 Name: solution

November 10, 2021

INSTRUCTIONS

Begin each problem in the space provided. If additional space is required, use the paper provided to you.

Work appearing on the backside of any exam page may NOT be graded.

In order for you to obtain maximum credit for a problem, the solution must be clearly presented and in accordance with the following guidelines.

— Coordinate systems used must be clearly identified.

— Wherever appropriate, free body diagrams must be drawn. These should be drawn separately from the given figures.

— Units must be clearly stated as part of the answer when numerical answers are presented.

If the solution does not follow a logical thought process, it will be assumed to be in error.

PROBLEM No. 1 (25 points)

Problem 1 consists of 10 questions. Each question is worth 2.5 points.

(a) Clearance fits are generally described as having a gap or space between the mating parts.

Which other type(s) of fits may also have a a gap or space, depending on the parts being manufactured to the allowable tolerance?

Select all that apply.

- [ ] Drive
- [ ] Force
- [ ] Interference
- [x] Running
- [x] Transition

See table 7-9

(b) Which fit has the smaller tolerance zone?

- ○ H9/d8
- ● H7/p6

(c) A steel shaft experiences high torque starts and stops. During running, the shaft experiences moderate shocks.

Which is the best choice for heat treating the shaft?

- ○ Annealing
- ○ Quenching
- ○ Tempering
- ● Case Hardening

See section 2-15 in text.

In a few words, justify your answer.

high torque means the shaft should resist torsional shear stress which is highest at the surface. case hardening would strengthen the outer surface but maintain ductility in the core; the ductility is needed for the moderate shocks.

(d) Contact stresses occur between two bodies with differing radii of curvature.

Describe a typical failure due to contact stress.

See section 3-19.

Failure can be cracks pits or flaking.

(e) According to Shigley, *the selection of a material for a machine part...is one of the most important decisions the designer is called on to make.*

M.F. Ashby developed diagrams to assist in rapidly narrowing and choosing groups of materials having similar properties.

You are designing a part with the following material requirements.

— Density (ρ) between 4 and 10 Mg/m^3
— Young's modulus (E) between 60 and 250 GPa
— Strength (S) spanning 25 to 1050 MPa

Select the material(s) that meet the design requirements.

- [ ] Aluminum alloys
- [ ] Copper alloys
- [ ] Lead alloys
- [ ] Magnesium alloys
- [x] Nickel alloys
- [ ] Carbon steels
- [ ] Stainless steels
- [ ] Titanium alloys
- [ ] Zinc alloys
- [ ] None of the above

See figures 2-24 and 2-27.

(f) A steel member has a Brinell of $H_b = 275$.

Estimate the ultimate strength of the steel in MPa.

See eqn. 2-36

$S_u = 3.4 H_B = 3.4 \cdot 275 = 935$ MPa

(g) The SAE 5W-30 oil in your car will perform like SAE 5 oil during winter driving in Indiana and like SAE 30 oil during summer driving in Indiana.

○ True
● False

In a few words, justify your answer.

the oil will perform like SAE 5 @ start-up in winter. when the engine has reached its operating temperature, the oil performs like SAE 30; the operating temperature is the same in summer and winter.

(h) To achieve thick film lubrication in some journal bearings, lubricants above SAE 70 are recommended. To achieve thin film lubrication in other journal bearings, lubricants below SAE 10 are recommended.

- [ ] True
- [x] False

In a few words, justify your answer.

Thin film is never recommended for journal bearing operation.

(i) You have been tasked with replacing all journal bearings in a machine with a combination of deep groove ball bearings and spherical roller bearings.

What will be the impact of the change? Select all that apply.

- [ ] The REBs will be more difficult to access and maintain.
- [x] The machine will be noisier.
- [ ] The rotating shaft(s) will need to be lengthened to accommodate the REBs.
- [x] The bearing life will be shorter.
- [x] The machine's speed will decrease.

(j) A journal bearing is to be used for a certain application. For a fixed journal diameter, a design team must now choose the journal bearing's length.

A team member suggests choosing a longer journal bearing. How will bearing performance be impacted by the choice of a longer bearing? Select all that apply.

- [x] The minimum film thickness will increase. Fig 12-15
- [ ] The temperature rise in the lubricant will increase. Fig 12-17
- [x] The film pressure will increase. Fig 12-20
- [ ] The lubricant flow rate will increase.
- [ ] The coefficient of friction will increase.

PROBLEM No. 2 (XX points)

A ball bearing is to be used to support the load from a helical gear drive, where the axial load is 6 kN and the radial load is 14 kN. The bearing's inner ring rotates.

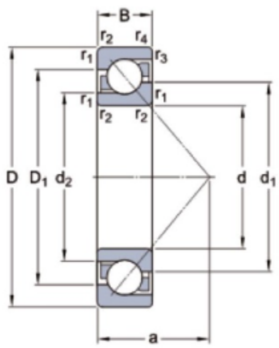
The bearing's bore diameter is $d = 20$ mm.

Determine the following.

(a) Choose a bearing from the catalog below. The catalog rating life is 10^6 cycles.

(b) For the bearing chosen, calculate the equivalent radial load (F_e).

(c) Is the bearing chosen expected to carry the load with 95% reliability for 10^7 cycles? If not, describe the next analysis step(s).



Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation	Dimension series to ISO 355 (ABMA)
d	D	T	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	–
15	35	11,75	18,5	14,6	1,43	17 000	20 000	0,055	▸ 30202	2CC
	42	14,25	27,7	20	2,08	15 000	18 000	0,094	▸ 30302	2FB
17	40	13,25	23,4	18,6	1,83	15 000	18 000	0,079	▸ 30203	2DB
	47	15,25	34,2	25	2,7	13 000	16 000	0,13	▸ 30303	2FB
	47	20,25	42,8	33,5	3,65	12 000	16 000	0,17	▸ 32303	2FD
20	42	15	29,7	27	2,65	13 000	16 000	0,099	▸ 32004 X	3CC
	47	15,25	34,1	28	3	12 000	15 000	0,12	▸ 30204	2DB
	52	16,25	41,9	32,5	3,55	12 000	14 000	0,17	▸ 30304	2FB
	52	22,25	54,3	45,5	5	11 000	14 000	0,23	▸ 32304	2FD
22	44	15	30,9	29	2,85	13 000	15 000	0,1	▸ 320/22 X	3CC
25	47	15	33,2	32,5	3,25	12 000	14 000	0,11	▸ 32005 X	4CC
	52	16,25	38,1	33,5	3,45	11 000	13 000	0,15	▸ 30205	3CC
	52	19,25	44,5	44	4,65	10 000	13 000	0,19	▸ 32205 B	5CD
	52	19,25	50,4	45,5	4,9	11 000	13 000	0,19	32205	2CD
	52	22	57,9	56	6	10 000	13 000	0,22	▸ 33205	2CE
	62	18,25	46,6	40	4,4	8 500	11 000	0,27	▸ 31305	7FB

choose one of these

[H]

a) choose 30304 w/ $C_{10} = 41.9$ kN and $C_0 = 32.5$ kN

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

Table 11–1 Equivalent Radial Load Factors for Ball Bearings

		$F_a/(VF_r) \le e$		$F_a/(VF_r) > e$	
F_a/C_0	e	X_1	Y_1	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 $F_a/C_0 < 0.014$.

$F_r = 14$ kN $\quad F_a = 6$ kN

$$\frac{F_a}{C_0} = \frac{6\text{ kN}}{32.5\text{ kN}} = 0.184$$

for $\frac{F_a}{C_0} = 0.17 \quad e = 0.34$

$$\frac{F_a}{VF_r} = \frac{6\text{ kN}}{1 \cdot 14\text{ kN}} = 0.428 > 0.34$$

$V=1$ for inner ring rotating

$i=2 \quad X_2 = 0.56 \quad Y_2 = 1.31$

$F_e = 0.56 \cdot 1 \cdot 14\text{ kN} + 1.31 \cdot 6\text{ kN} = 15.7$ kN

c) $a_1 F_R L_R^{1/a} = F_D L_D^{1/a}$

$a_1 = 0.64$ from notes for 95% reliability

$$F_R = \frac{F_D L_D^{1/a}}{a_1 L_R^{1/a}}$$

$L_R = 10^6$ cycles

$F_D = F_e = 15.7$ kN

$L_D = 10^7$ cycles

$a = 3$ for ball bearing

$$F_R = \frac{15.7\text{ kN} \cdot (10^7)^{1/3}}{0.64 \cdot (10^6)^{1/3}} = 52.9\text{ kN}$$

$F_R > C_{10}$ for bearing 30304 → will <u>not</u> support the load.

Next step: choose bearing 32304 and find F_e and F_R.

PROBLEM No. 3 (XX points)

A journal bearing supports a radial load of 10.8 kN.

The journal diameter is 60 mm. The journal rotates at 7200 rpm.

The bearing length is 60 mm and the bearing diameter is 60.06 mm.

The design requires a minimum film thickness h_0 of not less than 0.021 mm.

Determine the following.

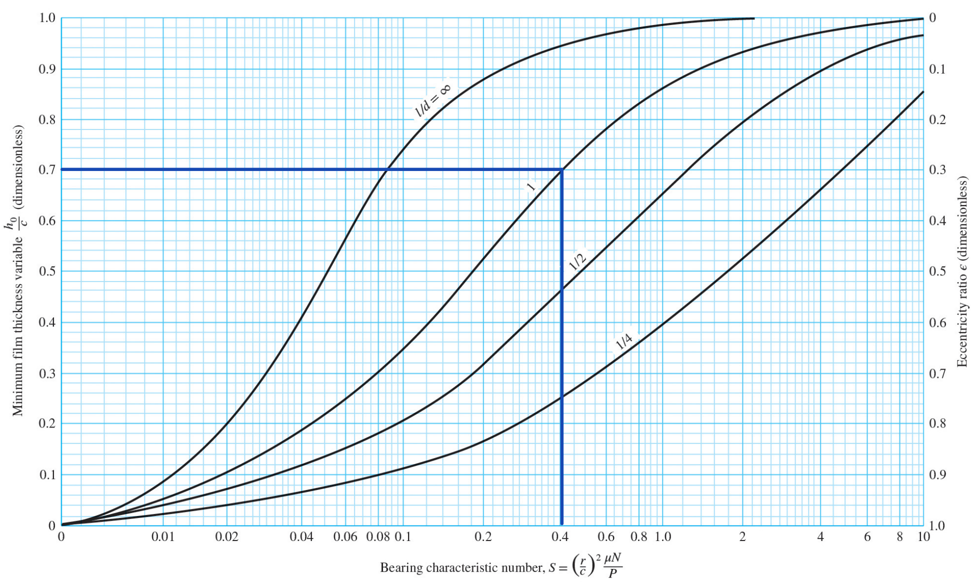
(a) The Sommerfeld number, S.

(b) The lubricant viscosity required (μ) in mPa·s.

(c) Use your result from part (b) to find the average lubricant temperature if the lubricant is SAE 30.

(d) Use the charts in the textbook to find the coefficient of friction, f.

(e) The power lost due to friction in W.



Source: A A. Raimondi and John Boyd, "A Solution for the Finite Journal Bearing and Its Application to Analysis and Design, Parts I, II, and III," Trans. ASLE. vol. 1, no. 1, in Lubrication Science and Technology, Pergamon, New York, 1958, pp. 159–209.

a) $l/d = 1$

$$C = \frac{60.06\text{ mm} - 60\text{ mm}}{2} = 0.03\text{ mm}$$

$$\frac{h_0}{c} = \frac{0.021\text{ mm}}{0.03\text{ mm}} = 0.7$$

From Fig. 12-15 $S = 0.4$

b) find μ from S

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P} \rightarrow \mu = \frac{SP}{N}\left(\frac{c}{r}\right)^2$$

$$P = \frac{W}{dl} = \frac{10800\text{ N}}{(0.06\text{ m})^2} = 3 \cdot 10^6\text{ Pa}$$

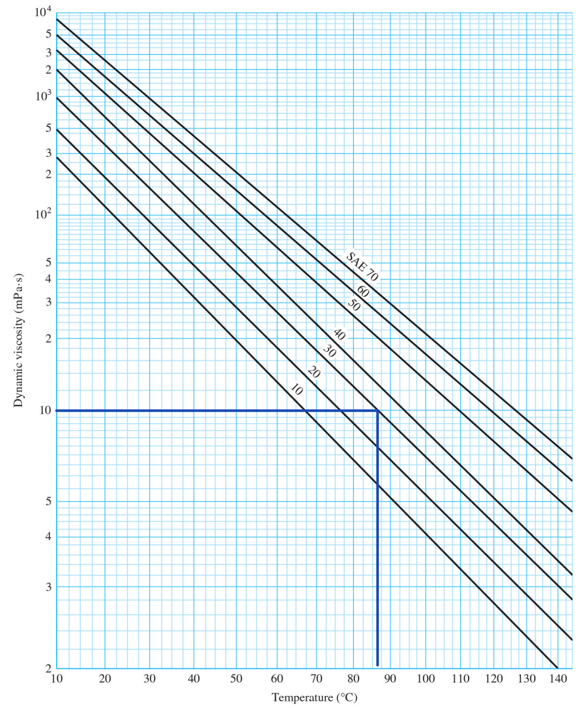
$$N = 7200\,\frac{\text{rev}}{\text{min}} \cdot \frac{\text{min}}{60\text{ s}} = 120\text{ rev/s}$$

$r = 30\text{ mm}$

$$\mu = \frac{0.4 \cdot 3 \cdot 10^6 \ N/m^2}{120 \ rev/s} \left(\frac{0.03 mm}{30 \ mm} \right)^2 = 10 \ mPa \cdot s$$

c) From Fig 12-3

temperature is about $86^\circ C$



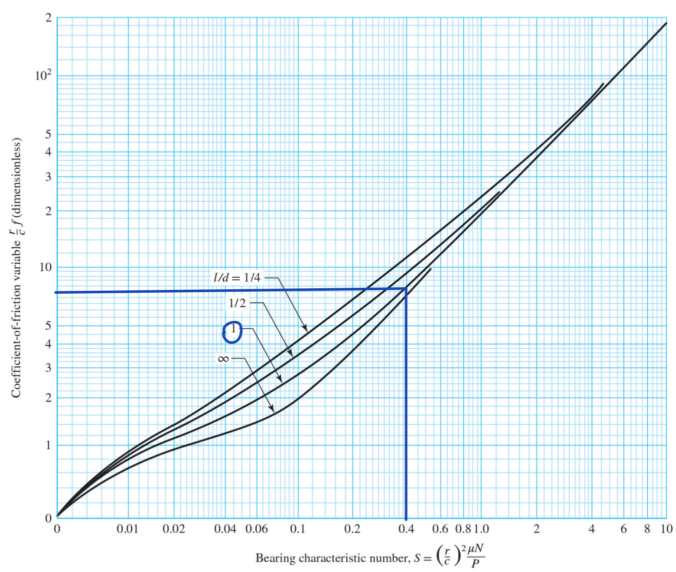
Source: Adapted from Figure 12–2.

d) From Fig 12-17

$$\frac{r}{c} f = 7.5$$

$$f = 7.5 \cdot \frac{c}{r} = 7.5 \cdot \frac{0.03}{30}$$

$$= 0.0075$$



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e) power lost $= T\omega = r f W \omega$

$$= 0.03 m \cdot 0.0075 \cdot 10800 \ N \cdot 7200 \ \frac{rev}{min} \cdot \frac{min}{60 s} \cdot \frac{2\pi \ rad}{rev} = 1.8 \ KW$$

PROBLEM No. 4 (XX points)

A 57-tooth spur gear is in mesh with a 23-tooth pinion.

The pinion is AISI 4140 nitrided grade 1 steel and rotates at 1000 rpm.

The gear is class 40 cast iron.

The diametral pitch is $P = 6$ teeth/inch and the pressure angle is $\phi = 20°$. The face width is 1.75 inch.

Assumptions and given information:

— The load is moderate shock and the power is smooth

— The gears are quality level 9

— The gears have uncrowned teeth, are straddle-mounted with bearings immediately adjacent, and are commercial enclosed gear units

— The gears have a backup ratio $m_B = 1.5$

— The reliability level is 99%

— The operating temperature is 200°F

— The pinion life is to be 10^8 revolutions

— Use $Y_N = 1.6831\,N^{-0.0323}$ and $Z_N = 2.466\,N^{-0.056}$

Determine the following.

(a) The diameters of the pinion and of the gear.

(b) Complete the table on the following page with the variables needed to analyze the gearset for bending and wear using the AGMA equations. Include dimensions, where applicable.

(c) Considering failure in the pinion due to bending, determine the transmitted load (W^t) in lbf for factor of safety $S_H = 1$.

(d) Using the result from (c), estimate the power capacity of the gearset in hp.

(e) Now consider failure in the gear due to bending.

○ The power capacity of the gearset will be larger than the capacity calculated in part (d).
○ The power capacity of the gearset will be smaller than the capacity calculated in part (d).

Briefly justify your answer: ____

a) $d_P = \dfrac{23 \text{ teeth}}{6 \text{ teeth/in}} = 3.833 \text{ in}$

$d_G = \dfrac{57 \text{ teeth}}{6 \text{ teeth/in}} = 9.5 \text{ in}$

b)

Variable	Pinion	Gear	
K_o	1.25 →		Fig 14-17
K_v	1.2 →		see below
K_s	1 →		
P_d	6 teeth/in →		given
F	1.75 in →		given
K_m	1.184	1.184	
K_B	1 →		for $m_B \geq 1.2$
J	0.35	~0.41	Fig 14-6 for $N_p = 23$ $N_G = 57$
S_t	40 kpsi	13 kpsi	
Y_N	0.9283	0.9560	
K_T	1 →		for $T < 250°$
K_R	1 →		for 99% reliability
C_p	2100 $\sqrt{psi}$ →		Table 14-8
d_P	3.833 in →		
C_f	1 →		
I	0.1145 →		see below
S_c	150 ksi	75 ksi	
Z_N	0.879	0.925	see below
C_H	1	1.012	

for K_v ...

$$V = \omega_p r_p = 1000 \frac{rev}{min} \cdot \frac{3.833 in}{2} \cdot \frac{ft}{12 in} \cdot \frac{2\pi rad}{rev} = 1003 \ ft/min$$

from Fig. 14-9 and $Q_v = 9$ $K_v = 1.2$

-or- from Eqn 14-27 and 14-28

$$K_v = \left(\frac{A + \sqrt{V}}{A}\right)^B = \left(\frac{76.88 + \sqrt{1003}}{76.88}\right)^{0.52} = 1.1965$$

very close.

$$A = 50 + 56(1-B) = 50 + 56(1 - 0.52) = 76.88$$

$$B = 0.25(12 - Q_v)^{2/3} = 0.25(12-9)^{2/3} = 0.52$$

for pinion $Y_N = 1.6831\,(10^8)^{-0.0323} = 0.9283$

for gear $N = 10^8 \cdot \frac{23}{57} = 4.04 \cdot 10^7$ cycles

$$Y_N = 1.6831\,(4.04 \cdot 10^7)^{-0.0323} = 0.9560$$

$$I = \frac{\cos\phi \sin\phi}{2m_N} \frac{m_G}{m_G+1} = \frac{\cos 20 \sin 20}{2} \cdot \frac{2.478}{3.478} = 0.1145$$

$\phi = 20^\circ$

$m_N = 1$ for spur gears

$m_G = \frac{N_G}{N_P} = \frac{57}{23} = 2.478$

for pinion $Z_N = 2.466\,(10^8)^{-0.056} = 0.879$

for gear $Z_N = 2.466\,(4.04 \cdot 10^7)^{-0.056} = 0.925$

c) $S_F = \dfrac{S_t Y_N / K_T K_R}{\sigma} \rightarrow \sigma = \dfrac{S_t Y_N / K_T K_R}{S_F}$

$$\sigma = \frac{40 \text{ kpsi} \cdot 0.9283 / 1.1}{1} = 36.99 \text{ kpsi}$$

$$\sigma = W^t K_o K_v K_s \frac{P_d}{F} \frac{K_m K_B}{J} \rightarrow W^t = \frac{\sigma}{K_o K_v K_s} \frac{F}{P_d} \frac{J}{K_m K_B}$$

$$W^t = \frac{36.99 \text{ kpsi}}{1.25 \cdot 1.2 \cdot 1} \frac{1.75 \text{ in}}{6 \text{ teeth/in}} \cdot \frac{0.35}{1.184 \cdot 1} = 2.13 \text{ kips}$$

$$= 2130 \text{ lbf}$$