MECH 325 - Midterm Book 1

Tuesday, October 23, 2018

Name: Solution

Student Number: ____________________

Circle your section

Section 101: 9:30am Tuesday and Thursday

Section 102: 11:30am Tuesday and Thursday

Signature: ____________________

Part 1 MC Mark _____ / 15

Part 1 SA Mark _____ / 15

Part 2 LA Mark _____ / 30

Total _____ / 60

Instructions

There are two parts to this exam with different instructions for each. Please read carefully.

Part 1 – Closed-Book (30 marks)

Multiple choice (15 marks: 1 marks per question). Complete all 14 questions by marking your response **in pencil in the computer score card**. Write your name and student number on the computer card and mark your student number in the "ID Field".

Short answer (15 marks: 2 or 3 marks per question). Complete all 6 questions by marking your response in this exam booklet.

After you have completed Part 1, hand in your booklet and Scantron scorecard and you will receive a handout for Part 2. You may not return to Part 1 after you hand it in.

Part 2 – Open-Book (30 marks)

Long answer (30 marks): Complete all parts to the questions for Part 2 by marking your responses in the separate exam booklet. This portion of the exam is open-book and open-notes.

Only non-programmable calculators may be used (or programmable calculators with purged memories).

Part 1A: Multiple Choice

Answer all 15 questions. Each is worth 1 mark. Choose the single best response – a given question may have more than one choice that is correct but marks will only be given for the *best* answer.

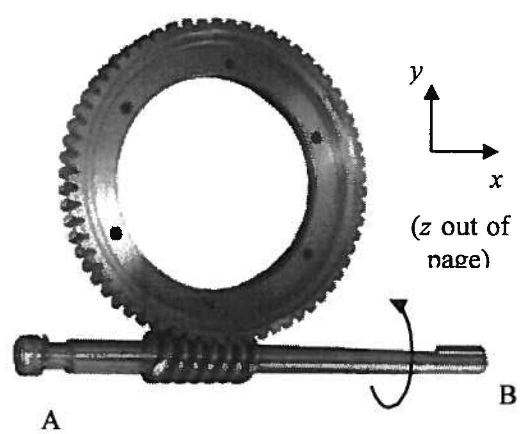
1. **What is the best description of the type of motion between the teeth of two properly meshed spur gears?**
   a) Purely rolling
   b) Some sliding when teeth engage and disengage, but predominantly rolling
   c) Approximately equal parts sliding and rolling
   d) Some rolling when teeth engage and disengage, but predominantly sliding
   e) Purely sliding

2. **Which of the following types of gears is appropriate for high-efficiency, smooth (low-noise) transmission of motion between parallel shafts with minimal axial loading on either shaft?**
   a) Bevel
   b) Spur
   c) Worm
   d) Herringbone
   e) Helical

3. **A spur gear has 40 teeth, a 14° pressure angle, and a pitch diameter of 2.0 in. What is the diametral pitch of the gear?**
   a) 0.05
   b) 1.42
   c) 2.86
   d) 7.0
   e) 20.0

4. **The worm gear shown has a 1" pitch diameter and has two threads. The gear has an 8" pitch diameter and 96 teeth. If the worm turns 24 revolutions in the direction indicated, how far does the gear turn?**

   a) 4 rotations clockwise
   b) 1 rotation clockwise
   c) ½ rotation clockwise
   d) 1 rotation counter-clockwise
   e) 4 rotations counter-clockwise



5 **A spur gearset has an input gear with 15 teeth and an output gear with 60 teeth. If 4 N·m of torque is supplied to the input gear at 50 rad/sec, approximately how much power is delivered by the output gear to the load?**

a) 16 W
b) 60 W
c) 75W
d) 200 W
e) 800 W

6 **Comparing a square vs. an ACME power screw, which of the following is true regarding the use of a split-nut:**

a. A split nut reduces friction on both types of power screw
b. A split nut allows for backlash adjustment on an ACME power screw
c. A split nut allows for backlash adjustment on a square power screw
d. A split nut is required when using a bushing for a friction collar on both types of power screw
e. A split nut is required when an ACME power screw is non-reversing

7 **A power screw engages with a nut with 10 threads. Approximately how much of the load is carried by the first 7 engaged threads, combined, in the power screw?**

a. 10%
b. 20%
c. 50%
d. 80%
e. 100%

8 **Rank the following flexible drives from lowest to highest power capacity. Assume approximately equal dimensions and operating conditions.**

a.	V-Belt	Flat belt	Roller Chain
b.	V-Belt	Roller Chain	Flat belt
c.	Flat belt	V-Belt	Roller Chain
d.	Flat belt	Roller Chain	V-Belt
e.	Roller Chain	Flat belt	V-Belt

9 **For roller chains the term *chordal speed variation* refers to:**

a. the speed of engagement of a chain with a sprocket
b. the relationship between the chain speed and the sprocket speed
c. the change in chain speed as the friction in the chain changes
d. the change in chain speed due to the slight rise and fall of the chain as it engages with the sprocket
e. the change in chain speed due to elongation of the chain

10 **When specifying the working load of wire rope, which of the statements is FALSE:**

a. one must consider the modulus of elasticity of the wire rope
b. one must take in account the maximum allowable bearing pressure of the wire rope on the sheave
c. one must consider the application in order to specify a safety factor
d. one must determine the maximum sheave diameter
e. one must consider loads caused by sudden stops and starts

11 A V-belt drive has a driving pulley operating at 200 rpm, a drivetrain value of $e = 0.25$, a nominal power of $H_{nom} = .75$ hp, a service factor of $K_s = 2.5$, and a design factor of $n_d = 1.5$. TWO belts are to be used in parallel. The following belts are available in the catalogue. Which is the smallest belt that will satisfy the design requirements?

a. Allowable power $H_a = 0.7$ hp
b. Allowable power $H_a = 1.1$ hp
c. Allowable power $H_a = 1.5$ hp
d. Allowable power $H_a = 2.9$ hp
e. Allowable power $H_a = 5.7$ hp

12 Which of the following is the best description of how a hydrodynamic bearing works?

a. The shaft moves by sliding against the bearing material
b. High pressure fluid is injected into the bearing to support the shaft
c. The rotating shaft pulls fluid to form a layer between it and the bearing
d. The shaft is supported by high pressure air fed through a porous material
e. Metal-on-metal contact is avoided by packing the bearing with a thick grease

13 Which of the following design or operating parameters should be increased in order to be able to reduce the required bushing length of a boundary-lubricated bearing in service? (assume all other parameters remain fixed)

a. Operating rotation speed, N
b. Applied force, F
c. Coefficient of friction, f_s
d. (PV) rating of the bushing
e. Wear factor, K

14 A tapered-roller bearing is best suited for:

a. Only large radial loads
b. Only large axial loads
c. Small radial loads, plus large axial loads in both directions
d. Large radial loads, plus large axial loads in both directions
e. Large radial loads, plus large axial loads in one direction only

15 A reasonable safety factor when specifying wire rope for a high-speed passenger elevator is

a. 2
b. 2 to 2.5
c. 3 to 4
d. 9 to 11
e. 20 to 30

Part 1B – Short-Answer Questions (15 marks)

Answer all questions in the spaces provided.

16. a) In gear design, what is meant by interference? How can it be avoided (2 marks)?

Interference relates to gear tooth contact that is not conjugate. It causes gear teeth to bind and cause excessive wear.

Interference is avoided by having the gear teeth engage between the tangents on the base circles formed by the pressure angle. One can select gears with greater number of teeth.

1 mk

1 mk

b) Two parallel shafts with 6 inch center distances are to be connected by a pair of gears having a diametrical pitch of 16 teeth/inch. The spur gears have a contact pressure angle of 24° and provide a gear reduction of 3.0. Determine the pitch diameter and number of teeth in the pinion gear (2 marks).

$r_p + r_G = 6''$ $\quad P = 16 \text{ teeth/inch}$ $\quad e = 3:1 = 1/3$

so $r_G / r_p = 3$

$\therefore 4r_p = 6''$ $\quad r_p = 1.5''$ and $r_G = 4.5''$

So $d_p = 3''$ and $d_G = 9''$

$3'' \times 16 = 48$ teeth

$9 \times 16 = 144$ teeth

1 mk right idea

1 mk correct solution

17.a) For synchronous power transmission with a flexible element, what is the principal *operating* difference between roller chains and timing belts? Exclude materials and secondary effects such as lubrication, dirt, and maximum capacity. (1 mark)

b) What is the simplest way to minimize this difference? (1 mark)

Roller chains are a positive drive system. No friction coefficient is required. Hence little pre-tension required

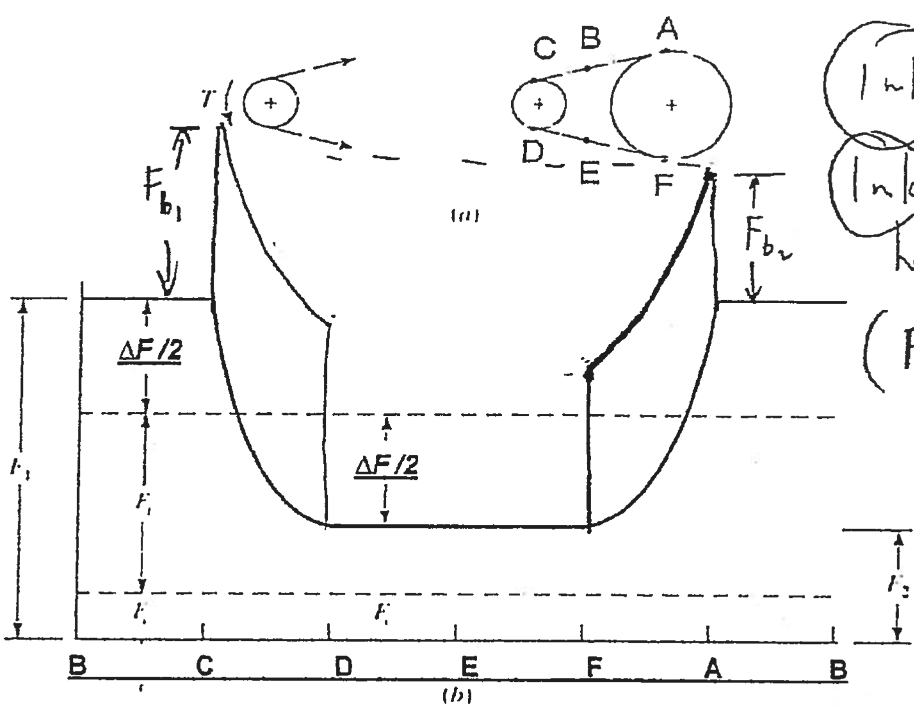
1 mk

Roller chains have chordal speed variation which does not occur with V-belts

Have sufficient number teeth on pinion sprocket. In general ≥ 17 teeth

1 mk

18. The diagram shown below is the tension profile for a flat belt. On this diagram, sketch the tension profile for a V-belt belt that is transmitting the same torque, but otherwise has the same linear speed, initial tension, pulley diameters, and belt weight per unit length. (2 marks)

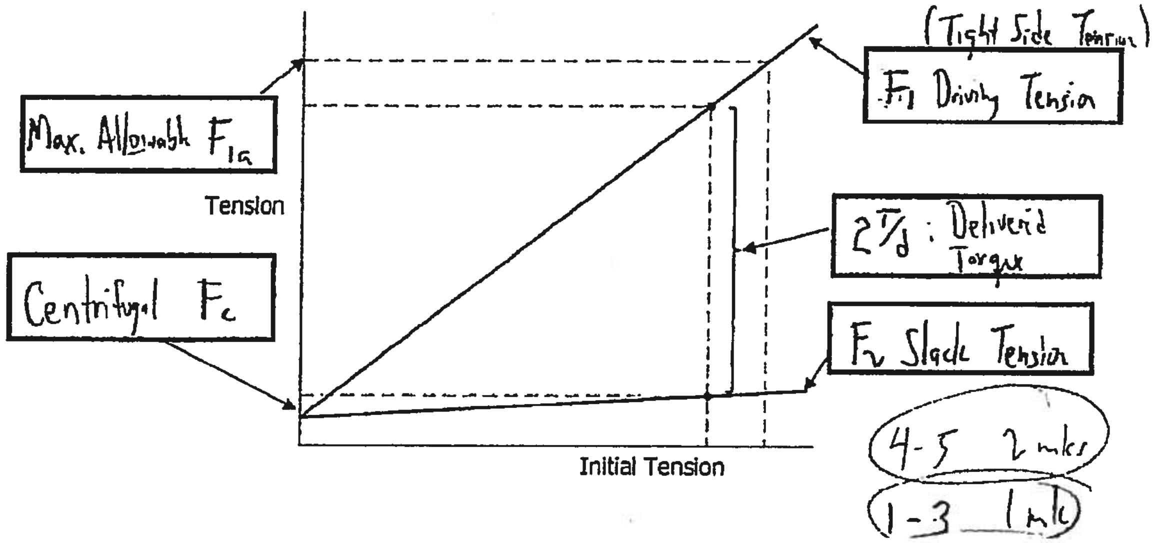


1 mk for sketch

1 mk for differing height

$(F_{b_1} > F_{b_2})$

19. **The relationship for flat belt tension versus initial tension is shown in the diagram below. Complete the diagram by giving the name or equation for each of the four quantities identified. Put your answers in the boxes provided. (2 marks)**



20. **A power screw is self-locking when the screw will not turn once a lifting torque has been removed. The equations for the raising torque, T_R, and the lowering torque, T_L for a power screw are given below:**

$$T_R = \frac{Fd_m}{2}\left(\frac{\pi f d_m + l}{\pi d_m - fl}\right) \quad T_L = \frac{Fd_m}{2}\left(\frac{\pi f d_m - l}{\pi d_m + fl}\right)$$

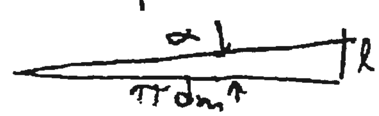
What is the condition for self-locking? How does it relate to the pitch angle? (3 marks)

Self locking when $\pi f d_m - l > 0$ (1 mk)

$\therefore \pi f d_m > l$

$f > l/\pi d_m$

We know from power screw geometry. Thread angle α is defined as



$\tan\alpha = l/\pi d_m$

$\therefore f > \tan\alpha$ (1 mk)

21. **a) Why is lubrication important in a chain drive train system? (1 mark)**
b) What is meant by "galling?" (1 mark)

a) Lubrication is important in chains in order to reduce friction and wear. Reduced wear means longer life (up to 300 times longer) and reduced friction can lead to higher loading and improved efficiency

b) Some metals (titanium, aluminium) will form a 'welded' bond when subject to contact under stress. These bonds will undergoe shear and fail causing metal erosion

MECH 325 - Midterm Book 2

Tuesday, October 23, 2018

Name: Solution

Student Number: ______________________

Circle your section

Section 101: 9:30am Tuesday and Thursday

Section 102: 11:30am Tuesday and Thursday

Signature: ______________________

Q20 _____ / 8

Q21 _____ / 10

Q22 _____ / 5

Q23 _____ / 7

Total _____ / 30

Instructions

Please read carefully.

Part 2 – Open-Book (30 pts)

Long answer: Complete all parts to the question by marking your response on the following pages. This portion of the exam is open-book and open-notes. **At the end of the exam, return this handout with your answers.**

This is a multi-part problem. For each part, use the parameter values given. Clearly indicate your final answer to each part by drawing a box around it. Separate different parts with a line drawn across the page.

Only non-programmable (or purged) calculators may be used.

Long Answer Problem (30 marks)

You are to design a set of spur gears to be used as gear box in a wood processing machine. An electric motor transmits 3.0 hp (2250W) to an 18-tooth pinion at 1750 rpm. The gear must rotate 460 ±5rpm Specifications for the gear box are as follows:

Electric Motor Speed	1750 rpm
Output Power	3.0 hp (2.25kW)
Diametrical Pitch	12 teeth/inch (module = 2mm.)
Number of Teeth on Pinion	18 teeth
Width of gears	1 inch (25 mm.)
Pressure Angle	20°
Operating Temperature	70°F (20°C)
Shaft Diameter	½ inch (12 mm.)

You can work in either Imperial or SI units. However, do not mix units as the conversions are not exact.

21. Determine a) the number of teeth on the gear and gear reduction ratio to achieve the required 460 ±5rpm output speed, b) the centre line distance from the pinion to the gear, c) the force transmitted from the pinion to the gear, and d) the radial load on the pinion shaft. **(8 marks).**

22. Determine all the AGMA factors for the gear tooth bending stress equation for the pinion and gear. Estimate the bending stress number for the pinion and gear. Justify all assumptions. Note: Some of the AGMA factors are given below. **(10 marks)**

Gear Quality	7
Design Lifetime	10^7 cycles
Reliability (both gears)	0.99
Size Factor, K_s (both gears)	1.0
Load Distribution Factor, K_m (both gears)	1.3
Stress Cycle Factor, Y_N (both gears)	1.0
Driven Load	Heavy Shock

23. Based on the results of the AGMA bending stress calculations, specify the material grade and subsequent hardness for the pinion and gear. Justify all assumptions. What is your safety factor for this gear? Explain **(5 marks)**

24. A set of bushings (boundary lubricated bearing) is proposed to support the pinion gear. Select a suitable material for this bushing. For this problem, assume that the load will be equally supported by the two bushings. The diameter of the motor shaft is ½ inch (12 mm.) Assume an operating temperature of 70°F (20°C) and a coefficient of friction, $f_s = 0.03$. Justify all assumptions. Do you support the proposal? How does this compare with a ball bearing of similar bore? Explain **(7 marks)**

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21. a) Required Gear Reduction $= \dfrac{1750\text{ rpm}}{460\text{ rpm}} = 3.804 : 1$

Number of teeth on gear $= N_G$ $\quad N_G = (N_P)(\text{gear reduction}) = (18)(3.804)$

$N_G = 68.47$ $\quad$ 1 mk

Round to integer value for teeth. Select 69 teeth so even pinion teeth mate with odd number gear (best practice) [68 will not be wrong]

Check gear ratio $\quad 1750\left(\tfrac{18}{69}\right) = 456.5\text{ rpm}$ OK. $\quad$ 1 mk

Final Reduction ratio $= 69/18 = 3.833 : 1$ $\quad$ 1 mk

b) Centre line distance $= \text{℄} = \tfrac{1}{2}P_{d\,\text{pinion}} + \tfrac{1}{2}P_{d\,\text{gear}}$ $\quad$ 1 mk

Diametrical pitch = 12

$\therefore \text{℄} = \tfrac{1}{2}\left(\dfrac{18\text{ teeth}}{12\text{ teeth/inch}} + \dfrac{69}{12}\right) = 3.625'' = 3\tfrac{5}{8}''$ $\quad$ 1 mk

if $N_G = 68$ $\quad \text{℄} = 3.583$ inches

metric $= 86$ mm.

c) Transmitted Force $= W_t$ $\quad$ recall Power = Force × velocity = Torque × ω

$1\text{ HP} = 550\text{ ft·lb/sec}$ or 33000 ft·lb/min

velocity = pitch line speed $= \omega r = V_t$

$V_t = \pi D_p n_p / 12 = \pi\left(18\text{ teeth}/12\text{ teeth/in} = 1.5\text{ in.}\right)(1750\text{ rpm})\left(\dfrac{1\text{ ft}}{12\text{ in.}}\right)$

$= 687.2\text{ ft/min}$ $\quad$ 1 mk

$W_t = \text{Power}/V_t = (3\text{ h.p})(33{,}000\text{ ft·lb/min}) \big/ 687.2\text{ ft/min}$

$W_t = 144.1\text{ lbs}$ $\quad$ 1 mk

| Metric $\quad P = T\omega \quad T = W_t \cdot r$

| $W_t = P/\omega r = \dfrac{2250\text{ W}}{\left[\dfrac{(1750\text{ rpm})(2\pi\text{ rad/rev})}{60}\right]\left[\dfrac{18\text{ teeth} \cdot 2\text{ mm/tooth}}{2}\right]}$

| $W_t = 682.2\text{ N}$

d) $W_R = W_t \tan\phi \quad \phi = 20°$ $\quad$ 1 mk

$W_R = (144.1)(\tan 20°)$

$W_R = 52.45\text{ lbs}$

$W_R = (682.2\text{ N})\tan 20° = 248.3\text{ N}$

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22. Determin all AGMA factors

Bending Stress Number $= \sigma = W^t K_o K_v K_s \dfrac{P_d}{F} \dfrac{K_m K_B}{J}$

Given: Size factor $= K_s = 1.0$

Load dist. factor $= K_m = 1.3$ (1 mk)

i) Overload factor $= K_o$

Driven load = heavy shock

Driven = electric motor = no driver shock

$\therefore K_o = 1.75$ (1 mk)

ii) Dynamic Factor $= K_v$

Using Fig 14-9 $V_t = 687.2$ ft/min ≈ 750 (1 mk)

$\underline{K_v \doteq 1.3}$ from Figure @ Gear quality = 7

One can also use $K_v = \left(\dfrac{A + \sqrt{V}}{A}\right)^B$ where $A = 50 + 56(1-B)$

and $B = 0.25(12 - \text{Gear Quali})^{2/3}$

so $B = 0.25(12-7)^{2/3} = 0.731 \Rightarrow A = 50 + 56(1-0.731) = 65.1$

$K_v = \left(\dfrac{65.1 + \sqrt{687.2}}{65.1}\right)^{0.731} = \underline{1.28 = K_v}$ (1 mk)

iii) Rim thickness factor $= K_B$

Since diameters are small $D_P = 1.5''$ $D_G = 69/12 = 5.75''$ (1 mk)

we can assume solid gears so $K_B = 1.0$ for both gears

iv) Geometry Factor, J

Use Figure 14-6 and assume full contact for teeth

$J_{pinion} \doteq 0.32$ (18 teeth into 70) (1 mk)

$J_{gear} \doteq 0.41$ (~70 into ~18 teeth) (1 mk)

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22. cont. Determine bending stress number

1 mk

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

From 21) $W_t = 144$ lbs $P_d = 12$ teeth/in F = Face Width = 1 inch.

$$\sigma_{pinion} = (144)(1.75)(1.3)(1.0)\left(\frac{12}{1}\right)\frac{(1.3)(1.0)}{0.32}$$

1 mk

$\sigma_p = 15.97$ ksi $\simeq 16$ ksi   123 MPa ← metric sol'n

$$\sigma_{gear} = (144)(1.75)(1.3)(1.0)\left(\frac{12}{1}\right)\frac{(1.3)(1.0)}{0.41}$$

1 mk

$\sigma_g = 12.46$ ksi ≈ 12 ksi $\simeq 90$ MPa a metric sol'n

23. Bending endurance strength

$$\sigma_{all} = \frac{S_t}{S_F} \frac{Y_N}{K_T K_R}$$

determine AGMA constants

- i) Temperature Factor $K_T = 1$ if $T < 250°F\ (120°C)$ 1 mk

we can assume temp. will not exceed 250°F so $K_T = 1.0$

ii) Reliability Factor $K_R = 1.00$ given $R = 0.99$ (Table 14-10) 1 mk

iii) Stress cycle Factor $Y_N = 1.00$ (given)

so Allowable Stress Number, $S_t = \sigma_{all} \times S_F$ (safety factor)

From 22) $\sigma_{max} = 16$ ksi so we want $S_t > 16$ ksi

From Table 14-3 Grade 1 gears are adequate as all values exceed 16 ksi

1 mk

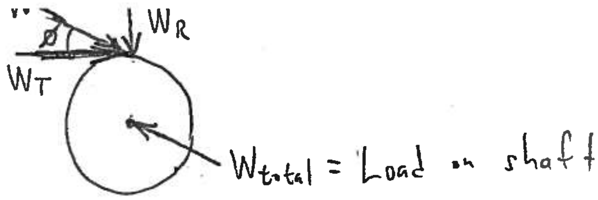
Choose Grade 1 "Through hardened" as minimum grade

From Fig 14-2 we can select a Brinell hardness of $H_B = 200$

which gives $S_t = 28$ ksi

(Table 14-3)

One can also choose Grade 1 Flame hardened type B pattern $S_t = 22$ ksi



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23. Cont.

For pinion $28\ ksi = 16 \times S_F \quad \therefore S_{F_p} = 1.75$ (1 mk)

For gear $28\ ksi = 12 \times S_F \quad \therefore S_{F_g} = 2.33$

Both safety factors are above 1.5 which one can consider as the minimum safety factor. Hence Grade 1 $H_B = 200$ Through Hardened is suitable (1 mk)

24. Proposal is to use bushings to support the shaft.

Given $D = \frac{1}{2}''$ $L = 1.0$ inch $f_s = 0.03$ $T_\infty = 70°F$ (1 mk)

From FBD) $W_R = 52.45$ lbs, $W_T = 144.1$ lbs $\therefore W_{total} = [52.45^2 + 144.1^2]^{\frac{1}{2}} = 153.4$ lbs

$P_{ave} = F/DL = \frac{1}{2}(153.4\ lbs) / \frac{1}{2}''(1'') = 153.4$ psi [each bushing has 50% load]

i) $P_{max} = 4/\pi\ P_{ave} = 4/\pi(153.4) = 195.3$ psi (1 mk)

From Table 12-7 Only rubber with $P_{max} = 50 psi < 195 psi$

ii) sliding velocity $V = \frac{\pi DN}{12} = \pi(\frac{1}{2}'')(1750\ rpm)/12 = 229.1\ ft/min$

From Table 12-7 Teflon & Teflon Fabric are eliminated (1 mk)

iii) $PV = (153.4\ psi)(229.1\ ft/min) = 35143$ (1 mk)

From 12-7, all but bronze & cast iron are eliminated

v) Temperature of lubricant $T_f = T_\infty + \frac{720 f_s F N}{J\ \hat{h}_{CR}\ L}$

For bushing $\hat{h}_{CR}$ = overall combined heat transfer coeff = $2.7 \frac{Btu}{h \cdot ft^2 \cdot °F}$

$J = 778\ ft \cdot lbf / Btu$

$\therefore T_f = 70°F + 720(0.03)(153.4^{\#}/2)(1750\ rpm) / (778)(2.7)(1'')$

$T_f = 70 + 1380 \simeq 1450°F$ all eliminated (2 mk)

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24. cont. If we compare the bushing with a bearing of similar size, we will see that a roller bearing is a better option.

bore $= \frac{1}{2}'' \approx 12\,mm$

From Table 11-2, Single Row 02 Deep-Groove Bearing

12mm bore $C_{10} = 6.89\,kN$ & $C_0 = 3.10\,kN$

$F = \frac{1}{2}(153.4\,lbs) = 76.7\,lbs = 76.7\,lb \times 0.454\,kg/lb = 34.8\,kg$

$F = (34.8\,kg)(9.81\,m/s^2) = 341.6\,N$

load $= 341.6 / C_{10} = 6890N = 0.05 \doteq 5\%$ loaded at C_{10}

If we include the load factor for heavy impact, say $a_f = 2.0$ to 3.0 our loading is still 10 to 15% of C_{10} load. We can expect a reasonable life for bearing.

$\therefore$ Choose roller bearings!