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 18th** you can register your group in the online Spreadsheet available at the following link

https://docs.google.com/spreadsheets/d/1BtuYyy3ZUYSMv2qG5f7ukU5_eGVCTN6719JysDELBKM/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 **13th of January 2023**.

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 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.

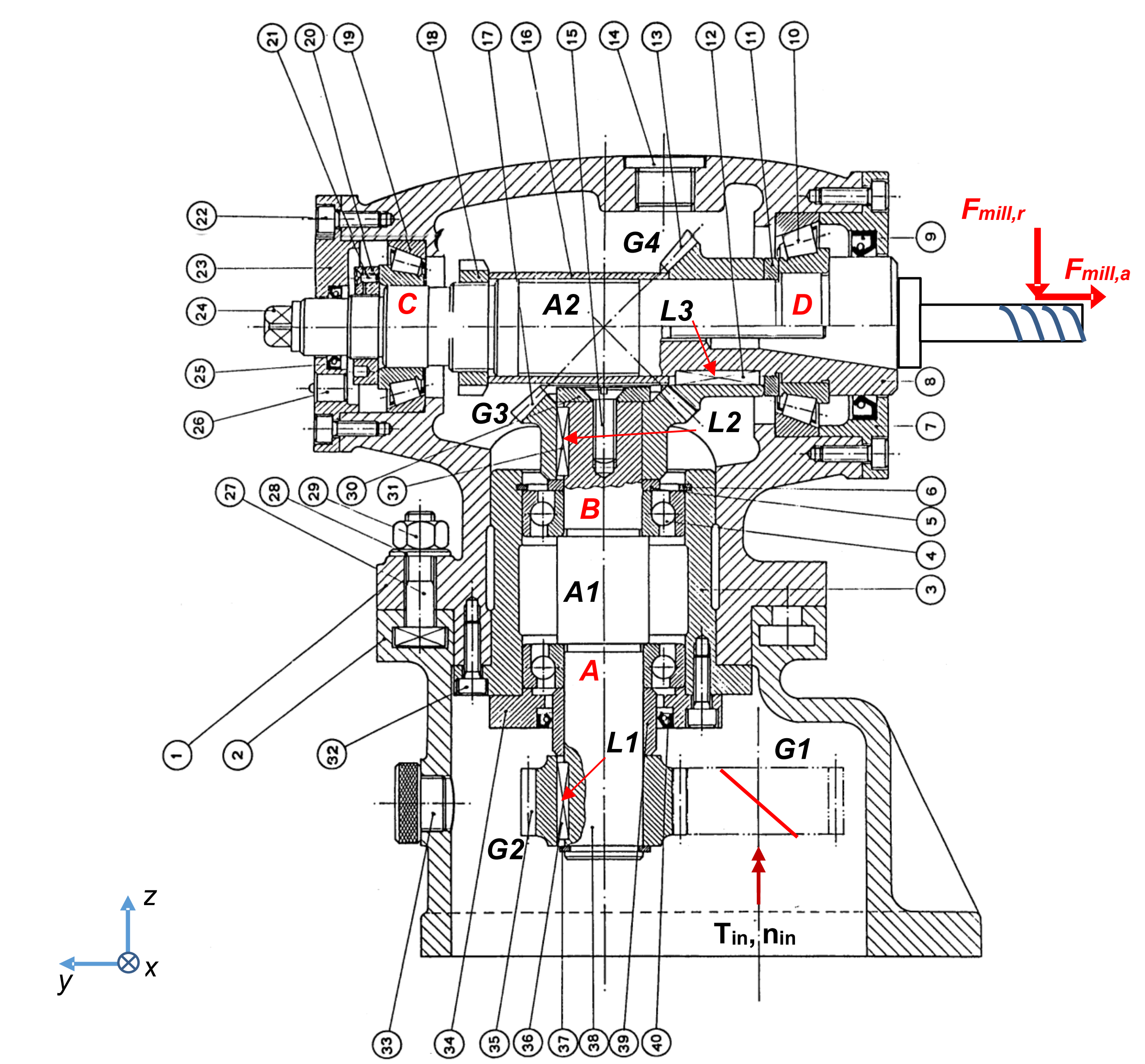


Figure 1. Gearbox drawing of a milling machine head

A1: gearbox input shaft; it receives power from the key **L1**.
A2: gearbox output shaft; it receives power from shaft **A1** by means of gear **G3** and transmits power to the user through the key **L3.**
G1-G2: helical gears.
G3-G4: bevel gears.
L1: key that connects shaft **A1** with gear **G2.**
L2: key that connects shaft **A1** with gear **G3.**
L3: key that connects shaft **A2** with gear **G4.**
$F_{mill,r}$ - $F_{mill,a}$: external forces applied to shaft **A2** by the milling tool.
A-B: angular contact ball bearings face-to-face arrangement (**SKF 7311 ACCBM**)
C-D: tapered roller bearings back-to-back arrangement (**SKF C:30311; D:30314**)

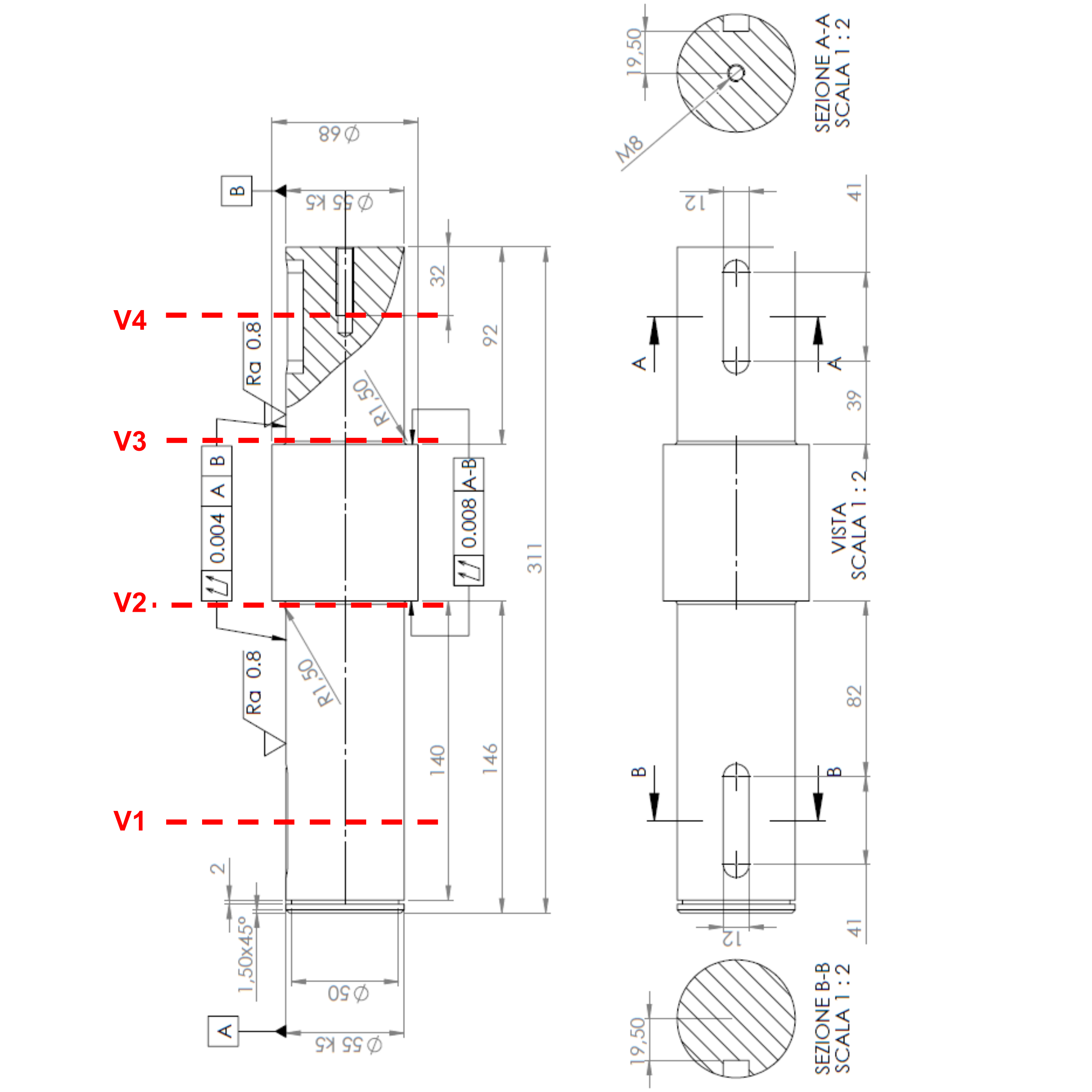


Figure 2. Shaft **A1** technical drawing

Consider:
- Section **V1** in the mid-section of the key **L1**;
- Section **V2** in the left shoulder of shaft **A1**;
- Section **V3** in the right shoulder of shaft **A1**;
- Section **V4** in the mid-section of the key **L2**;

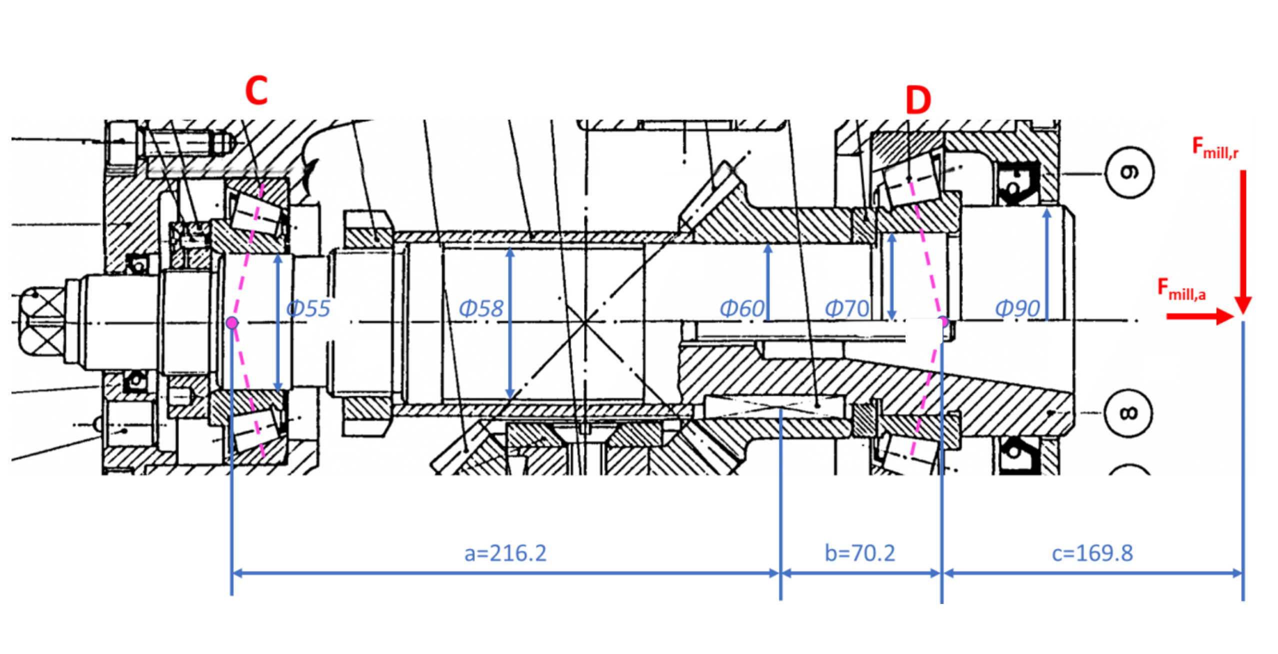


Figure 3. Shaft **A2** main dimensions

Consider:

- The forces $\mathbf{F}_{\mathbf{mill,r}}$ and $\mathbf{F}_{\mathbf{mill,a}}$ by the milling tool, are applied at distance c from the pressure centre of bearing **D**
- The values of the forces must be calculated as a function of the torque on shaft A2, as:

$$F_{mill,r} = \frac{T_{A2}}{0.0125}\ N$$

$$F_{mill,a} = 0.25 \cdot F_{mill,r}\ N$$

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 1.

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

A motor connected to **G1,** rotating at an angular speed of n_{in} rpm, provides a power P_{in} kW to shaft **A1** by mean of the engagement between **G1** and **G2** (see Annex 1 for input data). The gear ratio between **G1** and **G2** is $\tau_{12} = z_2/z_1 = 24/32$. The power is then transmitted to shaft **A2** by the engagement between two bevel gears **G3** and **G4** with a number of teeth $z_3 = z_4 = 24$.
The shaft **A1** is made of a C45 quenched and tempered steel ($\sigma_u = 780 \text{ MPa}$, $\sigma_y = 370 \text{ MPa}$, $\sigma_{D-1} = 380 \text{ MPa}$) and is supported by two angular ball bearings (SKF 7311 ACCBM) in a face-to-face configuration. Shaft **A2** is made of a 34CrNiMo6 quenched and tempered steel ($\sigma_u = 1050 \text{ MPa}$, $\sigma_y = 950 \text{ MPa}$, $\sigma_{D-1} = 520 \text{ MPa}$) and is supported by two tapered roller bearings in a back-to-back arrangement (SKF **C**:30311; **D**:30314). The four gears have the following characteristics:

	G1	**G2**	**G3**	**G4**
normal module, m_n (mm)	4	4	5	5
number of teeth, z	32	24	24	24
helix angle, ψ	21°	21°	-	-
pitch angle δ	-	-	45°	45°
normal pressure angle, ϕ_n	20°	20°	20°	20°
transmission accuracy level, Q_v	8	8	8	8
tooth face width (mm)	44	44	30	30
Ultimate strength (MPa)	1950	1950	1950	1950
Yield strength (MPa)	1450	1450	1450	1450
Hardness (HB)	525	525	525	525

Part 1

- Static verification and Fatigue analysis for infinite life for shaft **A1** in the sections shown in Figure 2
  - $\mathbf{V_1}$ in the mid-section of the key **L1**;
  - $\mathbf{V_2}$ in the left shoulder of shaft **A1**;
  - $\mathbf{V_3}$ in the right shoulder of shaft **A1**;
  - $\mathbf{V_4}$ in the mid-section of the key **L2;**

  considering the stress concentration and intensification factors where necessary.
- Bending and contact stress verification for gear **G2** 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 (A,B,C,D) 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
  - Oil lubrication (consider the ISO VG 68 lubricant for all the bearings verifications)
  - Reliability 90%

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.

SCHEME FOR THE TECHNICAL REPORT: STATIC VERIFICATION

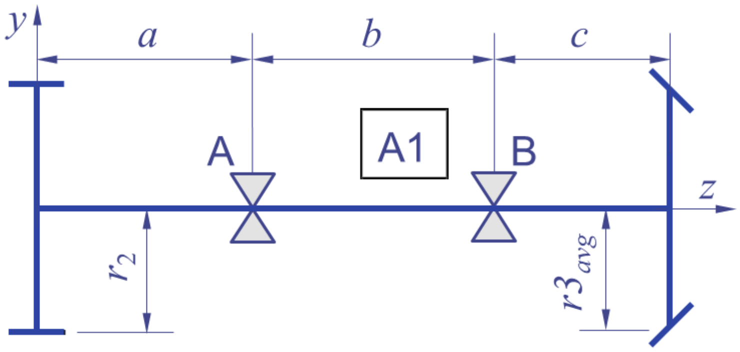
Team n.	02SXJJM Fundamentals of Machine Design date ....
	sheet n. ...../.....

Shaft A1: External forces	Shaft A2: External forces
$\begin{cases} T_{A1} = \cdots \text{Nm} \\ F_{t12} = \cdots \text{N} \\ F_{r12} = \cdots \text{N} \\ F_{a12} = \cdots \text{N} \end{cases}$	$\begin{cases} T_{A2} = \cdots \text{Nm} \\ F_{t34} = \cdots \text{N} \\ F_{r34} = \cdots \text{N} \\ F_{a34} = \cdots \text{N} \\ F_{mill,r} = \cdots \text{N} \\ F_{mill,a} = \cdots \text{N} \end{cases}$

Notes:

Assume a gear efficiency equal to 1.

Shaft A1: Reaction forces

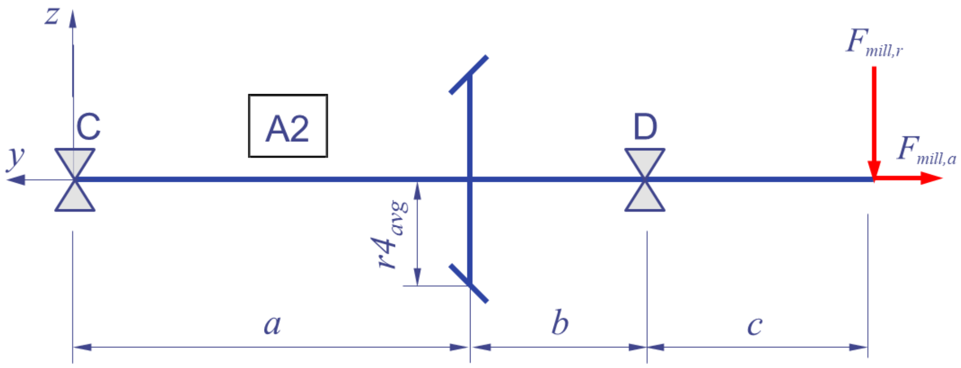


$$\begin{cases} a = \cdots \text{mm} \\ b = \cdots \text{mm} \\ c = \cdots \text{mm} \end{cases}$$

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

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

Shaft A2: Reaction forces



$$\begin{cases} a = \cdots \text{mm} \\ b = \cdots \text{mm} \\ c = \cdots \text{mm} \end{cases}$$

$$\begin{cases} R_{zC} = \cdots \text{N} \\ R_{zD} = \cdots \text{N} \end{cases}$$

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

Notes:

- Apply the reaction forces on the pressure center of the bearings.
- Forces from the gears of the system, can be considered applied on the midplane section of the corresponding key.
- For bevel gears, remember to apply forces on the average radius of the pitch cone.

Shaft A1: Axial reaction forces	Shaft A2: Axial reaction forces
$\begin{cases} R_{zA} = \cdots \text{N} \\ R_{zB} = \cdots \text{N} \end{cases}$	$\begin{cases} R_{yC} = \cdots \text{N} \\ R_{yD} = \cdots \text{N} \end{cases}$
***Notes**:* Use the SKF guidelines in the provided catalogue for the Axial force calculation on the two couple of bearings.	

Shaft A1 Internal loads	
Calculate the internal loads and the total bending moment $M_B(z) = \sqrt{M_x^2 + M_y^2}$ in the shaft A1 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 A1: 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 A1: 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 A1	
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 $SF_{s,min}$.	$\begin{cases} SF_{V1} = \cdots \\ SF_{V2} = \cdots \\ SF_{V3} = \cdots \\ SF_{V4} = \cdots \\ SF_{s,min} = \cdots \end{cases}$

SCHEME FOR THE TECHNICAL REPORT: FATIGUE VERIFICATION

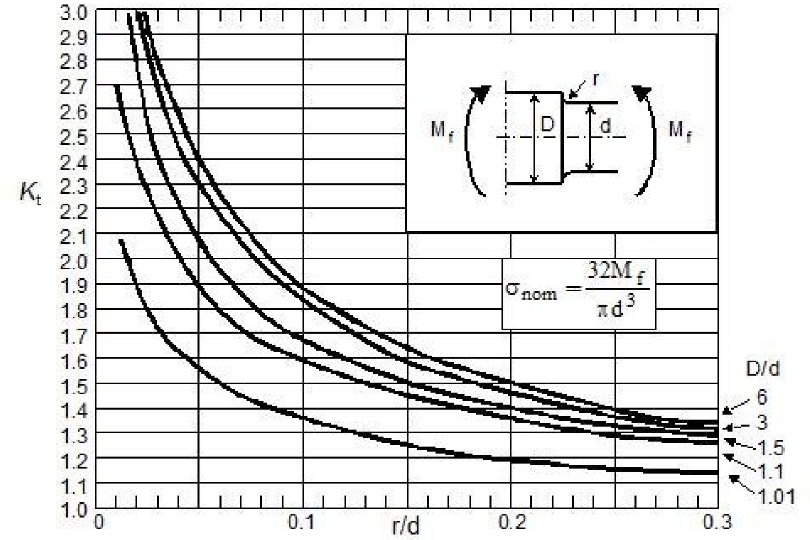
Team n.	02SXJJM Fundamentals of Machine Design date ....
	sheet n. ...../.....

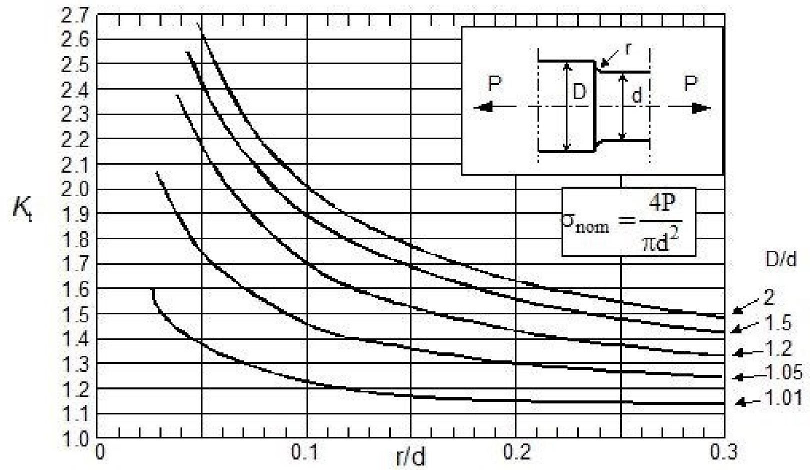
Evaluation of the fatigue stress on Shaft A1	
Evaluate, if present, the mean and alternate stress components due to the normal load, σ_m^N e σ_a^N, mean and alternate stress components due to the bending moment, $\sigma_m^{M_B}$ e $\sigma_a^{M_B}$, and the mean and alternate stress components due to 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.	$\begin{cases} \sigma_{a,eq} = \cdots \text{MPa} \\ \sigma_{m,eq} = \cdots \text{MPa} \end{cases}$ $SF_f = \cdots$

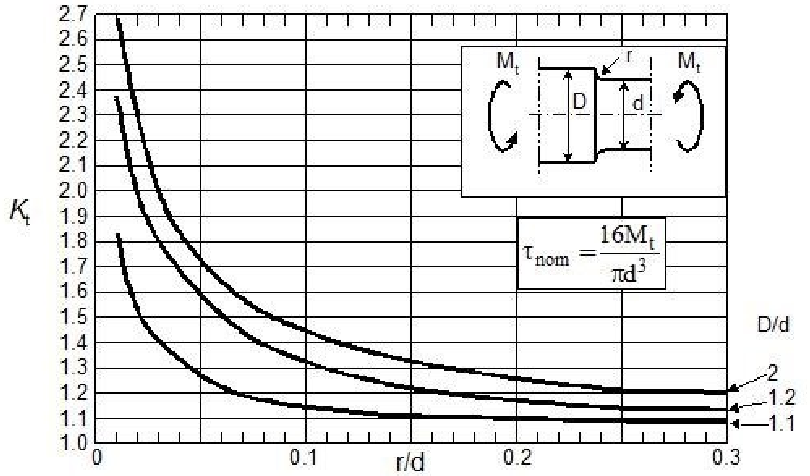
Reminder:

- **repeat the same procedure for sections V1, V2, V3 and V4**
- **Use the diagrams in the following pages for the coefficients evaluation.**
- **Use the Shigley Method to calculate the alternate and average equivalent stress components of the working point.**

Geometrical stress concentration factors Kt

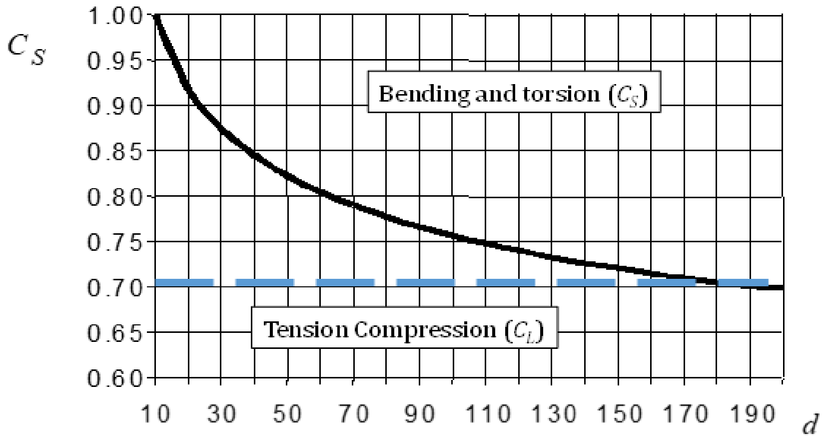




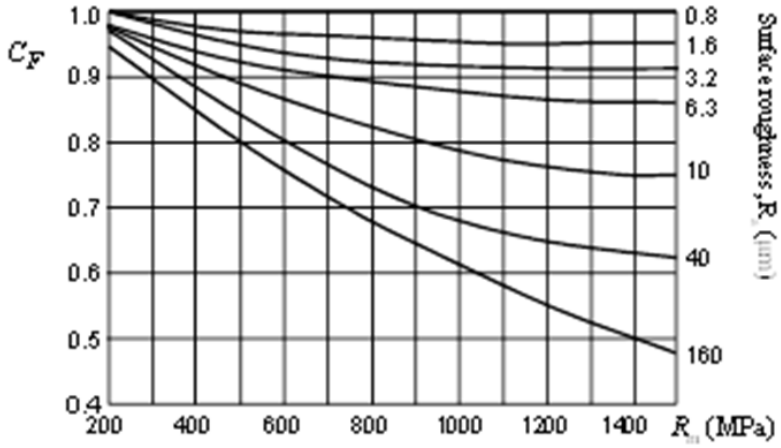


	Bending	Torsion
K_f for the keyseat	1.6	2.0

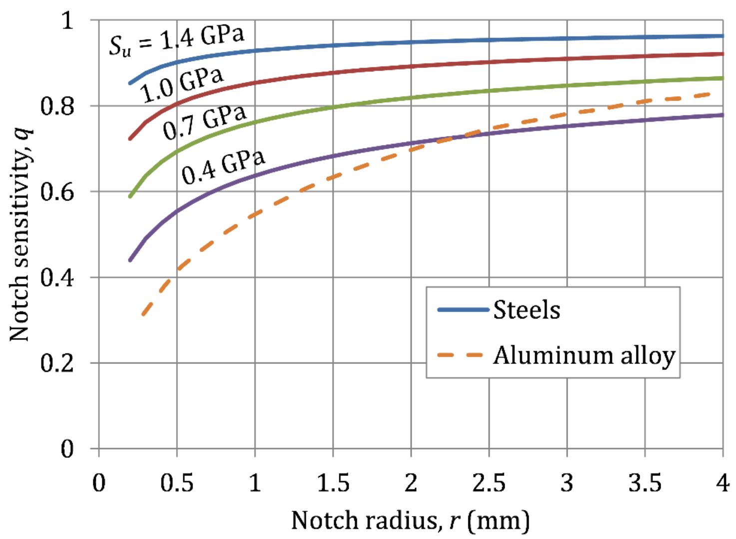
Size Effect Cs



Surface finish effect



Notch sensitivity q



SCHEME FOR THE TECHNICAL REPORT: GEAR TOOTH VERIFICATION

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

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

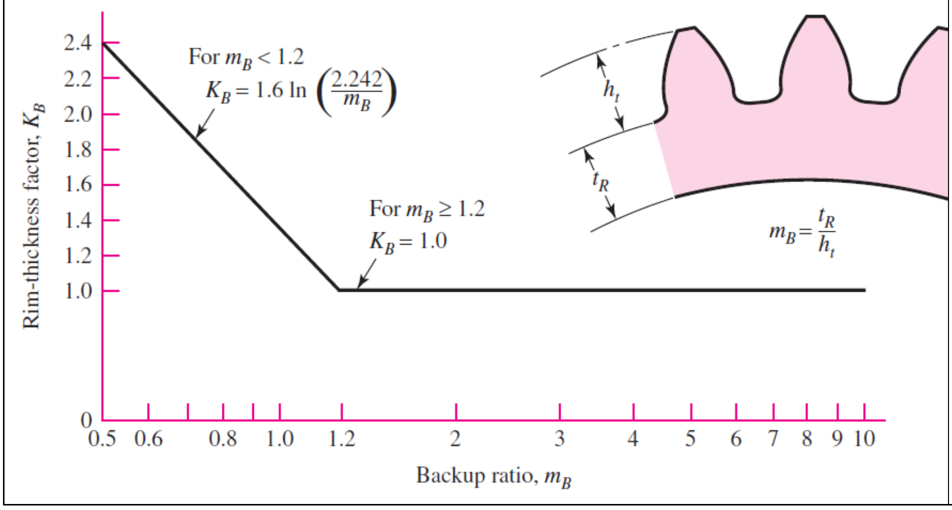
Overload factor K_0

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

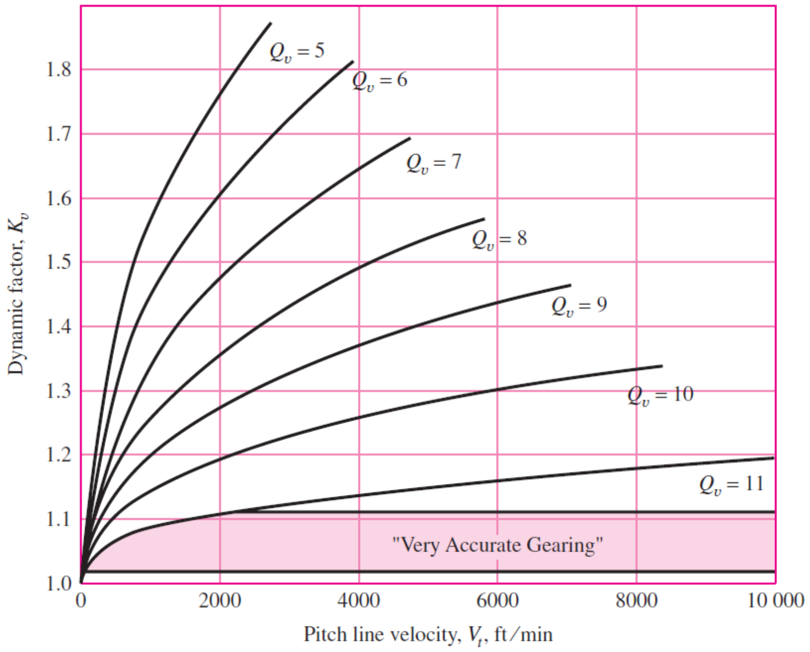
$K_0 = \cdots$

Rim-thickness factor K_B



$K_B = \cdots$

Dynamic factor K_v



$K_v = \ldots$

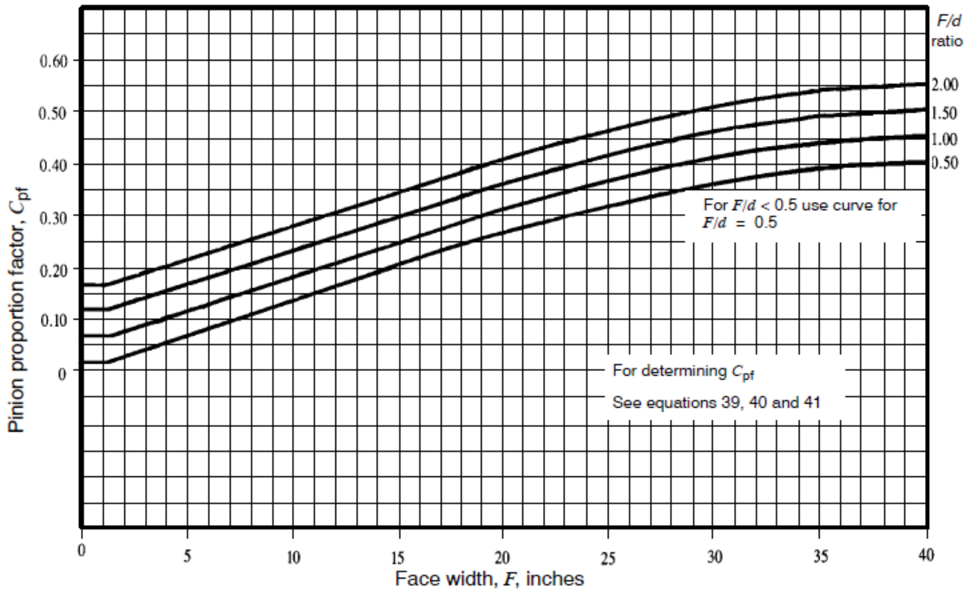
where Q_v is the transmission accuracy-level number and V_t is the peripheral linear speed of the coupling expressed in ft/min

Load distribution factor K_H

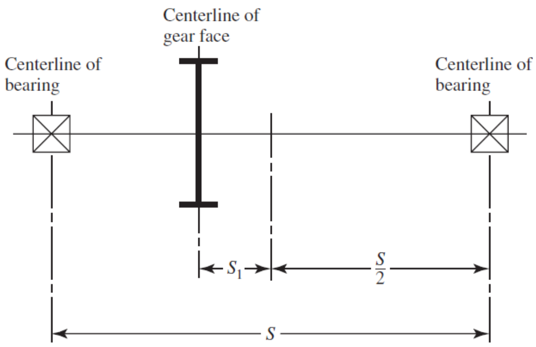
$$K_H = 1 + C_{mc}\left(C_{pf}C_{pm} + C_{ma}C_e\right),\ \text{where } C_e = 1.$$

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

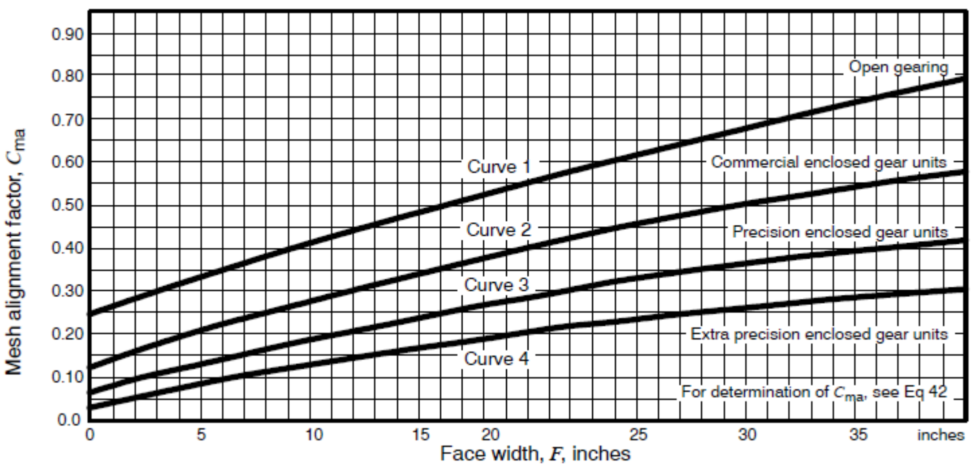
$$C_{mc} = \begin{cases} 1 & \text{for uncrowned teeth} \\ 0{,}8 & \text{for crowned teeth} \end{cases}$$



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

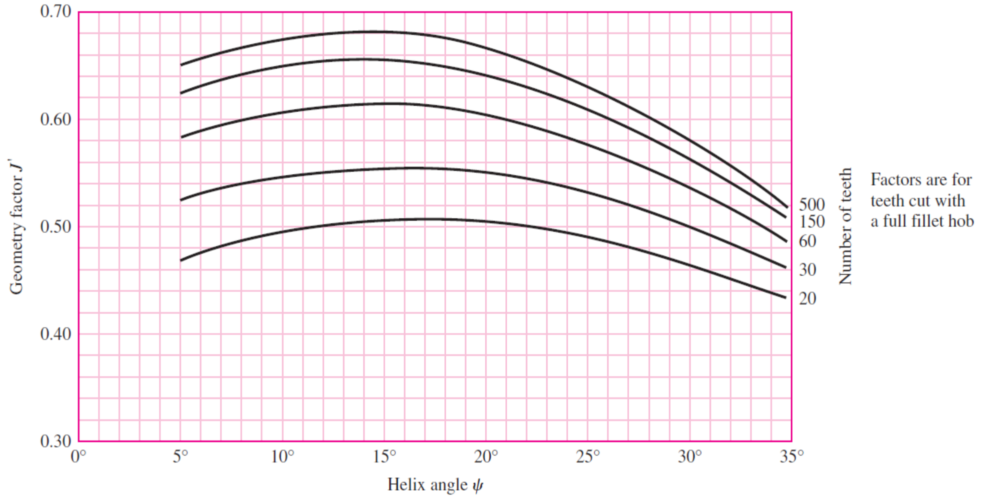
$$K_s = 0,843\left(b \cdot m_t\sqrt{Y}\right)^{0,0535}$$

Express b and mt in mm

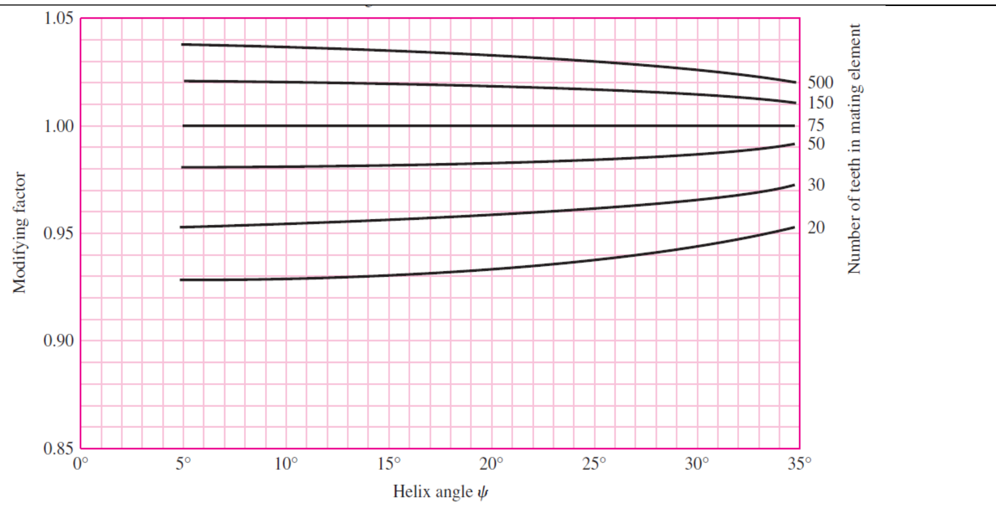
For helical gears

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

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



J'



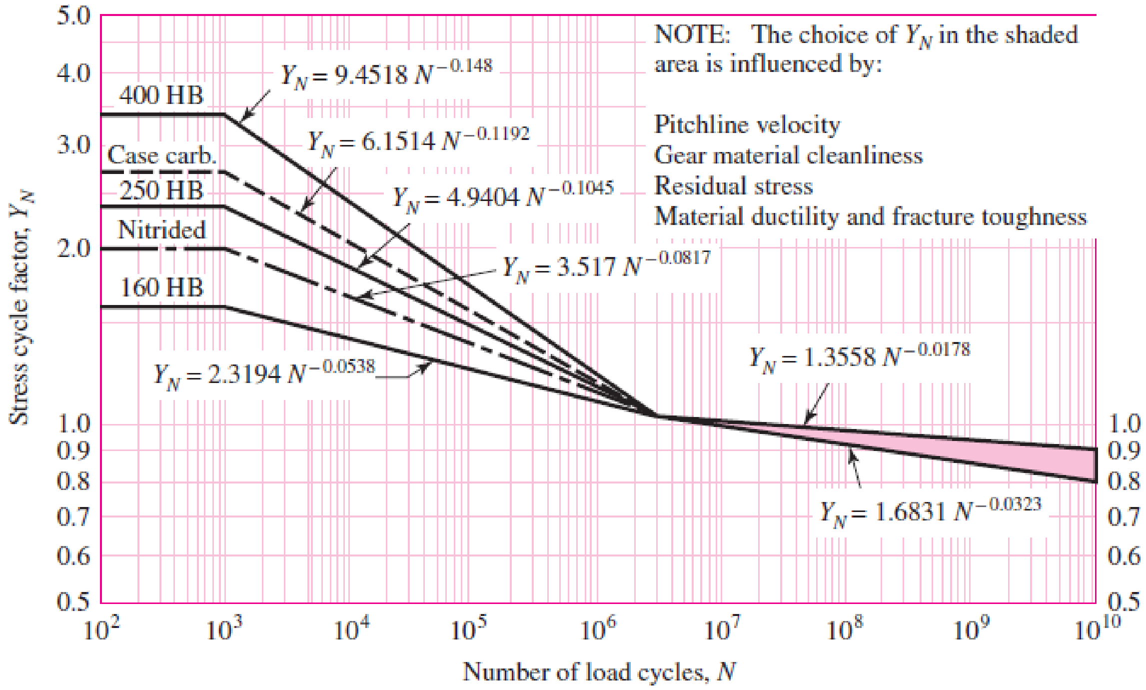
J''

Bending safety factor

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

Bending fatigue strength σ_{FP} ($S_{f,lim}$) = 840 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 $\tan[\phi_t] = \frac{\tan[\phi_n]}{\cos[\psi]}$.

$$Z_I = \begin{cases} \dfrac{\cos[\phi_t]\mathrm{sen}[\phi_t]}{2m_N} \cdot \dfrac{m_G}{m_G+1} & \text{external meshing} \\ \dfrac{\cos[\phi_t]\mathrm{sen}[\phi_t]}{2m_N} \cdot \dfrac{m_G}{m_G-1} & \text{internal meshing} \end{cases}$$

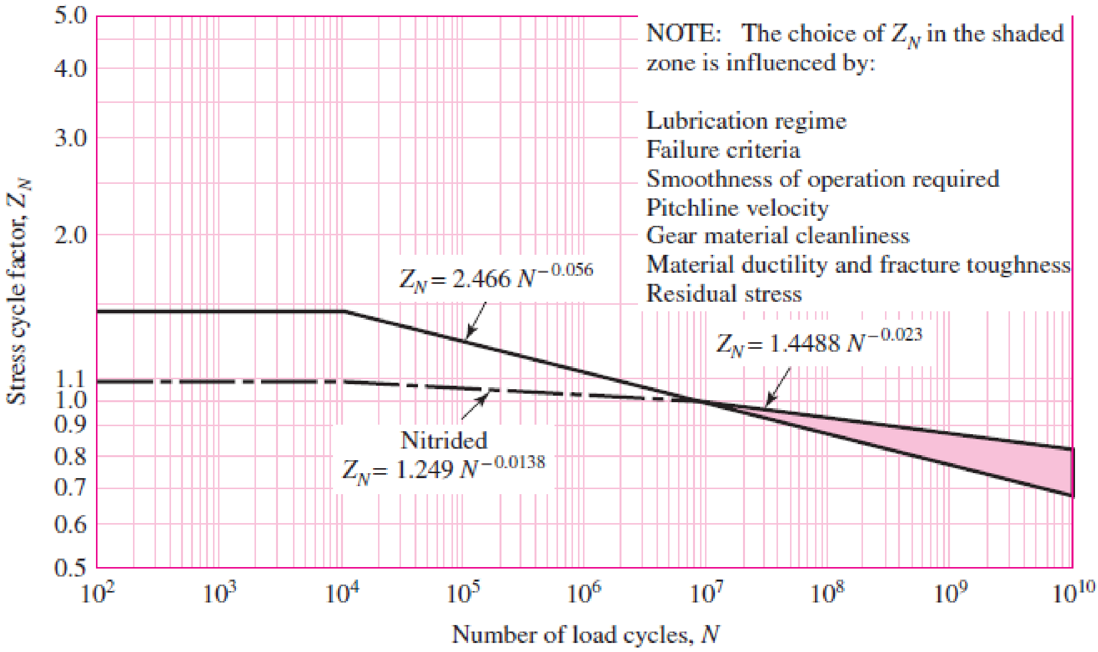
where $m_G = n_\mathbf{P}/n_\mathbf{G} = d_\mathbf{G}/d_\mathbf{P}$ is the gear ratio.

Wear safety factor (Hertzian contact)

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

Contact strength σ_{HP} *(S_c) = 1230 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 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
$$k = \frac{\nu}{\nu_1},$$ with: • ν lubricant viscosity at the given operating temperature • ν_1 minimum required lubricant viscosity for the given working conditions
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 k 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 SKF catalogue to find relevant tables and diagrams and for the equivalent load definitions**.
- **Repeat the verification for bearings on Shaft A1 and Shaft A2**

Annex 1: Group input data for gearbox analysis

Group No.	**Input Power P_{in}** (kW)	**Input speed n_{in}** (rpm)
1	33,2	1500
2	10,6	750
3	29	1000
4	32,5	1500
5	25	1000
6	16,7	750
7	15,9	750
8	32,8	1500
9	15	750
10	12,6	750
11	30,6	1500
12	38,6	1500
13	10,1	750
14	25,4	1000
15	34,4	1500
16	28,4	1000
17	31,7	1500
18	18,7	750
19	37,6	1500
20	31,5	1500
21	26,3	1000
22	14,2	750
23	21,2	1000
24	30,2	1500
25	23,2	1000
26	23	1000
27	28,5	1000
28	29,5	1000
29	28	1000
30	34,2	1500
31	25,7	1000
32	37,3	1500
33	19,6	750
34	12,7	750
35	19	750
36	13,4	750
37	34,9	1500
38	11,4	750
39	28,8	1000
40	26,4	1000

SUBMISSION TABLE FOR KEY CALCULATION RESULTS

	VALUE	UNIT OF MEASUREMENT
SHAFT ANALYSIS		
T_{A1}		Nm
$R_{x,A}$		N
$R_{y,A}$		N
$R_{x,B}$		N
$R_{y,B}$		N
$R_{z,A}$		N
$R_{z,B}$		N
Mx(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
$\sigma_{a,b\ nom}$ (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		-

- 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		
$Z_E \sqrt{\frac{F_t}{b \cdot d_{\boldsymbol{P}}} \cdot \frac{1}{Z_I}}$		MPa
$\sigma_{\max,pitting}$		MPa
S_H		-

	VALUE	UNIT OF MEASUREMENT
BEARING ANALYSIS		
$\mathbf{F_{r,A}}$		kN
$\mathbf{F_{a,A}}$		kN
$\mathbf{F_{r,B}}$		kN
$\mathbf{F_{a,B}}$		kN
$\mathbf{P_A}$		kN
$\mathbf{P_B}$		kN
$\mathbf{F_{r,C}}$		kN
$\mathbf{F_{a,C}}$		kN
$\mathbf{F_{r,D}}$		kN
$\mathbf{F_{a,D}}$		kN
$\mathbf{P_C}$		kN
$\mathbf{P_D}$		kN
$\mathbf{a_{skf,A}}$		-
$\mathbf{a_{skf,B}}$		-
$\mathbf{a_{skf,C}}$		-
$\mathbf{a_{skf,D}}$		-
$L_{10mh,A}$		Hours
$L_{10mh,B}$		Hours
$L_{10mh,C}$		Hours
$L_{10mh,D}$		Hours