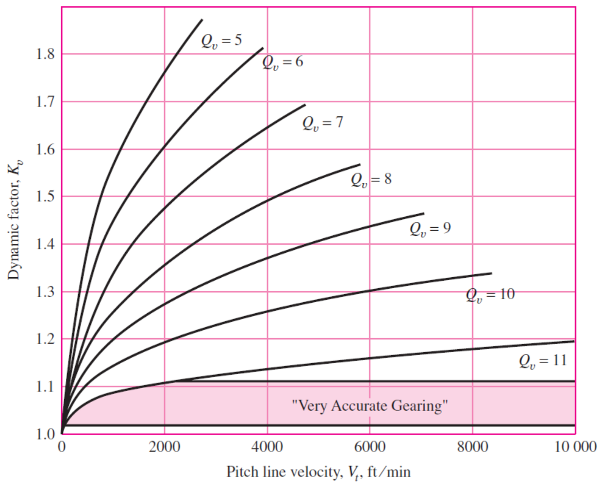
	Driven Machine		
Power source	Uniform	Moderate shock	Heavy shock
Uniform	1.00	1.25	1.75
Light shock	1.25	1.50	2.00
Medium shock	1.50	1.75	2.25

$K_0 = \cdots$

Rim-thickness factor K_B



$K_B = \cdots$

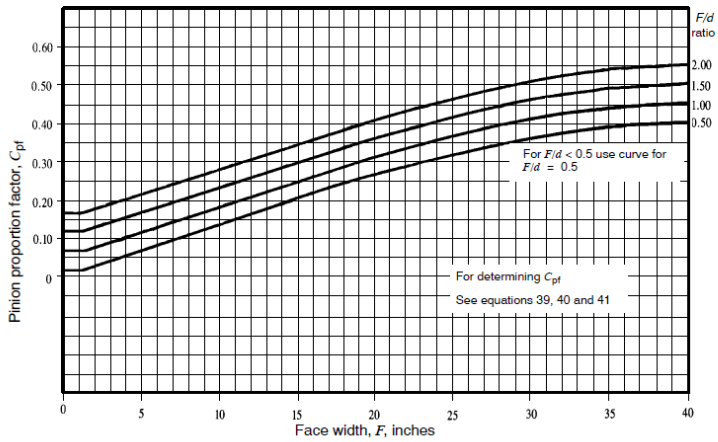
Dynamic factor K_v	
	$K_v\ = \ldots$ where $Q_v$is the transmission accuracy-level number and V_t is the peripheral linear speed of the coupling expressed in ft/min

Load distribution factor K_H

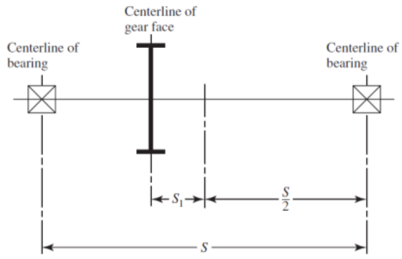
$$K_H = 1 + C_{mc}\left(C_{pf}C_{pm} + C_{ma}C_e\right), \text{ where } C_e = 1.$$

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

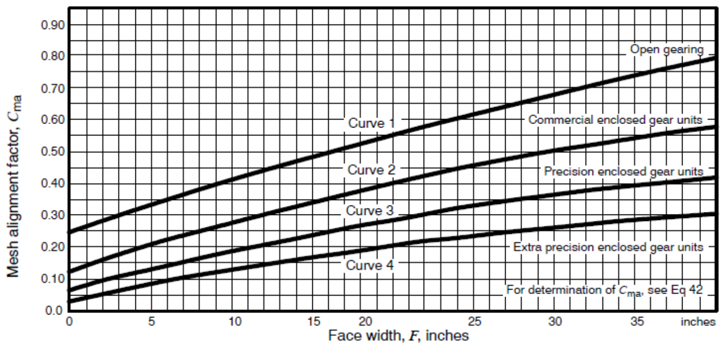
$$C_{mc} = \begin{cases} 1 & \text{for uncrowned teeth} \\ 0{,}8 & \text{for crowned teeth} \end{cases}$$



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



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



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

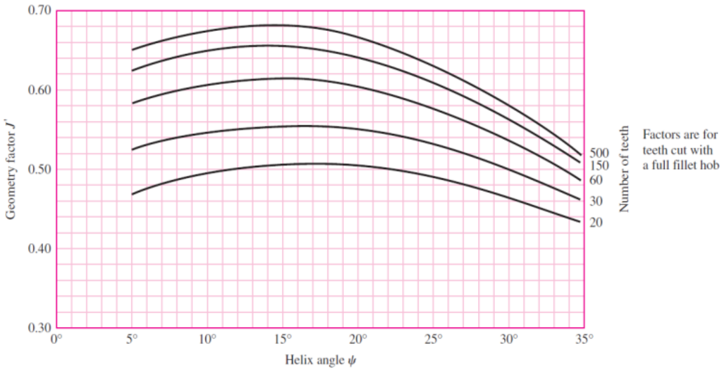
$$K_s = 0{,}843\left(b \cdot m_t\sqrt{Y}\right)^{0{,}0535}$$

Express b and mt in mm

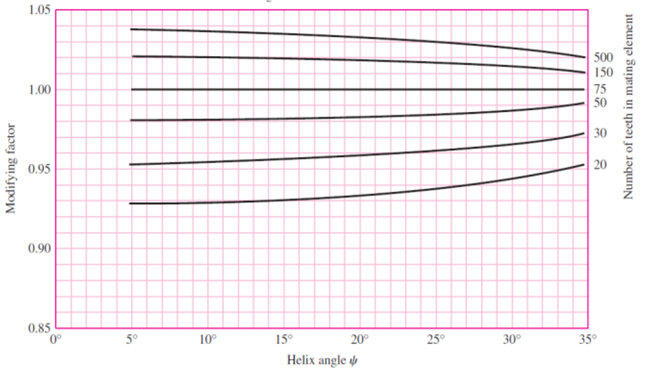
For helical gears

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

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



J'



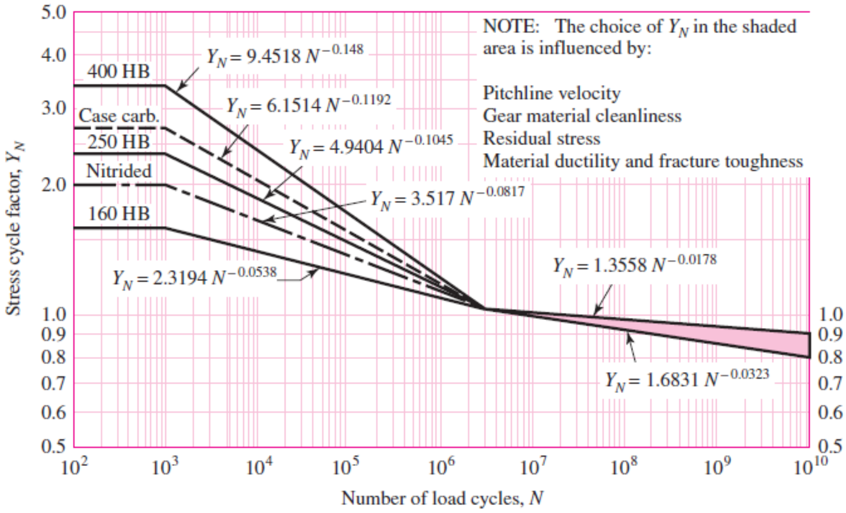
J''

Bending safety factor

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

Bending fatigue strength σ_{FP} ($S_{f,lim}$) = 860 MPa

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

$$\sigma_{\max,pitting} = Z_E\sqrt{F_t K_0 K_v K_s \cdot \frac{K_H}{b \cdot d_P} \cdot \frac{Z_R}{Z_I}}$$

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.

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

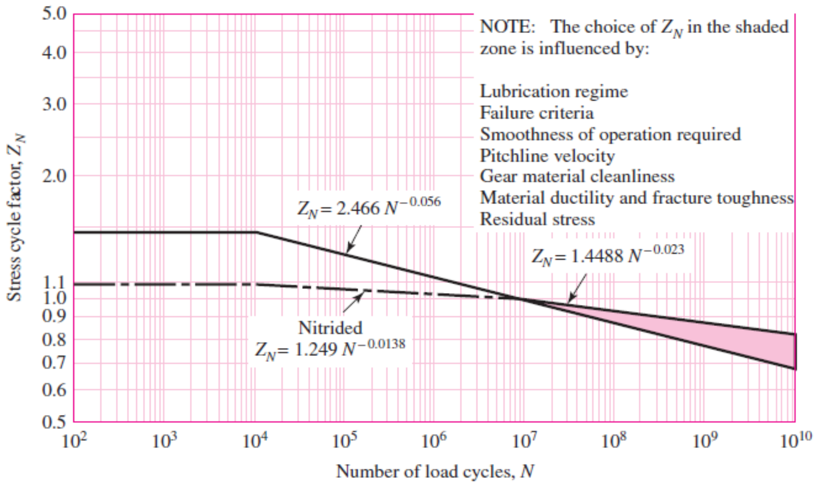
Surface strength geometry Z_I
$\begin{cases} r_{b_\mathbf{P}} = r_\mathbf{P}\cos[\phi_t] \\ r_{b_\mathbf{G}} = r_\mathbf{G}\cos[\phi_t] \end{cases}$,   where $r_\mathbf{P}$ e $r_\mathbf{G}$ are the pitch radii of the pinion **P** (the smaller gear) and the mating gear **G** respectively while $r_{b_\mathbf{P}}$ 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 $\tan[\phi_t] = \frac{\tan[\phi_n]}{\cos[\psi]}$. $Z_I = \begin{cases} \frac{\cos[\phi_t]\mathrm{sen}[\phi_t]}{2m_N}\cdot\frac{m_G}{m_G+1} & \text{external meshing} \\ \frac{\cos[\phi_t]\mathrm{sen}[\phi_t]}{2m_N}\cdot\frac{m_G}{m_G-1} & \text{internal meshing} \end{cases}$ where $m_G = n_\mathbf{P}/n_\mathbf{G} = d_\mathbf{G}/d_\mathbf{P}$ is the gear ratio.

Wear safety factor (Hertzian contact)

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

Contact strength σ_{HP} *(S_c) = 1470 MPa*

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 LIFE INVESTIGATION

Team n.	02SXJJM Fundamentals of Machine Design date ....
	sheet n. ...../.....

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)

$$L_{10m} = a_1 a_{SKF} \left(\frac{C}{P}\right)^p$$

Bearing life in operating hours

$$L_{10mh} = \frac{10^6}{60 \cdot n} L_{10m}$$

Evaluation of the contamination

Values for η_c are given for several levels of contamination

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

Remember to select the lubricant viscosity grade with the goal to ensure a value of the viscosity ratio κ as close as possible to 1 on the bearings of the output shaft A3 for the given operating temperature of 60°C.

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

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

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.

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, D, E and F).**

TIPS:

- **For the pair of roller bearings mounted on shaft A2 (support C), consider that the total load is equally split between the two roller bearings in the unit.**

Annex 1: Group input data for gearbox analysis

Group No.	Input Power P_{in} (kW)	Input speed n_{in} (rpm)
1	29,5	1000
2	34,5	1500
3	25,5	1250
4	20	750
5	31,5	1500
6	29	1250
7	36	1500
8	27	1250
9	19,5	750
10	30	1500
11	29,5	1250
12	21	750
13	22	750
14	26	1250
15	33	1500
16	24,5	1000
17	30	1250
18	21,5	750
19	25,5	1000
20	26,5	1000
21	35,5	1500
22	25	1000
23	28	1000
24	18,5	750
25	20,5	750
26	34	1500
27	30,5	1500
28	25	1250
29	32	1500
30	28,5	1000
31	19	750
32	26	1000
33	29	1000
34	27	1000
35	28	1250

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(z=V1)		Nm
Mx(z=V2)		Nm
Mx(z=V3)		Nm
Mx(z=V4)		Nm
My(z=V1)		Nm
My(z=V2)		Nm
My(z=V3)		Nm
My(z=V4)		Nm
N(z=V1)		N
N(z=V2)		N
N(z=V3)		N
N(z=V4)		N
$Mb_{tot,max}$		Nm
$Mt_{tot,max}$		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 ANALYSIS - BENDING		
$\frac{F_t}{b \cdot m_t \cdot Y_J}$		MPa
$\sigma_{\max,Fatigue}$		MPa
S_F		-
GEAR ANALYSIS - PITTING		
$Z_E \sqrt{\frac{F_t}{b \cdot d_P} \cdot \frac{1}{Z_I}}$		MPa
$\sigma_{\max,pitting}$		MPa
S_H		-

	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
$F_{r,E}$		kN
$F_{a,E}$		kN
$F_{r,F}$		kN
$F_{a,F}$		kN
P_E		kN
P_F		kN

** NOTE: Consider the values of the single roller bearing in the pair.*

	VALUE	UNIT OF MEASUREMENT
BEARING ANALYSIS: CORRECTED RATING LIFE		
ISO VG of the selected oil		mm^2/s
$\mathbf{a_{skf,A}}$		-
$\mathbf{a_{skf,B}}$		-
$\mathbf{a_{skf,C}}$		-
$\mathbf{a_{skf,D}}$		-
$\mathbf{a_{skf,E}}$		-
$\mathbf{a_{skf,F}}$		-
$L_{10mh,A}$		Hours
$L_{10mh,B}$		Hours
$L_{10mh,C}$		Hours
$L_{10mh,D}$		Hours
$L_{10mh,E}$		Hours
$L_{10mh,F}$		Hours