UNIVERSITI KUALA LUMPUR
KAMPUS CAWANGAN MALAYSIAN SPANISH INSTITUTE

FINAL EXAMINATION

OCTOBER 2025 SEMESTER

COURSE CODE	:	SCB35603 (V2)
COURSE TITLE	:	MACHINE COMPONENT DESIGN
PROGRAMME NAME	:	BACHELOR OF ENGINEERING TECHNOLOGY (HONS) IN MECHANICAL DESIGN
DATE	:	29 JANUARY 2026
TIME	:	9:00AM - 12:00PM
DURATION	:	3 HOURS

INSTRUCTIONS TO CANDIDATES

1. Please read the instructions given in the question paper CAREFULLY.
2. This question paper is printed on both sides of the paper.
3. This question paper consist of TWO sections.
4. Answer ALL questions for Section A.
5. Section B consist of four questions. Answer THREE (3) questions only.
6. Please write your answer on the answer booklet provided.
7. Please answer all questions in English only.
8. Please answer MCQ/EMQ questions using OMR sheet. ☐ *Tick if applicable*
9. Refer to the attached Formula/ Appendies. ☑ *Tick if applicable*

THERE ARE 10 PAGES OF QUESTIONS INCLUDING THIS PAGE

SECTION A (Total: 40 marks)

Answer ALL questions.

Please use the answer booklet provided.

Question 1

Theories of failure are used by engineers and designers to predict how a structure will fail under various loads. These theories help in predicting the capacity of a material to stand against the unlimited combinations of non-standard loads.

(a) Explain how the Distortion Energy Theory (Von Mises) works and its suitability for predicting yielding in ductile materials compared to the Maximum Normal Stress Theory.

(10 marks)

(b) A power transmission shaft failed after more than 10,000 hours of operation. Based on the figure below:
Refer Below - Figure1 : Shaft Failure .

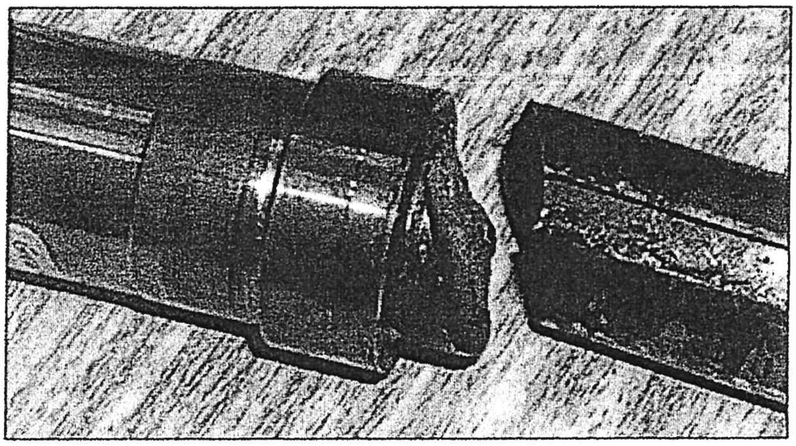


Figure 1: Shaft Failure

i. Identify a design factor that may have contributed to the failure.

(4 marks)

ii. Suggest a design improvement to enhance the shaft's fatigue life.

(6 marks)

Question 2

A power transmission shaft as shown in figure below is a mechanical component used to transmit torque and rotational motion from one part of a machine to another. It is a key element in many types of machinery and vehicles, serving as the mechanical link between a power source (like an engine or motor) and the driven component (like gears, wheels, pulleys, or gearboxes).

Refer Below - Figure2 : Transmission System .

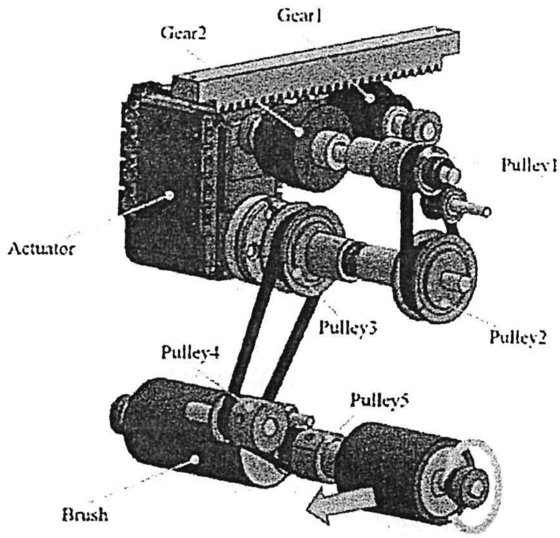


Figure 2: Transmission System

(a) Designing power transmission systems requires understanding of the components features and details. Sketch gear pressure angle diagram and describe three design considerations for gear drive.

(8 marks)

(b) Discuss the three important parameters necessary for V-belts drive for power transmission design.

(6 marks)

(c) A key and the keyway make up a keyed joint to secure the hub and the shaft to prevent relative movement between a power transmitting shaft and an attached component. For example, gear drives, pulleys or sprockets are connected securely using keys to the power transmitting shaft. Draw and describe a tangent key and explain its primary advantages over other types of keys.

(6 marks)

SECTION B (Total: 60 marks)

Answer THREE (3) questions only.

Please use the answer booklet provided.

Question 1

A power screw is a type of mechanical device used to convert rotational motion into linear motion, typically to transmit power or apply force. It is commonly used in lifting, clamping or positioning applications. The square thread power screw in figure below has a major diameter of 32 mm and a pitch of 4 mm with double threads. The provided parameter are: coefficient of friction, $f = f_c = 0.08$, collar diameter, $d_c = 40$ mm and force, $F = 6.4$ kN per screw.

Refer Below - Figure3 : Power Screw.

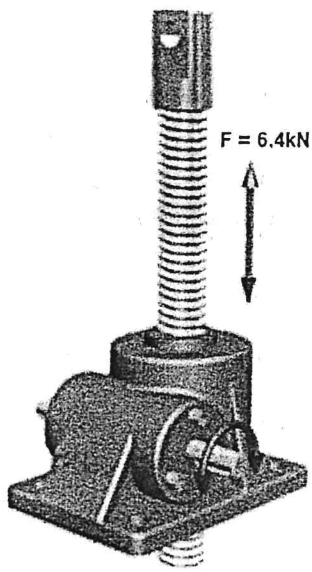


Figure 3: Power Screw

(a) Sketch square thread schematic diagram. Examine the thread depth, thread width, pitch diameter and lead.

(6 marks)

(b) Calculate the torque required to raise and lower the load.

(8 marks)

(c) Calculate the efficiency while lifting the load.

(2 marks)

(d) Calculate the body stresses and torsional.

(4 marks)

Question 2

In transmission design, a shaft is a rotating component that transmits power and torque between mechanical elements, while bearings support the shaft, reduce friction and ensure smooth, aligned rotation under various loads.

(a) A mild steel shaft is designed to transmit 100 kW of power at 300 rpm over a supported length of 3 meters. It supports two pulleys, each weighing 1500 N, positioned 1 meter from each end. Given a permissible shear stress of 60 MPa, sketch the free body diagram and calculate the required shaft diameter.

(12 marks)

(b) An SKF 6210 angular-contact ball bearing has an axial load Fa of 1780 N and a radial load Fr of 2225 N applied with the outer ring stationary. The basic static load rating C0 is 4450 N and the basic load rating C10 is 35155 N. Estimate the L10 life at a speed of 720 rev/min. (Note: 1lbf = 4.45N and refer Appendix 1 for e and Y2 value)

(8 marks)

Question 3

Gears and bearings are critical in mechanical design because gears enable precise transmission of power, torque, and speed between rotating shafts, often with changes in direction or magnitude, while bearings support these rotating elements, reduce friction, and ensure smooth, aligned motion, which together contribute to the efficiency, durability, and reliability of the overall mechanical system.

(a) A gearset consists of a 16-tooth pinion driving a 40-tooth gear. The diametral pitch is 2, and the addendum and dedendum are 1/P and 1.25/P, respectively. The gears are cut using a pressure angle of 20°. (1 inch = 25.4 mm)

i. Compute the circular pitch, the center distance and the radius of the base circles.

(4 marks)

ii. In mounting these gears, the center distance was incorrectly made 1/4 inch (25.4 mm) larger, compute the new pitch circle diameter for gear and pinion.

(4 marks)

iii. Compute the new values of the pressure angle.

(2 marks)

(b) A 50mm diameter shaft in figure below is supported by SKF bearing at both ends. The shaft carries a load of 10 kN at its center. The axial load on the bearing is 3 kN. The shaft is connected to an electric motor that rotates at 1440 rpm. Determine the suitable bearing for 1000 hours of operation.
Refer Below - Figure4 : Bearing Assembly.

(10 marks)

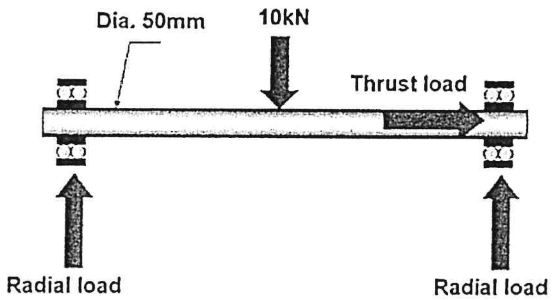


Figure 4: Bearing Assembly

Question 4

The theory of failures is important in component analysis because it ensures that engineered systems and structures perform safely and reliably under real-world conditions. Failure theories help predict *where*, *how*, and *when* a material or part is likely to fail based on applied stresses and material properties. A rotating shaft with a notch is more likely to fail at the notch due to stress concentration. The theory helps engineers reinforce or redesign that weak point.

(a) The cantilevered tube shown in figure below is to be made of 2014 aluminum alloy treated to obtain specified minimum yield strength of 276 MPa. We wish to select a size tube of 42 mm diameter and 5 mm thickness using a design factor n_d = 4. The bending load is F = 1.75 kN, the axial tension is P = 9.0 kN, and the torsion is T = 72 N.m. What is the realized factor of safety when using Distortion Energy Theory (DET)?

Refer Below - Figure5 : Cantilever Tube .

(10 marks)

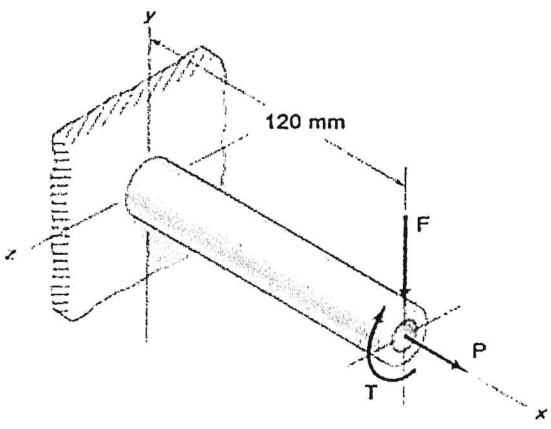
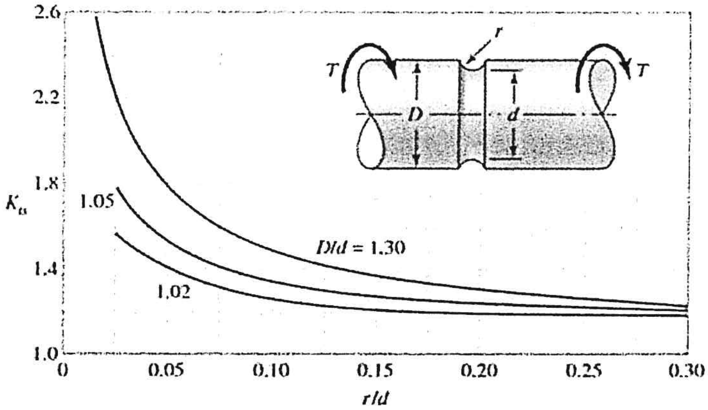


Figure 5: Cantilever Tube

(b) A solid circular bar with a diameter of 25 mm has a machined groove that is 2.5 mm deep and has a radius of 2.5 mm as illustrated in the figure below. The bar is made from AISI 1020 cold-drawn (CD) steel which has a shear yield strength given by $S_{su} = 0.67S_{ut}$. It is subjected to a fully reversed (purely alternating) torque of 200 N.m. For the S-N curve of this material, use the following parameters: fatigue strength reduction factor f = 0.9, theoretical shear stress concentration factor K_{ts} = 1.4 and notch sensitivity in shear q_{shear} = 0.94. Determine the number of cycles to failure.
Refer Below - Figure6 : Shaft component .

(10 marks)



Round bar profile:
d = 20 mm
D = 25 mm
r = 2.5 mm

Figure 6: Shaft component

END OF EXAMINATION PAPER

APPENDIX

APPENDIX

Mohr's Circle

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}{}^2}$$

$$\tau_1, \tau_2 = \pm\sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}{}^2}$$

$$\frac{\sigma_1}{S_t} - \frac{\sigma_3}{S_c} = \frac{1}{n}$$

Maximum tensile stress,

$$\sigma_{t(max)} = \frac{\sigma_t}{2} + \frac{1}{2}\left[\sqrt{(\sigma_t)^2 + 4\,\tau^2}\right]$$

Maximum compressive stress,

$$\sigma_{c(max)} = \frac{\sigma_c}{2} + \frac{1}{2}\left[\sqrt{(\sigma_c)^2 + 4\,\tau^2}\right]$$

Maximum shear stress,

$$\tau_{max} = \frac{1}{2}\left[\sqrt{(\sigma_t)^2 + 4\,\tau^2}\right]$$

Static & Fatique Failure Theories

$$\frac{\sigma_1}{S_{ut}} = \frac{1}{n} \quad \text{or} \quad \frac{\sigma_2}{-S_{uc}} = \frac{1}{n}$$

$$\tau_{\max} = \frac{\sigma_1 - \sigma_2}{2} = \frac{S_y}{2n}$$

$$\sigma^e = \sqrt{\sigma_1{}^2 + \sigma_2{}^2 - \sigma_1\sigma_2} = \frac{S_y}{n}$$

$$\sigma' = \left(\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2\right)^{1/2}$$

$$\sigma' = \frac{S_y}{n}$$

$$S'_e = \begin{cases} 0.5S_{ut} & S_{ut} \le 200 \text{ kpsi (1400 MPa)} \\ 100 \text{ kpsi} & S_{ut} > 200 \text{ kpsi} \\ 700 \text{ MPa} & S_{ut} > 1400 \text{ MPa} \end{cases}$$

$$S_e = k_a k_b k_c k_d k_e k_f S'_e$$

Surface factor (k_a) .... $k_a = aS_{ut}^b$

Table 6–2

Parameters for Marin Surface Modification Factor, Eq. (6–19)

Surface Finish	Factor a S_{ut}, kpsi	Factor a S_{ut}, MPa	Exponent b
Ground	1.34	1.58	–0.085
Machined or cold-drawn	2.70	4.51	–0.265
Hot-rolled	14.4	57.7	–0.718
As-forged	39.9	272.	–0.995

Size factor (k_b)

$$k_b = \begin{cases} (d/0.3)^{-0.107} = 0.879d^{-0.107} & 0.11 \le d \le 2 \text{ in} \\ 0.91d^{-0.157} & 2 < d \le 10 \text{ in} \\ (d/7.62)^{-0.107} = 1.24d^{-0.107} & 2.79 \le d \le 51 \text{ mm} \\ 1.51d^{-0.157} & 51 < d \le 254 \text{ mm} \end{cases}$$

Load factor (k_c)

$$k_c = \begin{cases} 1 & \text{bending} \\ 0.85 & \text{axial} \\ 0.59 & \text{torsion}^{17} \end{cases}$$

Temperature factor (k_d)

Table 6–4

Effect of Operating Temperature on the Tensile Strength of Steel.* (S_T = tensile strength at operating temperature; S_{RT} = tensile strength at room temperature; $0.099 \le \hat{\sigma} \le 0.110$)

Temperature, °C	S_T/S_{RT}	Temperature, °F	S_T/S_{RT}
20	1.000	70	1.000
50	1.010	100	1.008
100	1.020	200	1.020
150	1.025	300	1.024
200	1.020	400	1.018
250	1.000	500	0.995
300	0.975	600	0.963
350	0.943	700	0.927
400	0.900	800	0.872
450	0.843	900	0.797
500	0.768	1000	0.698
550	0.672	1100	0.567
600	0.549		

Reliability factor (k_e)

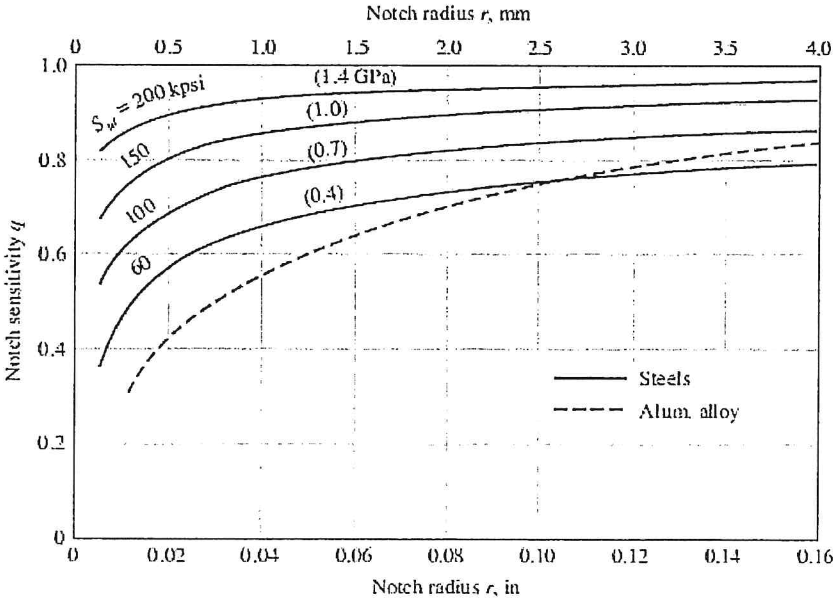
Table 6–5

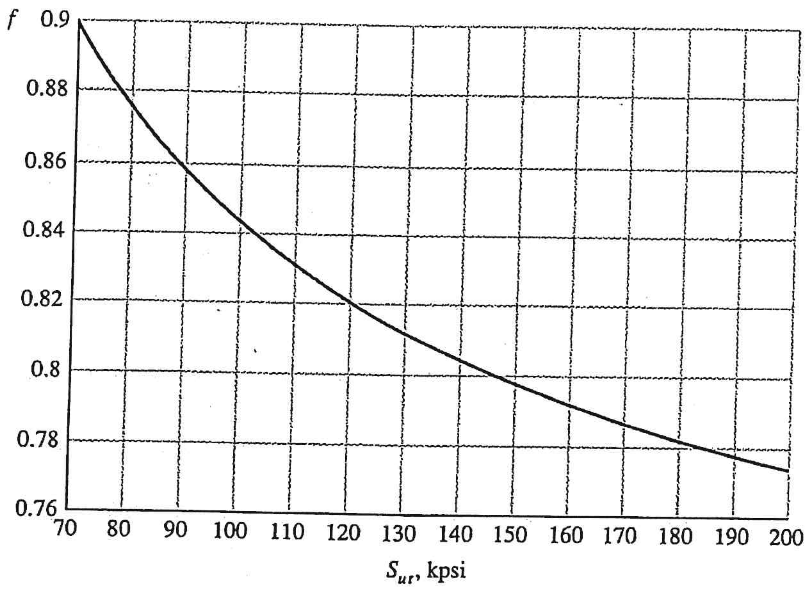
Reliability Factors k_e Corresponding to 8 Percent Standard Deviation of the Endurance Limit

Reliability, %	Transformation Variate z_a	Reliability Factor k_e
50	0	1.000
90	1.288	0.897
95	1.645	0.868
99	2.326	0.814
99.9	3.091	0.753
99.99	3.719	0.702
99.999	4.265	0.659
99.9999	4.753	0.620

Figure 6–20

Notch-sensitivity charts for steels and UNS A92024-T wrought aluminum alloys subjected to reversed bending or reversed axial loads. For larger notch radii, use the values of q corresponding to the $r = 0.16$-in (4-mm) ordinate. *(From George Sines and J. L. Waisman (eds.),* Metal Fatigue, *McGraw-Hill, New York. Copyright © 1969 by The McGraw-Hill Companies, Inc. Reprinted by permission.)*





Graph: Fatigue Fraction factor

Fatigue life calculation

$$N = \left(\frac{S_f}{a}\right)^{1/b} \qquad a = \frac{(fS_{ut})^2}{S_e}$$

$$b = -\frac{1}{3}\log\left(\frac{fS_{ut}}{S_e}\right)$$

For shearing

$$N = \left(\frac{S_f}{a}\right)^{1/b} \qquad a = \frac{(fS_{su})^2}{S_e}$$

$$b = -\frac{1}{3}\log\left(\frac{fS_{su}}{S_e}\right)$$

Soderberg $\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_y} = \frac{1}{n}$ **mod-Goodman** $\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n}$ **Gerber** $\frac{n\sigma_a}{S_e} + \left(\frac{n\sigma_m}{S_{ut}}\right)^2 = 1$

Shaft Design

$$q = \frac{K_f - 1}{K_t - 1} \quad \text{or} \quad q_{\text{shear}} = \frac{K_{fs} - 1}{K_{ts} - 1}$$

$$K_f = 1 + q(K_t - 1) \quad \text{or} \quad K_{fs} = 1 + q_{\text{shear}}(K_{ts} - 1)$$

Figure A-15-7

Round shaft with shoulder fillet in tension. $\sigma_0 = F/A$, where $A = \pi d^2/4$.

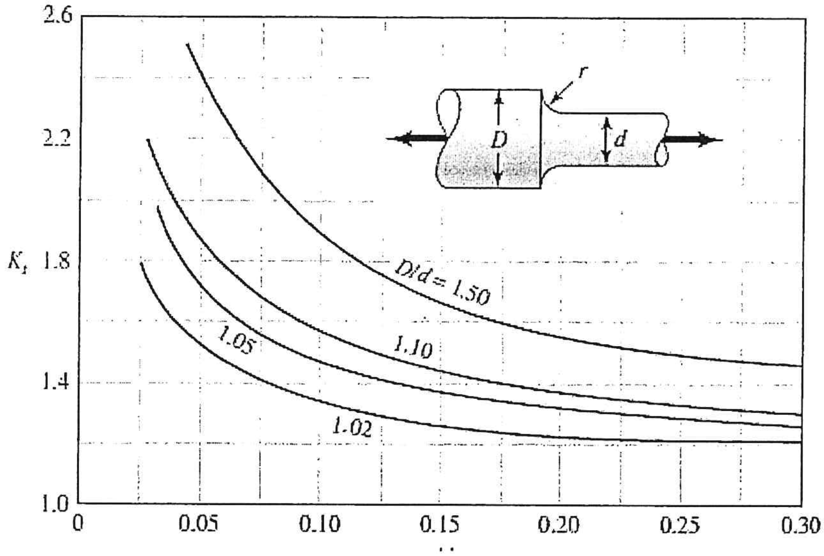


Figure A-15-8

Round shaft with shoulder fillet in torsion. $\tau_0 = Tc/J$, where $c = d/2$ and $J = \pi d^4/32$.

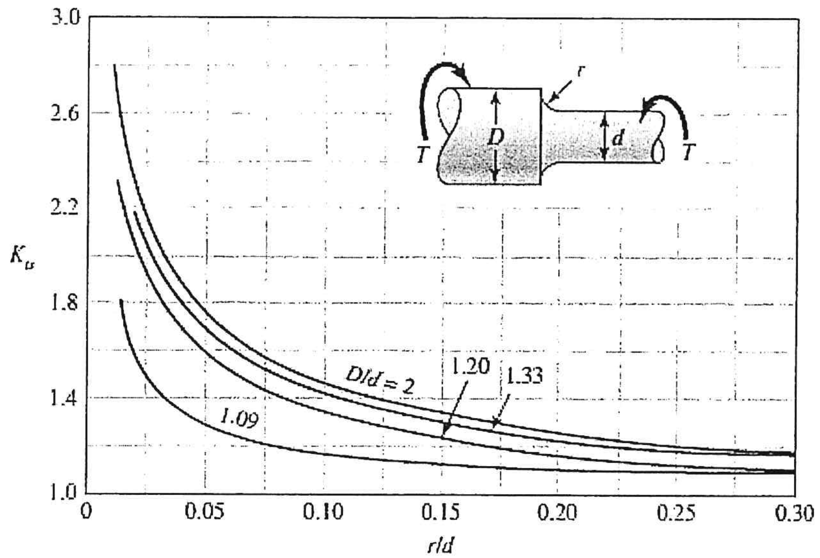
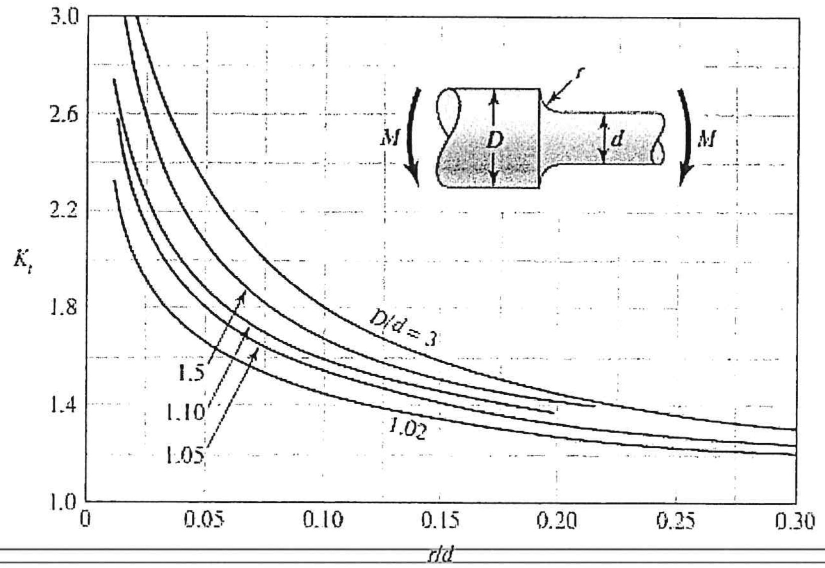


Figure A-15-9

Round shaft with shoulder fillet in bending. $\sigma_0 = Mc/I$, where $c = d/2$ and $I = \pi d^4/64$.



Fasteners & Joints Design

Figure 8–2

Basic profile for metric M and MJ threads.
d = major diameter
d_r = minor diameter
d_p = pitch diameter
p = pitch
$H = \frac{\sqrt{3}}{2} p$

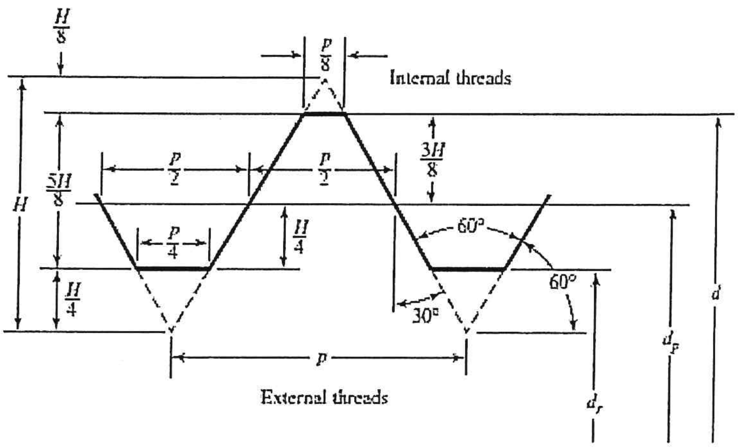
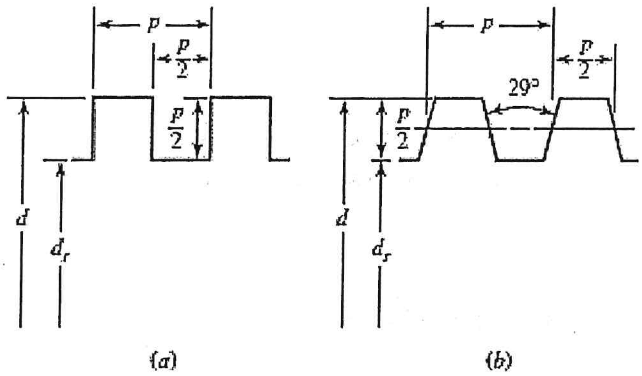


Figure 8–3

(a) Square thread; (b) Acme thread.

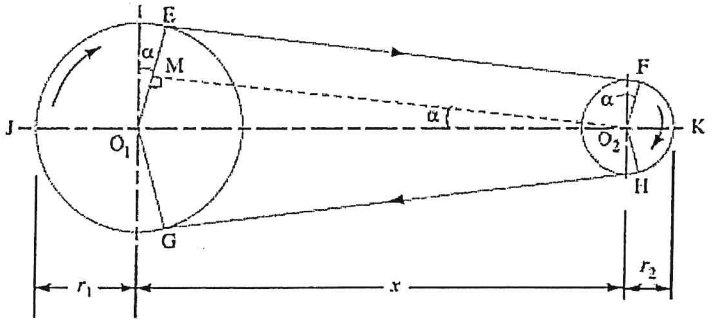


Lead, l = np

$$T_c = \frac{F f_c d_c}{2} \qquad T_R = \frac{F d_m}{2}\left(\frac{l + \pi f d_m}{\pi d_m - f l}\right) \qquad T_L = \frac{F d_m}{2}\left(\frac{\pi f d_m - l}{\pi d_m + f l}\right)$$

$$e = \frac{F l}{2\pi T_R} \qquad \tau = \frac{16 T_R}{\pi d_r^3} \qquad \sigma = -\frac{4F}{\pi d_r^2}$$

Puley & Belt



Torque $_{(driving)} = (T_1 - T_2).r_1$ *Torque* $_{(driven)} = (T_1 - T_2).r_2$

The relation between the tight side and slack side tensions,

The flat belt as

$$\log_e\left(\frac{T_1}{T_2}\right) = \mu.\theta \quad \text{or} \quad \frac{T_1}{T_2} = e^{\mu.\theta} \quad \text{or} \quad 2.3\log\left(\frac{T_1}{T_2}\right) = \mu.\theta$$

The V-belt as

$$2.3\log\left(\frac{T_1}{T_2}\right) = \mu.\theta \operatorname{cosec}\beta$$

Centrifugal force acting on the belt,

$$F_C = m.r.d\theta \times \frac{v^2}{r} = m.d\theta.v^2$$

Centrifugal tension,

$$T_C = m.v^2$$

Number of V-belts

$$= \frac{\text{Total power transmitted}}{\text{Power transmitted per belt}}$$

Power transmitted,

$$P = (T_1 - T_2)\,v \times n$$

equivalent twisting moment,

$$T_e = \sqrt{(K_m \times M)^2 + (K_t \times T)^2}$$

Rolling Contact Bearing

$$C_{10}(L_R n_R 60)^{1/a} = F_D(L_D n_D 60)^{1/a}$$

- C_{10}: catalog rating, lbf or kN
- L_R: rating life in hours
- n_R: rating speed, rev/min
- F_D: desired radial load, lbf or kN
- L_D: desired life, hours
- n_D: desired speed, rev/min

Solving for C_{10} gives

$$R = \exp\left[-\left(\frac{x - x_0}{\theta - x_0}\right)^b\right]$$

$$C_{10} \doteq F_D\left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{1/b}}\right]^{1/a} \qquad R \geq 0.90$$

$$F_e = X_i V F_r + Y_i F_a$$

$$C_{10} = F_D\left(\frac{L_D n_D 60}{L_R n_R 60}\right)^{1/a}$$

Table 11-1

Equivalent Radial Load Factors for Ball Bearings

F_a/C_0	e	$F_a/(VF_r) \le e$ X_1	Y_1	$F_a/(VF_r) > e$ X_2	Y_2
0.014*	0.19	1.00	0	0.56	2.30
0.021	0.21	1.00	0	0.56	2.15
0.028	0.22	1.00	0	0.56	1.99
0.042	0.24	1.00	0	0.56	1.85
0.056	0.26	1.00	0	0.56	1.71
0.070	0.27	1.00	0	0.56	1.63
0.084	0.28	1.00	0	0.56	1.55
0.110	0.30	1.00	0	0.56	1.45
0.17	0.34	1.00	0	0.56	1.31
0.28	0.38	1.00	0	0.56	1.15
0.42	0.42	1.00	0	0.56	1.04
0.56	0.44	1.00	0	0.56	1.00

*Use 0.014 if $F_a/C_0 < 0.014$.

Table 11-2

Dimensions and Load Ratings for Single-Row 02-Series Deep-Groove and Angular-Contact Ball Bearings

Bore, mm	OD, mm	Width, mm	Fillet Radius, mm	Shoulder Diameter, mm d_S	d_H	Load Ratings, kN Deep Groove C_{10}	C_0	Angular Contact C_{10}	C_0
10	30	9	0.6	12.5	27	5.07	2.24	4.94	2.12
12	32	10	0.6	14.5	28	6.89	3.10	7.02	3.05
15	35	11	0.6	17.5	31	7.80	3.55	8.06	3.65
17	40	12	0.6	19.5	34	9.56	4.50	9.95	4.75
20	47	14	1.0	25	41	12.7	6.20	13.3	6.55
25	52	15	1.0	30	47	14.0	6.95	14.8	7.65
30	62	16	1.0	35	55	19.5	10.0	20.3	11.0
35	72	17	1.0	41	65	25.5	13.7	27.0	15.0
40	80	18	1.0	46	72	30.7	16.6	31.9	18.6
45	85	19	1.0	52	77	33.2	18.6	35.8	21.2
50	90	20	1.0	56	82	35.1	19.6	37.7	22.8
55	100	21	1.5	63	90	43.6	25.0	46.2	28.5
60	110	22	1.5	70	99	47.5	28.0	55.9	35.5
65	120	23	1.5	74	109	55.9	34.0	63.7	41.5
70	125	24	1.5	79	114	61.8	37.5	68.9	45.5
75	130	25	1.5	86	119	66.3	40.5	71.5	49.0
80	140	26	2.0	93	127	70.2	45.0	80.6	55.0
85	150	28	2.0	99	136	83.2	53.0	90.4	63.0
90	160	30	2.0	104	146	95.6	62.0	106	73.5
95	170	32	2.0	110	156	109	69.5	121	85.0

Gear

$$P = \frac{N}{d}$$

$$m = \frac{d}{N}$$

$$p = \frac{\pi d}{N} = \pi m$$

$$pP = \pi$$

P = diametral pitch, teeth per inch
N = number of teeth
d = pitch diameter, in
m = module, mm
d = pitch diameter, mm
p = circular pitch

$$V = \pi d n / 12$$

V = pitch-line velocity, ft/min
d = gear diameter, in
n = gear speed, rev/min

$$W_t = 33\,000 \frac{H}{V}$$

W_t = transmitted load, lbf
H = power, hp
V = pitch-line velocity, ft/min

$$W_t = \frac{60\,000 H}{\pi d n}$$

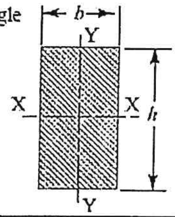
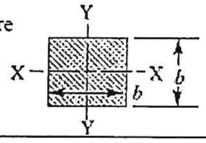
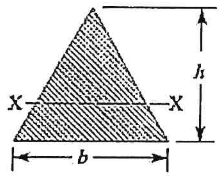
W_t = transmitted load, kN
H = power, kW
d = gear diameter, mm
n = speed, rev/min

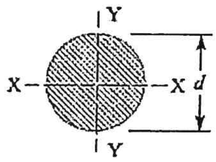
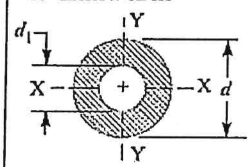
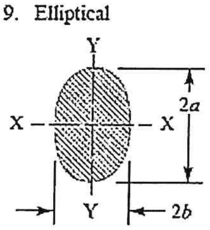
Table A–20

Deterministic ASTM Minimum Tensile and Yield Strengths for Some Hot-Rolled (HR) and Cold-Drawn (CD) Steels [The strengths listed are estimated ASTM minimum values in the size range 18 to 32 mm ($\frac{3}{4}$ to $1\frac{1}{4}$ in). These strengths are suitable for use with the design factor defined in Sec. 1–10, provided the materials conform to ASTM A6 or A568 requirements or are required in the purchase specifications. Remember that a numbering system is not a specification.] *Source:* 1986 SAE Handbook, p. 2.15.

1 UNS No.	2 SAE and/or AISI No.	3 Processing	4 Tensile Strength, MPa (kpsi)	5 Yield Strength, MPa (kpsi)	6 Elongation in 2 in, %	7 Reduction in Area, %	8 Brinell Hardness
G10060	1006	HR	300 (43)	170 (24)	30	55	86
		CD	330 (48)	280 (41)	20	45	95
G10100	1010	HR	320 (47)	180 (26)	28	50	95
		CD	370 (53)	300 (44)	20	40	105
G10150	1015	HR	340 (50)	190 (27.5)	28	50	101
		CD	390 (56)	320 (47)	18	40	111
G10180	1018	HR	400 (58)	220 (32)	25	50	116
		CD	440 (64)	370 (54)	15	40	126
G10200	1020	HR	380 (55)	210 (30)	25	50	111
		CD	470 (68)	390 (57)	15	40	131
G10300	1030	HR	470 (68)	260 (37.5)	20	42	137
		CD	520 (76)	440 (64)	12	35	149
G10350	1035	HR	500 (72)	270 (39.5)	18	40	143
		CD	550 (80)	460 (67)	12	35	163
G10400	1040	HR	520 (76)	290 (42)	18	40	149
		CD	590 (85)	490 (71)	12	35	170
G10450	1045	HR	570 (82)	310 (45)	16	40	163
		CD	630 (91)	530 (77)	12	35	179
G10500	1050	HR	620 (90)	340 (49.5)	15	35	179
		CD	690 (100)	580 (84)	10	30	197
G10600	1060	HR	680 (98)	370 (54)	12	30	201
G10800	1080	HR	770 (112)	420 (61.5)	10	25	229
G10950	1095	HR	830 (120)	460 (66)	10	25	248

Table : Properties of commonly used cross-sections.

Section	*Area* (*A*)	*Moment of inertia* (*I*)	*Distance from the neutral axis to the extreme fibre (y)	*Section modulus* $\left[Z = \frac{I}{y}\right]$	*Radius of gyration* $\left[k = \sqrt{\frac{I}{A}}\right]$
1. Rectangle 	bh	$I_{xx} = \frac{bh^3}{12}$ $I_{yy} = \frac{hb^3}{12}$	$\frac{h}{2}$ $\frac{b}{2}$	$Z_{xx} = \frac{bh^2}{6}$ $Z_{yy} = \frac{hb^2}{6}$	$k_{xx} = 0.289\ h$ $k_{yy} = 0.289\ b$
2. Square 	b^2	$I_{xx} = I_{yy} = \frac{b^4}{12}$	$\frac{b}{2}$	$Z_{xx} = Z_{yy} = \frac{b^3}{6}$	$k_{xx} = k_{yy} = 0.289\ b$
3. Triangle 	$\frac{bh}{2}$	$I_{xx} = \frac{bh^3}{36}$	$\frac{h}{3}$	$Z_{xx} = \frac{bh^2}{12}$	$k_{xx} = 0.2358\ h$

Section	(*A*)	(*I*)	(*y*)	$Z = \frac{I}{y}$	$k = \sqrt{\frac{I}{A}}$
7. Circle 	$\frac{\pi}{4} \times d^2$	$I_{xx} = I_{yy} = \frac{\pi d^4}{64}$	$\frac{d}{2}$	$Z_{xx} = Z_{yy} = \frac{\pi d^3}{32}$	$k_{xx} = k_{yy} = \frac{d}{2}$
8. Hollow circle 	$\frac{\pi}{4}(d^2 - d_1^2)$	$I_{xx} = I_{yy} = \frac{\pi}{64}(d^4 - d_1^4)$	$\frac{d}{2}$	$Z_{xx} = Z_{yy} = \frac{\pi}{32}\left(\frac{d^4 - d_1^4}{d}\right)$	$k_{xx} = k_{yy} = \frac{\sqrt{d^2 + d_1^2}}{4}$
9. Elliptical 	πab	$I_{xx} = \frac{\pi}{4} \times a^3 b$ $I_{yy} = \frac{\pi}{4} \times ab^3$	a b	$Z_{xx} = \frac{\pi}{4} \times a^2 b$ $Z_{yy} = \frac{\pi}{4} \times ab^2$	$k_{xx} = 0.5a$ $k_{yy} = 0.5b$