

MACHINE DESIGN II

FALL 2020

TITLE OF ASSIGNMENT: HOMEWORK 2

TOPICS COVERED IN HOMEWORK 2

1. Spur and Helical Gears (Chapter 14)
2. Fatigue Failure Resulting from Variable Loading (Chapter 6)

Grade:

Name:

This assignment is due at NOON on October 2nd, 2020. Complete the following problems from Shigley's Mechanical Engineering Design – 10th edition. There are a total of 10 problems in this homework.

1. Problem 14-2

A steel spur pinion has a diametral pitch of 10 teeth/in, 18 teeth cut full-depth with a 20° pressure angle, and a face width of 1 in. This pinion is expected to transmit 2 hp at a speed of 600 rev/min. Determine the bending stress.

2. Problem 14-6

A 20° full-depth steel spur pinion has 20 teeth and a module of 2 mm and is to transmit 0.5 kW at a speed of 200 rev/min. Find an appropriate face width if the bending stress is not to exceed 75 MPa.

3. Problem 14-12

A gear drive consists of a 16-tooth 20° steel spur pinion and a 48-tooth cast-iron gear having a pitch of 12 teeth/in. For a power input of 1.5 hp at a pinion speed of 700 rev/min, select a face width based on an allowable contact stress of 100 kpsi.

4. Problem 14-13

A gearset has a module of 5 mm, a 20° pressure angle, and a 24-tooth cast-iron spur pinion driving a 48-tooth cast-iron gear. The pinion is to rotate at 50 rev/min. What horsepower input can be used with this gearset if the contact stress is limited to 690 MPa and $F = 60$ mm?

5. Problem 6-9

For the interval $10^3 \leq N \leq 10^6$ cycles, develop an expression for the axial fatigue strength $(S'_f)_{ax}$ for the polished specimens of 4130 used to obtain Fig. 6-10. The ultimate strength is $S_{ut} = 125$ kpsi and the endurance limit is $(S'_e)_{ax} = 50$ kpsi.

6. Problem 6-11

Two steels are being considered for manufacture of as-forged connecting rods subjected to bending loads. One is AISI 4340 Cr-Mo-Ni steel capable of being heat-treated to a tensile strength of 260 kpsi. The other is a plain carbon steel AISI 1040 with an attainable S_{ut} of 113 kpsi. Each rod is to have a size giving an equivalent diameter d_e of 0.75 in. Determine the endurance limit for each material. Is there any advantage to using the alloy steel for this fatigue application?

7. Problem 6-13

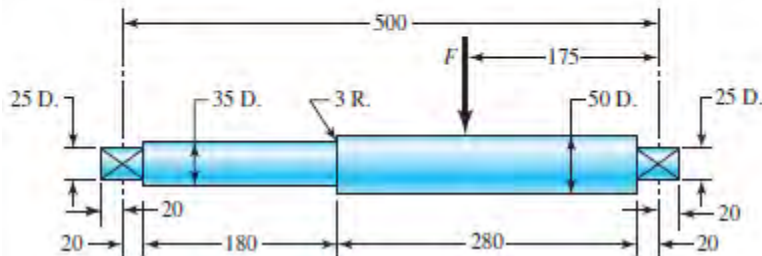
A solid square rod is cantilevered at one end. The rod is 0.6 m long and supports a completely reversing transverse load at the other end of ± 2 kN. The material is AISI 1080 hot-rolled steel. If the rod must support this load for 10^4 cycles with a design factor of 1.5, what dimension should the square cross section have? Neglect any stress concentrations at the support end.

8. Problem 6-16

Grade:

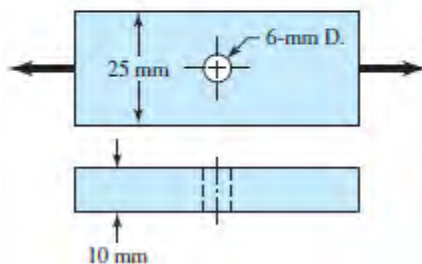
Name:

The rotating shaft shown in the figure is machined from AISI 1020 CD steel. It is subjected to a force of $F = 6 \text{ kN}$. Find the minimum factor of safety for fatigue based on infinite life. If the life is not infinite, estimate the number of cycles. Be sure to check for yielding.



9. Problem 6-25

The cold-drawn AISI 1040 steel bar shown in the figure is subjected to a completely reversed axial load fluctuating between 28 kN in compression to 28 kN in tension. Estimate the fatigue factor of safety based on achieving infinite life and the yielding factor of safety. If infinite life is not predicted, estimate the number of cycles to failure.



10. Problem 6-27

Using the modified Goodman criterion for infinite life, repeat Prob. 6-25 for each of the following loading conditions:

- (a) 0 kN to 28 kN
- (b) 12 kN to 28 kN
- (c) -28 kN to 12 kN

Homework requirement

- Print this page and use it as the cover.
- The submission should be each student's own work.
- The same day late submission will get a 10% penalty.
- Once the solution is posted, no more submission will be accepted.
- Write legibly. No credit will be given if the work is not readable.
- Take advantage of office hours.
 - Tuesdays & Thursdays 1 PM ~ 3 PM (tentative, may change in the week of June 1st)
 - Email me: japark@pvamu.edu to request a Zoom meeting if you cannot make it to the regular office hour due to any university excused reasons.

Problem 14-2

$$d = \frac{N}{P} = \frac{18}{10} = 1.8 \text{ in}$$

Table 14-2: $Y = 0.309$

$$\text{Eq. (13-34): } V = \frac{\pi dn}{12} = \frac{\pi(1.8)(600)}{12} = 282.7 \text{ ft/min}$$

$$\text{Eq. (14-4b): } K_v = \frac{1200 + 282.7}{1200} = 1.236$$

$$\text{Eq. (13-35): } W^t = 33\,000 \frac{H}{V} = 33\,000 \frac{2}{282.7} = 233.5 \text{ lbf}$$

$$\text{Eq. (14-7): } \sigma = \frac{K_v W^t P}{FY} = \frac{1.236(233.5)(10)}{1.0(0.309)} = 9340 \text{ psi} = 9.34 \text{ kpsi} \quad \text{Ans.}$$

Problem 14-6

$$d = mN = 2(20) = 40 \text{ mm}$$

Table 14-2: $Y = 0.322$

$$V = \frac{\pi dn}{60} = \frac{\pi(40)(10^{-3})(200)}{60} = 0.419 \text{ m/s}$$

$$\text{Eq. (14-6b): } K_v = \frac{6.1 + 0.419}{6.1} = 1.069$$

$$\text{Eq. (13-36): } W^t = \frac{60\,000H}{\pi dn} = \frac{60\,000(0.5)}{\pi(40)(200)} = 1.194 \text{ kN} = 1194 \text{ N}$$

$$\text{Eq. (14-8): } F = \frac{K_v W^t}{\sigma m Y} = \frac{1.069(1194)}{75(2.0)(0.322)} = 26.4 \text{ mm}$$

From Table 13-2, use $F = 28 \text{ mm}$. *Ans.*

Problem 14-12

$$d_p = \frac{16}{12} = 1.333 \text{ in}, \quad d_G = \frac{48}{12} = 4 \text{ in}$$

$$V = \frac{\pi(1.333)(700)}{12} = 244.3 \text{ ft/min}$$

$$\text{Eq. (14-4b): } K_v = \frac{1200 + 244.3}{1200} = 1.204$$

$$\text{Eq. (13-36): } W_t = 33\,000 \frac{H}{V} = 33\,000 \frac{1.5}{244.3} = 202.6 \text{ lbf}$$

Table 14-8: $C_p = 2100\sqrt{\text{psi}}$ [Note: Using Eq. (14-13) can result in wide variation in C_p due to wide variation in cast iron properties.]

$$\text{Eq. (14-12): } r_1 = \frac{1.333 \sin 20^\circ}{2} = 0.228 \text{ in}, \quad r_2 = \frac{4 \sin 20^\circ}{2} = 0.684 \text{ in}$$

Eq. (14-14):

$$\sigma_c = -2100 \left[\frac{1.204(202.6)}{F \cos 20^\circ} \left(\frac{1}{0.228} + \frac{1}{0.684} \right) \right]^{1/2} = -100(10^3)$$

$$F = \left(\frac{2100}{100(10^3)} \right)^2 \left[\frac{1.204(202.6)}{\cos 20^\circ} \right] \left(\frac{1}{0.228} + \frac{1}{0.684} \right) = 0.669 \text{ in}$$

Use $F = 0.75 \text{ in}$ *Ans.*

Problem 14-13

$$d_p = 5(24) = 120 \text{ mm}, \quad d_G = 5(48) = 240 \text{ mm}$$

$$V = \frac{\pi(120)(10^{-3})(50)}{60} = 0.3142 \text{ m/s}$$

$$\text{Eq. (14-6a): } K_v = \frac{3.05 + 0.3142}{3.05} = 1.103$$

$$W^t = \frac{60\,000H}{\pi d n} = \frac{60(10^3)H}{\pi(120)(50)} = 3.183H$$

where H is in kW and W^t is in kN

Table 14-8: $C_p = 163\sqrt{\text{MPa}}$ [Note: Using Eq. (14-13) can result in wide variation in C_p due to wide variation in cast iron properties].

$$\text{Eq. (14-12): } r_1 = \frac{120 \sin 20^\circ}{2} = 20.52 \text{ mm}, \quad r_2 = \frac{240 \sin 20^\circ}{2} = 41.04 \text{ mm}$$

$$\text{Eq. (14-14): } -690 = -163 \left[\frac{1.103(3.183)(10^3)H}{60 \cos 20^\circ} \left(\frac{1}{20.52} + \frac{1}{41.04} \right) \right]^{1/2}$$

$H = 3.94 \text{ kW}$ *Ans.*

Problem 6-9

Read from graph: $(10^3, 90)$ and $(10^6, 50)$. From $S = aN^b$

$$\log S_1 = \log a + b \log N_1$$

$$\log S_2 = \log a + b \log N_2$$

From which

$$\begin{aligned} \log a &= \frac{\log S_1 \log N_2 - \log S_2 \log N_1}{\log N_2 / N_1} \\ &= \frac{\log 90 \log 10^6 - \log 50 \log 10^3}{\log 10^6 / 10^3} \\ &= 2.2095 \end{aligned}$$

$$a = 10^{\log a} = 10^{2.2095} = 162.0 \text{ kpsi}$$

$$b = \frac{\log 50 / 90}{3} = -0.0851$$

$$(S_f)_{ax} = 162N^{-0.0851} \quad 10^3 \leq N \leq 10^6 \text{ in kpsi} \quad \text{Ans.}$$

Check:

$$[(S_f)_{ax}]_{10^3} = 162(10^3)^{-0.0851} = 90 \text{ kpsi}$$

$$[(S_f)_{ax}]_{10^6} = 162(10^6)^{-0.0851} = 50 \text{ kpsi}$$

The end points agree.

Problem 6-11

For AISI 4340 as-forged steel,

$$\text{Eq. (6-8): } S_e = 100 \text{ kpsi}$$

$$\text{Table 6-2: } a = 39.9, b = -0.995$$

$$\text{Eq. (6-19): } k_a = 39.9(260)^{-0.995} = 0.158$$

$$\text{Eq. (6-20): } k_b = \left(\frac{0.75}{0.30} \right)^{-0.107} = 0.907$$

Each of the other modifying factors is unity.

$$S_e = 0.158(0.907)(100) = 14.3 \text{ kpsi} \quad \text{Ans.}$$

For AISI 1040:

$$S'_e = 0.5(113) = 56.5 \text{ kpsi}$$

$$k_a = 39.9(113)^{-0.995} = 0.362$$

$$k_b = 0.907 \text{ (same as 4340)}$$

Each of the other modifying factors is unity

$$S_e = 0.362(0.907)(56.5) = 18.6 \text{ kpsi} \quad \text{Ans.}$$

Not only is AISI 1040 steel a contender, it has a superior endurance strength. *Ans.*

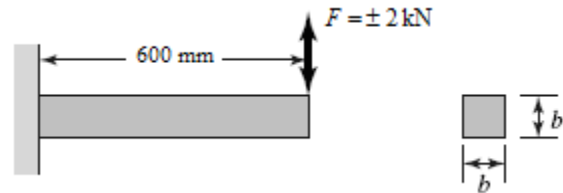
Problem 6-13

$L = 0.6 \text{ m}$, $F_a = 2 \text{ kN}$, $n = 1.5$, $N = 10^4$ cycles, $S_{ut} = 770 \text{ MPa}$, $S_y = 420 \text{ MPa}$ (Table A-20)

First evaluate the fatigue strength.

$$S'_e = 0.5(770) = 385 \text{ MPa}$$

$$k_a = 57.7(770)^{-0.718} = 0.488$$



Since the size is not yet known, assume a typical value of $k_b = 0.85$ and check later. All other modifiers are equal to one.

$$\text{Eq. (6-18): } S_e = 0.488(0.85)(385) = 160 \text{ MPa}$$

$$\text{In kpsi, } S_{ut} = 770/6.89 = 112 \text{ kpsi}$$

$$\text{Fig. 6-18: } f = 0.83$$

$$\text{Eq. (6-14): } a = \frac{(f S_{ut})^2}{S_e} = \frac{[0.83(770)]^2}{160} = 2553 \text{ MPa}$$

$$\text{Eq. (6-15): } b = -\frac{1}{3} \log \left(\frac{f S_{ut}}{S_e} \right) = -\frac{1}{3} \log \left(\frac{0.83(770)}{160} \right) = -0.2005$$

$$\text{Eq. (6-13): } S_f = aN^b = 2553(10^4)^{-0.2005} = 403 \text{ MPa}$$

Now evaluate the stress.

$$M_{\max} = (2000 \text{ N})(0.6 \text{ m}) = 1200 \text{ N} \cdot \text{m}$$

$$\sigma_a = \sigma_{\max} = \frac{Mc}{I} = \frac{M(b/2)}{b(b^3)/12} = \frac{6M}{b^3} = \frac{6(1200)}{b^3} = \frac{7200}{b^3} \text{ Pa, with } b \text{ in m.}$$

Compare strength to stress and solve for the necessary b .

$$n = \frac{S_f}{\sigma_a} = \frac{403(10^6)}{7200/b^3} = 1.5$$

$$b = 0.0299 \text{ m} \quad \text{Select } b = 30 \text{ mm.}$$

Since the size factor was guessed, go back and check it now.

$$\text{Eq. (6-25): } d_e = 0.808(hb)^{1/2} = 0.808b = 0.808(30) = 24.24 \text{ mm}$$

$$\text{Eq. (6-20): } k_b = \left(\frac{24.2}{7.62} \right)^{-0.107} = 0.88$$

Our guess of 0.85 was slightly conservative, so we will accept the result of

$$b = 30 \text{ mm. } \text{Ans.}$$

Checking yield,

$$\sigma_{\max} = \frac{7200}{0.030^3} (10^{-6}) = 267 \text{ MPa}$$

$$n_y = \frac{S_y}{\sigma_{\max}} = \frac{420}{267} = 1.57$$

Problem 6-16

From a free-body diagram analysis, the bearing reaction forces are found to be 2.1 kN at the left bearing and 3.9 kN at the right bearing. The critical location will be at the shoulder fillet between the 35 mm and the 50 mm diameters, where the bending moment is large, the diameter is smaller, and the stress concentration exists. The bending moment at this point is $M = 2.1(200) = 420 \text{ kN}\cdot\text{mm}$. With a rotating shaft, the bending stress will be completely reversed.

$$\sigma_{\text{rev}} = \frac{Mc}{I} = \frac{420 (35/2)}{(\pi/64)(35)^4} = 0.09978 \text{ kN/mm}^2 = 99.8 \text{ MPa}$$

This stress is far below the yield strength of 390 MPa, so yielding is not predicted. Find the stress concentration factor for the fatigue analysis.

$$\text{Fig. A-15-9: } r/d = 3/35 = 0.086, D/d = 50/35 = 1.43, K_t = 1.7$$

Get the notch sensitivity either from Fig. 6-20, or from the curve-fit Eqs. (6-34) and (6-35a). Using the equations, with $S_{ut} = 470 \text{ MPa} = 68.2 \text{ kpsi}$ and $r = 3 \text{ mm} = 0.118 \text{ in}$,

$$\sqrt{a} = 0.246 - 3.08(10^{-3})(68.2) + 1.51(10^{-5})(68.2)^2 - 2.67(10^{-8})(68.2)^3 = 0.09771$$

$$q = \frac{1}{1 + \frac{\sqrt{a}}{\sqrt{r}}} = \frac{1}{1 + \frac{0.09771}{\sqrt{0.118}}} = 0.78$$

$$\text{Eq. (6-32): } K_f = 1 + q(K_t - 1) = 1 + 0.78(1.7 - 1) = 1.55$$

$$\text{Eq. (6-8): } S'_e = 0.5S_{ut} = 0.5(470) = 235 \text{ MPa}$$

$$\text{Eq. (6-19): } k_a = aS_{ut}^b = 4.51(470)^{-0.265} = 0.88$$

$$\text{Eq. (6-24): } k_b = 1.24d^{-0.107} = 1.24(35)^{-0.107} = 0.85$$

$$\text{Eq. (6-26): } k_c = 1$$

$$\text{Eq. (6-18): } S_e = k_a k_b k_c S'_e = (0.88)(0.85)(1)(235) = 176 \text{ MPa}$$

$$n_f = \frac{S_e}{K_f \sigma_{\text{rev}}} = \frac{176}{1.55(99.8)} = 1.14 \quad \text{Infinite life is predicted.} \quad \text{Ans.}$$

Problem 6-25

Given: $F_{\text{max}} = 28 \text{ kN}$, $F_{\text{min}} = -28 \text{ kN}$. From Table A-20, for AISI 1040 CD,

$$S_{ut} = 590 \text{ MPa}, S_y = 490 \text{ MPa},$$

Check for yielding

$$\sigma_{\text{max}} = \frac{F_{\text{max}}}{A} = \frac{28\,000}{10(25-6)} = 147.4 \text{ N/mm}^2 = 147.4 \text{ MPa}$$

$$n_y = \frac{S_y}{\sigma_{\text{max}}} = \frac{490}{147.4} = 3.32 \quad \text{Ans.}$$

Determine the fatigue factor of safety based on infinite life

$$\text{Eq. (6-8):} \quad S'_e = 0.5(590) = 295 \text{ MPa}$$

$$\text{Eq. (6-19):} \quad k_a = aS_{ut}^b = 4.51(590)^{-0.265} = 0.832$$

$$\text{Eq. (6-21):} \quad k_b = 1 \quad (\text{axial})$$

$$\text{Eq. (6-26):} \quad k_c = 0.85$$

$$\text{Eq. (6-18):} \quad S_e = k_a k_b k_c S'_e = (0.832)(1)(0.85)(295) = 208.6 \text{ MPa}$$

$$\text{Fig. 6-20:} \quad q = 0.83$$

$$\text{Fig. A-15-1:} \quad d/w = 0.24, K_t = 2.44$$

$$K_f = 1 + q(K_t - 1) = 1 + 0.83(2.44 - 1) = 2.20$$

$$\sigma_a = K_f \left| \frac{F_{\text{max}} - F_{\text{min}}}{2A} \right| = 2.2 \left| \frac{28\,000 - (-28\,000)}{2(10)(25-6)} \right| = 324.2 \text{ MPa}$$

$$\sigma_m = K_f \frac{F_{\text{max}} + F_{\text{min}}}{2A} = 0$$

Note, since $\sigma_m = 0$, the stress is completely reversing, and

$$n_f = \frac{S_e}{\sigma_a} = \frac{208.6}{324.2} = 0.64 \quad \text{Ans.}$$

Since infinite life is not predicted, estimate the life from the S - N diagram. With $\sigma_m = 0$, the stress state is completely reversed, and the S - N diagram is applicable for σ_a .

$$S_{ut} = 590/6.89 = 85.6 \text{ kpsi}$$

$$\text{Fig. 6-18:} \quad f = 0.87$$

$$\text{Eq. (6-14):} \quad a = \frac{(f S_{ut})^2}{S_e} = \frac{[0.87(590)]^2}{208.6} = 1263$$

$$\text{Eq. (6-15): } b = -\frac{1}{3} \log \left(\frac{f S_{ut}}{S_e} \right) = -\frac{1}{3} \log \left(\frac{0.87(590)}{208.6} \right) = -0.1304$$

$$\text{Eq. (6-16): } N = \left(\frac{\sigma_{rev}}{a} \right)^{1/b} = \left(\frac{324.2}{1263} \right)^{\frac{1}{-0.1304}} = 33\,812 \text{ cycles}$$

$$N = 34\,000 \text{ cycles} \quad \text{Ans.}$$

Problem 6-27

$$S_{ut} = 590 \text{ MPa}, S_y = 490 \text{ MPa}$$

$$\text{(a) } F_{\max} = 28 \text{ kN}, F_{\min} = 0 \text{ kN}$$

Check for yielding

$$\sigma_{\max} = \frac{F_{\max}}{A} = \frac{28\,000}{10(25-6)} = 147.4 \text{ N/mm}^2 = 147.4 \text{ MPa}$$

$$n_y = \frac{S_y}{\sigma_{\max}} = \frac{490}{147.4} = 3.32 \quad \text{Ans.}$$

From Prob. 6-25: $S_e = 208.6 \text{ MPa}$, $K_f = 2.2$

$$\sigma_a = K_f \left| \frac{F_{\max} - F_{\min}}{2A} \right| = 2.2 \left| \frac{28\,000 - 0}{2(10)(25-6)} \right| = 162.1 \text{ MPa}$$

$$\sigma_m = K_f \frac{F_{\max} + F_{\min}}{2A} = 2.2 \left[\frac{28\,000 + 0}{2(10)(25-6)} \right] = 162.1 \text{ MPa}$$

For the modified Goodman criteria,

$$\frac{1}{n_f} = \frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{162.1}{208.6} + \frac{162.1}{590}$$

$$n_f = 0.95 \quad \text{Ans.}$$

Since infinite life is not predicted, estimate a life from the S-N diagram. First, find an equivalent completely reversed stress. For the modified Goodman criteria, see Ex. 6-12.

$$\sigma_{rev} = \frac{\sigma_a}{1 - (\sigma_m / S_{ut})} = \frac{162.1}{1 - (162.1 / 590)} = 223.5 \text{ MPa}$$

$$\text{Fig. 6-18: } f = 0.87$$

$$\text{Eq. (6-14): } a = \frac{(f S_{ut})^2}{S_e} = \frac{[0.87(590)]^2}{208.6} = 1263$$

$$\text{Eq. (6-15): } b = -\frac{1}{3} \log \left(\frac{f S_{ut}}{S_e} \right) = -\frac{1}{3} \log \left(\frac{0.87(590)}{208.6} \right) = -0.1304$$

Eq. (6-16): $N = \left(\frac{\sigma_{\text{rev}}}{a} \right)^{1/b} = \left(\frac{223.5}{1263} \right)^{-0.1304} = 586\,000 \text{ cycles} \quad \text{Ans.}$

(b) $F_{\text{max}} = 28 \text{ kN}, F_{\text{min}} = 12 \text{ kN}$

The maximum load is the same as in part (a), so

$$\sigma_{\text{max}} = 147.4 \text{ MPa}$$

$$n_y = 3.32 \quad \text{Ans.}$$

Factor of safety based on infinite life:

$$\sigma_a = K_f \left| \frac{F_{\text{max}} - F_{\text{min}}}{2A} \right| = 2.2 \left| \frac{28\,000 - 12\,000}{2(10)(25 - 6)} \right| = 92.63 \text{ MPa}$$

$$\sigma_m = K_f \frac{F_{\text{max}} + F_{\text{min}}}{2A} = 2.2 \left[\frac{28\,000 + 12\,000}{2(10)(25 - 6)} \right] = 231.6 \text{ MPa}$$

$$\frac{1}{n_f} = \frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{92.63}{208.6} + \frac{231.6}{590}$$

$$n_f = 1.20 \quad \text{Ans.}$$

(c) $F_{\text{max}} = 12 \text{ kN}, F_{\text{min}} = -28 \text{ kN}$

The compressive load is the largest, so check it for yielding.

$$\sigma_{\text{min}} = \frac{F_{\text{min}}}{A} = \frac{-28\,000}{10(25 - 6)} = -147.4 \text{ MPa}$$

$$n_y = \frac{S_{yc}}{\sigma_{\text{min}}} = \frac{-490}{-147.4} = 3.32 \quad \text{Ans.}$$

Factor of safety based on infinite life:

$$\sigma_a = K_f \left| \frac{F_{\text{max}} - F_{\text{min}}}{2A} \right| = 2.2 \left| \frac{12\,000 - (-28\,000)}{2(10)(25 - 6)} \right| = 231.6 \text{ MPa}$$

$$\sigma_m = K_f \frac{F_{\text{max}} + F_{\text{min}}}{2A} = 2.2 \left[\frac{12\,000 + (-28\,000)}{2(10)(25 - 6)} \right] = -92.63 \text{ MPa}$$

For $\sigma_m < 0$, $n_f = \frac{S_e}{\sigma_a} = \frac{208.6}{231.6} = 0.90 \quad \text{Ans.}$

Since infinite life is not predicted, estimate a life from the S-N diagram. For a negative mean stress, we shall assume the equivalent completely reversed stress is the same as the actual alternating stress. Get a and b from part (a).

Eq. (6-16):
$$N = \left(\frac{\sigma_{\text{rev}}}{a} \right)^{1/b} = \left(\frac{231.6}{1263} \right)^{\frac{1}{-0.1304}} = 446\,000 \text{ cycles} \quad \text{Ans.}$$

Above Average

Grade:

Name: Aamna Jangda¹

This assignment is due at NOON on October 2st, 2020. Complete the following problems from Shigley's Mechanical Engineering Design – 10th edition. There are a total of 10 problems in this homework.

1. Problem 14-2

A steel spur pinion has a diametral pitch of 10 teeth/in, 18 teeth cut full-depth with a 20° pressure angle, and a face width of 1 in. This pinion is expected to transmit 2 hp at a speed of 600 rev/min. Determine the bending stress.

2. Problem 14-6

A 20° full-depth steel spur pinion has 20 teeth and a module of 2 mm and is to transmit 0.5 kW at a speed of 200 rev/min. Find an appropriate face width if the bending stress is not to exceed 75 MPa.

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A gearset has a module of 5 mm, a 20° pressure angle, and a 24-tooth cast-iron spur pinion driving a 48-tooth cast-iron gear. The pinion is to rotate at 50 rev/min. What horsepower input can be used with this gearset if the contact stress is limited to 690 MPa and $F = 60$ mm?

5. Problem 6-9

For the interval $10^3 \leq N \leq 10^6$ cycles, develop an expression for the axial fatigue strength $(S'_f)_{ax}$ for the polished specimens of 4130 used to obtain Fig. 6-10. The ultimate strength is $S_{ut} = 125$ kpsi and the endurance limit is $(S'_e)_{ax} = 50$ kpsi.

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A solid square rod is cantilevered at one end. The rod is 0.6 m long and supports a completely reversing transverse load at the other end of ± 2 kN. The material is AISI 1080 hot-rolled steel. If the rod must support this load for 10^4 cycles with a design factor of 1.5, what dimension should the square cross section have? Neglect any stress concentrations at the support end.

8. Problem 6-16

Summary of Comments on MCEG4043-HW2.pdf

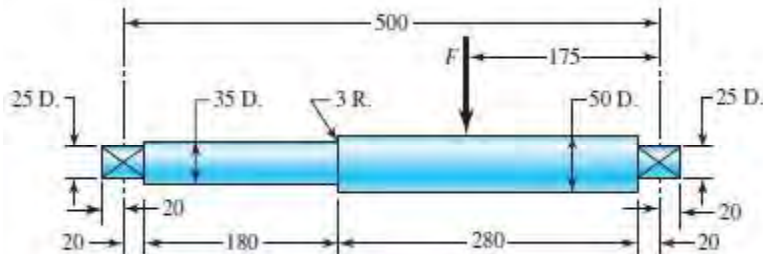
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 Number: 1 Author: Amna J Date: Indeterminate
Aamna Jangda

Grade:

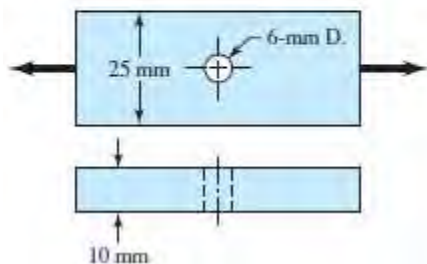
Name: Aamna Jangda¹

The rotating shaft shown in the figure is machined from AISI 1020 CD steel. It is subjected to a force of $F = 6$ kN. Find the minimum factor of safety for fatigue based on infinite life. If the life is not infinite, estimate the number of cycles. Be sure to check for yielding.



9. Problem 6-25

The cold-drawn AISI 1040 steel bar shown in the figure is subjected to a completely reversed axial load fluctuating between 28 kN in compression to 28 kN in tension. Estimate the fatigue factor of safety based on achieving infinite life and the yielding factor of safety. If infinite life is not predicted, estimate the number of cycles to failure.



10. Problem 6-27

Using the modified Goodman criterion for infinite life, repeat Prob. 6-25 for each of the following loading conditions:

- (a) 0 kN to 28 kN
- (b) 12 kN to 28 kN
- (c) -28 kN to 12 kN

Homework requirement

- Print this page and use it as the cover.
- The submission should be each student's own work.
- The same day late submission will get a 10% penalty.
- Once the solution is posted, no more submission will be accepted.
- Write legibly. No credit will be given if the work is not readable.
- Take advantage of office hours.
 - Tuesdays & Thursdays 1 PM ~ 3 PM (tentative, may change in the week of June 1st)
 - Email me: japark@pvamu.edu to request a Zoom meeting if you cannot make it to the regular office hour due to any university excused reasons.



 Number: 1 Author: Amna J Date: Indeterminate
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 Number: 2 Author: japark Subject: Sticky Note Date: 12/15/2020 12:46:26 PM
Neat work as always! Graded 6-11 and 6-27.
So what is your verdict for 6-11?

10/10

HW #2

14-2) Given

$$P = \text{diametral pitch} = 10 \text{ teeth/in}$$

$$N = \text{number of teeth} = 18 \text{ teeth}$$

$$n = \text{speed} = 600 \text{ rev/min}$$

$$h = \text{horsepower} = 2 \text{ hp}$$

$$d = \frac{N}{P} = \frac{18}{10} = 1.8 \text{ in}$$

$$\text{Table 14-2} \Rightarrow 18 \text{ teeth, } Y = 0.309$$

$$V = \frac{\pi d n}{12} = \frac{\pi (1.8)(600)}{12} = 282.7 \text{ ft/min}$$

$$K_v = \frac{1200 + 282.7}{1200} = 1.236$$

$$W^t = 33000 \frac{H}{V} = 33000 \frac{2}{282.7} = 233.5 \text{ lbf}$$

$$\sigma = \frac{K_v W^t P}{F Y} = \frac{1.236 (233.5)(10)}{1.0(0.309)}$$

$$= 9340 \text{ psi} = \boxed{9.34 \text{ kpsi}}$$

$$14-6) d = mN = 2(20) = 40 \text{ mm}$$

$$Y = 0.322$$

$$V = \frac{\pi d n}{60} = \frac{\pi (40)(10^{-3})(200)}{60} = 0.419 \text{ m/s}$$

$$K_v = \frac{6.1 + 0.419}{6.1} = 1.069$$

$$W^t = \frac{60000 H}{\pi d n} = \frac{60000(0.5)}{\pi (40)(200)} = 1.194 \text{ kN} = 1194 \text{ N}$$

$$14-12) d_p = \frac{N_p}{P} \Rightarrow d_p = \frac{16}{12} = 1.333 \text{ in}$$

$$d_g = \frac{N_g}{P} \Rightarrow \frac{48}{12} = 4 \text{ in}$$

$$V = \frac{\pi d_p n}{12} = \frac{\pi 1.333 \times 700}{12} = 244.285 \text{ ft/min}$$

$$K_v = \frac{1200 + V}{1200} = \frac{1200 + 244.285}{1200} = 1.204$$

$$W^t = 33000 \frac{H}{V} \Rightarrow 33000 \frac{1.5}{244.285} = 202.632 \text{ hp}$$

$$r_1 = \frac{d_p \sin \phi}{2} \Rightarrow \frac{1.333 \sin 20^\circ}{2} = 0.228 \text{ in}$$

$$r_2 = \frac{d_g \sin \phi}{2} \Rightarrow \frac{4 \sin 20^\circ}{2} = 0.684 \text{ in}$$

14-12 con'd $\sigma_c = -C_p \left[\frac{K_v W_t}{F \cos \phi} \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \right]^{1/2}$

$$-100 \times 10^3 = -2100 \left[\frac{1.204(202.632)}{F \cos 20^\circ} \left(\frac{1}{0.228} + \frac{1}{0.684} \right) \right]^{1/2}$$

$$(100 \times 10^3)^2 = (2100)^2 \left[\frac{1.204(202.632)}{F \cos 20^\circ} \left(\frac{1}{0.228} + \frac{1}{0.684} \right) \right]$$

$$F = \frac{(2100)^2}{(100 \times 10^3)^2} \left[\frac{1.204(202.632)}{\cos 20^\circ} \left(\frac{1}{0.228} + \frac{1}{0.684} \right) \right]$$

$$= 0.669 \text{ in}$$

14-13) $d_p = 5(24) = 120 \text{ mm}$ $d_g = 5(48) = 240 \text{ mm}$

$$V = \frac{\pi (120)(10^{-3})(50)}{60} = 0.3142 \text{ m/s}$$

$$K_v = \frac{3.05 + 0.3142}{3.05} = 1.103$$

$$W_t = \frac{60000 H}{\pi d_n} = \frac{60(10^3)(H)}{\pi (120)(50)} = 3.183 H$$

From table: $C_p = 163 \text{ MPa}$

$$r_1 = \frac{120 \sin 20^\circ}{2} = 20.52 \text{ mm} \quad r_2 = \frac{240 \sin 20^\circ}{2} = 41.04 \text{ mm}$$

$$-690 = -163 \left[\frac{1.103(3.183)(10^3) H}{60 \cos 20^\circ} \left(\frac{1}{20.52} + \frac{1}{41.04} \right) \right]^{1/2}$$

$$H = 3.94 \text{ kW}$$

6-9) From graph

$$N = 10^3 \quad S_f = 90 \text{ kpsi} \quad S = aN^b$$

$$N = 10^6 \quad S_f = 50 \text{ kpsi}$$

$$\log S_1 = \log a + b \log N_1$$

$$\log S_2 = \log a + b \log N_2$$

$$\log a = \frac{\log S_1 \log N_2 - \log S_2 \log N_1}{\log N_2 / N_1}$$

$$= \frac{\log 90 \log 10^6 - \log 50 \log 10^3}{\log 10^6 / 10^3} = -2.2095$$

$$a = 10^{1.054} = 10^{2.2095} = 162.0 \text{ kpsi}$$

$$b = \frac{\log 50 / 90}{3} = -0.0851$$

$$(S_f)_{2x} = 162 N^{-0.0851}$$

6-11) For AISI 4340 $\Rightarrow S_{ut} = 260 \text{ kpsi}$

$$S_{ut} > 200 \text{ kpsi} \quad \text{is } S_e' = 100 \text{ kpsi}$$

$$\text{Table: } a = 39.9, \quad b = -0.995$$

$$K_a = a S_{ut}^b \Rightarrow 39.9 \times 260^{-0.995} \Rightarrow K_a = 0.1578$$

$$K_b = 0.879 d^{-0.107} \quad d = 6.75 \quad K_b = 0.879 (0.75)^{-0.107}$$

$$S_e = K_a K_b S_e' \Rightarrow 0.1378 \times 0.9065 \times 100 = 12.43 \text{ kpsi} \quad K_c = 0.9065$$

$$\text{Carbon steel} \Rightarrow 113 \text{ kpsi} \quad S_{ut} \leq 200 \text{ kpsi} \quad \text{is } S_e' = 0.5 S_{ut}$$

$$S_e' = 0.5 (113) = 56.5 \text{ kpsi} \quad K_a = a S_{ut}^b = 39.9 \times 113^{-0.995} = 0.2615$$

$$K_b = 0.879 (0.75)^{-0.107} \Rightarrow K_b = 0.9065 \quad S_e = K_a K_b S_e'$$

$$S_e = 0.2615 \times 0.9065 \times 56.5 \Rightarrow 13.517 \text{ kpsi} \quad \text{Carbon is greater}$$

6-13) $L = 0.6 \text{ m}$ $F_a = 2 \text{ kN}$ $n = 1.5$ $N = 10^4$ $S_{ut} = 770 \text{ MPa}$

$S_y = 420 \text{ MPa}$

$S'_a = 0.5(770) = 385$

$k_a = 57.7(770)^{-0.718} = 0.488$

$F = 2 \text{ kN}$



$S_e = 0.488(.85)(385) = 160 \text{ MPa}$

$S_{ut} = 770 / 6.84 = 112 \text{ kpsi}$

$f = 0.83$

$a = \frac{(f S_{ut})^2}{S_e} = \frac{(0.83 \cdot 770)^2}{160} = 2553 \text{ MPa}$

$b = -\frac{1}{3} \log \left(\frac{f S_{ut}}{S_e} \right) = -\frac{1}{3} \log \left(\frac{0.83(770)}{160} \right) = -.2005$

$S_f = a N^b = 2553(10^4)^{-.2005} = 403 \text{ MPa}$

$M_{max} = 2000 \text{ N} (0.6 \text{ m}) = 1200 \text{ N}\cdot\text{m}$

$\sigma_c = \sigma_{max} = \frac{Mc}{I} = \frac{6M}{b^3} = \frac{6(1200)}{b^3} = \frac{7200}{b^3} \text{ Pa}$

$n = \frac{S_f}{\sigma_c} = \frac{403(10^6)}{7200/b^3} = 1.5$

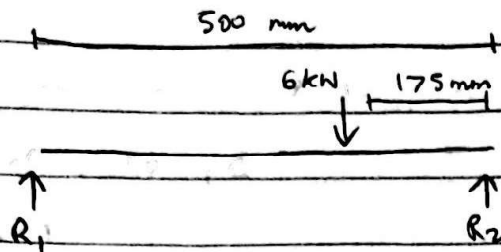
$b = 0.0299 \text{ m}$ $b = 30 \text{ mm}$

$d_c = 0.808(hb)^{1/2} = 0.808(30) = 24.24 \text{ mm}$

$k_s = \left(\frac{24.2}{7.62} \right)^{-1.07} = 0.88$

$b = 30 \text{ mm}$

6-16)



$$\sum F_y = 0 \quad R_1 + R_2 = 6$$

$$\sum M_R = 0 \Rightarrow R_2(500) = 6(325)$$

$$R_2 = 3.9 \text{ kN} \quad R_1 = 2.1 \text{ kN}$$

$$S_{ut} = 470 \text{ MPa} \quad S_{ut} < 1400 \text{ MPa}$$

$$S_y = 390 \text{ MPa} \quad S_e' = 0.5 S_{ut} = 0.5(470) = 235 \text{ MPa}$$

$$a = 4.51 \quad b = -0.265 \quad k_a = a S_{ut} = 4.51(470)^{-0.265} = 0.883$$

$$k_b = \left(\frac{d}{7.62}\right)^{-0.107} = \left(\frac{35}{7.62}\right)^{-0.107} = 0.85 \quad 0.11 \leq d \leq 2 \text{ in}$$

$$\text{Bending factor } k_c = 1 \quad k_d = 1 \quad k_e = 1$$

$$S_e = k_a k_b k_c k_d k_e S_e' = 0.883 \times 0.85 \times 235 = 176.4 \text{ MPa}$$

Table
A-15

$$\frac{r}{d} = \frac{3}{35} = 0.0857$$

$$\frac{D}{d} = \frac{50}{35} = 1.428 \quad k_f = 1.7$$

$$r = 3 \text{ mm} = .12 \text{ in} \quad S_{ut} = 470 \text{ MPa} = 68.16 \text{ kpsi}$$

$$\sqrt{a} = 0.246 - 3.08 \times 10^{-3} S_{ut} + 1.51 \times 10^{-5} S_{ut}^2 - 2.67 \times 10^{-8} S_{ut}^3$$

$$\sqrt{a} = 0.246 - 3.08 \times 10^{-3} \times 68.16 + 1.51 \times 10^{-5} \times 68.16^2$$

$$\hookrightarrow -2.67 \times 10^{-8} \times 68.16^3 = 0.0977 \sqrt{\text{in}}$$

$$= 0.0977 \sqrt{25.4 \text{ mm}} = 0.4927 \sqrt{\text{mm}}$$

$$k_f = 1 + \frac{(1.7-1)}{1 + (0.4927/\sqrt{3})} = 1.54$$

$$M = R_1 x = 2.1(200) = 420 \text{ Nm}$$

$$\frac{I}{c} = \frac{\pi d^3}{32} = \frac{\pi (35^3)}{32} = 4.269 \times 10^3 \text{ mm}^3$$

$$\sigma_{rev} = k_f \frac{M}{I/c} \Rightarrow 1.54 \frac{420}{4.269 \times 10^3} = 153.67 \text{ MPa}$$

$$n_f = \frac{S_e}{k_f \sigma} = \frac{176.4}{1.54 \times 153.67} = 1.13 \quad \boxed{n_f = 1.13}$$

$$6-29) F_{max} = 28 \text{ kN} \quad F_{min} = -28 \text{ kN}$$

$$S_{ur} = 590 \text{ MPa} \quad S_y = 490 \text{ MPa}$$

$$\sigma_{max} = \frac{F_{max}}{A} = \frac{28000}{10(25-6)} = 147.4 \text{ N/mm}^2 = 147.4 \text{ MPa}$$

$$n_y = \frac{S_y}{\sigma_{max}} = \frac{490}{147.4} = 3.32$$

$$S_e = 0.5 \cdot 590 = 295 \text{ MPa}$$

$$k_a = a S_{ut}^b = 4.51 (590)^{-0.85} = 0.832$$

$$k_b = 1 \quad k_c = 0.85$$

$$S_r = k_a k_b k_c S_e = (0.832)(1)(0.85)(295) = 208.6 \text{ MPa}$$

$$q = 0.83 \quad d_w = 24 \quad k_t = 2.44$$

$$k_f = 1 + q(k_t - 1) = 1 + 0.83(2.44 - 1) = 2.20$$

$$\sigma_r = K_f \left| \frac{F_{max} - F_{min}}{2A} \right| = 2.2 \left| \frac{28000 - (-28000)}{2(10 \times 25 - 6)} \right| = 324.2 \text{ MPa}$$

$$\sigma_m = K_f \cdot \frac{F_{max} + F_{min}}{2A} = 0 \quad \frac{1}{n_f} = \frac{\sigma}{S} + \frac{\sigma}{S_u} = \frac{324.2}{208.6} + \frac{0}{590}$$

$$n_f = 0.64$$

$$S_t = 590 / 6.89 = 85.6 \text{ ksi} \quad f = 0.87$$

$$a = \frac{(f S_{ut})^2}{S_e} = \frac{(82.9)^2}{208.6} = 1263$$

$$b = \frac{1}{3} \log \left(\frac{f S_{ut}}{S_e} \right) = \frac{1}{3} \log \left(\frac{82.9}{208.6} \right) = -0.134$$

$$N = \left(\frac{\sigma_r}{a} \right)^{1/b} = \left(\frac{324.2}{1263} \right)^{1/-0.134} = 33812 \text{ cycles}$$

6-27. $S_{ut} = 590 \text{ MPa}$ $S_y = 490 \text{ MPa}$
 $F_{max} = 28 \text{ kN}$ $F_{min} = 0 \text{ kN}$

$$\sigma_{max} = \frac{F_{max}}{A} = \frac{28000}{10(25-6)} = 147.4 \text{ N/mm}^2 = 147.4 \text{ MPa}$$

$$n_y = \frac{S_y}{\sigma_{max}} = \frac{490}{147.4} = \boxed{3.32}$$

$$\sigma_a = K_f \left| \frac{F_{max} - F_{min}}{2A} \right| = 2.2 \left| \frac{28000 - 0}{2(10)(25-6)} \right| = 162.1 \text{ MPa}$$

$$\sigma_m = K_f \frac{F_{max} + F_{min}}{2A} = 2.2 \left[\frac{28000 + 0}{2(10)(25-6)} \right] = 162.1 \text{ MPa}$$

$$\frac{1}{n_f} = \frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{162.1}{208.6} + \frac{162.1}{590} = \boxed{n_f = 0.95}$$

$$\sigma_{rev} = \frac{\sigma_a}{1 - (S_e/S_{ut})} = \frac{162.1}{1 - 162.1/590} = 223.5 \text{ MPa} \neq 0.57$$

$$c = \frac{(F_y)^n}{S_e} = \frac{(0.87)(590)^2}{208.6} = 1263$$

$$b = -1/3 \log \left(\frac{f S_{ut}}{S_e} \right) = -1/3 \log \left(\frac{0.87(590)}{208.6} \right) = -0.1304$$

$$N = \left(\frac{\sigma_{rev}}{a} \right)^{1/b} = \left(\frac{223.5}{1263} \right)^{1/-0.1304} = \boxed{586000 \text{ cycles}}$$

b) $F_{max} = 28 \text{ kN}$ $F_{min} = 12 \text{ kN}$ $\sigma_a = 92.63 \text{ MPa}$ $\sigma_m = 231.6 \text{ MPa}$

$$\frac{1}{n_f} = \frac{92.63}{208.6} + \frac{231.6}{590} = \boxed{n_f = 1.2}$$

8-27 c) $F_{max} = 12 \text{ kN}$ $F_{min} = -28 \text{ kN}$

con't $\sigma_{min} = \frac{F_{min}}{A} = \frac{-28000}{10(25-6)} = -147.4 \text{ MPa}$

$n_s = \frac{S_{yc}}{\sigma_{min}} = \frac{-490}{-147.4} = \boxed{3.32}$

$\sigma_a = K_f \left| \frac{F_{max} - F_{min}}{2A} \right|$

$\sigma_a = 231.6 \text{ MPa}$ $\sigma_{min} = -92.63 \text{ MPa}$

$\sigma_{min} < 0$ $n_f = \frac{S_e}{\sigma} = \frac{208.6}{231.6} = \boxed{0.90}$

$N = \left(\frac{\sigma_{min}}{a} \right)^{1/b} = \left(\frac{-231.6}{-126.3} \right)^{1/-1.24} = \boxed{446000 \text{ cycles}}$

Average



Machine 2 HW #2 Don Brodard

1.) 14-2:

Given: $p = 10 \text{ t/in}$ Find Bending stress:

$$\phi = 20^\circ$$

$$n_p = 600 \text{ rpm}$$

$$H = 2 \text{ hp}$$

$$N_p = 18$$

$$b = 1 \text{ in}$$

$$d = N/p = 18/10 = 1.8 \text{ in}$$

$$Y_{\text{for } N_p = 18} \text{ is } 0.309$$

$$V = \frac{\pi d n_p}{12} \Rightarrow \frac{\pi (1.8)(600)}{12} = 282.7 \text{ ft/min}$$

$$K_v = \frac{1200 + 282.7}{1200} = 1.24$$

$$W^t = \frac{T}{(d/2)} = \frac{(63025 H)}{(\frac{n_p d}{2})} = \frac{63025 \cdot 2}{600(1.5/2)} = 233.4 \text{ lbf}$$

$$\sigma = F Y = \frac{K_v W^t P}{1 \cdot 0.309} = \boxed{9366.21 \text{ psi}}$$

2.) 14-6:

Given: $\phi = 20^\circ$ Find F :

$$N_p = 20 \text{ teeth}$$

$$m = 2 \text{ mm}$$

$$H = 0.5 \text{ kW}$$

$$n_p = 200 \text{ rpm}$$

$$\sigma \leq 75 \text{ MPa}$$

$$d = m N_p = 2 \cdot 20 = 40 \text{ mm}$$

$$Y_{20} = 0.322$$

$$V = \frac{\pi d n_p}{60} = \frac{\pi \cdot 0.04 \cdot 200}{60} = 0.418 \text{ m/s}$$

$$V = \frac{\pi d n_p}{60} = \frac{\pi \cdot 0.04 \cdot 200}{60} = 0.418 \text{ m/s}$$

$$K_v = \frac{6.1 + V}{6.1} = \frac{6.1 + 0.418}{6.1} = 1.069$$

$$W^t = \frac{60 H}{\pi d n_p} = \frac{60 \cdot 500}{\pi \cdot 0.04 \cdot 200} = 1194 \text{ N}$$

$$\sigma = \frac{F_m Y}{F} \Rightarrow F = \frac{K_v W^t}{75 \cdot 2 \cdot 0.322} = \frac{1.069 \cdot 1194}{75 \cdot 2 \cdot 0.322} = 26.4 \text{ mm}$$

Summary of Comments on HW_2_Don_Bradford.pdf

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Number: 1 Author: japark Subject: Sticky Note Date: 12/15/2020 12:48:10 PM

Neat work as always! Graded 6-11 and 6-27.
So what is your verdict for 6-11?

6/10

3.) 14-12:

Given: $N_{p1} = 16$ Find: Force width

$$\phi = 20^\circ$$

$$N_{p2} = 48$$

$$p = 12$$

$$H = 1.5$$

$$d_p = N_{p1}/p = 16/12 = 1.333 \text{ in}$$

$$d_g = N_{p2}/p = 48/12 = 4 \text{ in}$$

$$n_1 = 700 \text{ rpm}$$

$$V = \frac{\pi d_p n}{12} = \frac{\pi (1.333) \cdot 700}{12} = 244.285 \text{ ft/min}$$

$$\sigma = 120 \text{ kpsi}$$

$$K_v = \frac{1200}{1200 + \frac{V}{1200}} = \frac{1200}{1200 + \frac{244.285}{1200}} = 1.204$$

$$W^t = 33000 \frac{H}{V} = 33000 \cdot \left(\frac{1.5}{244.285} \right) = 202.632 \text{ lbf}$$

$$C_p = 2100 \sqrt{\text{psi}}$$

$$r_1 = \frac{d_p \sin \phi}{2} = \frac{1.333 \sin 20^\circ}{2} = 0.228 \text{ in}$$

$$r_2 = \frac{d_g \sin \phi}{2} = \frac{4 \sin 20^\circ}{2} = 0.684 \text{ in}$$

$$\sigma = -C_p \left(\frac{K_v W^t}{F \cos \phi} \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \right)^{1/2} \Rightarrow F = \frac{C_p^2}{\sigma^2} \left(\frac{K_v W^t}{\cos \phi} \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \right)$$

$$F = \frac{2100^2}{(120 \times 10^3)^2} \left(\frac{1.204 (202.632)}{\cos 20^\circ} \left(\frac{1}{0.228} + \frac{1}{0.684} \right) \right) = 0.669 \text{ in}$$

4.) 14-13:

Given: $m = 5 \text{ mm}$ Find: Max horsepower input

$$\phi = 20^\circ$$

$$N_p = 24$$

$$d_p = m N_p = 5 \cdot 24 = 120 \text{ mm}$$

$$C_p = 163$$

$$N_g = 48$$

$$d_g = m N_g = 5 \cdot 48 = 240 \text{ mm}$$

$$n_p = 50 \text{ rpm}$$

$$V = \pi d_p n_p = \frac{1}{60} \cdot \pi \cdot 0.12 \cdot 50 = 0.3142 \text{ m/s}$$

$$\sigma = 690$$

$$K_v = \frac{3.05 + V}{3.05} = \frac{3.05 + 0.3142}{3.05} = 1.103$$

$$F = 60 \text{ mm}$$

$$W^t = \frac{60H}{\pi d_p n_p} = \frac{60H}{\pi \cdot 0.120 \cdot 50} = 3.183H \quad r_1 = \frac{d_p \sin \phi}{2} = \frac{120 \sin 20^\circ}{2} = 20.52$$

$$r_2 = \frac{d_g \sin 20^\circ}{2} = \frac{240 \sin 20^\circ}{2} = 41.04$$

$$\sigma = -C_p \left[\frac{K_v W^t \left(\frac{1}{r_1} + \frac{1}{r_2} \right)}{F \cos \phi} \right]^{1/2} \Rightarrow -690 = 163 \left[\frac{1.103(3.183H) \left(\frac{1}{20.52} + \frac{1}{41.04} \right)}{60 \cos 20^\circ} \right]^{1/2}$$

$$\Rightarrow 690^2 = 163^2 \left(\frac{3510.844(H)}{60 \cos 20^\circ} \right) \Rightarrow H = 3.94 \text{ kW}$$

5.) 6-9:

Given: Find an expression for axial fatigue strength

$$10^3 \leq N \leq 10^6 \quad @ N = 10^3 \quad S_f = 90 \text{ Kpsi}$$

$$S_{ut} = 125 \text{ Kpsi} \quad @ N = 10^6 \quad S_f = 50 \text{ Kpsi}$$

$$S'_f = 50 \text{ Kpsi} \quad (S'_f)_{ax} = a N^b \quad (1)$$

$$\log S'_f = \log a N^b = \log a + b \log N$$

$$\log 90 = \log a + b \log 10^3 \Rightarrow b = \frac{\log 90 - \log a}{\log 10^3} \quad (2)$$

$$\log S'_f = \log a + b \log N \Rightarrow \log 50 = \log a + \left(\frac{\log 90 - \log a}{\log 10^3} \right) \log 10^6$$

$$\log a = \frac{\log 90 \log 10^6 - \log 50 \log 10^3}{\log 10^6 / \log 10^3}$$

$$\log a = 2.209 \Rightarrow a = 161.80 \text{ Kpsi}$$

6.) 6-11:

given:

#1 = AISI 4340 $S_{ut} = 260$ kpsi

#2 = AISI 1040 $S_{ut} = 113$ kpsi

$d_e = 0.75$ in

$a = 39.9$ $b = -0.995$

$$K_a = a S_{ut}^b = 39.9 \times 260^{-0.995} = 0.1578$$

$$K_b = 0.879 d_e^{-0.107} = 0.879 (0.75)^{-0.107} = 0.9065$$

$$S_e = K_a \cdot K_b \cdot S_e' = 0.1578 \times 0.9065 \times 100 = 14.304 \text{ kpsi}$$

AISI 1040: ~

$$S_{ut} \leq 200 \Rightarrow S_e' = 0.5 S_{ut} = 0.5 (113) = 56.5 \text{ kpsi}$$

$a = 39.9$ $b = -0.995$

$$K_a = 39.9 (113)^{-0.995} = 0.3615$$

$$K_b = 0.879 (0.75)^{-0.107} = 0.9065$$

$$S_e = 0.3615 \times 0.9065 \times 56.5 = 18.517 \text{ kpsi}$$

AISI 1040 is best choice
It has a higher "Se"

7.) 6-13:

$$S_{ut} = 770 \text{ MPa} \cdot 0.145 = 111.65 \text{ kpsi}$$

$$f = 0.83$$

$$S_{ut} < 1400 \text{ MPa}$$

$$S_e' = 0.5 S_{ut} \Rightarrow 0.5 (770) = 385 \text{ MPa}$$

$$K_a = 57.7 (770)^{-0.78} = 0.488$$

$$K_b \approx 0.85$$

$$S_e = 0.488 \times 0.85 \times 385 = 160 \text{ MPa}$$

$$a = \frac{(0.83 (770))^2}{160} = 2553 \text{ MPa}$$

$$b = -\frac{1}{8} \log \left(\frac{0.83 (770)}{160} \right) = -0.225$$

$$S_F = a N^b = 2553 (10^4)^{-0.225} = 403 \text{ MPa}$$

$$M_{max} = 2000 \cdot 0.6 = 1200$$

$$\sigma_{max} = \frac{6 (1200)}{b^3} = \frac{7200}{63}$$

$$n = S_e / \sigma_{max}$$

$$403 \times 10^6 \Rightarrow b = 29.9 \text{ mm}$$

$$1.5 = 7200 / b^3$$

$$d_e' = 0.808 (h_b)^{1/2}$$

$$d_e = 0.808 (30) = 24.2 \text{ mm}$$

$$K_b = \left(\frac{24.2}{4.62} \right)^{-1.57} = 0.88$$

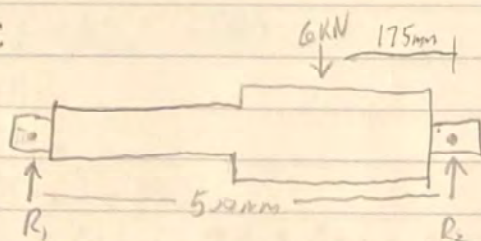
$$11253 \text{ Square} = 30 \text{ mm} \times 30 \text{ mm}$$

$$\sigma_{max} = 720 / 0.03 = 260 \text{ MPa}$$

$$n_y = 420 / 260 = 1.57$$

8.) 6-16:

FBD



$$\sum F_y = 0; R_1 + R_2 = 6 \quad (1)$$

$$\sum M_{R_1} = 0; R_2 \times 500 = 6 \times 325 \Rightarrow R_2 = 3.9 \text{ kN} \Rightarrow R_1 = 6 - 3.9 = 2.1 \text{ kN}$$

AISI 1020 CD:

$$S_{ut} = 470 \text{ MPa} \quad S_{ut} < 1400 \text{ MPa} \Rightarrow S_e' = 0.5 S_{ut} = 0.5(470) = 235 \text{ MPa}$$

$$S_y = 390 \text{ MPa} \quad \alpha = 4.51 \quad b = -0.265$$

$$K_a = \alpha S_{ut}^b = 4.51(470)^{-0.265} = 0.833$$

$$K_b = (d/7.62)^{-0.107} = (35/7.62)^{-0.107} = 0.85$$

$$K_c = 1, K_d = 1, K_e = 1 \quad S_e = 0.833 \cdot 0.85 \cdot 235 = 176.4 \text{ MPa}$$

$$r/b = 3/35 = 0.0857$$

$$D/d = 50/35 = 1.428 \quad K_f = 1.7$$

$$r = 3 \text{ mm} = 0.12 \text{ in} \quad S_{ut} = 470 \text{ MPa} = 68.16 \text{ kpsi}$$

$$\sqrt{a} = 0.246 - 3.08 \times 10^{-3} \cdot 68.16 + 1.51 \times 10^{-5} \cdot 68.16^2 - 2.07 \times 10^{-8} \cdot 68.16^3$$

$$= 0.4927 \text{ mm}$$

$$K_f = 1 + \frac{K_t - 1}{1 + \frac{\sqrt{a}}{r}} = 1 + \frac{(1.7 - 1)}{1 + \frac{0.4927}{0.12}} = 1.54$$

$$M = R_1 x = 2.1 \times 200 = 420 \text{ N}\cdot\text{m}$$

$$I/c = \frac{\pi d^3}{32} = \frac{\pi 35^3}{32} = 4.209 \times 10^6 \text{ mm}^4$$

$$\sigma_{rev} = K_f \frac{M}{I/c} = \frac{1.54 \cdot 420}{4.209 \times 10^6} = 153.6 \text{ MPa}$$

$$n_f = \frac{S_e}{K_f \sigma_{rev}} = \frac{176.4}{1.54 \cdot 153.6} = 1.13$$

9.) 6-25; AISI 1040 steel

$$S_{ut} = 590 \text{ MPa}$$

$$S_y = 470 \text{ MPa}$$

$$A = 10(25-6) = 190 \text{ mm}^2$$

$$\sigma_{max} = F_{max} / A = 28 \text{ kN} / 190 = 147.4 \text{ MPa}$$

$$n_y = S_y / \sigma_{max} = 470 / 147.4 = 3.2$$

$$S_e' = 0.5 S_{ut} = 0.5(590) = 295 \text{ MPa}$$

$$a = 4.51 \quad b = 0.205$$

$$k_a = a S_{ut}^b = 4.51 (590)^{-0.205} = 0.831, \quad k_c = 0.85$$

$$S_e = 0.831 \cdot 0.85 \cdot 295 = 208.4 \text{ MPa}$$

$$c/w = 6/25 = 0.24 \Rightarrow k_e = 2.45$$

$$Q = 0.8$$

$$K_f = 1 + Q(k_e - 1) = 1 + 0.8(2.45 - 1) = 2.16$$

$$\sigma_a = K_f \left| \frac{F_{max} - F_{min}}{2A} \right| = 2.16 \left| \frac{(28 \text{ kN}) - (-28 \text{ kN})}{2(190)} \right| = 318.3 \text{ MPa}$$

$$\sigma_m = 0$$

$$1/n_F = \sigma_a / S_e + \sigma_m / S_{ut} = (318.3 / 208.3 + 0 / 590)^{-1} = 0.654$$

$$\sigma_{rev} = \sigma_a / (1 - \sigma_m / S_{ut}) = 318.3$$

$$S_F = a N^b$$

$$a = \left(\frac{S_F}{S_e} \right)^2 = \left(\frac{0.87 \times 590}{208.3} \right)^2 = 1264.4$$

$$b = -1/3 \log(S_F / S_e) = -1/3 \log(0.87 \times 590 / 208.3) = -0.1306$$

$$S_F = 1264.4 N^{-0.1306} = 318.3$$

$$N = \left(\frac{\sigma_{rev}}{a} \right)^{1/b} = \left(\frac{318.3}{1264.4} \right)^{1/0.1306} = 38744 \text{ cycles}$$

Below Average

Grade:

Name: *Asher Carson*  

This assignment is due at NOON on October 2nd, 2020. Complete the following problems from Shigley's Mechanical Engineering Design – 10th edition. There are a total of 10 problems in this homework.

1. Problem 14-2

A steel spur pinion has a diametral pitch of 10 teeth/in, 18 teeth cut full-depth with a 20° pressure angle, and a face width of 1 in. This pinion is expected to transmit 2 hp at a speed of 600 rev/min. Determine the bending stress.

2. Problem 14-6

A 20° full-depth steel spur pinion has 20 teeth and a module of 2 mm and is to transmit 0.5 kW at a speed of 200 rev/min. Find an appropriate face width if the bending stress is not to exceed 75 MPa.

3. Problem 14-12

A gear drive consists of a 16-tooth 20° steel spur pinion and a 48-tooth cast-iron gear having a pitch of 12 teeth/in. For a power input of 1.5 hp at a pinion speed of 700 rev/min, select a face width based on an allowable contact stress of 100 kpsi.

4. Problem 14-13

A gearset has a module of 5 mm, a 20° pressure angle, and a 24-tooth cast-iron spur pinion driving a 48-tooth cast-iron gear. The pinion is to rotate at 50 rev/min. What horsepower input can be used with this gearset if the contact stress is limited to 690 MPa and $F = 60$ mm?

5. Problem 6-9

For the interval $10^3 \leq N \leq 10^6$ cycles, develop an expression for the axial fatigue strength $(S'_f)_{ax}$ for the polished specimens of 4130 used to obtain Fig. 6-10. The ultimate strength is $S_{ut} = 125$ kpsi and the endurance limit is $(S'_e)_{ax} = 50$ kpsi.

6. Problem 6-11

Two steels are being considered for manufacture of as-forged connecting rods subjected to bending loads. One is AISI 4340 Cr-Mo-Ni steel capable of being heat-treated to a tensile strength of 260 kpsi. The other is a plain carbon steel AISI 1040 with an attainable S_{ut} of 113 kpsi. Each rod is to have a size giving an equivalent diameter d_e of 0.75 in. Determine the endurance limit for each material. Is there any advantage to using the alloy steel for this fatigue application?

7. Problem 6-13

A solid square rod is cantilevered at one end. The rod is 0.6 m long and supports a completely reversing transverse load at the other end of ± 2 kN. The material is AISI 1080 hot-rolled steel. If the rod must support this load for 10^4 cycles with a design factor of 1.5, what dimension should the square cross section have? Neglect any stress concentrations at the support end.

8. Problem 6-16

Summary of Comments on Asher Carson HW 2 MCEG 4043 Y02.pdf

Page: 1

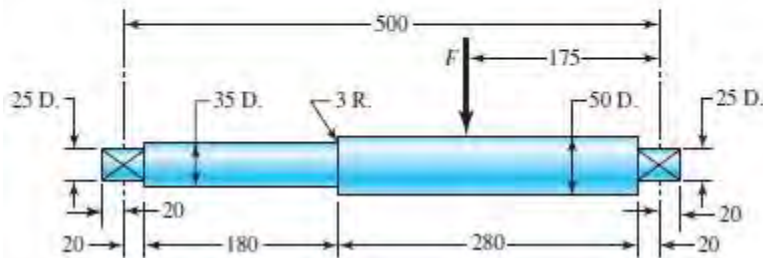
 Number: 1 Author: japark Subject: Sticky Note Date: 12/15/2020 12:48:53 PM
Graded 6-11 and 6-27. Half of the homework is missing (-5) Late (-10%)

4.5/10

Grade:

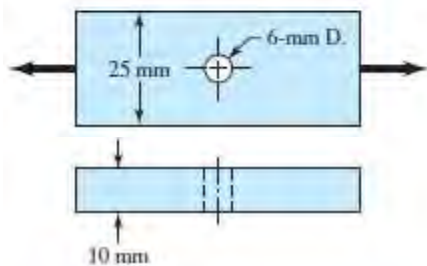
Name:

The rotating shaft shown in the figure is machined from AISI 1020 CD steel. It is subjected to a force of $F = 6 \text{ kN}$. Find the minimum factor of safety for fatigue based on infinite life. If the life is not infinite, estimate the number of cycles. Be sure to check for yielding.



9. Problem 6-25

The cold-drawn AISI 1040 steel bar shown in the figure is subjected to a completely reversed axial load fluctuating between 28 kN in compression to 28 kN in tension. Estimate the fatigue factor of safety based on achieving infinite life and the yielding factor of safety. If infinite life is not predicted, estimate the number of cycles to failure.



10. Problem 6-27

Using the modified Goodman criterion for infinite life, repeat Prob. 6-25 for each of the following loading conditions:

- (a) 0 kN to 28 kN
- (b) 12 kN to 28 kN
- (c) -28 kN to 12 kN

Homework requirement

- Print this page and use it as the cover.
- The submission should be each student's own work.
- The same day late submission will get a 10% penalty.
- Once the solution is posted, no more submission will be accepted.
- Write legibly. No credit will be given if the work is not readable.
- Take advantage of office hours.
 - Tuesdays & Thursdays 1 PM ~ 3 PM (tentative, may change in the week of June 1st)
 - Email me: japark@pvamu.edu to request a Zoom meeting if you cannot make it to the regular office hour due to any university excused reasons.

Problem 14-2

$$\begin{aligned}
 1.) \quad P &= 10 \frac{\text{teeth}}{\text{in}} & H &= 2 \text{ hp} & P &= \frac{N}{d} & m &= \frac{d}{N} \\
 \phi &= 20^\circ & F &= 1 \text{ in} & d &= \frac{N}{P} & m &= 1.6 \text{ in} \\
 n_p &= 600 \frac{\text{rev}}{\text{min}} & N &= 18 & & & & 16 \text{ teeth}
 \end{aligned}$$

$$\sigma = \frac{M}{I/c} = \frac{K_v W^o P}{F Y} \quad d = 1.6 \text{ teeth} \quad m = 0.1 \text{ in}$$

$$Y = 0.309, N = 19$$

$$V = \frac{11.8 n_p}{12} = \frac{12(1.6 \text{ in}) (600 \frac{\text{rev}}{\text{min}})}{12} \quad K_v = \frac{1200 + V}{1200}$$

$$V = 242.74 \frac{\text{ft}}{\text{min}} \quad K_v = 1.2356 \frac{\text{ft}}{\text{min}}$$

$$W_b = 33,000 \left(\frac{H}{V} \right)^{1/4}$$

$$W_b = 33,000 \left(\frac{2 \text{ hp}}{242.74 \frac{\text{ft}}{\text{min}}} \right) \quad \sigma = \frac{(1.2356 \frac{\text{ft}}{\text{min}}) (233.4272 \text{ lb}) (10 \frac{\text{teeth}}{\text{in}})}{(1 \text{ in}) (0.309)}$$

$$W_b = 233.4272 \text{ lb}$$

$$\sigma = 1334.2152 \text{ psi}$$

Problem 14-6

$$\begin{aligned}
 2.) \quad \phi &= 20^\circ & m &= d & N &= 20, Y = 0.322 \\
 & & N & & & \\
 \text{Steel} & & m \cdot N &= d \\
 N_p &= 20 \text{ teeth} & & & & \\
 m &= 2 \text{ mm} & & & & \\
 H &= 0.5 \text{ kW} & 2 \text{ mm} (20 \text{ teeth}) &= d_1 \\
 n_p &= 200 \frac{\text{rev}}{\text{min}} & 40 \text{ mm} &= d \\
 \sigma &= 75 \text{ MPa} & 40 \text{ mm} \times \frac{1 \text{ m}}{1000 \text{ mm}} &= d \\
 & & 0.04 \text{ m} &= d
 \end{aligned}$$

$$V = \frac{\pi d n_p}{12}$$

$$V = \frac{\pi (0.04 \text{ m}) (200 \frac{\text{rev}}{\text{min}})}{60}$$

$$V = 0.4189 \frac{\text{m}}{\text{s}}$$

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

$$W_t = \frac{60,000 (0.5 \text{ kW})}{\pi (0.04 \text{ m}) (200 \frac{\text{rev}}{\text{min}})}$$

$$W_t = 1193.662 \text{ N}$$

$$K_v = \frac{6.1 + V}{6.1}$$

$$K_v = \frac{6.1 + 0.4189 \frac{\text{m}}{\text{s}}}{6.1}$$

$$K_v = 1.0687$$

$$F_m Y (\sigma) (K_v W_t) F_m Y$$

$$\frac{F_m Y \sigma}{m Y \sigma} = K_v W_t$$

$$F = \frac{K_v W_t}{m Y \sigma}$$

$$F = \frac{(1.0687)(1193.662 \text{ N})}{(2 \text{ mm})(0.322)(75 \text{ MPa})}$$

$$F = 26.4105 \text{ mm}$$

$$\begin{aligned}
 3.) \quad N_p &= 16 \text{ teeth} & N_g &= 48 \text{ teeth} & V &= \frac{r_d p n}{12} \\
 \phi_p &= 20^\circ & P &= 12 \text{ teeth/in} & & \\
 \text{steel} & & H &= 1.5 \text{ hp} & V &= \frac{12(1.333 \text{ in})(700 \frac{\text{rev}}{\text{min}})}{12} \\
 n_p &= 700 \frac{\text{rev}}{\text{min}} & & & & \\
 \sigma_c &= 100 \text{ kpsi} & \frac{100 \text{ kpsi}}{1 \text{ kpsi}} &= 10 \times 10^5 \text{ psi} & V &= 244.346 \frac{\text{ft}}{\text{min}}
 \end{aligned}$$

$$p = \frac{N}{d} \quad dg = \frac{N_g}{P} \quad Kv = \frac{1200 + V}{1200}$$

$$d = \frac{N}{p} \quad dg = \frac{48 \text{ teeth}}{12 \text{ teeth/in}} \quad Kv = 1.2036$$

$$\begin{aligned}
 d_p &= \frac{N_p}{p} = \frac{16 \text{ teeth}}{12 \frac{\text{teeth}}{\text{in}}} & dg &= 4 \text{ in} & W_t &= 33,000 \frac{\text{ft}}{\text{min}} \\
 d_p &= 1.333 \text{ in} & & & W_t &= 33,000 / 1.5 \text{ hp} \\
 & & & & & (244.346 \frac{\text{ft}}{\text{min}})
 \end{aligned}$$

$$W_t = 202.5615 \text{ hp}$$

$$C_p = 2100 \sqrt{\text{psi}} \quad (\text{Table 14-8})$$

$$(\sigma_c)^2 = \left[C_p \left[\frac{K_v W_t^2}{F \cos \phi} \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \right]^{\frac{1}{2}} \right]^2 \quad (r_1 = d_p \sin \phi \quad r_2 = dg \sin \phi)$$

$$F(\sigma_c)^2 = -C_p^2 \left[\frac{K_v W_t^2}{F \cos \phi} \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \right]^2 \quad r_1 = \frac{(1.333 \text{ in}) \sin(20^\circ)}{2} \quad r_2 = \frac{(4) \sin(20^\circ)}{2}$$

$$F(\sigma_c)^2 = -C_p^2 \left[\frac{K_v W_t^2}{\cos \phi} \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \right]^2 \quad r_1 = 0.228 \quad r_2 = 0.684$$

$$F = \frac{C_p^2}{\sigma_c^2} \left[\frac{K_v W_t^2}{\cos \phi} \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \right]$$

$$F = 0.6692 \text{ in}$$

$$4.) m = 5 \text{ mm}$$

$$\phi = 20^\circ$$

$$N_p = 24$$

$$N_g = 48$$

$$n_p = 50 \frac{\text{rev}}{\text{min}}$$

$$\sigma_c = 690 \text{ MPa}$$

$$F = 60 \text{ mm}$$

$$C_p = 163 \sqrt{\text{MPa}}$$

$$(\sigma_c)^2 \left(C_p \left[K_v W^b \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \right]^{\frac{1}{2}} \right)^2$$

$$\frac{\sigma_c^2}{W^b} = -C_p^2 \left[K_v W^b \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \right]^{\frac{1}{2}} \cdot \frac{1}{W^b}$$

$$\frac{\sigma_c^2}{W^b} = -C_p^2 \left[K_v \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \right]^{\frac{1}{2}}$$

$$\left(\frac{1}{W^b} \right)^{-1} = \left[\frac{-C_p^2}{\sigma_c^2} \left[K_v \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \right]^{\frac{1}{2}} \right]^{-1}$$

$$W^b = \left[\frac{-C_p^2}{\sigma_c^2} \left[K_v \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \right]^{\frac{1}{2}} \right]^{-1}$$

$$W^b = 12531.2926$$

$$m = d$$

$$IV$$

$$N/m = d$$

$$N/p_m = d_p$$

$$(24 \text{ teeth}) (5 \text{ mm}) = d_p$$

$$120 \text{ mm} = d_p$$

$$N_g m = d_g$$

$$(48 \text{ teeth}) (5 \text{ mm}) = d_g$$

$$240 \text{ mm} = d_g$$

$$20 \text{ mm} \times \frac{1 \text{ m}}{1000 \text{ mm}} = 0.02 \text{ m}$$

$$240 \text{ mm} \times \frac{1 \text{ m}}{1000 \text{ mm}} = 0.24 \text{ m}$$

$$K_v = 3.05 + 0.3142$$

$$3.05$$

$$V = \frac{\pi d_p n_p}{60}$$

$$K_v = 1.103$$

$$V = \frac{\pi (0.12 \text{ m}) (50 \frac{\text{rev}}{\text{min}})}{60}$$

$$W^b = 60 \text{ H}$$

$$\frac{\pi (0.12) (50)}{60}$$

$$V = 0.3142 \frac{\text{m}}{\text{s}}$$

$$W^b = 3.1431 \text{ H}$$

$$r_1 = \frac{d_p \sin \phi}{2}$$

$$r_2 = \frac{d_g \sin \phi}{2}$$

$$3.1431 \text{ H} = 12531.2926$$

$$r_1 = \frac{120 \sin(20^\circ)}{2}$$

$$r_2 = \frac{240 \sin(20^\circ)}{2}$$

$$H = 3936.626 \text{ W}$$

$$H = 3.9366 \text{ kW}$$

$$r_1 = 20.5212 \text{ mm}$$

$$r_2 = 41.0424 \text{ mm}$$

$$5.) 10^3 \leq N \leq 10^6$$

$$S_f = aN^b$$

$$S_{ut} = 125 \text{ kpsi}$$

$$N = 10^3$$

$$(S_f)_{ax} = 50 \text{ kpsi}$$

$$\log(S_f) = \log(aN^b)$$

$$\log(S_f) = \log(a) + \log(N^b)$$

$$N = 10^3, S_f = 90 \text{ kpsi}$$

$$\log(S_f) = \log(a) + b \log(N)$$

$$N = 10^6, S_f = 50 \text{ kpsi}$$

$$\log(90) = \log(a) + b \log(10^3)$$

$$\frac{\log(90) - \log(a)}{\log(10^3)} = \frac{b \log(10^3)}{\log(10^3)}$$

$$N = 10^6$$

$$\log(50 \text{ kpsi}) = \log a + b \log(10^6) \quad b = \frac{\log(90) - \log(a)}{\log(10^3)}$$

$$\log(10^3) \cdot \log(50 \text{ kpsi}) = \left[\log a + \frac{\log(90) - \log(a)}{\log(10^3)} \log(10^6) \right] \log(10^3)$$

$$\log(10^3) \cdot \log(50 \text{ kpsi}) = \log a \cdot \log(10^3) + \frac{\log(90) - \log(a)}{\log(10^3)} \log(10^6) \log(10^3)$$

$$\log(a) = \frac{\log(50) \log(10^3) - \log(90) \log(10^6)}{\log(10^3)}$$

$$\log(a) = 2.209$$

$$a = 161.80 \text{ kpsi}$$

$$b = \frac{\log(90) - \log(161.80)}{\log(10^3)}$$

$$b = -0.08491$$

$$b = -0.085$$

$$(S_f)_{ax} = 161.4 N^{-0.085}$$

6.) AISI 4340 Cr-Mo-Ni Steel

$$S_{ut} = 260 \text{ kpsi}$$

$$a = 39.9, b = -0.995$$

AISI 1040 carbon steel

$$S_{ut} = 113 \text{ kpsi}$$

$$(k_a) = a S_{ut}^b$$

$$d_c = 0.75 \text{ in}$$

$$(k_a) = 39.9 (260)^{-0.995}$$

$$S_{ut} > 200 \text{ kpsi}, S'_c = 100 \text{ kpsi} \quad k_a = 0.1578$$

$$k_b = 0.879 d^{-0.107} \quad 0.11 \leq d \leq 2 \text{ in}$$

$$k_b = 0.879 (0.75)^{-0.107}$$

$$S_{ut} \leq 200 \text{ kpsi}, S'_c = 0.8 (S_{ut})$$

$$k_b = 0.9065$$

$$S'_c = 113 \text{ kpsi} (0.8)$$

AISI 4340

$$S'_c = 56.5 \text{ kpsi}$$

$$S_c = k_a k_b S'_c$$

$$S_c = (0.1578) (0.9065) (100 \text{ kpsi})$$

$$S_c = 14.3046 \text{ kpsi}$$

AISI 1040

$$S_c = k_a k_b S'_c$$

$$S'_c = 56.5 \text{ kpsi}$$

$$S_c = (0.3615) (0.9065) (56.5 \text{ kpsi})$$

$$k_a = 39.9 (113)^{-0.995}$$

$$S_c = 18.5168 \text{ kpsi}$$

$$k_a = 0.3615$$

$$k_b = 0.879 d^{-0.107} \quad 0.11 \leq d \leq 2 \text{ in}$$

$$k_b = 0.879 (0.75)^{-0.107}$$

$$k_b = 0.9065$$

The Carbon Steel has a stronger endurance limit

MACHINE DESIGN II

FALL 2020

TITLE OF ASSIGNMENT: HOMEWORK 3

TOPICS COVERED IN HOMEWORK 3

1. Shaft and Shaft Components (Chapter 14)

Grade:

Name:

This assignment is due at 8 am on Oct 28th. Complete the following problems from Shigley's Mechanical Engineering Design – 10th edition.

1. Problem 7-1
2. Problem 7-4
3. Problem 7-11
4. Problem 7-18-(a)
5. Problem 7-19
6. Problem 7-23-(a)

Homework requirement

- Print this page and use it as the cover.
- The submission should be each student's own work.
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- You may also bring the questions related to this homework to the Friday live sessions.

Problem 7-1**(a)** DE-Gerber, Eq. (7-10):

$$A = \sqrt{4(K_f M_a)^2 + 3(K_{fs} T_a)^2} = \sqrt{4[(2.2)(70)]^2 + 3[(1.8)(45)]^2} = 338.4 \text{ N} \cdot \text{m}$$

$$B = \sqrt{4(K_f M_m)^2 + 3(K_{fs} T_m)^2} = \sqrt{4[(2.2)(55)]^2 + 3[(1.8)(35)]^2} = 265.5 \text{ N} \cdot \text{m}$$

$$d = \left\{ \frac{8(2)(338.4)}{\pi(210)(10^6)} \left[1 + \left(1 + \left[\frac{2(265.5)(210)(10^6)}{338.4(700)(10^6)} \right]^2 \right)^{1/2} \right] \right\}^{1/3}$$

$$d = 25.85 (10^{-3}) \text{ m} = 25.85 \text{ mm} \quad \text{Ans.}$$

(b) DE-elliptic, Eq. (7-12) can be shown to be

$$d = \left(\frac{16n}{\pi} \sqrt{\frac{A^2}{S_e^2} + \frac{B^2}{S_y^2}} \right)^{1/3} = \left(\frac{16(2)}{\pi} \sqrt{\frac{(338.4)^2}{[(210)(10^6)]^2} + \frac{(265.5)^2}{[(560)(10^6)]^2}} \right)^{1/3}$$

$$d = 25.77 (10^{-3}) \text{ m} = 25.77 \text{ mm} \quad \text{Ans.}$$

(c) DE-Soderberg, Eq. (7-14) can be shown to be

$$d = \left[\frac{16n}{\pi} \left(\frac{A}{S_e} + \frac{B}{S_y} \right) \right]^{1/3} = \left[\frac{16(2)}{\pi} \left(\frac{338.4}{210(10^6)} + \frac{265.5}{560(10^6)} \right) \right]^{1/3}$$

$$d = 27.70 (10^{-3}) \text{ m} = 27.70 \text{ mm} \quad \text{Ans.}$$

(d) DE-Goodman: Eq. (7-8) can be shown to be

$$d = \left[\frac{16n}{\pi} \left(\frac{A}{S_e} + \frac{B}{S_{ut}} \right) \right]^{1/3} = \left[\frac{16(2)}{\pi} \left(\frac{338.4}{210(10^6)} + \frac{265.5}{700(10^6)} \right) \right]^{1/3}$$

$$d = 27.27 (10^{-3}) \text{ m} = 27.27 \text{ mm} \quad \text{Ans.}$$

Criterion	d (mm)	Compared to DE-Gerber	
DE-Gerber	25.85		
DE-Elliptic	25.77	0.31% Lower	Less conservative
DE-Soderberg	27.70	7.2% Higher	More conservative
DE-Goodman	27.27	5.5% Higher	More conservative

Problem 7-4

We have a design task of identifying bending moment and torsion diagrams which are preliminary to an industrial roller shaft design. Let point C represent the center of the span of the roller.

$$F_C^y = 30(8) = 240 \text{ lbf}$$

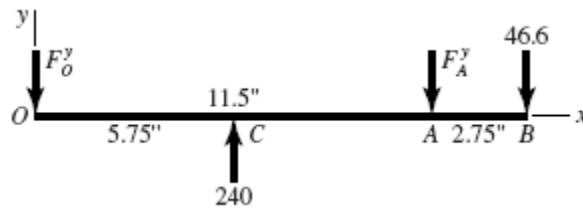
$$F_C^z = 0.4(240) = 96 \text{ lbf}$$

$$T = F_C^z(2) = 96(2) = 192 \text{ lbf} \cdot \text{in}$$

$$F_B^z = \frac{T}{1.5} = \frac{192}{1.5} = 128 \text{ lbf}$$

$$F_B^y = F_B^z \tan 20^\circ = 128 \tan 20^\circ = 46.6 \text{ lbf}$$

(a) xy -plane



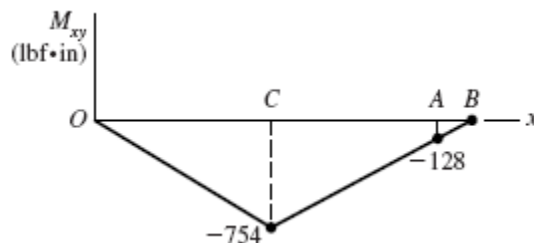
$$\Sigma M_O = 240(5.75) - F_A^y(11.5) - 46.6(14.25) = 0$$

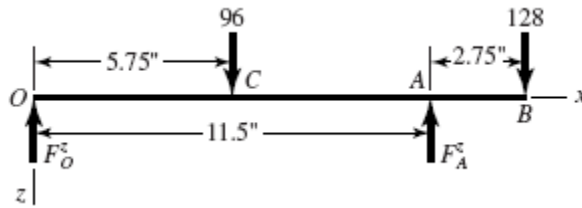
$$F_A^y = \frac{240(5.75) - 46.6(14.25)}{11.5} = 62.3 \text{ lbf}$$

$$\Sigma M_A = F_O^y(11.5) - 46.6(2.75) - 240(5.75) = 0$$

$$F_O^y = \frac{240(5.75) + 46.6(2.75)}{11.5} = 131.1 \text{ lbf}$$

Bending moment diagram:



xz-plane

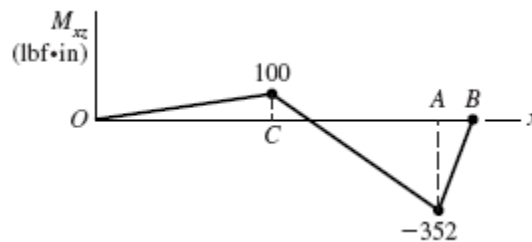
$$\Sigma M_O = 0 = 96(5.75) - F_A^z(11.5) + 128(14.25)$$

$$F_A^z = \frac{96(5.75) + 128(14.25)}{11.5} = 206.6 \text{ lbf}$$

$$\Sigma M_A = 0 = F_O^z(11.5) + 128(2.75) - 96(5.75)$$

$$F_O^z = \frac{96(5.75) - 128(2.75)}{11.5} = 17.4 \text{ lbf}$$

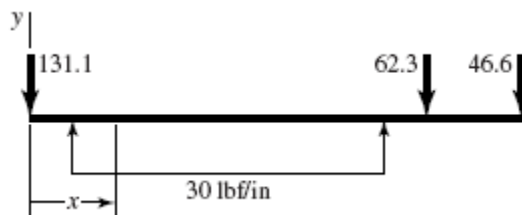
Bending moment diagram:



$$M_C = \sqrt{100^2 + (-754)^2} = 761 \text{ lbf} \cdot \text{in}$$

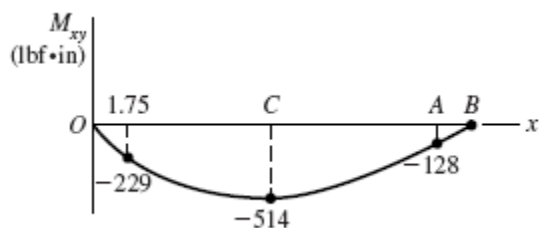
$$M_A = \sqrt{(-128)^2 + (-352)^2} = 375 \text{ lbf} \cdot \text{in}$$

Torque: The torque is constant from C to B, with a magnitude previously obtained of 192 lbf·in.

(b) *xy-plane*

$$M_{xy} = -131.1x + 15\langle x - 1.75 \rangle^2 - 15\langle x - 9.75 \rangle^2 - 62.3\langle x - 11.5 \rangle^1$$

Bending moment diagram:

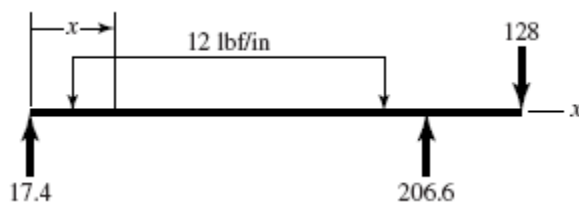


$M_{\max} = -516 \text{ lbf} \cdot \text{in}$ and occurs at 6.12 in.

$$M_C = 131.1(5.75) - 15(5.75 - 1.75)^2 = 514 \text{ lbf} \cdot \text{in}$$

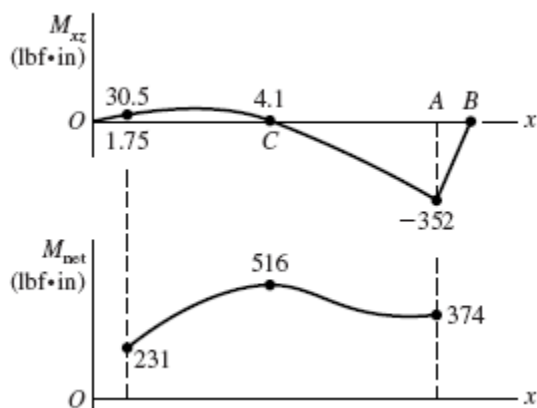
This is reduced from $754 \text{ lbf} \cdot \text{in}$ found in part (a). The maximum occurs at $x = 6.12$ in rather than C , but it is close enough.

xz-plane



$$M_{xz} = 17.4x - 6\langle x - 1.75 \rangle^2 + 6\langle x - 9.75 \rangle^2 + 206.6\langle x - 11.5 \rangle^1$$

Bending moment diagram:



$$\text{Let } M_{\text{net}} = \sqrt{M_{xy}^2 + M_{xz}^2}$$

Plot $M_{\text{net}}(x)$, $1.75 \leq x \leq 11.5$ in

$M_{\max} = 516 \text{ lbf} \cdot \text{in}$ at $x = 6.25$ in

Torque: The torque rises from 0 to $192 \text{ lbf} \cdot \text{in}$ linearly across the roller, then is constant to B . Ans.

Problem 7-11**7-7 through 7-16**

These are design problems, which can have many acceptable designs. See the solution for Prob. 7-17 for an example of the design process.

7-17 (a) One possible shaft layout is shown in part (e). Both bearings and the gear will be located against shoulders. The gear and the motor will transmit the torque through the keys. The bearings can be lightly pressed onto the shaft. The left bearing will locate the shaft in the housing, while the right bearing will float in the housing.

(b) From summing moments around the shaft axis, the tangential transmitted load through the gear will be

$$W_t = T / (d / 2) = 2500 / (4 / 2) = 1250 \text{ lbf}$$

The radial component of gear force is related by the pressure angle.

$$W_r = W_t \tan \phi = 1250 \tan 20^\circ = 455 \text{ lbf}$$

$$W = (W_r^2 + W_t^2)^{1/2} = (455^2 + 1250^2)^{1/2} = 1330 \text{ lbf}$$

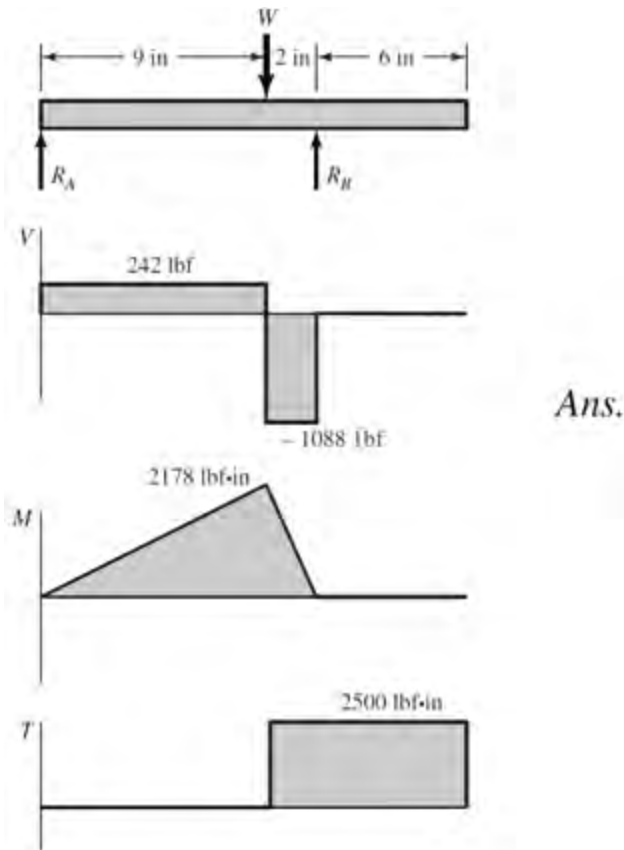
Reactions R_A and R_B , and the load W are all in the same plane. From force and moment balance,

$$R_A = 1330(2 / 11) = 242 \text{ lbf}$$

$$R_B = 1330(9 / 11) = 1088 \text{ lbf}$$

$$M_{\max} = R_A(9) = 242(9) = 2178 \text{ lbf} \cdot \text{in}$$

Shear force, bending moment, and torque diagrams can now be obtained.



(c) Potential critical locations occur at each stress concentration (shoulders and keyways). To be thorough, the stress at each potentially critical location should be evaluated. For now, we will choose the most likely critical location, by observation of the loading situation, to be in the keyway for the gear. At this point there is a large stress concentration, a large bending moment, and the torque is present. The other locations either have small bending moments, or no torque. The stress concentration for the keyway is highest at the ends. For simplicity, and to be conservative, we will use the maximum bending moment, even though it will have dropped off a little at the end of the keyway.

(d) At the gear keyway, approximately 9 in from the left end of the shaft, the bending is completely reversed and the torque is steady.

$$M_a = 2178 \text{ lbf} \cdot \text{in} \quad T_m = 2500 \text{ lbf} \cdot \text{in} \quad M_m = T_a = 0$$

From Table 7-1, p. 365, estimate stress concentrations for the end-milled keyseat to be $K_t = 2.14$ and $K_{ts} = 3.0$. For the relatively low strength steel specified (AISI 1020 CD), roughly estimate notch sensitivities of $q = 0.75$ and $q_s = 0.80$, obtained by observation of Figs. 6-20 and 6-21, assuming a typical radius at the bottom of the keyseat of $r/d = 0.02$, and a shaft diameter of up to 3 inches.

$$\text{Eq. (6-32):} \quad K_f = 1 + 0.75(2.14 - 1) = 1.9$$

$$K_{fs} = 1 + 0.8(3.0 - 1) = 2.6$$

Eq. (6-19): $k_a = 2.70(68)^{-0.265} = 0.883$

For estimating k_b , guess $d = 2$ in.

Eq. (6-20) $k_b = (2 / 0.3)^{-0.107} = 0.816$

Eq. (6-18) $S_e = 0.883(0.816)(0.5)(68) = 24.5$ kpsi

Selecting the DE-Goodman criteria for a conservative first design,

$$\text{Eq. (7-8): } d = \left[\frac{16n}{\pi} \left\{ \frac{\left[4(K_f M_a)^2 \right]^{1/2}}{S_e} + \frac{\left[3(K_{fs} T_m)^2 \right]^{1/2}}{S_{ut}} \right\} \right]^{1/3}$$

$$d = \left[\frac{16(1.5)}{\pi} \left\{ \frac{\left[4(1.9 \cdot 2178)^2 \right]^{1/2}}{24\,500} + \frac{\left[3(2.6 \cdot 2500)^2 \right]^{1/2}}{68\,000} \right\} \right]^{1/3}$$

$d = 1.57$ in *Ans.*

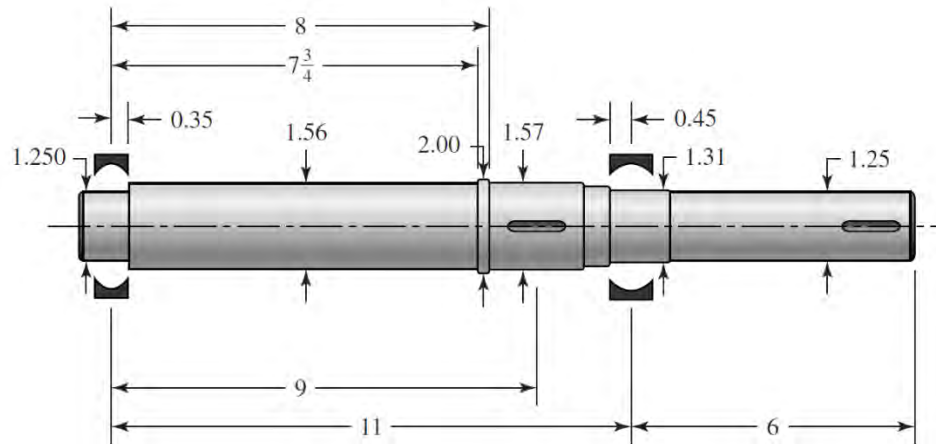
With this diameter, the estimates for notch sensitivity and size factor were conservative, but close enough for a first iteration until deflections are checked. Check yielding with this diameter.

$$\text{Eq. (7-15): } \sigma'_{\max} = \left[\left(\frac{32K_f M_a}{\pi d^3} \right)^2 + 3 \left(\frac{16K_{fs} T_m}{\pi d^3} \right)^2 \right]^{1/2}$$

$$\sigma'_{\max} = \left[\left(\frac{32(1.9)(2178)}{\pi(1.57)^3} \right)^2 + 3 \left(\frac{16(2.6)(2500)}{\pi(1.57)^3} \right)^2 \right]^{1/2} = 18\,389 \text{ psi} = 18.4 \text{ kpsi}$$

$$n_y = S_y / \sigma'_{\max} = 57 / 18.4 = 3.1 \quad \text{Ans.}$$

(e) Now estimate other diameters to provide typical shoulder supports for the gear and bearings (p. 364). Also, estimate the gear and bearing widths.



(f) Entering this shaft geometry into beam analysis software (or Finite Element software), the following deflections are determined:

Left bearing slope:	0.000 532 rad
Right bearing slope:	− 0.000 850 rad
Gear slope:	− 0.000 545 rad
Right end of shaft slope:	− 0.000 850 rad
Gear deflection:	− 0.001 45 in
Right end of shaft deflection:	0.005 10 in

Comparing these deflections to the recommendations in Table 7-2, everything is within typical range except the gear slope is a little high for an uncrowned gear.

(g) To use a non-crowned gear, the gear slope is recommended to be less than 0.0005 rad. Since all other deflections are acceptable, we will target an increase in diameter only for the long section between the left bearing and the gear. Increasing this diameter from the proposed 1.56 in to 1.75 in, produces a gear slope of − 0.000 401 rad. All other deflections are improved as well.

Problem 7-18

(a) Use the distortion-energy elliptic failure locus. The torque and moment loadings on the shaft are shown in the solution to Prob. 7-17.

Candidate critical locations for strength:

- Left seat keyway
- Right bearing shoulder
- Right keyway

Table A-20 for 1030 HR: $S_{ut} = 68$ kpsi, $S_y = 37.5$ kpsi, $H_B = 137$

Eq. (6-8): $S'_e = 0.5(68) = 34.0$ kpsi

Eq. (6-19): $k_a = 2.70(68)^{-0.265} = 0.883$

$$k_c = k_d = k_e = 1$$

Left keyway

See Table 7-1 for keyway stress concentration factors,

$$\left. \begin{array}{l} K_t = 2.14 \\ K_{ts} = 3.0 \end{array} \right\} \text{Profile keyway}$$

For an end-mill profile keyway cutter of 0.010 in radius, estimate notch sensitivities.

Fig. 6-20: $q = 0.51$

Fig. 6-21: $q_s = 0.57$

Eq. (6-32): $K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.57(3.0 - 1) = 2.1$

$$K_f = 1 + 0.51(2.14 - 1) = 1.6$$

Eq. (6-20): $k_b = \left(\frac{1.875}{0.30} \right)^{-0.107} = 0.822$

Eq. (6-18): $S_e = 0.883(0.822)(34.0) = 24.7 \text{ kpsi}$

Eq. (7-11):
$$\frac{1}{n_f} = \frac{16}{\pi(1.875^3)} \left\{ 4 \left[\frac{1.6(2178)}{24\,700} \right]^2 + 3 \left[\frac{2.1(2500)}{37\,500} \right]^2 \right\}^{\frac{1}{2}}$$

$$n_f = 3.5 \quad \text{Ans.}$$

Right bearing shoulder

The text does not give minimum and maximum shoulder diameters for 03-series bearings (roller). Use $D = 1.75 \text{ in.}$

$$\frac{r}{d} = \frac{0.030}{1.574} = 0.019, \quad \frac{D}{d} = \frac{1.75}{1.574} = 1.11$$

Fig. A-15-9: $K_t = 2.4$

Fig. A-15-8: $K_{ts} = 1.6$

Fig. 6-20: $q = 0.65$

Fig. 6-21: $q_s = 0.70$

Eq. (6-32): $K_f = 1 + 0.65(2.4 - 1) = 1.91$

$$K_{fs} = 1 + 0.70(1.6 - 1) = 1.42$$

$$M = 2178 \left(\frac{0.453}{2} \right) = 493 \text{ lbf} \cdot \text{in}$$

Eq. (7-11):
$$\frac{1}{n_f} = \frac{16}{\pi(1.574^3)} \left[4 \left(\frac{1.91(493)}{24\,700} \right)^2 + 3 \left(\frac{1.42(2500)}{37\,500} \right)^2 \right]^{1/2}$$

$$n_f = 4.2 \quad \text{Ans.}$$

Right keyway

Use the same stress concentration factors as for the left keyway. There is no bending moment, thus Eq. (7-11) reduces to:

$$\frac{1}{n_f} = \frac{16\sqrt{3}K_{fs}T_m}{\pi d^3 S_y} = \frac{16\sqrt{3}(2.1)(2500)}{\pi(1.5^3)(37\,500)}$$

$$n_f = 2.7 \quad \text{Ans.}$$

Yielding

Check for yielding at the left keyway, where the completely reversed bending is maximum, and the steady torque is present. Using Eq. (7-15), with $M_m = T_a = 0$,

$$\sigma'_{\max} = \left[\left(\frac{32K_f M_a}{\pi d^3} \right)^2 + 3 \left(\frac{16K_{fs} T_m}{\pi d^3} \right)^2 \right]^{1/2}$$

$$= \left[\left(\frac{32(1.6)(2178)}{\pi(1.875)^3} \right)^2 + 3 \left(\frac{16(2.1)(2500)}{\pi(1.875)^3} \right)^2 \right]^{1/2}$$

$$= 8791 \text{ psi} = 8.79 \text{ kpsi}$$

$$n_y = \frac{S_y}{\sigma'_{\max}} = \frac{37.5}{8.79} = 4.3 \quad \text{Ans.}$$

Check in smaller diameter at right end of shaft where only steady torsion exists.

$$\sigma'_{\max} = \left[3 \left(\frac{16K_{fs} T_m}{\pi d^3} \right)^2 \right]^{1/2}$$

$$= \left[3 \left(\frac{16(2.1)(2500)}{\pi(1.5)^3} \right)^2 \right]^{1/2}$$

$$= 13\,722 \text{ psi} = 13.7 \text{ kpsi}$$

$$n_y = \frac{S_y}{\sigma'_{\max}} = \frac{37.5}{13.7} = 2.7 \quad \text{Ans.}$$

Problem 7-19

The most likely critical locations for fatigue are at locations where the bending moment is high, the cross section is small, stress concentration exists, and torque exists. The two-plane bending moment diagrams, shown in the solution to Prob. 3-72, indicate decreasing moments in both planes to the left of A and to the right of C, with combined values at A and C of $M_A = 5324 \text{ lbf}\cdot\text{in}$ and $M_C = 6750 \text{ lbf}\cdot\text{in}$. The torque is constant between A and B, with $T = 2819 \text{ lbf}\cdot\text{in}$. The most likely critical locations are at the stress concentrations near A and C. The two shoulders near A can be eliminated since the shoulders near C have the same geometry but a higher bending moment. We will consider the following potentially critical locations:

- keyway at A
- shoulder to the left of C
- shoulder to the right of C

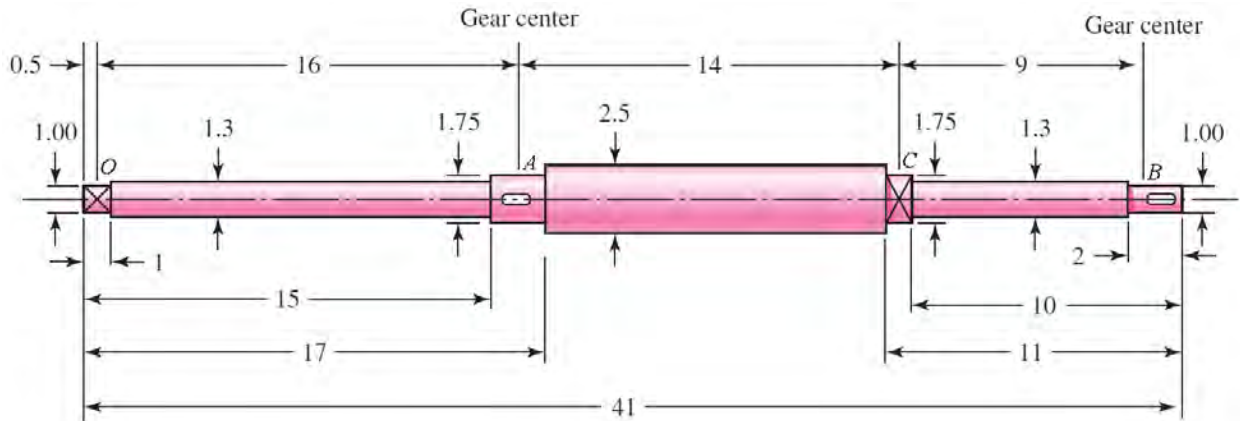


Table A-20: $S_{ut} = 64$ kpsi, $S_y = 54$ kpsi

Eq. (6-8): $S'_e = 0.5(64) = 32.0$ kpsi

Eq. (6-19): $k_a = 2.70(64)^{-0.265} = 0.897$
 $k_c = k_d = k_e = 1$

Keyway at A

Assuming $r/d = 0.02$ for typical end-milled keyway cutter (p. 365), with $d = 1.75$ in,
 $r = 0.02d = 0.035$ in.

Table 7-1: $K_t = 2.14$, $K_{ts} = 3.0$

Fig. 6-20: $q = 0.65$

Fig. 6-21: $q_s = 0.71$

Eq. (6-32): $K_f = 1 + q(K_t - 1) = 1 + 0.65(2.14 - 1) = 1.7$

$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.71(3.0 - 1) = 2.4$

Eq. (6-20): $k_b = \left(\frac{1.75}{0.30}\right)^{-0.107} = 0.828$

Eq. (6-18): $S_e = 0.897(0.828)(32) = 23.8$ kpsi

We will choose the DE-Gerber criteria since this is an analysis problem in which we would like to evaluate typical expectations.

Using Eq. (7-9) with $M_m = T_a = 0$,

$$A = \sqrt{4(K_f M_a)^2} = \sqrt{4[(1.7)(5324)]^2} = 18\,102 \text{ lbf} \cdot \text{in} = 18.10 \text{ kip} \cdot \text{in}$$

$$B = \sqrt{3(K_{fs} T_m)^2} = \sqrt{3[(2.4)(2819)]^2} = 11\,718 \text{ lbf} \cdot \text{in} = 11.72 \text{ kip} \cdot \text{in}$$

$$\begin{aligned}
 \frac{1}{n} &= \frac{8A}{\pi d^3 S_e} \left\{ 1 + \left[1 + \left(\frac{2BS_e}{AS_{ut}} \right)^2 \right]^{1/2} \right\} \\
 &= \frac{8(18.10)}{\pi(1.75^3)(23.8)} \left\{ 1 + \left[1 + \left(\frac{2(11.72)(23.8)}{(18.10)(64)} \right)^2 \right]^{1/2} \right\} \\
 n &= 1.3
 \end{aligned}$$

Shoulder to the left of C

$$r/d = 0.0625 / 1.75 = 0.036, \quad D/d = 2.5 / 1.75 = 1.43$$

Fig. A-15-9: $K_t = 2.2$

Fig. A-15-8: $K_{ts} = 1.8$

Fig. 6-20: $q = 0.71$

Fig. 6-21: $q_s = 0.76$

$$\text{Eq. (6-32): } K_f = 1 + q(K_t - 1) = 1 + 0.71(2.2 - 1) = 1.9$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.76(1.8 - 1) = 1.6$$

$$\text{Eq. (6-20): } k_b = \left(\frac{1.75}{0.30} \right)^{-0.107} = 0.828$$

$$\text{Eq. (6-18): } S_e = 0.897(0.828)(32) = 23.8 \text{ kpsi}$$

For convenience, we will use the full value of the bending moment at C, even though it will be slightly less at the shoulder. Using Eq. (7-9) with $M_m = T_a = 0$,

$$A = \sqrt{4(K_f M_a)^2} = \sqrt{4[(1.9)(6750)]^2} = 25\,650 \text{ lbf} \cdot \text{in} = 25.65 \text{ kip} \cdot \text{in}$$

$$B = \sqrt{3(K_{fs} T_m)^2} = \sqrt{3[(1.6)(2819)]^2} = 7812 \text{ lbf} \cdot \text{in} = 7.812 \text{ kip} \cdot \text{in}$$

$$\begin{aligned}
 \frac{1}{n} &= \frac{8A}{\pi d^3 S_e} \left\{ 1 + \left[1 + \left(\frac{2BS_e}{AS_{ut}} \right)^2 \right]^{1/2} \right\} \\
 &= \frac{8(25.65)}{\pi(1.75^3)(23.8)} \left\{ 1 + \left[1 + \left(\frac{2(7.812)(23.8)}{(25.65)(64)} \right)^2 \right]^{1/2} \right\} \\
 n &= 0.96
 \end{aligned}$$

Shoulder to the right of C

$$r/d = 0.0625 / 1.3 = 0.048, \quad D/d = 1.75 / 1.3 = 1.35$$

Fig. A-15-9: $K_t = 2.0$

Fig. A-15-8: $K_{ts} = 1.7$

Fig. 6-20: $q = 0.71$

Fig. 6-21: $q_s = 0.76$

Eq. (6-32): $K_f = 1 + q(K_t - 1) = 1 + 0.71(2.0 - 1) = 1.7$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.76(1.7 - 1) = 1.5$$

Eq. (6-20): $k_b = \left(\frac{1.3}{0.30}\right)^{-0.107} = 0.855$

Eq. (6-18): $S_e = 0.897(0.855)(32) = 24.5$ kpsi

For convenience, we will use the full value of the bending moment at C, even though it will be slightly less at the shoulder. Using Eq. (7-9) with $M_m = T_a = 0$,

$$A = \sqrt{4(K_f M_a)^2} = \sqrt{4[(1.7)(6750)]^2} = 22\,950 \text{ lbf} \cdot \text{in} = 22.95 \text{ kip} \cdot \text{in}$$

$$B = \sqrt{3(K_{fs} T_m)^2} = \sqrt{3[(1.5)(2819)]^2} = 7324 \text{ lbf} \cdot \text{in} = 7.324 \text{ kip} \cdot \text{in}$$

$$\begin{aligned} \frac{1}{n} &= \frac{8A}{\pi d^3 S_e} \left\{ 1 + \left[1 + \left(\frac{2BS_e}{AS_{ut}} \right)^2 \right]^{1/2} \right\} \\ &= \frac{8(22.95)}{\pi(1.3^3)(24.5)} \left\{ 1 + \left[1 + \left(\frac{2(7.324)(24.5)}{(22.95)(64)} \right)^2 \right]^{1/2} \right\} \\ n &= 0.45 \end{aligned}$$

The critical location is at the shoulder to the right of C, where $n = 0.45$ and finite life is predicted. *Ans.*

Though not explicitly called for in the problem statement, a static check for yielding is especially warranted with such a low fatigue factor of safety. Using Eq. (7-15), with $M_m = T_a = 0$,

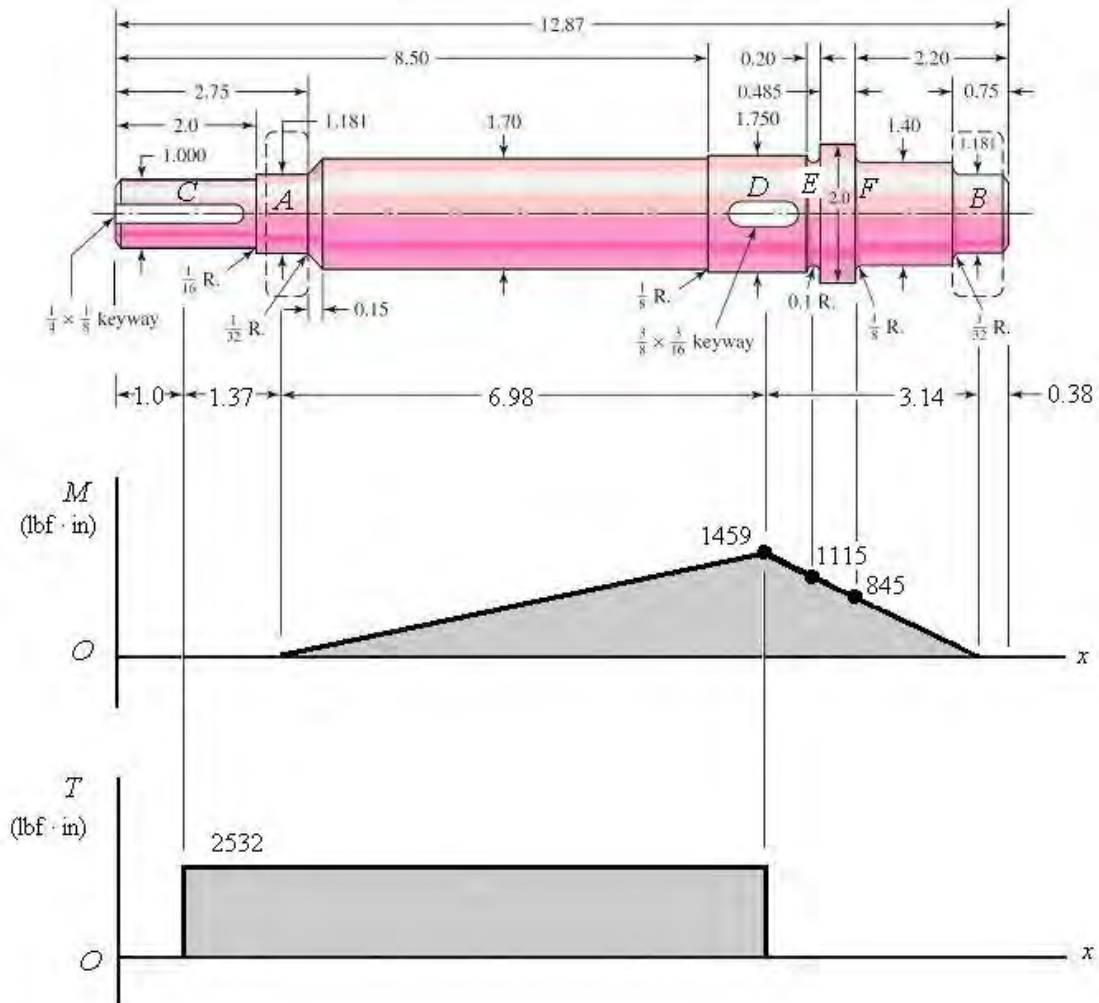
$$\begin{aligned} \sigma'_{\max} &= \left[\left(\frac{32K_f M_a}{\pi d^3} \right)^2 + 3 \left(\frac{16K_{fs} T_m}{\pi d^3} \right)^2 \right]^{1/2} \\ &= \left[\left(\frac{32(1.7)(6750)}{\pi(1.3)^3} \right)^2 + 3 \left(\frac{16(1.5)(2819)}{\pi(1.3)^3} \right)^2 \right]^{1/2} = 55\,845 \text{ psi} = 55.8 \text{ kpsi} \\ n &= \frac{S_y}{\sigma'_{\max}} = \frac{54}{55.8} = 0.97 \end{aligned}$$

This indicates localized yielding is predicted at the stress-concentration, though after localized cold-working it may not be a problem. The finite fatigue life is still likely to be the failure mode that will dictate whether this shaft is acceptable.

It is interesting to note the impact of stress concentration on the acceptability of the proposed design. This problem is linked with several previous problems (see Table 1-2, p. 34) in which the shaft was considered to have a constant diameter of 1.25 in. In each of the previous problems, the 1.25 in diameter was more than adequate for deflection, static, and fatigue considerations. In this problem, even though practically the entire shaft has diameters larger than 1.25 in, the stress concentrations significantly reduce the anticipated fatigue life.

Problem 7-23

- (a) Label the approximate locations of the effective centers of the bearings as A and B , the fan as C , and the gear as D , with axial dimensions as shown. Since there is only one gear, we can combine the radial and tangential gear forces into a single resultant force with an accompanying torque, and handle the statics problem in a single plane. From statics, the resultant reactions at the bearings can be found to be $R_A = 209.9$ lbf and $R_B = 464.5$ lbf. The bending moment and torque diagrams are shown, with the maximum bending moment at D of $M_D = 209.9(6.98) = 1459$ lbf·in and a torque transmitted from D to C of $T = 633(8/2) = 2532$ lbf·in. Due to the shaft rotation, the bending stress on any stress element will be completely reversed, while the torsional stress will be steady. Since we do not have any information about the fan, we will ignore any axial load that it would introduce. It would not likely contribute much compared to the bending anyway.



Potentially critical locations are identified as follows:

- Keyway at C, where the torque is high, the diameter is small, and the keyway creates a stress concentration.
- Keyway at D, where the bending moment is maximum, the torque is high, and the keyway creates a stress concentration.
- Groove at E, where the diameter is smaller than at D, the bending moment is still high, and the groove creates a stress concentration. There is no torque here, though.
- Shoulder at F, where the diameter is smaller than at D or E, the bending moment is still moderate, and the shoulder creates a stress concentration. There is no torque here, though.
- The shoulder to the left of D can be eliminated since the change in diameter is very slight, so that the stress concentration will undoubtedly be much less than at D.

Table A-20: $S_{ut} = 68$ kpsi, $S_y = 57$ kpsi

Eq. (6-8): $S'_e = 0.5(68) = 34.0$ kpsi

Eq. (6-19): $k_a = 2.70(68)^{-0.265} = 0.883$

Keyway at C

Since there is only steady torsion here, only a static check needs to be performed. We'll use the maximum shear stress theory.

$$\tau = \frac{Tr}{J} = \frac{2532(1.00/2)}{\pi(1.00^4)/32} = 12.9 \text{ kpsi}$$

$$\text{Eq. (5-3): } n_y = \frac{S_y/2}{\tau} = \frac{57/2}{12.9} = 2.21$$

Keyway at D

Assuming $r/d = 0.02$ for typical end-milled keyway cutter (p. 365), with $d = 1.75$ in, $r = 0.02d = 0.035$ in.

Table 7-1: $K_t = 2.14$, $K_{ts} = 3.0$

Fig. 6-20: $q = 0.66$

Fig. 6-21: $q_s = 0.72$

$$\text{Eq. (6-32): } K_f = 1 + q(K_t - 1) = 1 + 0.66(2.14 - 1) = 1.8$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.72(3.0 - 1) = 2.4$$

$$\text{Eq. (6-20): } k_b = \left(\frac{1.75}{0.30}\right)^{-0.107} = 0.828$$

$$\text{Eq. (6-18): } S_e = 0.883(0.828)(34.0) = 24.9 \text{ kpsi}$$

We will choose the DE-Gerber criteria since this is an analysis problem in which we would like to evaluate typical expectations.

Using Eq. (7-9) with $M_m = T_a = 0$,

$$A = \sqrt{4(K_f M_a)^2} = \sqrt{4[(1.8)(1459)]^2} = 5252 \text{ lbf} \cdot \text{in} = 5.252 \text{ kip} \cdot \text{in}$$

$$B = \sqrt{3(K_{fs} T_m)^2} = \sqrt{3[(2.4)(2532)]^2} = 10\,525 \text{ lbf} \cdot \text{in} = 10.53 \text{ kip} \cdot \text{in}$$

$$\begin{aligned} \frac{1}{n} &= \frac{8A}{\pi d^3 S_e} \left\{ 1 + \left[1 + \left(\frac{2BS_e}{AS_{ut}} \right)^2 \right]^{1/2} \right\} \\ &= \frac{8(5.252)}{\pi(1.75^3)(24.9)} \left\{ 1 + \left[1 + \left(\frac{2(10.53)(24.9)}{(5.252)(68)} \right)^2 \right]^{1/2} \right\} \end{aligned}$$

$$n = 3.59$$

Ans.

Groove at E

We will assume Figs. A-15-14 is applicable since the 2 in diameter to the right of the groove is relatively narrow and will likely not allow the stress flow to fully develop. (See Fig. 7-9 for the stress flow concept.)

$$r/d = 0.1 / 1.55 = 0.065, \quad D/d = 1.75 / 1.55 = 1.13$$

Fig. A-15-14: $K_t = 2.1$

Fig. 6-20: $q = 0.76$

Eq. (6-32): $K_f = 1 + q(K_t - 1) = 1 + 0.76(2.1 - 1) = 1.8$

Eq. (6-20): $k_b = \left(\frac{1.55}{0.30}\right)^{-0.107} = 0.839$

Eq. (6-18): $S_e = 0.883(0.839)(34) = 25.2 \text{ kpsi}$

Using Eq. (7-9) with $M_m = T_a = T_m = 0$,

$$A = \sqrt{4(K_f M_a)^2} = \sqrt{4[(1.8)(1115)]^2} = 4122 \text{ lbf} \cdot \text{in} = 4.122 \text{ kip} \cdot \text{in}$$

$$B = 0$$

$$\begin{aligned} \frac{1}{n} &= \frac{8A}{\pi d^3 S_e} \left\{ 1 + \left[1 + \left(\frac{2BS_e}{AS_{ut}} \right)^2 \right]^{1/2} \right\} \\ &= \frac{8(4.122)}{\pi(1.55^3)(25.2)} \left\{ 1 + \left[1 + (0)^2 \right]^{1/2} \right\} \\ n &= 4.47 \quad \text{Ans.} \end{aligned}$$

Shoulder at F

$$r/d = 0.125 / 1.40 = 0.089, \quad D/d = 2.0 / 1.40 = 1.43$$

Fig. A-15-9: $K_t = 1.7$

Fig. 6-20: $q = 0.78$

Eq. (6-32): $K_f = 1 + q(K_t - 1) = 1 + 0.78(1.7 - 1) = 1.5$

Eq. (6-20): $k_b = \left(\frac{1.40}{0.30}\right)^{-0.107} = 0.848$

Eq. (6-18): $S_e = 0.883(0.848)(34) = 25.5 \text{ kpsi}$

Using Eq. (7-9) with $M_m = T_a = T_m = 0$,

$$A = \sqrt{4(K_f M_a)^2} = \sqrt{4[(1.5)(845)]^2} = 2535 \text{ lbf} \cdot \text{in} = 2.535 \text{ kip} \cdot \text{in}$$

$$B = 0$$

$$\begin{aligned}\frac{1}{n} &= \frac{8A}{\pi d^3 S_e} \left\{ 1 + \left[1 + \left(\frac{2BS_e}{AS_{ut}} \right)^2 \right]^{1/2} \right\} \\ &= \frac{8(2.535)}{\pi(1.40^3)(25.5)} \left\{ 1 + \left[1 + (0)^2 \right]^{1/2} \right\} \\ n &= 5.42 \quad \text{Ans.}\end{aligned}$$

Above Average

Grade:

Name: Chinagorom Louis

This assignment is due at 8 am on Oct 28th. Complete the following problems from Shigley's Mechanical Engineering Design – 10th edition.

1. Problem 7-1 ✓
2. Problem 7-4 ✓
3. Problem 7-11 ✓
4. Problem 7-18-(a) ✓
5. Problem 7-19 ✓
6. Problem 7-23-(a) ✓

**Homework requirement**

- Print this page and use it as the cover.
- The submission should be each student's own work.
- The same day late submission will get a 10% penalty.
- Once the solution gets posted, no more submission will get accepted.
- Write legibly. No credit will be given if the work is not readable.
- Take advantage of office hours.
 - Tuesdays & Thursdays 1 PM ~ 3 PM
 - Email me: japark@pvamu.edu to request a Zoom meeting if you cannot make it to the regular office hour due to any university excused reasons.
- You may also bring the questions related to this homework to the Friday live sessions.

Summary of Comments on Louis- MCEG 4043 HW3

Page: 1



Number: 1 Author: japark Subject: Sticky Note Date: 12/15/2020 12:50:15 PM
Graded 7-1 and 7-18. I see your hard work. Thanks!

10/10

Chinagorom Louis MCG 4043 - HW3

Problem 7-1

Given

Shaft w/ bending & torsion $M_a = 70 \text{ Nm}$, $T_a = 45 \text{ Nm}$

$M_m = 55 \text{ Nm}$, $T_m = 35 \text{ Nm}$

$S_u = 700 \text{ MPa}$, $S_y = 560 \text{ MPa}$, $S_e = 210 \text{ MPa}$

$k_t = 2.2$, $k_{ts} = 1.8$, $n = 2.0$

Find

a) min d using DE-Gerber

$$d = \left(\frac{8nA}{\pi S_e} \left\{ 1 + \left[1 + \left(\frac{2BS_e}{A S_{ut}} \right)^2 \right]^{1/2} \right\} \right)^{1/3}$$

$$A = \sqrt{4(k_t M_a)^2 + 3(k_{ts} T_a)^2}$$

$$= \sqrt{4(2.2 \times 70 \text{ Nm})^2 + 3(1.8 \times 45 \text{ Nm})^2} = 338.45$$

$$B = \sqrt{4(k_t M_m)^2 + 3(k_{ts} T_m)^2}$$

$$= \sqrt{4(2.2 \times 55 \text{ Nm})^2 + 3(1.8 \times 35 \text{ Nm})^2} = 265.46$$

$$d = \left(\frac{8(2.0)338.45}{\pi \times 210 \text{ MPa}} \left\{ 1 + \left[1 + \left(\frac{2 \times 265.46}{338.45 \times 770 \text{ MPa}} \right)^2 \right]^{1/2} \right\} \right)^{1/3}$$

$$\boxed{d = 2.54 \text{ m}}$$

b) min d using DE-ASME Elliptic

$$d = \left\{ \frac{16n}{\pi} \left[4 \left(\frac{k_t M_a}{S_e} \right)^2 + 3 \left(\frac{k_{ts} T_a}{S_e} \right)^2 + 4 \left(\frac{k_t M_m}{S_y} \right)^2 + 3 \left(\frac{k_{ts} T_m}{S_y} \right)^2 \right]^{1/2} \right\}^{1/3}$$

$$d = \left\{ \frac{16 \times 2}{\pi} \left[4 \left(\frac{2.2 \times 70}{210} \right)^2 + 3 \left(\frac{1.8 \times 45}{210} \right)^2 + 4 \left(\frac{2.2 \times 55}{560} \right)^2 + 3 \left(\frac{1.8 \times 35}{560} \right)^2 \right]^{1/2} \right\}^{1/3}$$

$$\boxed{d = 2.58 \text{ m}}$$

c) DE-Soderberg

$$d = \left(\frac{16n}{\pi} \left\{ \frac{1}{S_e} \left[4(k_t M_a)^2 + 3(k_{ts} T_a)^2 \right]^{1/2} + \frac{1}{S_y} \left[4(k_t M_m)^2 + 3(k_{ts} T_m)^2 \right]^{1/2} \right\} \right)^{1/3}$$

$$d = \left(\frac{16 \times 2}{\pi} \left\{ \frac{1}{210} \left[4(2.2 \times 70)^2 + 3(1.8 \times 45)^2 \right]^{1/2} + \frac{1}{560} \left[4(2.2 \times 55)^2 + 3(1.8 \times 35)^2 \right]^{1/2} \right\} \right)^{1/3}$$

$$\boxed{d = 2.77 \text{ m}}$$

d) DE-Goodman

$$d = \left(\frac{16n}{\pi} \left\{ \frac{1}{S_e} \left[4(k_{ts} T_m)^2 + 3(k_{ts} T_a)^2 \right]^{1/2} + \frac{1}{S_{ut}} \left[4(k_{ts} T_m)^2 + 3(k_{ts} T_a)^2 \right]^{1/2} \right\} \right)^{1/3}$$

$$d = \left(\frac{16 \times 2}{\pi} \left\{ \frac{1}{210} \left[4(2.2 \times 70)^2 + 3(1.8 \times 95)^2 \right]^{1/2} + \frac{1}{700} \left[4(2.2 \times 55)^2 + 3(1.8 \times 35)^2 \right]^{1/2} \right\} \right)^{1/3}$$

$$d = 2.73 \text{ m}$$

Each of the criterions produce similar minimum diameter values. From L $\rightarrow$ G Gerber < ASME Elliptic < Goodman < Soderberg.

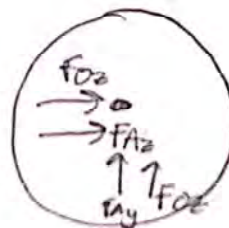
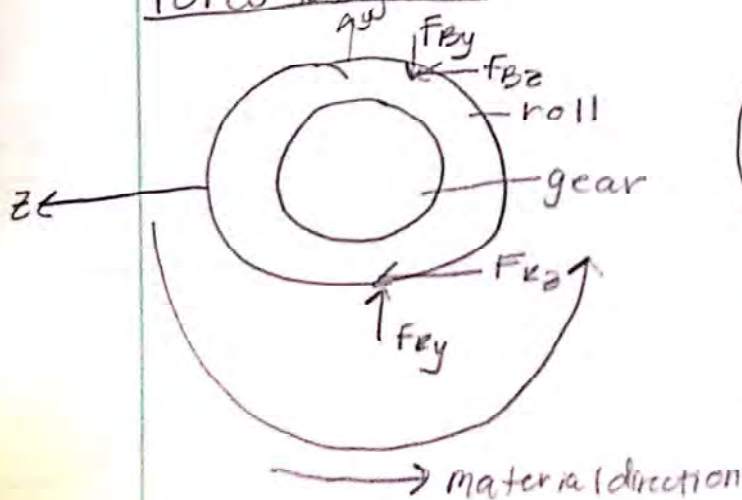
Problem 7-4

Given
 $n = 300 \text{ rev/min}$, driven by force F , $R = 3 \text{ in}$, $N = 30 \text{ lb/in}$, $\mu = 0.40$

Find:

(a) moment & shear diagrams modeling roll force as concentrated force @ center of roll

Force Diagram



$$F_{By} = 30 \text{ lb/in} \times 8 \text{ in} = 240 \text{ lb}$$

$$F_{Bz} = 0.4 \times 240 = 96 \text{ lb}$$

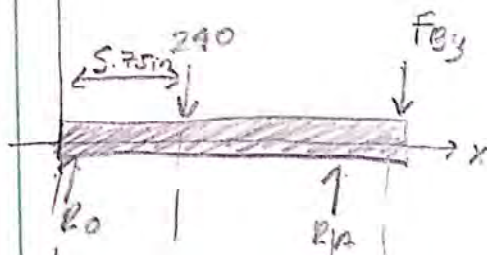
$$\sum T = F_{Bz} \times \frac{3}{2} - F_{K2} \times \frac{4}{2} = 0$$

$$F_{Bz} = 128 \text{ lb}$$

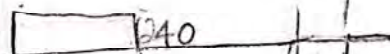
$$F_{By} = F_{Bz} \times \tan(20^\circ) = 46.6 \text{ lb}$$

(a) xy plane analysis:

Concentrated load



$V(\text{lbf})$

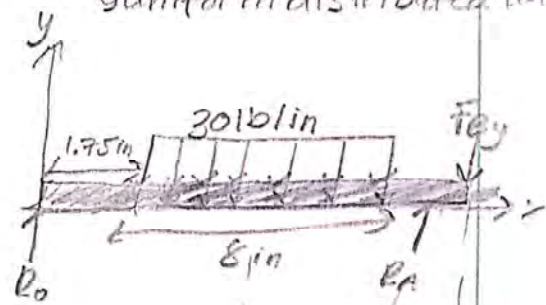


$M(\text{lbf}\cdot\text{in})$



(b) xy plane analysis:

Uniform distributed load



$V(\text{lbf})$

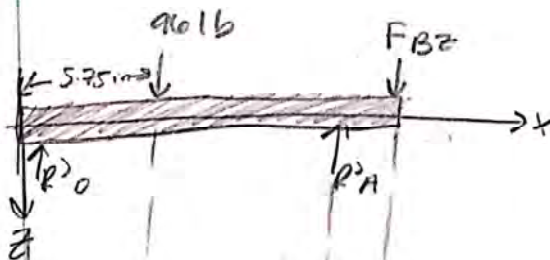


$M(\text{lbf}\cdot\text{in})$



xz plane analysis:

Concentrated load



$V(\text{lbf})$

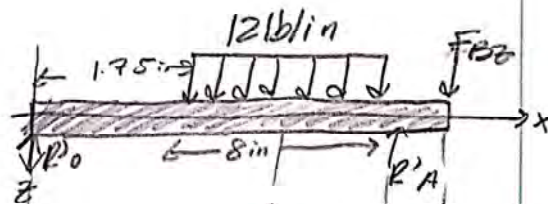


$M(\text{lbf}\cdot\text{in})$



xz plane analysis:

Uniform distributed load



$V(\text{lbf})$

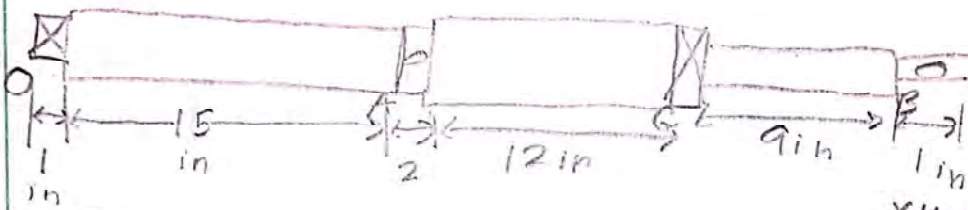


$M(\text{lbf}\cdot\text{in})$



Problem 7-11 - 3-72 p. 152.

(a) proposed shaft design



$$F_A = 300 \text{ lbf} = F_{A_t}$$

$$d_A = 15 \text{ in}$$

$$T_m = T = 300 \text{ lbf} \times 15 = 4500 \text{ lbf} \cdot \text{in}$$

$$F_{A_r} = 300 \tan 20^\circ = 671 \text{ lbf}$$

$$F_{B_t} = \frac{T}{r} = \frac{671}{7.5} = 89.5 \text{ lbf}$$

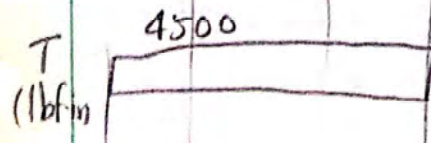
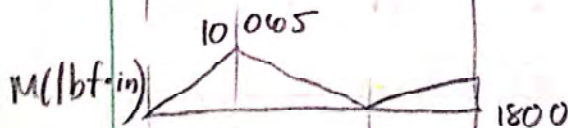
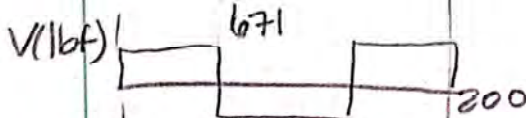
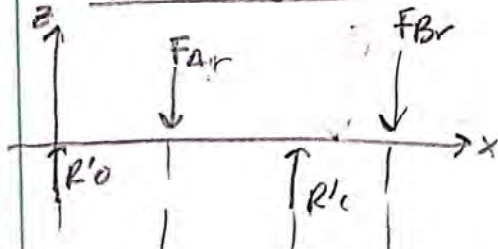
$$\sum \mathcal{M}_0 = F_{A_t} \times 16 - F_{B_t} \times 39 + R_c \times 30 = 0 \text{ (lbf)}$$

$$R_c = 276.35 \text{ lbf}$$

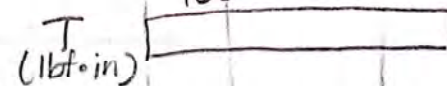
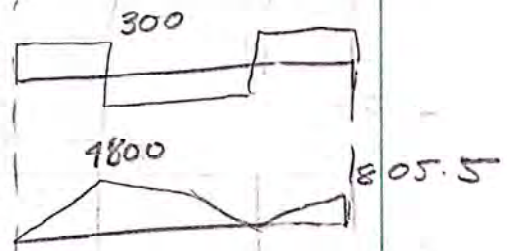
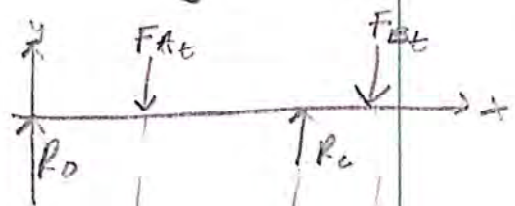
$$\uparrow \sum F_y = R_0 + R_c - F_{A_t} - F_{B_t}$$

$$R_0 = 113.15 \text{ lbf}$$

xy analysis



xy analysis



$$F_{B_r} = 89.5 \tan(20^\circ) = 200 \text{ lbf}$$

$$\sum \mathcal{M}_0 = -F_{A_r} \times 16 - F_{B_r} \times 39 + R'_c \times 30 = 0$$

$$R'_c = 618 \text{ lbf}$$

$$\uparrow \sum F_y = R'_0 + R'_c - F_{A_r} - F_{B_r} = 0$$

$$R'_0 = 253 \text{ lbf}$$

(b) Suitable material for shaft
 AISI 1018 CD steel

(c) determine critical diameters $\rightarrow n = 1.5$

Assume $\rightarrow$ Goodman Criteria

$$d = \left(\frac{16n}{\pi} \left[\frac{1}{S_e} [4(K_f M_a)^2 + 3(K_{fs} T_a)^2]^{1/2} + \frac{1}{S_{ut}} [4(K_f M_m)^2 + 3(K_{fs} T_m)^2]^{1/2} \right)^{1/3}$$

$$M_a = M_{TOTAL} = \sqrt{(M_{xy})^2 + (M_{xz})^2}$$

$$M_{xy} = 9800 + 1805.5$$

$$= \sqrt{(16605.5)^2 + (11865)^2}$$

$$M_{xz} = 10065 + 1800$$

$$M_a = 13578 \text{ lbf}\cdot\text{in}$$

Table A-20

$$S_{ut} = 64 \text{ kpsi}$$

$$S_e' = 0.5(64) = 32 \text{ kpsi}$$

$$S_e = k_a k_b k_c \times S_e'$$

$$k_a = a S_{ut}^b = 2.70(64)^{-0.265} = 0.90$$

$$k_b = 0.91(8)^{-0.157} = 0.65 = 0.65$$

$$k_c = 0.59$$

$$S_e = 0.90 \times 0.65 \times 0.59 \times 32 = 11.04 \text{ kpsi}$$

Fig 6-20

$$K_f = 1 + q(K_t - 1)$$

$$q = 0.70 \sim 0.72$$

assume

Table 7-1 $\rightarrow$ Key seat end mills

$$K_t = 2.14 \quad K_{ts} = 3.0$$

Fig 6-21

$$K_{fs} = 1 + q_s(K_{ts} - 1)$$

$$q_s = 0.69 \sim 0.71$$

$$K_{fs} = 1 + 0.69(3.0 - 1)$$

$$K_{fs} = 2.38$$

$$K_f = 1 + 0.70(2.14 - 1) = 1.80$$

$$d = \left(\frac{16(1.5)}{\pi} \left[\frac{1}{11.04 \times 10^3} [4(1.8 \times 13578)^2]^{1/2} + \frac{1}{6.9 \times 10^3} [3(238 \times 4500)^2]^{1/2} \right)^{1/3}$$

$$d = 12.8 \text{ in}$$

Problem 7-18(a)

Given

1030HR, ball bearing - left, cylindrical roller bearing - right

Find

- (a) n_{min} by evaluating @ critical locations
DE-ASME Elliptic fatigue criterion

Solution

$$\frac{1}{n} = \frac{16}{\pi d^3} \left[4 \left(\frac{K_t M_a}{S_a} \right)^2 + 3 \left(\frac{K_{ts} T_a}{S_a} \right)^2 + 4 \left(\frac{K_t M_m}{S_y} \right)^2 + 3 \left(\frac{K_{ts} T_m}{S_y} \right)^2 \right]^{1/2}$$

① $T = F_t \times r \rightarrow F = \frac{T}{r} = \frac{2500 \text{ lbf}}{4 \text{ in}}$

$F_t = 625 \text{ lbf/in}$

$F_r = F_t \tan(20^\circ) = 625 \tan(20^\circ) =$

$F_r = 227.5 \text{ lbf/in}$

Calculate:

$K_t, M_a, K_{ts}, M_m, T_a, T_m,$
 S_a, S_y

Find
 S_{ut}, S_y

Given

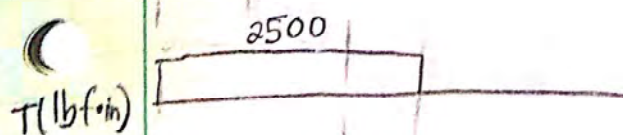
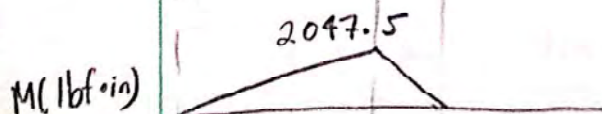
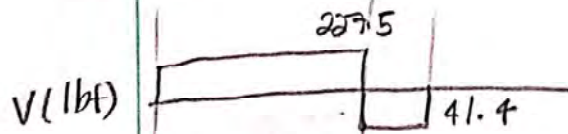
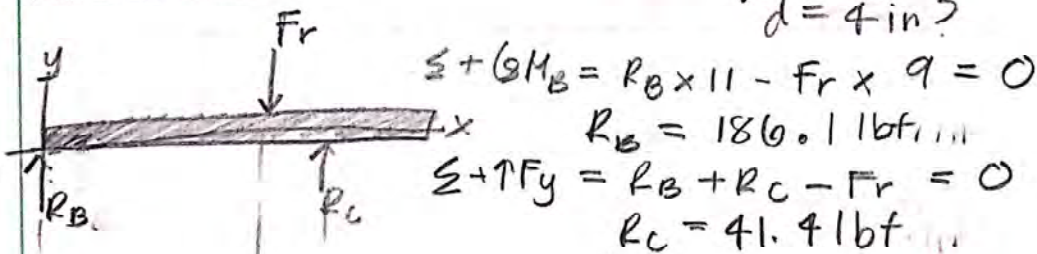
$T = 2500 \text{ lbf}\cdot\text{in}$

$n = 1200$

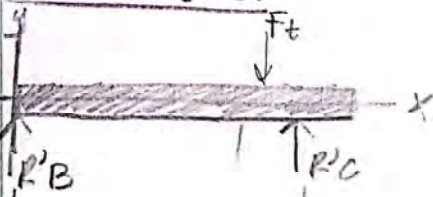
$\phi = 20^\circ$

$d = 4 \text{ in}$

② xy analysis:



② XZ Analysis:



$$\sum \circlearrowleft M_B = R'_B \times 11 - F_t \times 9 = 0$$

$$R'_B = 511.4 \text{ lbf}$$

$$\sum \uparrow F_y = R'_B + R'_C - F_t = 0$$

$$R'_C = 113.6 \text{ lbf}$$

$$\textcircled{4} M_{\text{total}} = \sqrt{M_{xy}^2 + M_{xz}^2}$$

$$= \sqrt{(2047.5)^2 + (5625)^2}$$

$$M_{\text{total}} = M_a = 5986 \text{ lbf} \cdot \text{in}$$

$$T = T_m = 2500 \text{ lbf} \cdot \text{in}$$

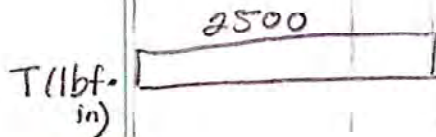
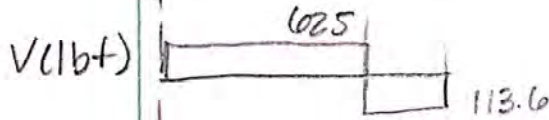


Table A-20

$$S_y = 37.5 \text{ kpsi}$$

$$S_{ut} = 68 \text{ kpsi}$$

$$S_e' = 0.5(68) = 34 \text{ kpsi}$$

$$S_e = K_a K_b K_c S_e'$$

$$K_a = a S_{ut}^b = 14.4(68)^{-0.718} = 0.70$$

$$K_b = 0.91 d^{-0.157} = 0.91(4)^{-0.157} = 0.73$$

$$K_c = 0.59$$

$$\rightarrow S_e = 0.70 \times 0.73 \times 0.59 \times 34 = 10.25 \text{ kpsi}$$

Fig 6-20

$$k_t = 1 + q(k_t - 1)$$

$$q = 0.62 \sqrt{0.65}$$

$$k_{ts} = 1 + q_s(k_{ts} - 1)$$

$$q_s = 0.73 \sqrt{0.76}$$

Table 7-1:

Assume:

Shoulder fillet - Sharp $\rightarrow K_{t,t} = 2.7, K_{t,s} = 2.2$

$$k_t = 1 + 0.62(2.7 - 1) = 2.1$$

$$K_{ts} = 1 + 0.73(2.2 - 1) = 1.9$$

$$\frac{1}{n} = \frac{16}{\pi(4)^3} \left[4 \left(\frac{2.1 \times 5986}{10.25 \times 10^3} \right)^2 + 3 \left(\frac{1.9 \times 2500}{37.5 \times 10^3} \right)^2 \right]^{1/2}$$

$$\frac{1}{n} = 0.196 \rightarrow 1 = 0.196n \rightarrow \boxed{n_{min} = 5.10}$$

Problem 7-19

Given

AISI 1018 CD steel, keyseat endmills

Find

$n_{min} \rightarrow$ DE-Gerber

Solution

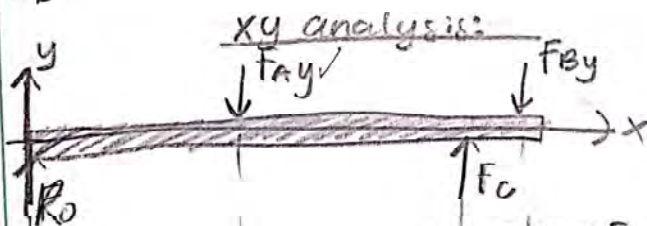
From 3-72 pg. 152

$$F_{Ay} = 300 \text{ lbf}, \phi = 20^\circ$$

$$d_A = 20 \text{ in}$$

$$F_B = ?$$

$$d_B = 8 \text{ in}$$

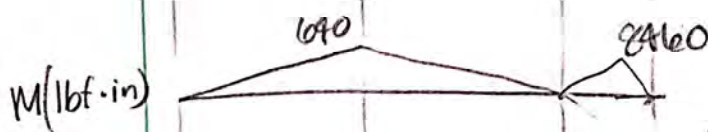


$$\sum \mathcal{M}_O = -F_{Ay} \times 16 - F_{By} \times 39 + F_C \times 30 = 0$$

$$F_C = 940 \text{ lbf}$$

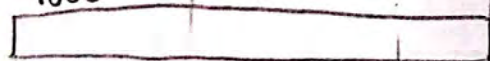
$$\sum \uparrow F_y = R_0 - F_{Ay} - F_{By} + F_C = 0$$

$$R_0 = 40 \text{ lbf}$$



T (lbf·in)

$$4800$$



M (lbf·in)

$$4800$$

T (lbf·in)

$$4800$$

$$\frac{1}{n} = \frac{8A}{\pi d^3 S_e} \left[1 + \left[1 + \left(\frac{2BS_e}{AS_e} \right)^2 \right]^{1/2} \right]$$

$$A = \sqrt{4(K+M_e)^2 + 3(K+T_e)^2}$$

$$B = \sqrt{4(K+M_m)^2 + 3(K+T_m)^2}$$

$$F_{Ar} = 300 \text{ lbf} \tan(20^\circ)$$

$$F_{Ar} = 109.2 \text{ lbf}$$

$$T = F_{Ay} \times r$$

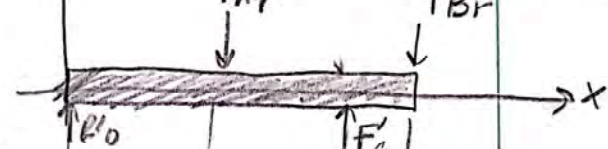
$$T = 300 \times 16 = 4800$$

$$F_{By} = \frac{T}{r} = \frac{4800}{8} = 600 \text{ lbf}$$

xz analysis

$$F_{Br} = 600 \tan(20^\circ)$$

$$F_{Br} = 218.4 \text{ lbf}$$

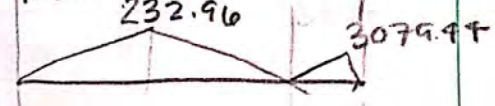
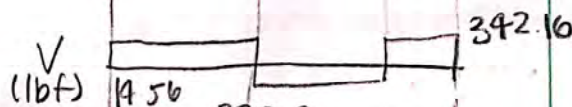


$$\sum \mathcal{M}_O = F_{Ar} \times 16 - F_{Br} \times 39 + F_C \times 30$$

$$F'_C = 342.16 \text{ lbf}$$

$$\sum \uparrow F_y = R'_0 - F_{Ar} - F_{Br} + F'_C = 0$$

$$R'_0 = 14.56$$



$$4800$$

$$M_{total} = \sqrt{M_{xy}^2 + M_{xz}^2}$$

$$= \sqrt{(9100)^2 + (-3312.4)^2}$$

$$M_{total} = M_a = 9684 \text{ lbf} \cdot \text{in}$$

$$T = T_m = 4800$$

Table A-20

$$S_y = 54 \text{ kpsi}$$

$$S_{ut} = 64 \text{ kpsi}$$

$$S_e' = 0.5(64) = 32 \text{ kpsi}$$

$$S_e = k_a k_b k_c \times S_e'$$

$$k_a = a S_{ut}^{-b} = 2.70(64)^{-0.265} = 0.90$$

$$k_b = 0.91(8)^{-0.157} = 0.65$$

$$k_c = 0.59$$

$$S_e = 0.90 \times 0.65 \times 0.59 \times 32 = 11.04 \text{ kpsi}$$

Fig 6-21

$$k_{fs} = 1 + q_s(k_{ts} - 1)$$

$$q = 0.69 \sim 0.71$$

Fig 6-20 $k_f = 1 + q(k_t - 1)$
 $q = 0.70 \sim 0.72$

Table 7-1

$$k_b = 2.14 \quad k_{ts} = 3.0$$

$$k_f = 1 + 0.70(2.14 - 1)$$

$$k_f = 1.80$$

$$k_{fs} = 1 + 0.69(3.0 - 1)$$

$$k_{fs} = 2.38$$

$$A = \sqrt{4(1.80 \times 9684)^2}$$

$$A = 34862.4$$

$$B = \sqrt{3(2.38 \times 4800)^2}$$

$$B = 19786.9$$

$$\frac{1}{n} = \frac{8(34862.4)}{\pi(8)^3 \times 11.04 \times 10^3} \leq 1 + \left[1 + \left(\frac{2 \times 19786.9 \times 11.04 \times 10^3}{34862.4 \times 64 \times 10^3} \right)^2 \right]^{1/2}$$

$$\frac{1}{n} = .0317 \rightarrow 1 = 0.0269n \rightarrow \boxed{n_{min} = 31.5}$$

Problem 7-23(a)

Given

Driven by right keyway, support = 2 deep groove ball bearings, AISI 1020 CD Steel, $F_r = 230 \text{ lb}$, $F_t = 633 \text{ lbf}$, $d = 8 \text{ in}$

Find

$n_{\min} \rightarrow$ DE Gerber failure criterion

Solution

$$\frac{1}{n} = \frac{8A}{\pi d^3 S_e} \left\{ 1 + \left[1 + \left(\frac{2BS_e}{AS_{ut}} \right)^2 \right]^{1/2} \right\}$$

$$A = \sqrt{4(k_f M_o)^2 + 3(k_{fs} T_o)^2}$$

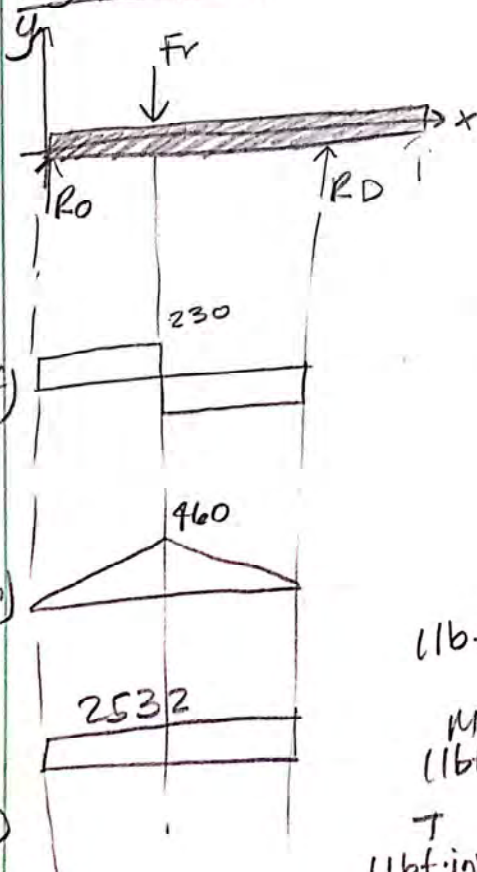
$$B = \sqrt{4(k_f M_o)^2 + 3(k_{fs} T_m)^2}$$

Find

$S_e, S_{ut}, k_f, k_{fs}, M_o, T_m$

$$T_m = F_t \times r = 633 \text{ lbf} \times 4 \text{ in} = 2532 \text{ lbf}\cdot\text{in}$$

xy plane analysis



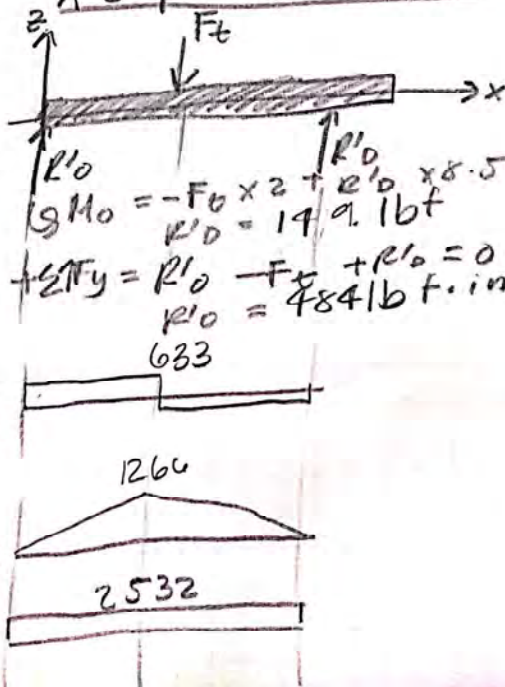
$$\sum \mathcal{M}_o = F_r \times 2 + R_D \times 8.50 = 0$$

$$R_D = 54.1$$

$$+\sum F_y = R_o - F_r + R_D = 0$$

$$R_o = 175.9 \text{ lbf}\cdot\text{in}$$

xz plane analysis



$$\sum \mathcal{M}_o = -F_t \times 2 + R'_D \times 8.50 = 0$$

$$R'_D = 149.1 \text{ lbf}$$

$$+\sum F_y = R'_o - F_t + R'_D = 0$$

$$R'_o = 484 \text{ lbf}\cdot\text{in}$$

$$M_{total} = M_a = \sqrt{(M_{xy})^2 + (M_{xz})^2} \\ = \sqrt{(460)^2 + (1260)^2} = 1317 \text{ lb-in}$$

Table A-20:

$$S_{uc} = 68 \text{ kpsi}$$

$$S_e' = 0.5(68) = 34 \text{ kpsi}$$

$$S_e = K_a K_b K_c \times S_e'$$

$$K_a = a S_{uc}^b = 2.70(68)^{-0.265} = 0.88$$

$$K_b = 0.91(8)^{-0.157} = 0.66$$

$$K_c = 0.59$$

$$S_e = 0.88 \times 0.66 \times 0.59 \times 34 = 11.65 \text{ kpsi}$$

$$K_t = 1 + q(K_t - 1)$$

$$K_{ts} = 1 + q_s(K_{ts} - 1)$$

Fig 6-20 $q = 0.62 \sim 0.65$ Fig 6-21 $q_s = 0.73 \sim 0.76$

Table 7-1

Assume:

end mill keyseat

$$K_t = 2.14$$

$$K_{ts} = 3.0$$

$$K_t = 1 + 0.62(2.14 - 1)$$

$$K_{ts} = 1 + 0.73(3.0 - 1)$$

$$K_{ts} = 2.46$$

$$K_t = 1.71$$

$$A = \sqrt{4(K_t M_a)^2} = \sqrt{4(1.71 \times 1317)^2} = 4607$$

$$B = \sqrt{3(K_{ts} T_m)^2} = \sqrt{3(2.46 \times 2532)^2} = 10788$$

$$\frac{1}{n} = \frac{8(4607)}{\pi(8)^3 \times 11.65 \times 10^3} \left\{ 1 + \left[1 + \left(\frac{2 \times 10788 \times 11.65 \times 10^3}{4607 \times 68 \times 10^3} \right)^2 \right]^{1/2} \right\}$$

$$\frac{1}{n} = 0.0041 \rightarrow 1 = 0.0045n \rightarrow n_{min} = 222.81$$

Average

Grade:

Name: Iyanuloluwa Olalumade

This assignment is due at 8 am on Oct 28th. Complete the following problems from Shigley's Mechanical Engineering Design – 10th edition.

1. Problem 7-1
2. Problem 7-4
3. Problem 7-11
4. Problem 7-18-(a)
5. Problem 7-19
6. Problem 7-23-(a)



Homework requirement

- Print this page and use it as the cover.
- The submission should be each student's own work.
- The same day late submission will get a 10% penalty.
- Once the solution gets posted, no more submission will get accepted.
- Write legibly. No credit will be given if the work is not readable.
- Take advantage of office hours.
 - Tuesdays & Thursdays 1 PM ~ 3 PM
 - Email me: japark@pvamu.edu to request a Zoom meeting if you cannot make it to the regular office hour due to any university excused reasons.
- You may also bring the questions related to this homework to the Friday live sessions.

Summary of Comments on HW3

Page: 1

 Number: 1 Author: japark Subject: Sticky Note Date: 12/15/2020 12:51:42 PM
Graded 7-1 and 7-18. 7-11 and 19 are missing (-2).

8/10

Yamufunwa
Olahumode

7.1 a) DE - Gerber Eq. (7-10)

$$A = \sqrt{4(K_f M_a)^2 + 3(K_{fs} T_a)^2} = \sqrt{4[(2.2)(70)]^2 + 3[(1.8)(45)]^2} = 338.4 \text{ Nm}$$

$$B = \sqrt{4(K_f M_m)^2 + 3(K_{fs} T_m)^2} = \sqrt{4[(2.2)(55)]^2 + 3[(1.8)(35)]^2} = 265.5 \text{ Nm}$$

$$d = \left[\frac{8(2)(338.4)}{\pi (210)(10^6)} \left[1 + \left[1 + \frac{2(265.5)(210)(10^6)}{338.4(700)(10^6)} \right]^2 \right]^{\frac{1}{2}} \right]^{\frac{1}{3}}$$

$$d = 25.85 (10^{-3}) \text{ m} = 25.85 \text{ mm}$$

b) DE - elliptic Eq. (7-12) can be shown to be

$$d = \left[\frac{16n}{\pi} \left(\frac{A}{S_e} + \frac{B}{S_y} \right) \right]^{\frac{1}{3}} = \left[\frac{16(2)}{\pi} \left(\frac{338.4}{210(10^6)} + \frac{265.5}{560(10^6)} \right) \right]^{\frac{1}{3}}$$

$$d = 25.77 (10^{-3}) \text{ m} = 25.77 \text{ mm}$$

c) DE - Soderberg Eq. (7-14) can be shown to be

$$d = \left[\frac{16n}{\pi} \left(\frac{A}{S_e} + \frac{B}{S_y} \right) \right]^{\frac{1}{3}} = \left[\frac{16(2)}{\pi} \left(\frac{338.4}{210(10^6)} + \frac{265}{560(10^6)} \right) \right]^{\frac{1}{3}}$$

$$d = 27.70 (10^{-3}) \text{ m} = 27.70 \text{ mm}$$

d) DE - Goodman: Eq. (7-8)

$$d = \left[\frac{16n}{\pi} \left(\frac{A}{S_e} + \frac{B}{S_{ut}} \right) \right]^{\frac{1}{3}} = \left[\frac{16(2)}{\pi} \left(\frac{338.4}{210(10^6)} + \frac{265.5}{700(10^6)} \right) \right]^{\frac{1}{3}}$$

$$d = 27.27 (10^{-3}) \text{ m} = 27.27 \text{ mm}$$

DE - Gerber - 25.85

DE - Elliptic - 25.77

DE - Soderberg - 27.70

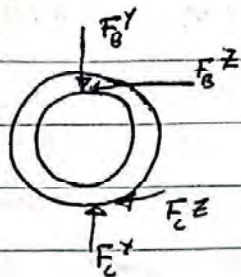
DE - Goodman 27.27

0.31% Lower Less conservative

7.2% Higher More conservative

5.5% Higher more conservative

7.4



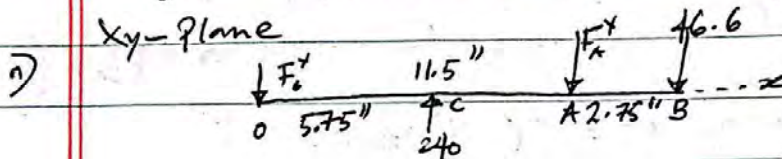
$$F_C^Y = 30(8) = 240 \text{ lb}$$

$$F_C^Z = 0.4(240) = 96 \text{ lb}$$

$$T = F_C^Z(2) = 96(2) = 192 \text{ lb} \cdot \text{in}$$

$$F_B^Z = \frac{T}{1.5} = \frac{192}{1.5} = 128 \text{ lb}$$

$$F_B^Y = F_B^Z \tan 20^\circ = 128 \tan 20^\circ = 46.6 \text{ lb}$$



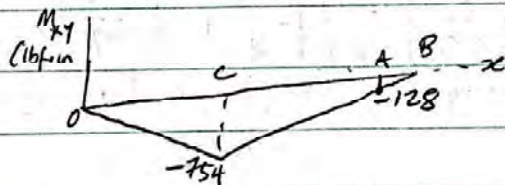
$$\sum M_O = 240(5.75) - F_A^Y(11.5) - 46.6(14.25) = 0$$

$$F_A^Y = \frac{240(5.75) - 46.6(14.25)}{11.5} = 62.3 \text{ lb}$$

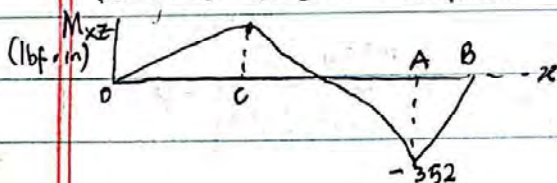
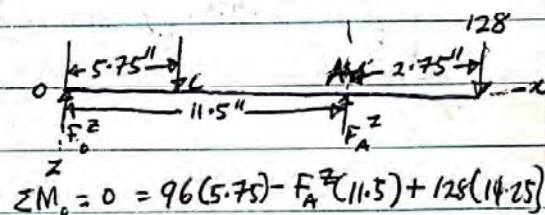
$$\sum M_A = F_O^Y(11.5) - 46.6(2.75) - 240(5.75) = 0$$

$$F_O^Y = \frac{240(5.75) + 46.6(2.75)}{11.5} = 131.1 \text{ lb}$$

BENDING MOMENT DIAGRAM



BENDING MOMENT DIAGRAM

 xz plane

$$\sum M_O = 0 = 96(5.75) - F_A^Z(11.5) + 128(14.25)$$

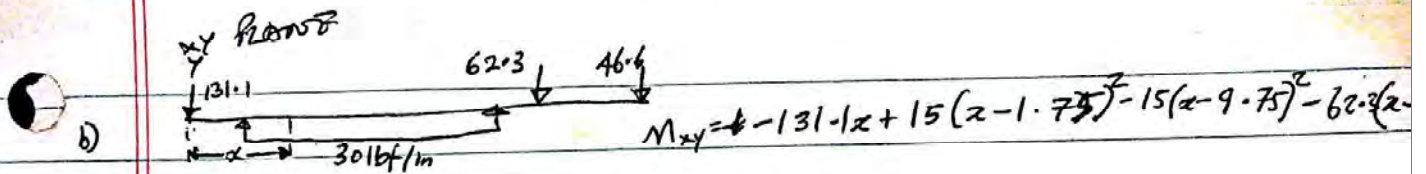
$$F_A^Z = 206.6 \text{ lb}$$

$$\sum M_A = 0 = F_O^Z(11.5) + 128(2.75) - 96(5.75)$$

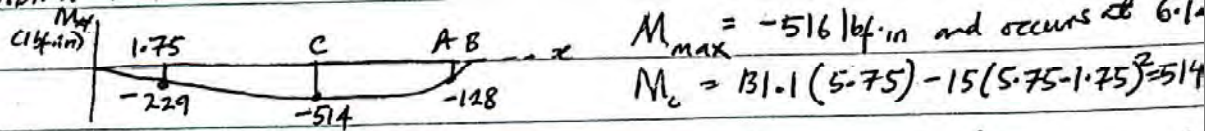
$$F_O^Z = 17.4 \text{ lb}$$

$$M_C = \sqrt{100^2 + (-754)^2} = 761 \text{ lb} \cdot \text{in}$$

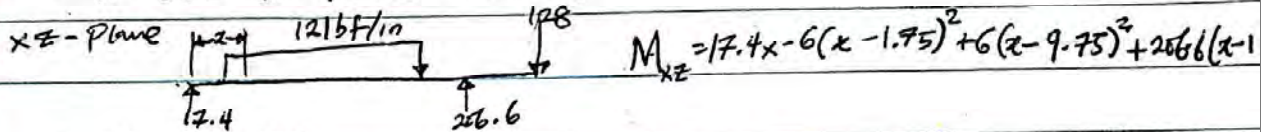
$$M_A = \sqrt{(-128)^2 + (-352)^2} = 375 \text{ lb} \cdot \text{in}$$



BENDING MOMENT DIAGRAM



This is reduced from 754 lb/in found in Part (a). The maximum occurs at $x = 6.12$ in rather than C.

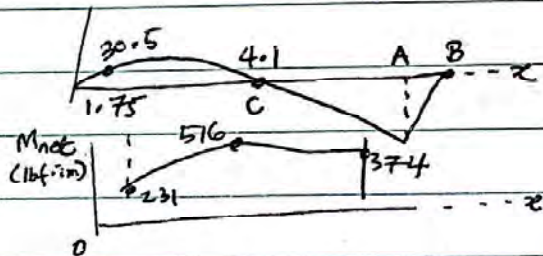


BENDING MOMENT

LET $M_{\text{net}} = \sqrt{M_{xy}^2 + M_{xz}^2}$

Plot $M_{\text{net}}(x)$, $1.75 \leq x \leq 11.75$

$M_{\max} = 516 \text{ lb/in}$ at $x = 6.25 \text{ in}$



7-18 a) USE the distortion-energy elliptic failure locus. The torque and moment loadings on the shaft are shown in the solution to Prob. 7-17

Table A-20 For 1030 HR $S_{ut} = 68 \text{ kpsi}$, $S_y = 37.5 \text{ kpsi}$, $f_B = 1/37$

Eq (6-8):

$S_e = 0.5(68) = 34.0 \text{ kpsi}$

Eq (6-19)

$k_a = 2.70(68)^{-0.265} = 0.583$

$k_c = k_d = k_e = 1$

$$S_{ut} = 64 \text{ ksi} \quad S_y = 54 \text{ ksi}$$

$$S_e = 0.5(64) = 32 \text{ ksi}$$

$$k_a = 2.70(64)^{-0.265} = 0.897$$

$$\text{with } d = 1.75, \delta = 0.02d = 0.035 \text{ in}$$

$$k_t = 2.14, k_{fs} = 3.0$$

$$q = 0.65$$

$$q_s = 0.71$$

$$k_t = 1 + q(k_t - 1) = 1 + 0.65(2.14 - 1) = 1.7$$

$$k_{fs} = 1 + q_s(k_{fs} - 1) = 1 + 0.71(3.0 - 1) = 2.4$$

$$k_b = \left(\frac{1.75}{0.30} \right)^{0.107} = 0.828$$

$$S_e = 0.897(0.828)(32) = 23.8 \text{ ksi}$$

$$A = \sqrt{4(k_t M_a)^2} = \sqrt{4[1.7(5324)]^2} = 18102 \text{ lbf in} = 18.10 \text{ kip in}$$

$$B = \sqrt{3(k_{fs} T_m)^2} = \sqrt{3[2.4(2819)]^2} = 11.72 \text{ kip in}$$

$$\frac{1}{n} = \frac{8A}{\pi d^3 S_e} \left[1 + \left[1 + \left(\frac{2BS_e}{AS_{ut}} \right)^2 \right]^{1/2} \right]$$

$$n = 1.3$$

$$A = \sqrt{4(k_t M_a)^2} = \sqrt{4[(1.7)(6750)]^2} = 25.65 \text{ kip in}$$

$$B = \sqrt{3(k_{fs} T_m)^2} = \sqrt{3[(1.6)(2819)]^2} = 7.812 \text{ kip in}$$

$$n = \frac{8(25.65)}{\pi(1.75^3)(23.8)} \left[1 + \left[1 + \left(\frac{2(7.812)(23.8)}{25.65(64)} \right)^2 \right]^{1/2} \right]$$

$$n = 0.96$$

$$\text{Required } \sigma_{max} = \left[\left(\frac{32(1.7)(6750)}{\pi(1.3)^3} \right)^2 + 3 \left(\frac{16(1.5)(2819)}{\pi(1.3)} \right)^2 \right]^{1/2}$$

$$= 55.8 \text{ ksi}$$

$$n = \frac{S_y}{\sigma_{max}} = \frac{54}{55.8} = 0.97$$

7-23

$$S_{ut} = 68 \text{ kpsi}, S_y = 57 \text{ kpsi}$$

$$S_e = 0.5(68) = 34 \text{ kpsi}$$

$$k_a = 2.70(68)^{-0.265} = 0.883$$

$$C = \frac{T_e}{J} = \frac{2532(1.00/2)}{\pi(1.00^4)/32} = 12.9 \text{ kpsi}$$

$$n_y = \frac{S_y/2}{C} = \frac{57/2}{12.9} = 2.21$$

$$k_t = 2.14, k_{ts} = 3.0$$

$$q = 0.66$$

$$q_s = 0.72$$

$$k_t = 1 + q(k_t - 1) = 1 + 0.66(2.14 - 1) = 1.8$$

$$k_{ts} = 1 + q_s(k_{ts} - 1) = 1 + 0.72(3.0 - 1) = 2.4$$

$$k_b = \left(\frac{1.75}{0.30}\right)^{-0.107} = 0.822$$

$$S_e = 0.883(0.822)(34.0) = 24.9 \text{ kpsi}$$

$$A = \sqrt{4(k_t M_a)^2} = \sqrt{4[1.8(1459)]^2} = 5.252 \text{ kip.in}$$

$$B = \sqrt{3[(2.4)(2532)]^2} = 10.53 \text{ kip.in}$$

$$n = \frac{8(5.252)}{\pi(1.75^3)(24.9)} \left[1 + \left[1 + \left[\frac{2(10.53)(24.9)}{(5.252)(68)} \right]^2 \right]^{\frac{1}{2}} \right]$$

$$= 3.59$$

Below Average

HW 3



①

7-1)

Gurbar criterion: $A = \sqrt{4(K_f \cdot M_n)^2 + 3(K_{fs} \cdot T_n)^2}$

$$A = \sqrt{4(2.2 \times 70)^2 + 3(1.8 \times 45)^2} = \sqrt{94864 + 19683} = 338.45 \text{ N}\cdot\text{m}$$

$$(Eqn 7-10) \quad B = \sqrt{4(K_f \cdot M_n)^2 + 3(K_{fs} \cdot T_n)^2} = \sqrt{4(2.2 \times 5.5)^2 + 3(1.8 \times 35)^2}$$

$$B = \sqrt{70741} = 265.4637 \text{ N}\cdot\text{m}$$

$$d = \left(\frac{8 \cdot n \cdot A}{\pi \cdot S_e} \left\{ 1 + \left[1 + \left(\frac{2 \cdot B \cdot S_e}{A \cdot S_{ut}} \right)^2 \right]^{1/2} \right\} \right)^{1/3}$$

$$d = \left(\frac{8 \cdot 2 \cdot 338.45 \times 10^3}{\pi \cdot 210} \left\{ 1 + \left[1 + \left(\frac{2 \cdot 265.4637 \times 10^3 \times 210}{338.45 \times 10^3 \times 700} \right)^2 \right]^{1/2} \right\} \right)^{1/3}$$

$$d = \left(8208.1 \left\{ 1 + \left[1 + (0.4706)^2 \right]^{1/2} \right\} \right)^{1/3}$$

$$d = (8208.1 \times 2.1052)^{1/3} = 25.85 \text{ mm}$$

(minimum accept diameter of shaft)

$$b) \quad d = \left\{ \frac{16 \cdot n}{\pi} \left[4 \left(\frac{K_f \cdot M_n}{S_e} \right)^2 + 3 \left(\frac{K_{fs} \cdot T_n}{S_e} \right)^2 + 4 \left(\frac{K_r \cdot M_n}{S_y} \right)^2 + 3 \left(\frac{K_{rs} \cdot T_n}{S_y} \right)^2 \right] \right\}^{1/2}$$

$$d = \left\{ \frac{16 \cdot 2}{\pi} \left[4 \left(\frac{2.2 \times 70 \times 10^3}{210} \right)^2 + 3 \left(\frac{1.8 \times 45 \times 10^3}{210} \right)^2 + 4 \left(\frac{2.2 \times 55 \times 10^3}{560} \right)^2 + 3 \left(\frac{1.8 \times 35 \times 10^3}{560} \right)^2 \right] \right\}^{1/2}$$

$$d = \left\{ 10.1559 \left[4(133)^2 + 3(386)^2 + 4(216)^2 + 3(112.5)^2 \right] \right\}^{1/2}$$

$$d = 25.77 \text{ mm} \quad (\text{min. diameter of shaft acceptable})$$

Summary of Comments on CamScanner 10-28-2020 09.15.05

Page: 1

 Number: 1 Author: japark Subject: Sticky Note Date: 12/15/2020 12:52:44 PM
Graded 7-1 and 7-18 (-4). 7-11, 23 are missing (-2). Late -10%

3.6/10

$$\begin{aligned}
 c) \quad d &= \left\{ \frac{16 \times 2}{\pi} \left(\frac{1}{210} \left[4(2.2 \times 70 \times 10^3)^2 + 3(1.8 \times 45 \times 10^3)^2 \right]^{1/2} + \frac{1}{560} \left[4(2.2 \times 55 \times 10^3)^2 + 3(1.8 \times 35 \times 10^3)^2 \right]^{1/2} \right) \right\}^{1/3} \\
 &= \left\{ 10.18 \times \left(\frac{1}{210} \left[4(154000)^2 + 3(81000)^2 \right]^{1/2} + \frac{1}{560} \left[4(121000)^2 + 3(63000)^2 \right]^{1/2} \right) \right\}^{1/3} \\
 &= \left\{ 10.18 \times (1611.65 + 474.04) \right\}^{1/3} = 27.69 \text{ mm}
 \end{aligned}$$

$$\begin{aligned}
 d) \quad d &= \left\{ \frac{16 \times 2}{\pi} \left(\frac{1}{210} \left[4(2.2 \times 70 \times 10^3)^2 + 3(1.8 \times 45 \times 10^3)^2 \right]^{1/2} + \frac{1}{700} \left[4(2.2 \times 55 \times 10^3)^2 + 3(1.8 \times 35 \times 10^3)^2 \right]^{1/2} \right) \right\}^{1/3} \\
 d &= \left\{ 10.18 \left(\frac{1}{210} \left[4(154000)^2 + 3(81000)^2 \right]^{1/2} + \frac{1}{700} \left[4(121000)^2 + 3(63000)^2 \right]^{1/2} \right) \right\}^{1/3} \\
 d &= \left\{ 10.18 \times (1611.65 + 379.233) \right\}^{1/3} \\
 d &= 27.27 \text{ mm}
 \end{aligned}$$

7-4)

$$F_e^y = F_N \times L = 30 \times 8 = 240 \text{ lbf}$$

$$F_e^z = 0.4 \times 240 = 96 \text{ lbf} ; T = F_e^z \times d/2 = 96 \times \left(\frac{4}{2}\right) = 192$$

$$\Delta F_B^z = T / d/2 = 192 / (3/2) = 128 \text{ lbf} ; F_B^y = F_B^z \tan 20^\circ = 46.6 \text{ lbf}$$

$$a) \sum M_O = 0 ; 240 \times 5.75 - F_A^y \times 11.5 - 46.6 \times 14.25 = 0$$

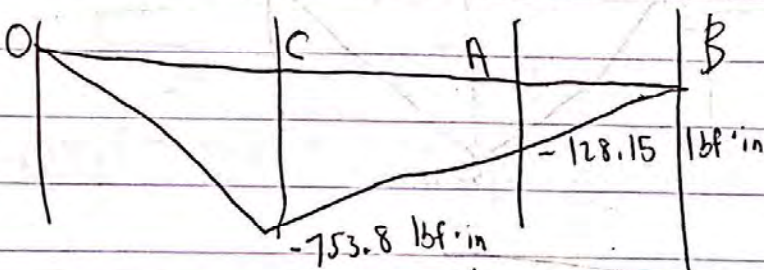
$$F_A^y = \frac{240 \times 5.75 - 46.6 \times 14.25}{11.5} = 62.15 \text{ lbf}$$

$$\sum M_A = F_O^y \times 11.5 - 46.6 \times 2.75 - 240 \times 5.75 = 0 ;$$

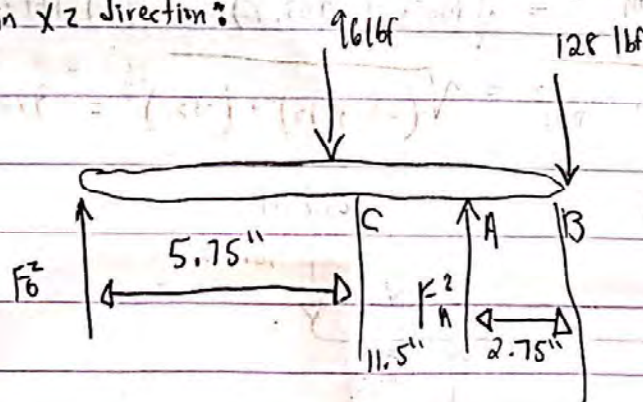
$$F_O^y = \frac{46.6 \times 2.75 + 240 \times 5.75}{11.5} = 131.1 \text{ lbf}$$

$$\Delta M_O = 0 ; M_C = -F_O^y \times 5.75 = -131.1 \times 5.75 = -753.8 \text{ lbf}\cdot\text{in}$$

$$M_A = -F_B^y \times 2.75 = -46.6 \times 2.75 = -128.15 \text{ lbf}\cdot\text{in}$$



FBD of roller in xz direction:



$$\sum M_o = 0 : 96 \times 5.75 - F_A^2 \times 11.5 + 128 \times 14.25 = 0$$

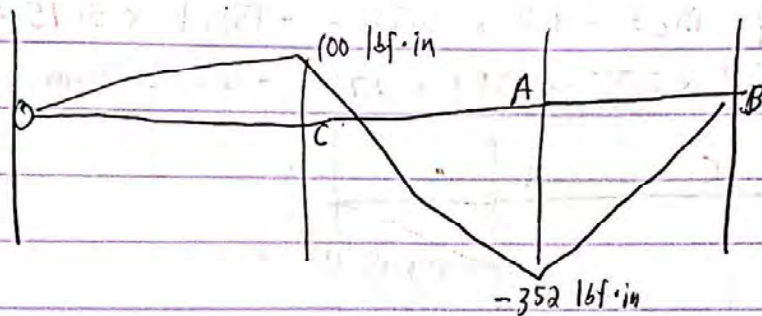
$$F_A^2 = \frac{96 \times 5.75 + 128 \times 14.25}{11.5} = 206.6 \text{ lbf}$$

$$\sum M_A = 0 : F_o^2 \times 11.5 + 128 \times 2.75 - 96 \times 5.75 = 0$$

$$F_o^2 = \frac{96 \times 5.75 - 128 \times 2.75}{11.5} = 17.4 \text{ lbf}$$

$$\sum M_o = 0 ; M_c = F_o^2 \times 5.75 = 17.4 \times 5.75 = 100 \text{ lbf}\cdot\text{in}$$

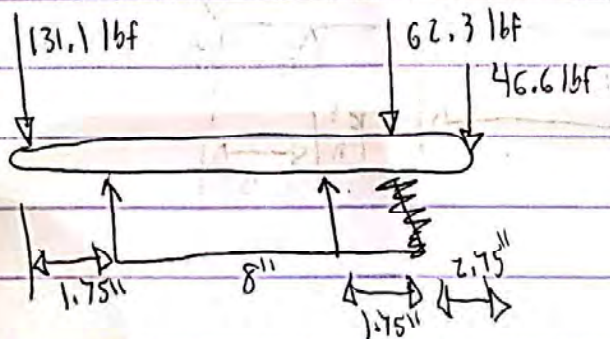
$$M_A = F_D^2 \times 2.75 = -128 \times 2.75 = -352 \text{ lbf}\cdot\text{in} ; M_B = 0$$



$$M_c = \sqrt{M_{cxy}^2 + M_{cxz}^2} = \sqrt{100^2 + (-753.7)^2} = 761 \text{ lbf}\cdot\text{in}$$

$$M_p = \sqrt{M_{pxy}^2 + M_{pxz}^2} = \sqrt{(-128.15)^2 + (-352)^2} = 375 \text{ lbf}\cdot\text{in}$$

b)

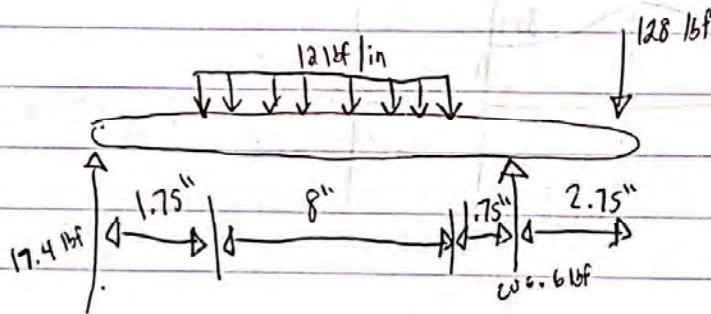
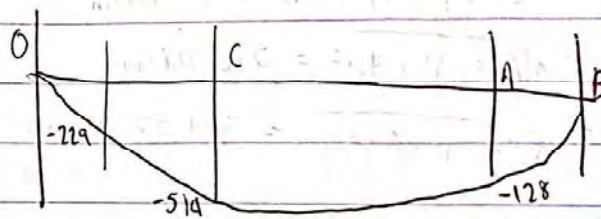


$$\square M_{xy} = -131.1x + 15 \times (x-1.75)^2 ; \frac{dM_{xy}}{dx} = 0$$

$$-131.1 + (15 \times 2 \times (x-1.75)) = 0 ; x = 6.12 \text{ in}$$

$$M_{\max} = -131.3 \times 6.12 + 15 \times (6.12 - 1.75)^2 = -516 \text{ lbf}\cdot\text{in}$$

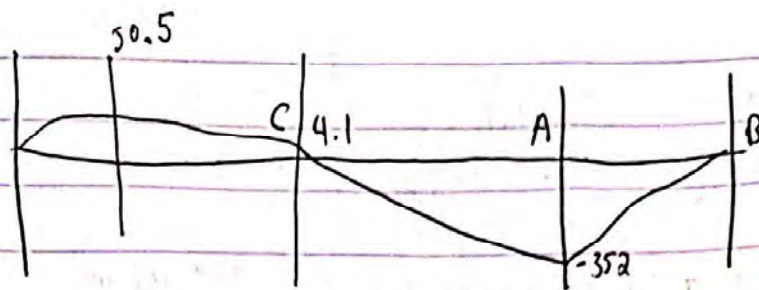
$$M_c = -131.1 \times 5.75 + 15 \times (5.75 - 1.75)^2 = 514 \text{ lbf}\cdot\text{in}$$



$$M_{xz} = 17.4x - 6 \times (x-1.75)^2 ; M_{1.75} = 17.4 \times 1.75 = 30.45 \text{ lbf}\cdot\text{in}$$

$$M_c = 17.4 \times 5.75 - 6 \times (5.75 - 1.75)^2 = 4.01 \text{ lbf}\cdot\text{in}$$

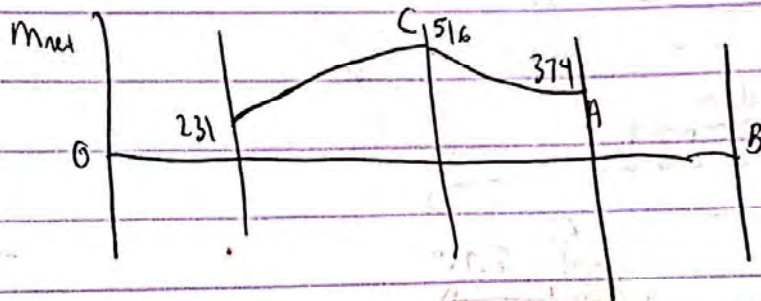
$$M_A = -128 \times 2.75 = -352 \text{ lbf}\cdot\text{in}$$



$$M_{net@1.75} = \sqrt{M_{xy}^2 + M_{xz}^2} = \sqrt{(-229)^2 + 30.5^2} = 231 \text{ lbf}\cdot\text{in}$$

$$M_c = \sqrt{M_{xy}^2 + M_{xz}^2} = \sqrt{(-516)^2 + 4.1^2} = 516 \text{ lbf}\cdot\text{in}$$

$$M_c = \sqrt{M_{xy}^2 + M_{xz}^2} = \sqrt{(-188)^2 + (-352)^2} = 374.55 \text{ lbf}\cdot\text{in}$$



~~Handwritten scribbles~~

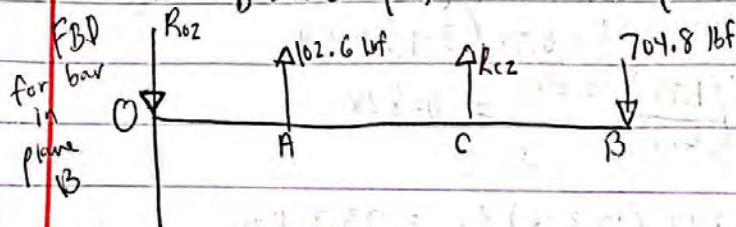
$$7-19) \sum M_x = 0; -300 \cos 20(10) + F_B \cos 20(9) = 0; F_B = 750 \text{ lbf}$$

$$\sum M_o = 0; 300 \cos 20(16) - 750 \sin 20(39) + R_{cy}(30) = 0; R_{cy} = 183.1 \text{ lbf}$$

$$\sum F_y = 0; R_{cy} + 300 \cos 20 + 183 - 750 \sin 20 = 0; R_{cy} = -208.1 \text{ lbf}$$

$$M_A = R_{cy}(16) = -208.5 \times 16 = -3336 \text{ lbf}\cdot\text{in}$$

$$M_C = F_B \sin 20^\circ(9) = 750 \sin 20^\circ(9) = 256.5 \times 9 = 2308.5 \text{ lbf}\cdot\text{in}$$



$$\sum M_o = 0; 300 \sin 20(16) - R_{cz}(30) - 750 \cos 20(39) = 0; R_{cz} = -861 \text{ lbf}$$

$$\sum F_z = 0; R_{o2} - 300 \sin 20 - 861 + 750 \cos 20 = 0; R_{o2} = 259 \text{ lbf}$$

$$M_A = 259.3 \times 16 = -4149 \text{ lbf}\cdot\text{in}$$

$$M_C = 704.8 \times 9 = 6343 \text{ lbf}\cdot\text{in}$$

$$M_A = \sqrt{3336^2 + (-4149)^2} = 5324 \text{ lbf}\cdot\text{in}$$

$$M_C = \sqrt{(-2308)^2 + (-6343)^2} = 6750 \text{ lbf}\cdot\text{in} \quad (\text{critical location @ C})$$

$$T = (300 \cos 20^\circ)10 = 2819 \text{ lbf}\cdot\text{in}$$

$$\text{Moment } (M_a) = 5324 \text{ lbf}\cdot\text{in} \quad \{ \text{Torque } (T_m) = 2819 \text{ lbf}\cdot\text{in}$$

$$(M_m) = T_a = 0$$

for AISI 1018 CD steel: $S_{ut} = 64 \text{ ksi}$ $S_y = 54 \text{ ksi}$

$$S_e' = 0.5 S_{ut} = 0.5 (64) = 32 \text{ ksi}$$

From table "parameters in Marin surface condition": $a = 2.7$, $b = -0.265$

$$K_a = a S_{ut}^b = 2.7 (64)^{-0.265} = 0.897$$

Keyway at A: $r = 0.02 \times 1.75 = 0.035 \text{ in}$

From table "First iteration estimates for stress": $K_t = 2.14$ & $K_{ts} = 3$

Determine fatigue stress conc. factor: $K_f = 1 + q(K_t - 1) = 1 + 0.65(2.14 - 1) = 1.7$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.71(3 - 1) = 2.4$$

$$K_b = \left(\frac{d}{0.3}\right)^{-0.107} = \left(\frac{1.75}{0.3}\right)^{-0.107} = 0.828$$

$$S_e = K_a K_b S_e' = 0.897 (0.828) 32 = 23.8 \text{ ksi}$$

$$\square \text{ DE Gerber criteria eqn: } \left(\frac{1}{n}\right) = \frac{8A}{\pi d^3 S_e} \left\{ 1 + \left[1 + \left(\frac{2BS_e}{A S_{ut}} \right)^2 \right]^{1/2} \right\}$$

$$A = \sqrt{4(K_f M_a)^2} = \sqrt{4(1.7(5324))^2} = 18.1 \text{ kip.in}$$

$$B = \sqrt{3(K_{fs} T_m)^2} = \sqrt{3(2.4(2819))^2} = 11.7 \text{ kip.in}$$

$$\left(\frac{1}{n}\right) = \frac{8(18.1)}{\pi (1.75)^3 23.8} \left\{ 1 + \left[1 + \left(\frac{2(11.7)23.8}{(18.1)64} \right)^2 \right]^{1/2} \right\}$$

$n = 1.3$

For shoulder bearings: Shoulder to the left of C: $D = 2.5 \text{ in} \rightarrow$

$$\frac{r}{d} = \frac{0.0625}{1.75} = 0.036 ; \quad \frac{D}{d} = \frac{2.5}{1.75} = 1.43 ;$$

$$K_t = 2.2 \text{ \& } K_b = 1.8$$

From figure round shaft with shoulder fillet in torsion: $\frac{r}{d} = \frac{0.0625}{1.75}$

$$\square q_s = 0.76 ; \quad q = 0.71 ; \quad r = 0.16 \text{ in} ; \quad K_f = 1 + q(K_t - 1)$$

MACHINE DESIGN II

FALL 2020

TITLE OF ASSIGNMENT: HOMEWORK 4

TOPICS COVERED IN HOMEWORK 4

1. Rolling Contact Bearings (Chapter 11)
2. Clutches, Brakes, Couplings, and Flywheels (Chapter 16)

Grade:

Name:

This assignment is due at 8 am on Nov 16th. Complete the following problems from Shigley's Mechanical Engineering Design – 10th edition.

1. Problem 11-6
2. Problem 11-7
3. Problem 11-22
4. Problem 16-1
5. Problem 16-2
6. Problem 16-7

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- Print this page and use it as the cover.
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- Take advantage of office hours.
 - Tuesdays & Thursdays 1 PM ~ 3 PM
 - Email me: japark@pvamu.edu to request a Zoom meeting if you cannot make it to the regular office hour due to any university excused reasons.

Problem 11-6

For the straight cylindrical roller bearing specified with a service factor of 1, $R = 0.95$ and $F_R = 20$ kN.

$$x_D = \frac{L_D}{L_R} = \frac{60L_D n_D}{L_{10}} = \frac{60(8000)950}{10^6} = 456$$

$$C_{10} = 20 \left\{ \frac{456}{0.02 + 4.439 [\ln(1/0.95)]^{1/1.483}} \right\}^{3/10} = 145 \text{ kN} \quad \text{Ans.}$$

Problem 11-7

Both bearings need to be rated in terms of the same catalog rating system in order to compare them. Using a rating life of one million revolutions, both bearings can be rated in terms of a Basic Load Rating.

$$\text{Eq. (11-3):} \quad C_A = F_A \left(\frac{L_A}{L_R} \right)^{1/a} = F_A \left(\frac{L_A n_A 60}{L_R} \right)^{1/a} = 2.0 \left[\frac{(3000)(500)(60)}{10^6} \right]^{1/3}$$

$$= 8.96 \text{ kN}$$

Bearing B already is rated at one million revolutions, so $C_B = 7.0$ kN. Since $C_A > C_B$, bearing A can carry the larger load. Ans.

Problem 11-22

$$F_r = 8 \text{ kN}, \quad R = 0.9, \quad L_D = 10^9 \text{ rev}$$

$$\text{Eq. (11-3):} \quad C_{10} = F_D \left(\frac{L_D}{L_R} \right)^{1/a} = 8 \left(\frac{10^9}{10^6} \right)^{1/3} = 80 \text{ kN}$$

From Table 11-2, select the 85 mm bore. Ans.

Problem 16-1

Given: $r = 300/2 = 150$ mm, $a = R = 125$ mm, $b = 40$ mm, $f = 0.28$, $F = 2.2$ kN, $\theta_1 = 0^\circ$, $\theta_2 = 120^\circ$, and $\theta_a = 90^\circ$. From which, $\sin \theta_a = \sin 90^\circ = 1$.

$$\text{Eq. (16-2):}$$

$$M_f = \frac{0.28 p_a (0.040)(0.150)}{1} \int_{0^\circ}^{120^\circ} \sin \theta (0.150 - 0.125 \cos \theta) d\theta$$

$$= 2.993(10^{-4}) p_a \text{ N}\cdot\text{m}$$

$$\text{Eq. (16-3): } M_N = \frac{p_a(0.040)(0.150)(0.125)}{1} \int_{0^\circ}^{120^\circ} \sin^2 \theta d\theta = 9.478(10^{-4}) p_a \text{ N}\cdot\text{m}$$

$$c = 2(0.125 \cos 30^\circ) = 0.2165 \text{ m}$$

$$\text{Eq. (16-4): } F = \frac{9.478(10^{-4}) p_a - 2.993(10^{-4}) p_a}{0.2165} = 2.995(10^{-3}) p_a$$

$$p_a = F / [2.995(10^{-3})] = 2200 / [2.995(10^{-3})] \\ = 734.5(10^3) \text{ Pa for cw rotation}$$

$$\text{Eq. (16-7): } 2200 = \frac{9.478(10^{-4}) p_a + 2.993(10^{-4}) p_a}{0.2165}$$

$$p_a = 381.9(10^3) \text{ Pa for ccw rotation}$$

A maximum pressure of 734.5 kPa occurs on the RH shoe for cw rotation. *Ans.*

(b) RH shoe:

Eq. (16-6):

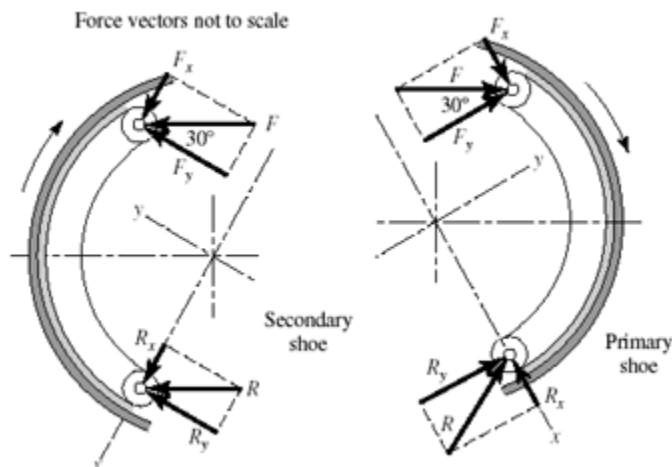
$$T_R = \frac{0.28(734.5)10^3(0.040)0.150^2(\cos 0^\circ - \cos 120^\circ)}{1} = 277.6 \text{ N}\cdot\text{m} \quad \text{Ans.}$$

LH shoe:

$$T_L = 277.6 \frac{381.9}{734.5} = 144.4 \text{ N}\cdot\text{m} \quad \text{Ans.}$$

$$T_{\text{total}} = 277.6 + 144.4 = 422 \text{ N}\cdot\text{m} \quad \text{Ans.}$$

(c)



RH shoe: $F_x = 2200 \sin 30^\circ = 1100 \text{ N}$, $F_y = 2200 \cos 30^\circ = 1905 \text{ N}$

Eqs. (16-8): $A = \left(\frac{1}{2} \sin^2 \theta \right)_{0^\circ}^{120^\circ} = 0.375$, $B = \left(\frac{\theta}{2} - \frac{1}{4} \sin 2\theta \right)_0^{2\pi/3 \text{ rad}} = 1.264$

Eqs. (16-9): $R_x = \frac{734.5(10^3)0.040(0.150)}{1} [0.375 - 0.28(1.264)] - 1100 = -1007 \text{ N}$
 $R_y = \frac{734.5(10^3)0.040(0.150)}{1} [1.264 + 0.28(0.375)] - 1905 = 4128 \text{ N}$
 $R = [(-1007)^2 + 4128^2]^{1/2} = 4249 \text{ N} \quad \text{Ans.}$

LH shoe: $F_x = 1100 \text{ N}$, $F_y = 1905 \text{ N}$

Eqs. (16-10): $R_x = \frac{381.9(10^3)0.040(0.150)}{1} [0.375 + 0.28(1.264)] - 1100 = 570 \text{ N}$
 $R_y = \frac{381.9(10^3)0.040(0.150)}{1} [1.264 - 0.28(0.375)] - 1905 = 751 \text{ N}$
 $R = (570^2 + 751^2)^{1/2} = 959 \text{ N} \quad \text{Ans.}$

Problem 16-2

Given: $r = 300/2 = 150 \text{ mm}$, $a = R = 125 \text{ mm}$, $b = 40 \text{ mm}$, $f = 0.28$, $F = 2.2 \text{ kN}$, $\theta_1 = 15^\circ$, $\theta_2 = 105^\circ$, and $\theta_a = 90^\circ$. From which, $\sin \theta_a = \sin 90^\circ = 1$.

Eq. (16-2):

$$M_f = \frac{0.28 p_a (0.040)(0.150)}{1} \int_{15^\circ}^{105^\circ} \sin \theta (0.150 - 0.125 \cos \theta) d\theta = 2.177(10^{-4}) p_a$$

$$\text{Eq. (16-3): } M_N = \frac{p_a (0.040)(0.150)(0.125)}{1} \int_{15^\circ}^{105^\circ} \sin^2 \theta d\theta = 7.765(10^{-4}) p_a$$

$$c = 2(0.125) \cos 30^\circ = 0.2165 \text{ m}$$

$$\text{Eq. (16-4): } F = \frac{7.765(10^{-4}) p_a - 2.177(10^{-4}) p_a}{0.2165} = 2.581(10^{-3}) p_a$$

$$\begin{aligned} \text{RH shoe: } p_a &= 2200 / [2.581(10^{-3})] = 852.4 (10^3) \text{ Pa} \\ &= 852.4 \text{ kPa on RH shoe for cw rotation } \text{Ans.} \end{aligned}$$

$$\text{Eq. (16-6): } T_R = \frac{0.28(852.4)10^3(0.040)(0.150^2)(\cos 15^\circ - \cos 105^\circ)}{1} = 263 \text{ N ?m}$$

LH shoe:

$$2200 = \frac{7.765(10^{-4}) p_a + 2.177(10^{-4}) p_a}{0.2165}$$

$$p_a = 479.1(10^3) \text{ Pa} = 479.1 \text{ kPa on LH shoe for cw rotation } \text{Ans.}$$

$$T_L = \frac{0.28(479.1)10^3(0.040)(0.150^2)(\cos 15^\circ - \cos 105^\circ)}{1} = 148 \text{ N ?m}$$

$$T_{\text{total}} = 263 + 148 = 411 \text{ N ?m } \text{Ans.}$$

Comparing this result with that of Prob. 16-1, a 2.6% reduction in torque is obtained by using 25% less braking material.

Problem 16-7

Preliminaries: $\theta_2 = 180^\circ - 30^\circ - \tan^{-1}(3/12) = 136^\circ$, $\theta_1 = 20^\circ - \tan^{-1}(3/12) = 6^\circ$,
 $\theta_a = 90^\circ$, $\sin \theta_a = 1$, $a = (3^2 + 12^2)^{1/2} = 12.37 \text{ in}$, $r = 10 \text{ in}$, $f = 0.30$, $b = 2 \text{ in}$, $p_a = 150 \text{ psi}$.

$$\text{Eq. (16-2): } M_f = \frac{0.30(150)(2)(10)}{\sin 90^\circ} \int_{6^\circ}^{136^\circ} \sin \theta (10 - 12.37 \cos \theta) d\theta = 12\,800 \text{ lbf ?in}$$

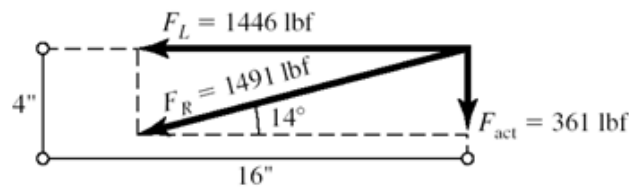
$$\text{Eq. (16-3): } M_N = \frac{150(2)(10)(12.37)}{\sin 90^\circ} \int_{6^\circ}^{136^\circ} \sin^2 \theta d\theta = 53\,300 \text{ lbf ?in}$$

LH shoe:

$$c_L = 12 + 12 + 4 = 28 \text{ in}$$

Now note that M_f is cw and M_N is ccw. Thus,

$$F_L = \frac{53\,300 - 12\,800}{28} = 1446 \text{ lbf}$$



Eq. (16-6): $T_L = \frac{0.30(150)(2)(10)^2(\cos 6^\circ - \cos 136^\circ)}{\sin 90^\circ} = 15\,420 \text{ lbf} \cdot \text{in}$

RH shoe:

$$M_N = 53\,300 \frac{p_a}{150} = 355.3 p_a, \quad M_f = 12\,800 \frac{p_a}{150} = 85.3 p_a$$

On this shoe, both M_N and M_f are ccw. Also,

$$c_R = (24 - 2 \tan 14^\circ) \cos 14^\circ = 22.8 \text{ in}$$

$$F_{\text{act}} = F_L \sin 14^\circ = 361 \text{ lbf} \quad \text{Ans.}$$

$$F_R = F_L / \cos 14^\circ = 1491 \text{ lbf}$$

Thus, $1491 = \frac{355.3 + 85.3}{22.8} p_a \Rightarrow p_a = 77.2 \text{ psi}$

Then, $T_R = \frac{0.30(77.2)(2)(10)^2(\cos 6^\circ - \cos 136^\circ)}{\sin 90^\circ} = 7940 \text{ lbf} \cdot \text{in}$

$$T_{\text{total}} = 15\,420 + 7940 = 23\,400 \text{ lbf} \cdot \text{in} \quad \text{Ans.}$$

Above Average

Grade:

Name:

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Summary of Comments on homework four

Page: 1



Number: 1 Author: japark Subject: Sticky Note Date: 12/15/2020 12:53:52 PM
Graded 11-6 and 16-2. Include FBD for brake problems.

10/10

Name Habeeb Ayangbade

11-6 Given $L_D = 8000h$
 $N = 950 \text{ rev/min}$

$$W = 20 \text{ kN}$$

$$R_D = 0.95$$

$$F_D = 20 \text{ kN}$$

$$X_D = \frac{60 L_D N_D}{Z_{10}}$$

$$= \frac{60 (8000) (950)}{10^6}$$

$$X_D = 456$$

$$C_{10} = a_f F_D \left[\frac{X_D}{x_0 + (8 - x_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{1/4}} \right]^{1/a}$$

$$C_{10} = (1)(20) \left[\frac{456}{0.02 + (4.439) \left(\ln \left(\frac{1}{0.95} \right) \right)^{1/4.439}} \right]^{3/10}$$

$$C_{10} = (1)(20) \left[\frac{456}{0.619} \right]^{3/10}$$

$$C_{10} = 144.95 \text{ kN}$$

$$C_{10} = 145 \text{ kN}$$

Choose a 03-150mm with $C_{10} = 151 \text{ kN}$.

11-7

$$F_R \cdot (L_R n_R b_o)^{1/4} = F_D \cdot (L_D n_D b_o)^{1/4}$$

$$2 \cdot (3000 \times 500 \times 60)^{1/3} = F_D \cdot (L_D n_D b_o)^{1/3}$$

$$F_D = \frac{228.94}{(L_D n_D)^{1/3}} \text{ kN}$$

Consider bearing B

$$F_R \cdot (L_R n_R b_o)^{1/4} = F_D \cdot (L_D n_D b_o)^{1/4}$$

$$7 \cdot (10^6)^{1/3} = F_D \cdot (L_D n_D b_o)^{1/3}$$

$$F_D = \frac{178.8}{(L_D n_D)^{1/3}} \text{ kN}$$

From the above results $F_{Da} > F_{Db}$
Therefore, the bearing A can carry the larger load.

11-22

$$X_D = \frac{L_0}{L_R}$$

$$X_D = \frac{10^7}{10^6} = 1000$$

For ball bearing $a=3$

Weibull parameters to be $X_0 = 0.02$

$$(b - X_0) = 4.439 \quad b = 1.483$$

$$C_{10} = q_f F_c \left[\frac{X_D}{X_0 + (b - X_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{1/b}} \right]^{1/a}$$

$$C_{10} = 1 \times 8 \left[\frac{1000}{0.02 + (4.439) \left(\ln \left(\frac{1}{0.9} \right) \right)^{1/1.483}} \right]^{1/3}$$

$$= 1 \times 8 \left[\frac{1000}{0.993348} \right]^{1/3}$$

$$= 80.178 \text{ kN}$$

From Table 11-2 Therefore, the Smallest bore size that can satisfy the given condition is 95 mm.

16-1 Given

$$d_m = 300 \text{ mm}$$

$$R = 125 \text{ mm}$$

$$b = 40 \text{ mm}$$

$$F = 2.2 \text{ kN}$$

$$f = 0.28$$

$$P_a = \frac{F}{0.003}$$

$$= \frac{2.2 \times 10^3}{0.003}$$

$$= 734 \text{ kPa}$$

$$a) M_f = \frac{f P_a b r}{\sin \theta_a} \int_{\theta_1}^{\theta_2} \sin \theta (r - a \cos \theta) d\theta$$

$$\text{where } \theta_1 = 0^\circ$$

$$\theta_2 = 120^\circ, \theta_a = 90^\circ, a = 125$$

$$M_f = \frac{0.28 \times P_a \times 0.04 \times 0.15}{\sin 90} \int_0^{120} \sin \theta (0.15 - 0.125 \cos \theta) d\theta$$

$$= 1.68 \times 10^{-3} P_a \left[(-0.15 \cos \theta)_0^{120} + 0.125 \left(\frac{1}{4} \cos 2\theta \right)_0^{120} \right]$$

$$M_f = 2.99 \times 10^{-4} P_a \text{ N}\cdot\text{m}$$

$$M_N = \frac{P_a b r a}{\sin \theta_a} \int_{\theta_1}^{\theta_2} \sin^2 \theta d\theta$$

$$= \frac{P_a \times 0.04 \times 0.15 \times 0.125}{\sin 90} \left[\frac{\theta}{2} - \frac{1}{4} \sin 2\theta \right]_0^{120}$$

$$= 7.5 \times 10^{-4} P_a \left[\frac{2\pi}{3} - \frac{1}{4} \sin 240^\circ \right]$$

$$= 9.47 \times 10^{-4} P_a \text{ N}\cdot\text{m}$$

$$F = \frac{M_N - M_f}{c} \quad \text{where } c = 2x a \cos \theta$$

$$= 2 \times 0.125 \cos 30^\circ$$

$$= 0.216 \text{ m}$$

$$F = \frac{9.47 \times 10^{-4} P_a - 2.99 \times 10^{-4} P_a}{0.216}$$

$$= 0.003 P_a \text{ N}$$

16-1

$$F = \frac{M_N + M_f}{c}$$

$$= \frac{9.47 \times 10^{-4} \text{ Pa} + 2.99 \times 10^{-4} \text{ Pa}}{0.216}$$

$$F = 0.0057 \text{ Pa}$$

$$P_a = \frac{F}{0.0057}$$

$$\frac{2.2 \times 10^3}{0.0057}$$

$$= 386 \text{ kPa}$$

$$P_{\max} = 734 \text{ kPa ccw}$$

$$\textcircled{b} T_R = \frac{f p_a b r^2 (\cos \theta_1 - \cos \theta_2)}{\sin \theta_1}$$

$$= \frac{0.28 \times 734 \times 10^3 \times 0.04 \times 0.15^2 (\cos 0^\circ - \cos 120^\circ)}{\sin 90^\circ}$$

$$= 277.5 \text{ N}\cdot\text{m}$$

$$T_L = \frac{f p_a b r^2 (\cos \theta_1 - \cos \theta_2)}{\sin \theta_1}$$

$$= \frac{0.28 \times 386 \times 10^3 \times 0.04 \times 0.15^2 (\cos 0^\circ - \cos 120^\circ)}{\sin 90^\circ}$$

$$T_L = 145.9 \text{ N}\cdot\text{m}$$

$$T_{\text{total}} = T_R + T_L$$

$$= 277.5 + 145.9$$

$$= 423.4 \text{ N}\cdot\text{m}$$

② force in horizontal direction

$$F_x = 2.2 \sin 30 = 1.1 \text{ kN}$$

force in vertical direction

$$F_y = 2.2 \cos 30 = 1.905 \text{ kN}$$

$$R_x = \frac{P_a b r}{\sin \theta_1} (A - f_B) - F_x$$

$$R_y = \frac{P_a b r}{\sin \theta_1} (B + f_A) - F_y$$

$$\text{where } A = \int_0^{\theta_2} \sin \theta \cos \theta d\theta$$

$$= \left(-\frac{1}{2} \cos 2\theta \right)_0^{2\pi/3}$$

$$= 0.375$$

$$B = \int_0^{\theta_2} \sin^2 \theta d\theta$$

$$= \left(\frac{\theta}{2} - \frac{1}{4} \sin 2\theta \right)_0^{2\pi/3}$$

$$= 1.264$$

$$\therefore R_x = \frac{734 \times 10^3 \times 0.04 \times 0.15 (0.375 - 0.28 \times 1.264)}{\sin 90^\circ}$$

$$1.1 \times 10^3$$

$$= -1.011 \text{ kN}$$

$$\therefore R_y = \frac{734 \times 10^3 \times 0.04 \times 0.15 (1.264 + 0.28 \times 0.375)}{\sin 90^\circ}$$

$$1.905 \times 10^3$$

$$= 4.12 \text{ kN}$$

$$R = \sqrt{(-1.01)^2 + 4.12^2}$$

$$R = 4.24 \text{ kN}$$

16-1

$$\therefore R_x = \frac{386 \times 10^3 \times 0.04 \times 0.15}{\sin 90^\circ} (0.375 + 0.28 \times 1.264) - 1.1 \times 10^3$$

$$= 0.59 \text{ kN}$$

$$R_y = \frac{386 \times 10^3 \times 0.04 \times 0.05}{\sin 90^\circ} (1.264 - 0.28 \times 0.375) - 1.905 \times 10^3$$

$$= -0.78 \text{ kN}$$

$$R = \sqrt{0.59^2 + 0.78^2}$$

$$R = 0.98 \text{ kN}$$

16-2

Given $d_{\text{rim}} = 300 \text{ mm}$

$$R = 125 \text{ mm}$$

$$b = 40 \text{ mm}$$

$$\bar{F} = 2.2 \text{ kN}$$

$$f = 0.28$$

$$M_f = \frac{f p_a b}{\sin \theta_a} \int_{\theta_1}^{\theta_2} \sin \theta (r - a \cos \theta) d\theta$$

$$\text{Where } \theta_1 = 15^\circ$$

$$\theta_2 = 105^\circ$$

$$\theta_a = 90^\circ$$

$$a = 125 \text{ mm}$$

$$M_f = \frac{0.28 \times p_a \times 0.04 \times 0.15}{\sin 90} \int_{15}^{105} \sin \theta (0.15 - 0.125 \cos \theta) d\theta$$

$$1.68 \times 10^{-3} p_a \left[(-0.15 \cos \theta)_{15}^{105} + 0.125 \left(\frac{1}{4} \cos 2\theta \right)_{15}^{105} \right]$$

$$= 0.2185 \times 10^{-4} p_a \text{ N.m}$$

$$M_N = \frac{p_a b r a}{\sin \theta_a} \int_{\theta_1}^{\theta_2} \sin^2 \theta d\theta$$

$$= \frac{p_a \times 0.04 \times 0.15 \times 0.125}{\sin 90} \left[\frac{\theta}{2} - \frac{1}{4} \sin 2\theta \right]_{15}^{105}$$

$$= 7.76 \times 10^{-4} p_a \text{ N.m}$$

$$F = \frac{M_N - M_f}{c}$$

$$\text{Where } c = 2 \times a \cos \theta$$

$$= 2 \times 0.125 \cos 30$$

$$= 0.216 \text{ m}$$

$$\therefore F = \frac{7.76 \times 10^{-4} p_a - 0.2185 \times 10^{-4} p_a}{0.216}$$

$$= 0.0035 p_a \text{ N}$$

$$p_a = \frac{F}{0.0035}$$

$$= \frac{2.2 \times 10^3}{0.0035}$$

$$= 628.6 \text{ kPa}$$

$$F = \frac{M_N + M_f}{c}$$

$$F = \frac{7.76 \times 10^{-4} p_a + 0.2185 \times 10^{-4} p_a}{0.216}$$

$$= 0.0037 p_a$$

$$p_a = \frac{F}{0.0037}$$

$$= \frac{2.2 \times 10^3}{0.0037}$$

$$= 594.6 \text{ kPa}$$

$$p_{\text{max}} = 628.6 \text{ kPa}$$

$$16-2 \quad T_R = \frac{f_p a b r^2 (\cos \theta_1 - \cos \theta_2)}{\sin \theta_1}$$

$$= \frac{0.28 \times 628.6 \times 10^3 \times 0.04 \times 0.15^2 \times (\cos 15^\circ - \cos 105^\circ)}{\sin 90^\circ}$$

$$= 194 \text{ N}\cdot\text{m}$$

$$T_L = \frac{f_p a b r^2 (\cos \theta_1 - \cos \theta_2)}{\sin \theta_1}$$

$$= \frac{0.28 \times 594 \times 10^3 \times 0.04 \times 0.15^2 \times (\cos 15^\circ - \cos 105^\circ)}{\sin 90^\circ}$$

$$= 183.5 \text{ N}\cdot\text{m}$$

$$T_{\text{total}} = T_L + T_R$$

$$= 194 + 183.5 = 377.5 \text{ N}\cdot\text{m}$$

16-7 Find the angles θ_1 , θ_2 and θ_a

$$\theta_1 = 20 - \tan^{-1} \left(\frac{3}{12} \right) = 6^\circ$$

$$\theta_2 = 180 - 30 - \tan^{-1} \left(\frac{3}{12} \right) = 136^\circ$$

$$\theta_a = 90^\circ$$

$$a = \sqrt{3^2 + 12^2} = 12.37 \text{ in}$$

$$M_f = \frac{f p_a b r}{\sin \theta_a} \int_{\theta_1}^{\theta_2} \sin \theta (r - a \cos \theta) d\theta$$

$$M_f = \frac{(0.3)(150)(2)(10)}{\sin 90} \int_6^{136} \sin \theta (10 - 12.37 \cos \theta) d\theta$$

$$= 900 \left(\frac{1}{12.37} \frac{(10 - 12.37 \cos \theta)^2}{2} \right) \Big|_6^{136}$$

$$= \frac{900}{24.74} (10 - 12.37 \cos 136) - (10 - 12.37 \cos 6) = 12799.5 \text{ lbf} \cdot \text{in}$$

$$M_N = \frac{p_a b r a}{\sin \theta_a} \int_{\theta_1}^{\theta_2} \sin^2 \theta d\theta$$

$$M_N = \frac{150 \times 2 \times 10 \times 12.37}{\sin 90} \int_6^{136} \sin^2 \theta d\theta$$

$$= 37110 \left(\frac{\theta}{2} - \frac{\sin 2\theta}{4} \right) \Big|_6^{136}$$

$$= 37110 \left(\frac{136}{2} \times \frac{\pi}{180} - \frac{\sin^2(136)}{4} \right) - \left(\frac{6}{2} \times \frac{\pi}{180} - \frac{\sin^2(6)}{4} \right)$$

$$= 37110 (1.4367 - 0.00038195)$$

$$= 53301.763 \text{ lbf} \cdot \text{in}$$

$$C_L = 12 + 12 + 4 = 28 \text{ in}$$

$$F_L = \frac{M_N - M_f}{C_L}$$

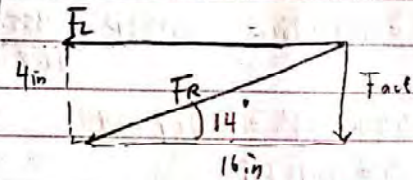
$$= \frac{53301.763 - 12799.51}{28}$$

$$= 1446.51 \text{ lbf}$$

$$T_L = \frac{f p_a b r^2 (\cos \theta_1 - \cos \theta_2)}{\sin \theta_a}$$

$$T_L = \frac{0.30 \times 150 \times 2 \times 10^2 \times (\cos 6 - \cos 136)}{\sin 90}$$

$$= 15428.81 \text{ lbf} \cdot \text{in}$$



16-7

$$\begin{aligned} F_{\text{ax}} &= F_L \cdot \sin 14^\circ \\ &= 1446.5 \sin 14^\circ \\ &= 349.74 \text{ lbf} \end{aligned}$$

$$\begin{aligned} F_R &= \frac{F_L}{\cos 14^\circ} \\ &= \frac{1446.5}{\cos 14^\circ} \\ &= 1490.78 \text{ lbf} \end{aligned}$$

$$C_r = (24 - 2 \tan 14^\circ) \times \cos 14^\circ = 22.8 \text{ in}$$

$$\begin{aligned} F_R &= \frac{M_N + M_T}{C_r} \\ &= 1490.78 = \frac{85.33 P_a + 355.345 P_a}{22.8} \end{aligned}$$

$$440.675 P_a = 33989.784$$

$$P_a = 77.13 \text{ psi}$$

$$T_L = \frac{f P_a b r^2 (\cos \theta_1 - \cos \theta_2)}{\sin \theta_a}$$

$$T_R = \frac{0.30 \times 77.13 \times 2 \times 10^2 (\cos 6^\circ - \cos 136^\circ)}{\sin 90^\circ}$$

$$= 7931.41 \text{ lbf} \cdot \text{in}$$

$$\begin{aligned} T_{\text{total}} &= T_L + T_R \\ &= 15424.8 + 7931.41 \\ &= 23356.21 \text{ lbf} \cdot \text{in} \end{aligned}$$

Average

Grade:

Name: ABDULLAH ELAHEE

This assignment is due at 8 am on Nov 16th. Complete the following problems from Shigley's Mechanical Engineering Design – 10th edition.

1. Problem 11-6
2. Problem 11-7
3. Problem 11-22
4. Problem 16-1
5. Problem 16-2
6. Problem 16-7



Homework requirement

- Print this page and use it as the cover.
- The submission should be each student's own work.
- The same day late submission will get a 10% penalty.
- Once the solution gets posted, no more submission will get accepted.
- Write legibly. No credit will be given if the work is not readable.
- Take advantage of office hours.
 - Tuesdays & Thursdays 1 PM ~ 3 PM
 - Email me: japark@pvamu.edu to request a Zoom meeting if you cannot make it to the regular office hour due to any university excused reasons.

Summary of Comments on Epson iPrint

Page: 1



Number: 1

Author: japark

Subject: Sticky Note

Date: 12/15/2020 12:55:06 PM

Graded 11-6 and 16-2. Include FBD for brake problems.

Late (-10%)

9/10

11-6) Given;

$$L_D = 8000 \text{ h}$$

$$W = 20 \text{ kN}$$

$$F_D = 20 \text{ kN}$$

$$R_D = 0.95$$

$$N = 950 \text{ rev/min}$$

$$X_D = \frac{60 L_D N_P}{L_{10}} = \frac{60 \times 8000 \times 950}{10^6} = 456$$

$$C_{10} = a_f F_D \left[\frac{X_D}{X_0 + (\Theta - X_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{1/b}} \right]^{1/a} \quad \text{--- ①}$$

$$a = \frac{10}{3} \quad \text{S-R 03 - Serie Ball Bearing}$$

$$X_0 = 0.02, (\Theta - X_0) = 4.439, b = 1.483, a_f = 1$$

Sub in eq. ①

$$C_{10} = 20 \left[\frac{456}{0.02 + 4.439 \ln \left(\frac{1}{0.95} \right)^{1/1.483}} \right]^{3/10} = 145 \text{ kN}$$

choose 03-150 mm with $C_{10} = 151 \text{ kN}$

Bearing A

$$11-7) F_R (L_R n_R 60)^{1/4} = F_D (L_D n_D 60)^{1/4}$$

$$2(3000 \times 500 \times 60)^{1/3} = F_D (L_D n_D 60)^{1/3}$$

$$F_D = \frac{228.94}{(L_D n_D)^{1/3}} \text{ kN}$$

Bearing B

$$7 \times (10^6)^{1/3} = F_D (L_D n_D 60)^{1/3}; F_D = \frac{178}{(L_D n_D)^{1/3}} \text{ kN}, \text{ Therefore Bearing}$$

A can carry larger load

$$11-22) X_D = \frac{L_D}{L_R} = \frac{10^9}{10^6} = 1000$$

ball bearing $a = 3$

$$C_{10} = a_f F_e \left[\frac{X_D}{X_0 + (0 - X_0) \ln \left(\frac{1}{R_D} \right)^{1/b}} \right]^{1/a} = 8 \times \left[\frac{1000}{0.02 + 4.439 \ln \left(\frac{1}{0.9} \right)^{1/1.483}} \right]^{1/3}$$

$$= 80.178 \text{ kN}$$

From Table 11-2

smallest bore size is 85 mm

16-1) Given:

$$d_{rim} = 300 \text{ mm}$$

$$b = 40 \text{ mm}$$

$$f = 0.28$$

$$R = 125 \text{ mm}$$

$$F = 2.2 \text{ kN}$$

$$a) M_f = \frac{f P_a b r}{\sin \Theta_a} \int_{\Theta_1}^{\Theta_2} \sin \Theta (r - a \cos \Theta) d\Theta$$

$$\Theta_1 = 0^\circ, \Theta_2 = 120^\circ, \Theta_a = 90^\circ, a = 125 \text{ mm}$$

$$M_f = \frac{0.28 \times P_a \times 0.04 \times 0.15}{\sin 90} \int_0^{120} \sin \Theta (0.15 - 0.125 \cos \Theta) d\Theta$$

$$= 2.99 \times 10^{-4} P_a \text{ N.m}$$

$$M_n = \frac{P_a b r a}{\sin \Theta_a} \int_{\Theta_1}^{\Theta_2} \sin^2 \Theta d\Theta = \frac{P_a \times 0.04 \times 0.15 \times 0.125}{\sin 90} \left[\frac{\Theta}{2} - \frac{1}{4} \sin 2\Theta \right]_0^{120}$$

$$= 9.47 \times 10^{-4} P_a \text{ N.m}$$

$$C = 2 \times 0.125 \cos 30 = 0.216 \text{ m}$$

$$F = \frac{M_n - M_f}{C} = \frac{9.49 \times 10^{-4} P_a - 2.99 \times 10^{-4} P_a}{0.216} = 0.003 P_a \text{ N}$$

$$P_a = \frac{F}{0.003} = \frac{2.2 \times 10^3}{0.003} = 734 \text{ kPa}$$

$$16-1) F = \frac{M_n + M_f}{c} = \frac{9.47 \times 10^{-4} P_a + 2.99 \times 10^{-4} P_a}{0.216}$$

$$F = 0.0057 P_a; P_a = \frac{2.2 \times 10^3}{0.0057} = 386 \text{ kPa}$$

$$P_{\max} = 734 \text{ kPa ccw}$$

$$b) T_R = \frac{f P_a b r^2 (\cos \theta_1 - \cos \theta_2)}{\sin \theta_a} = \frac{0.28 \times 734 \times 10^3 \times 0.04 \times 0.15^2 \times (\cos 0^\circ - \cos 120^\circ)}{\sin 90^\circ}$$

$$= \boxed{277.5 \text{ N}\cdot\text{m}}$$

LH shoe:

$$T_L = \frac{0.28 \times 386 \times 10^3 \times 0.04 \times 0.15^2 \times (\cos 0^\circ - \cos 120^\circ)}{\sin 90^\circ} = \boxed{145.9 \text{ N}\cdot\text{m}}$$

$$T_{\text{total}} = 277.5 + 145.9 = \boxed{423.4 \text{ N}\cdot\text{m}}$$

RH shoe:

$$F_x = 2.2 \sin 30^\circ = 1.1 \text{ kN}$$

$$F_y = 2.2 \cos 30^\circ = 1.905 \text{ kN}$$

$$R_x = \frac{P_a b r}{\sin \theta_a} (A - fB) - F_x; A = \int_{\theta_1}^{\theta_2} \sin \theta \cos \theta d\theta = \left(-\frac{1}{2} \cos 2\theta \right)_0^{25/3} = 0.375$$

$$R_y = \frac{P_a b r}{\sin \theta_a} (B + fA) - F_y; B = \int_{\theta_1}^{\theta_2} \sin^2 \theta d\theta = \left(\frac{\theta}{2} - \frac{1}{4} \sin 2\theta \right)_0^{25/3} = 1.264$$

$$R_x = \frac{734 \times 10^3 \times 0.04 \times 0.15}{\sin 90^\circ} ((0.375 - 0.28 \times 1.264) - 1.1 \times 10^3) = -1.01 \text{ kN}$$

$$R_y = \frac{734 \times 10^3 \times 0.04 \times 0.15}{\sin 90^\circ} = (1.264 \times 0.28 + 0.375) - 1.905 \times 10^3 = -4.12 \text{ kN}$$

$$R = \sqrt{1.01^2 + 4.12^2} = \boxed{4.24 \text{ kN}}$$

LH shoe,

$$F_x = 1.1 \text{ kN}; F_y = 1.904 \text{ kN}$$

$$A = 0.375, B = 1.264$$

$$R_x = \frac{386 \times 10^3 \times 0.04 \times 0.15}{\sin 90^\circ} (0.375 + 0.28 \times 1.264) - 1.1 \times 10^3 = 0.59 \text{ kN}$$

$$R_y = \frac{386 \times 10^3 \times 0.04 \times 0.15}{\sin 90^\circ} (1.264 - 0.28 \times 0.375) - 1.905 \times 10^3 = 0.78$$

$$R = \sqrt{0.59^2 + 0.78^2} = \boxed{0.98 \text{ kN}}$$

16-2) A Given:

$$d_{\text{rim}} = 300 \text{ mm}$$

$$b = 40 \text{ mm}$$

$$f = 0.28$$

$$R = 125 \text{ mm}$$

$$F = 2.2 \text{ kN}$$

$$M_f = \frac{0.28 \times P_a \times 0.04 \times 0.15}{\sin 90} \int_{15}^{105} \sin \theta (0.15 - 0.125 \cos \theta) d\theta$$

$$= 0.2185 \times 10^{-4} P_a \text{ N.m}$$

$$M_n = \frac{P_a \times 0.04 \times 0.15 \times 0.125}{\sin 90} \int_{15}^{105} \sin^2 \theta d\theta = 7.76 \times 10^{-4} P_a \text{ N.m}$$

$$C = 0.216$$

$$F = \frac{7.76 \times 10^{-4} P_a - 0.2185 \times 10^{-4} P_a}{0.216} = 0.0035 P_a \text{ N}$$

$$P_a = \frac{2.2 \times 10^3}{0.0035} = 628.6 \text{ kPa}$$

$$F = \frac{M_n + M_f}{C} = \frac{7.76 \times 10^{-4} P_a + 0.2185 \times 10^{-4} P_a}{0.216} = 0.0037 P_a$$

$$P_a = \frac{2.2 \times 10^3}{0.0037} = 594.6 \text{ kPa} ; P_{\text{max}} = 628.6 \text{ kPa CW}$$

$$T_R = \frac{0.28 \times 628.6 \times 10^3 \times 0.04 \times 0.15^2 \times (\cos 15^\circ - \cos 105^\circ)}{\sin 90} = 194 \text{ N.m}$$

$$T_L = \frac{0.28 \times 594.6 \times 10^3 \times 0.04 \times 0.15^2 \times (\cos 15^\circ - \cos 105^\circ)}{\sin 90} = 183.5 \text{ N.m}$$

$$T_{\text{total}} = 377.5 \text{ N.m}$$

16-7)

$$\theta_1 = 20^\circ - \tan^{-1}\left(\frac{3}{12}\right) = 6^\circ$$

$$\theta_2 = 180^\circ - 30^\circ - \tan^{-1}\left(\frac{3}{12}\right) = 136^\circ$$

$$\theta_3 = 90^\circ$$

$$r = \sqrt{3^2 + 12^2} = 12.37 \text{ in}$$

$$M_f = \frac{0.3 \times 150 \times 2 \times 10}{\sin 90^\circ} \int_6^{136} \sin \theta (10 - 12.37 \cos \theta) d\theta$$

$$= 12799.5 \text{ lbf.in}$$

$$M_N = \frac{150 \times 2 \times 10 \times 12.37}{\sin 90^\circ} \int_6^{136} \sin^2 \theta d\theta = 53301.76 \text{ lbf.in}$$

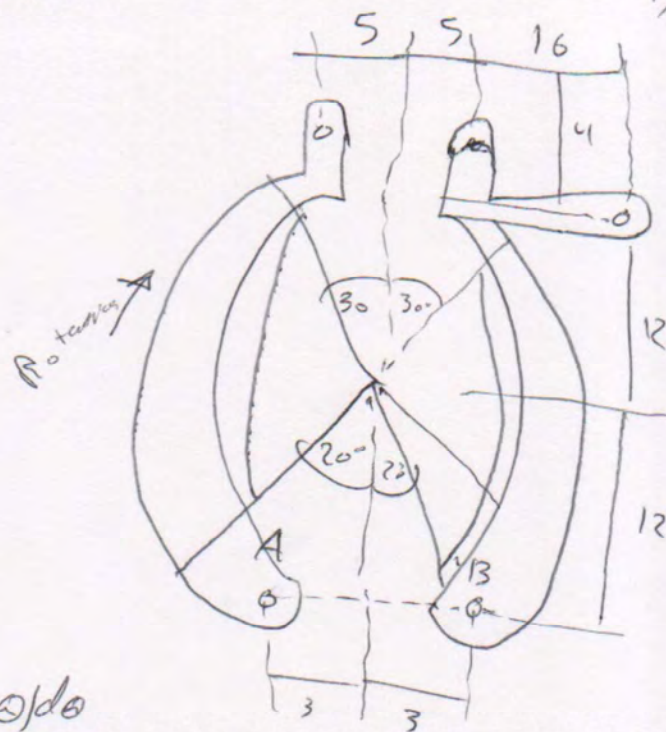
$$L = 12 + 12 + 4 = 28 \text{ in}$$

$$F_L = \frac{53301.76 + 12799.51}{28} = 1446.51 \text{ lbf}$$

$$T_L = \frac{0.3 \times 150 \times 2 \times 10^2 \times (\cos 6^\circ - \cos 136^\circ)}{\sin 90^\circ} = 15424.8 \text{ lbf.in}$$

$$M_f = (M_f)_{LH} \left(\frac{P_4}{150} \right) = (12799.5) \left(\frac{P_4}{150} \right) = 85.33 P_4$$

$$M_N = (M_N)_{LH} \left(\frac{P_4}{150} \right) = (53301.76) \left(\frac{P_4}{150} \right) = 355.345 P_4$$

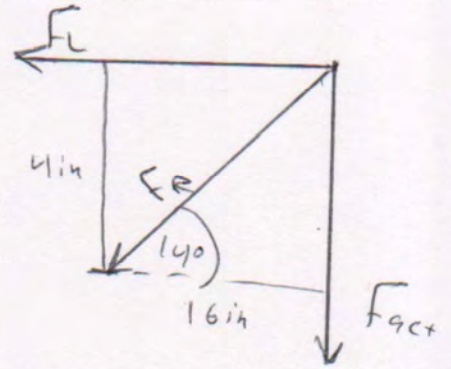


16-7)

$$F_{act} = F_L \sin 14^\circ = 1446.5 \sin 14$$

$$= \boxed{349.94 \text{ lbf}}$$

$$F_R = \frac{F_L}{\cos 14} = \frac{1446.5}{\cos 14^\circ} = 1490.78 \text{ lbf}$$



$$C_R = (24 - 2 \tan 14) \times \cos 14 = 22.8$$

$$F_R = \frac{M_n + M_f}{C_R} \Rightarrow 1490.78 = \frac{85.33 P_a + 355.34 P_a}{22.8}$$

$$P_a = 77.13 \text{ psi}$$

$$T_R = \frac{0.3 \times 77.13 \times 2 \times 10^3 (\cos 6^\circ - \cos 136^\circ)}{\sin 90^\circ} = 7931.41 \text{ lbf-in}$$

$$T_{total} = \boxed{23356.21 \text{ lbf-in}}$$

Below Average

FIVE STAR.
★★★★★

$$11-6) C_{10} = F_R = F_D \left(\frac{L_D}{L_R} \right)^{1/a} = \left(\frac{L_D n_D 60}{L_R} \right)^{1/a} F_D$$

$$\begin{aligned} F_D &= 20 \text{ kN} \\ L_D &= 8000 \text{ h} \\ n_D &= 950 \text{ rpm} \\ L_R &= 10^6 \\ a &= 10/3 \end{aligned}$$

FIVE STAR.
★★★★★

$$\begin{aligned} C_{10} &= F_D \left(\frac{L_D n_D 60}{L_R} \right)^{1/a} \\ &= 20 \text{ kN} \left(\frac{(8000 \text{ h})(950 \text{ rpm})60}{10^6 \text{ rpm}} \right)^{1/(10/3)} \end{aligned}$$

FIVE STAR.
★★★★★

$$C_{10} = 126 \text{ kN}$$

FIVE STAR.
★★★★★

Summary of Comments on 2020-11-16 07:55

Page: 1

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Graded 11-6 and 16-2 (-3). 11-22, 16-1, and 16-7 are missing -3

4/10

11-7) Bearing A:

$$a = 3$$

$$F_A = 2 \text{ kN}$$

$$L_A = 3000 \text{ h}$$

$$n_A = 500 \text{ rpm}$$

Bearing B:

$$a = 3$$

$$F_B = 7 \text{ kN}$$

$$L_B = 10^6 \text{ cycles}$$

$$C_A = F_A \left(\frac{L_A n_A 60}{L_R} \right)^{1/a}$$

$$= 2 \text{ kN} \left(\frac{(3000 \text{ h})(500 \text{ rpm}) 60}{10^6} \right)^{1/3}$$

$$C_A = 8.963 \text{ kN}$$

@ 10^6 cycles, bearing A can carry the larger load.

VI. PROJECT

MACHINE DESIGN II

FALL 2020

TITLE OF ASSIGNMENT: PROJECT

TOPICS COVERED IN PROJECT

1. Gears – General
2. Shafts and Shaft Components
3. Bearing
4. Fatigue Analysis (Variable Loading)

Gear and driveshaft design for automotive powertrain system

Presentation: 8~9:50 AM on Nov 23, 2020

Report due date: 12 PM on Nov 23, 2020, for full credit (**No extension will be granted**)

Overview

Transmission is a vital part of the ICE-powered vehicles. Gear systems in transmission delivers power from the engine to the wheels via rotary motion of a shaft or shafts. The goal of this project is to design a multi-speed gearbox based on a dyno graph by performing detailed analysis and design of the shafts (chapter 7), gears (chapters 13 and 14), and rolling element bearings (chapter 11). The values in the dyno graph are from the crank. The power transmission case study in chapter 18 serves as guidance for this design exercise. Student groups can assume that torque converter, final drive, axle shafts and their components operate 100% efficiency and have infinite life. **The coefficient of friction of tires can be assumed to be 0.5.** Gear changing mechanism is not required in this project.

Design specification and constraints

The purpose of this project is to **design a transmission that has 6 forward and 1 reverse speeds with a minimum safety factor of 2.5 for all components.**

Gear surface wear (pitting) can be ignored.

Refer to data sheet to obtain values for power, torque, vehicle weight, final drive ratio, wheel & tire size, and engine characteristics.

The vehicle needs to **climb up a 40% grade hill (starting from the specified RPM).**

The reverse gear can have the same gear ratio as the first forward gear.

Shafts need to have **infinite life** based on the **DE-Goodman failure criterion.**

Bearings are expected to have at least **20,000 hours of life.**

You need to have at least **three design iterations** to improve your shaft design.

Smaller gearbox is preferred due to overall packaging constraints.

If in doubt, please ask your Dr. Park! He WILL help you.

Machine elements to be designed and/or selected

Gear ratio, gear geometry, shafts (gear and drive shafts), other machine elements discussed in this class (e.g., bearings and keys). The design should include material decisions based on appropriate analysis. Use the following for your reference. Shaft design for stress (see Example 7-2, pg. 366), Shaft design for deflection (see Example 7-3, pg. 372), Bearing selection (see Case Study Part 7, pg. 947), Key design (see Case Study Part 8, pg. 948).

Components listed in the following should be included in your project report

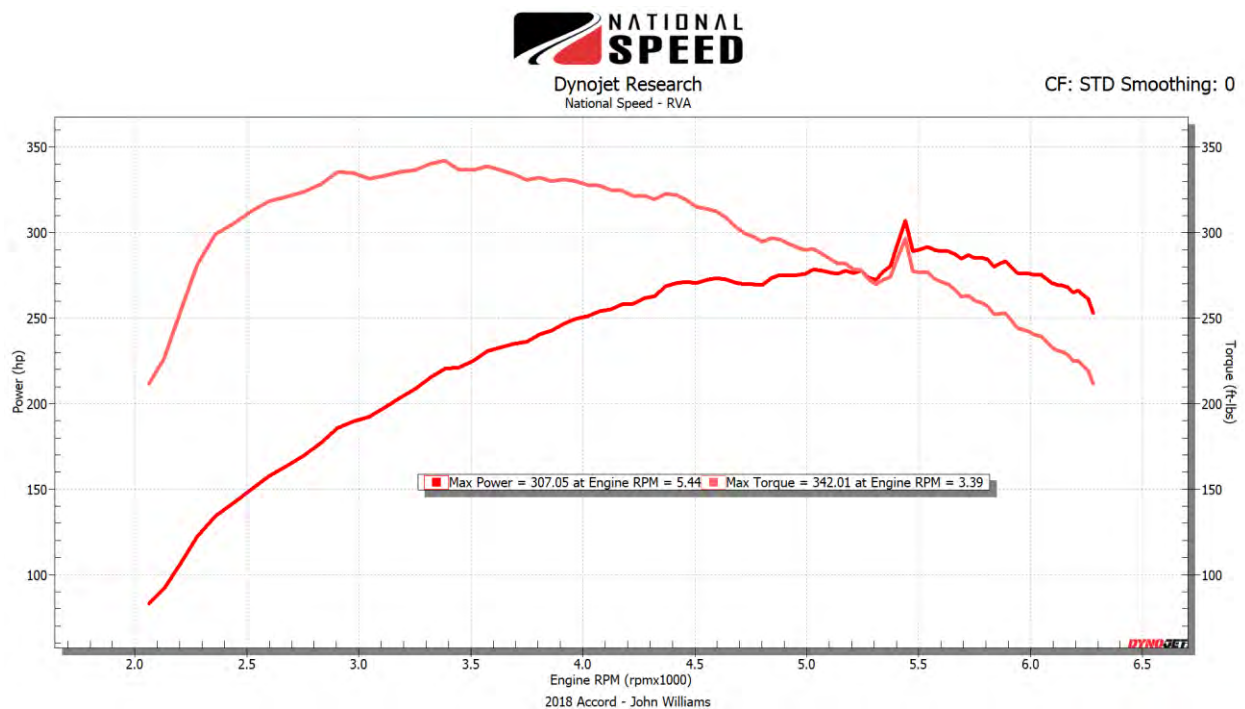
- A hand calculation of the components (e.g., gear ratio, bending & contact stress, shaft diameter) with proper description. Screenshot of handwritten equations are NOT accepted.
- Ansys stress and deflection analysis for gears and shafts. Material properties, boundary conditions, mesh, stress and displacement plots with legible legends. Maximum stress and displacement locations with values.
- 3D models of shafts, spur gears, and bearings made in NX. White background please!
- 3D assembly of your overall design & animation (the animation can be used in presentation)

- The 2D drawing of shafts gears keys and bearing with detailed dimensions made in NX. All necessary dimensions should be included that any person can redraw parts just based on your 2D drawing.
- Compare the calculated gear ratios with actual factory gear ratio values. Comment why there is such discrepancies.

Project Groups

Group 1	Group 2	Group 3	Group 4	Group 5	Group 6

Vehicle #1 (Mid-size sedan-2018 Honda Accord 2.0t)



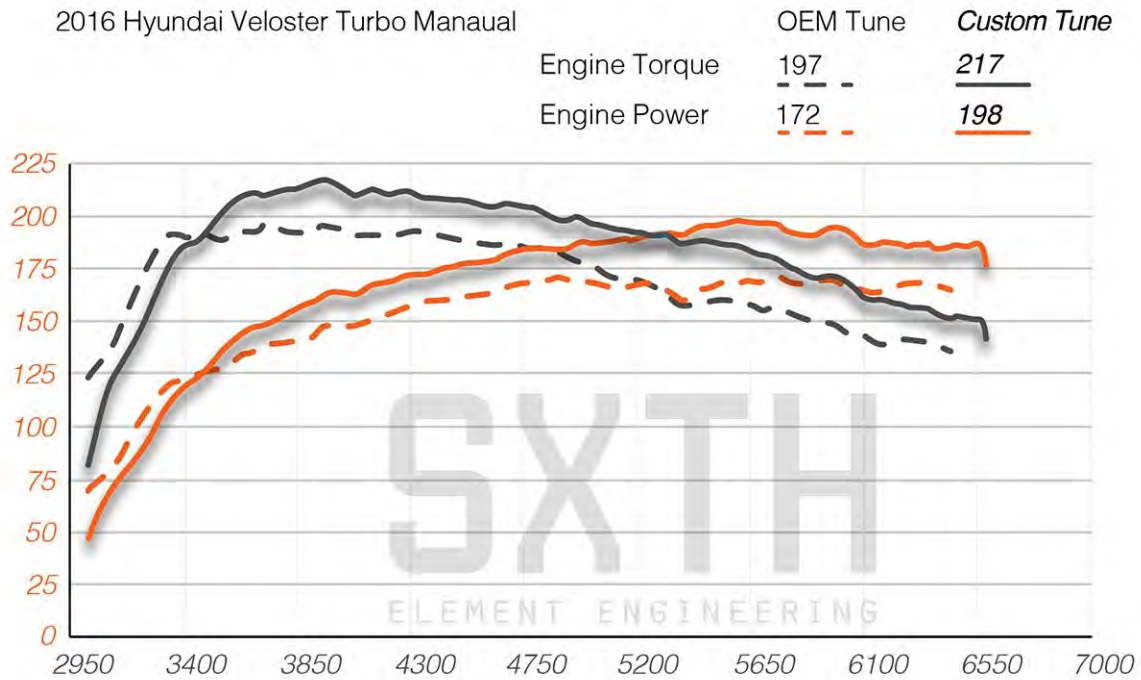
Vehicle weight: 3300 lb

Tire size: 225/50/R17

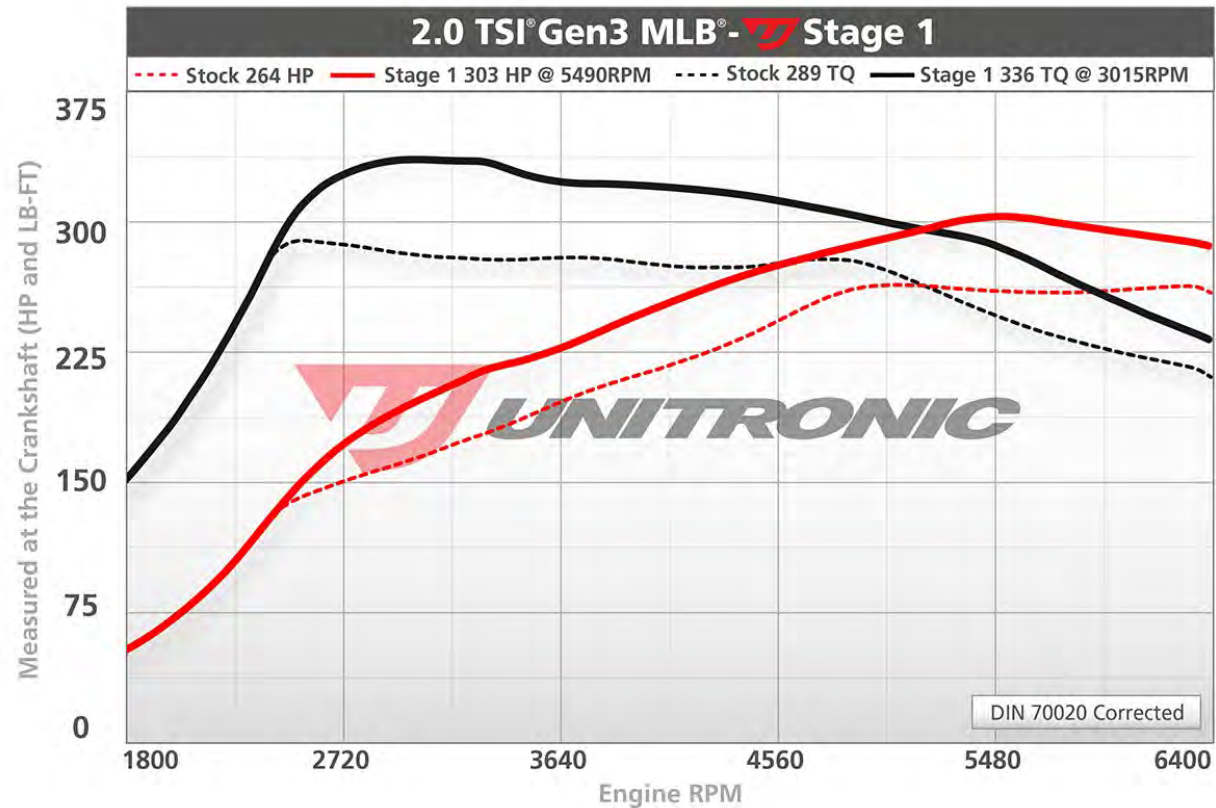
Final drive: 3.55

Constraint:

- 85 mph should take place at 2500 RPM (highest gear)
- Initial effort to climb 40% grade hill is made at 2500 RPM (lowest gear)

Vehicle #2 (Compact sports hatch- 2016 Hyundai Veloster turbo)

Vehicle #3 (Compact sedan - 2017 Audi A4)



Use stock values for power and torque (dashed line)

Vehicle weight: 3550 lb

Tire size: 225/50/R17

Final drive: 4.27

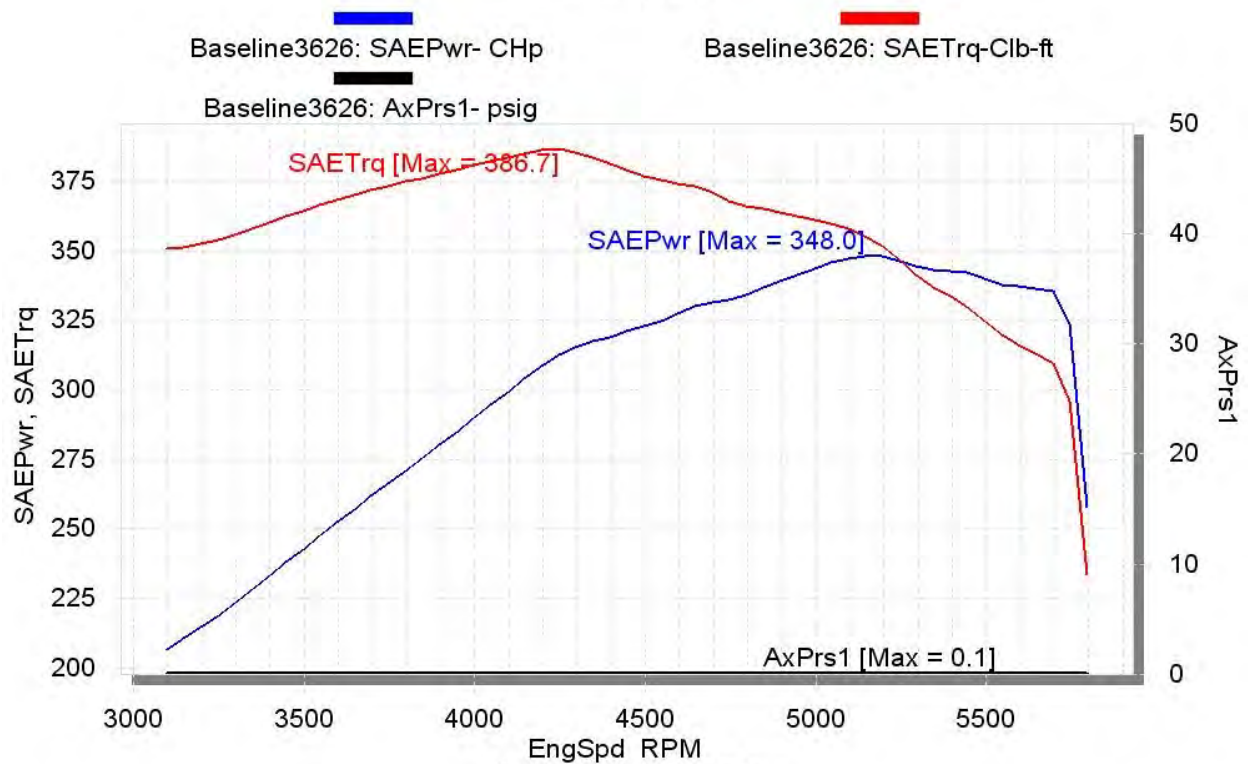
Constraint:

- 65 mph should take place at 2000 RPM (highest gear)
- Initial effort to climb 40% grade hill is made at 2000 RPM (lowest gear)

Vehicle #4 (Pickup truck – 2016 Ram 1500)

Arrington Performance

Baseline3626,



06/29/16

Arrington Performance
SuperFlow WinDyn™ V2.8

10:00:04

Vehicle weight: 5500 lb

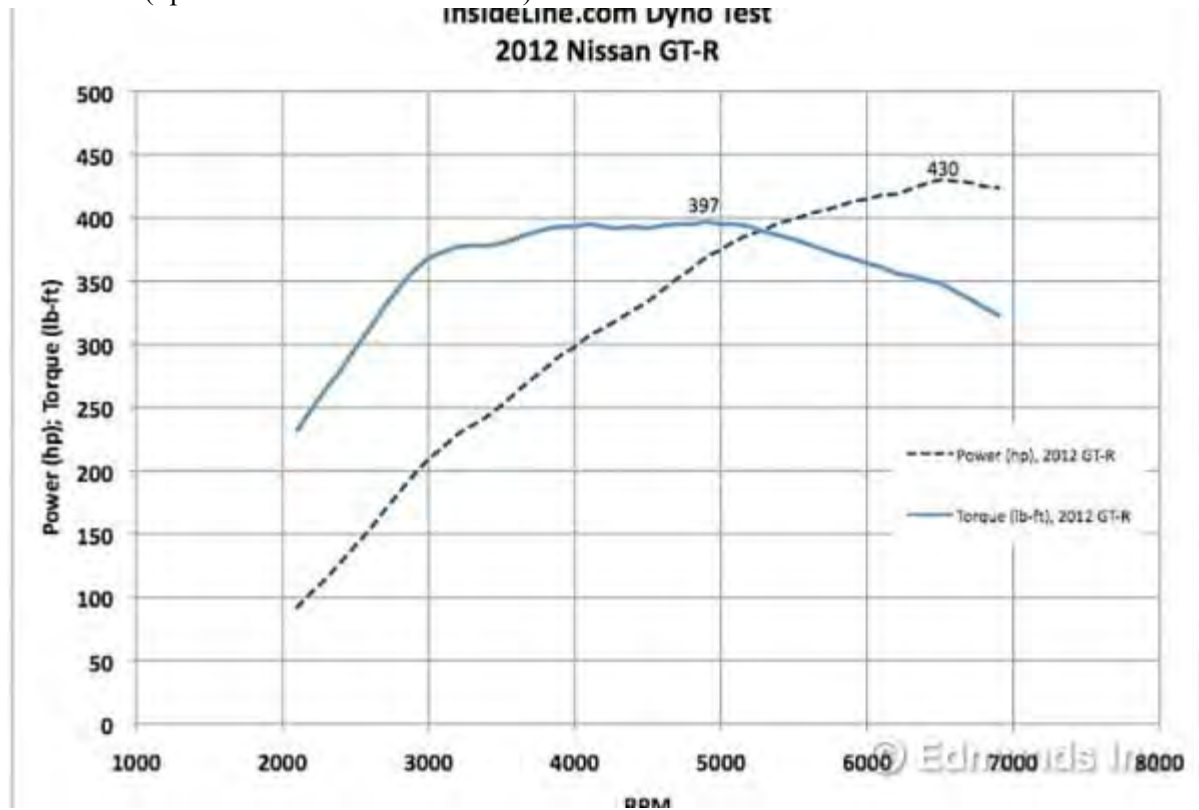
Tire size: 275/60/R20

Final drive: 3.21

Constraint:

- 90 mph should take place at 3200 RPM (highest gear)
- Initial effort to climb 40% grade hill is made at 3000 RPM (lowest gear)

Vehicle #5 (Sports car – 2012 Nissan GT-R)



Vehicle weight: 3800 lb

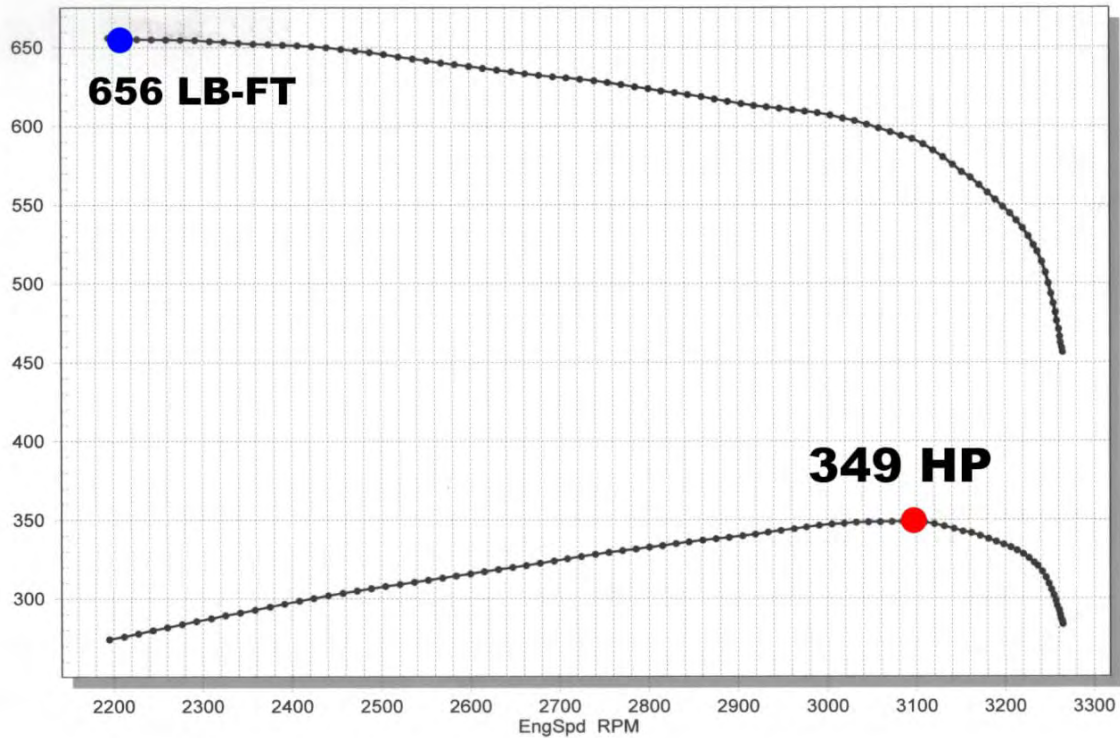
Tire size: 285/35/R20

Final drive: 3.70

Constraint:

- 90 mph should take place at 2500 RPM (highest gear)
- Initial effort to climb 40% grade hill is made at 2100 RPM (lowest gear)

Vehicle #6 (Heavy duty pickup truck – 2011 Silverado 2500 duramax)



Vehicle weight: 7400 lbs

Tire size: 265-70/R18

Final drive: 3.73

Constraint:

- 90 mph should take place at 2700 RPM (highest gear)
- Initial effort to climb 40% grade hill is made at 2200 RPM (lowest gear)

Due 9/30:

Hand calculation of ratio for the highest gear. Must consider all parameters provided (engine speed, final drive, tire size, etc.). Write a computer code that correlates the engine speed and ground speed based on your preliminary hand calculation. MATLAB, Python, or C++ are accepted. Both hand calculations and codes need to be submitted.

Due 10/12:

Hand calculation of ratio for the lowest gear. Must consider all driving conditions provided (hill grade, vehicle weight, tire size, etc.). The document you submit must have all free body diagrams. Write a computer code that correlates the engine speed and the torque or the transmitted force on the drive shaft (propeller shaft) – that connects the transmission and the differential (final drive) based on your hand calculation. Both hand calculations and codes need to be submitted.

Due 10/28:

Size analysis on all the gears (forward and reverse) based on your assumption on the diametral pitch (English unit) or module (SI unit). Number of teeth, pitch radii should be calculated. Need to consider all design constraints such as interference, etc. Write a computer code that provides diameter and radii upon necessary inputs (e.g., gear ratio, diametral pitch). Both hand calculations and codes need to be submitted.

Force analysis on all gears (bending and contact). Based on your force values, choose the face width along with the materials for all gears. All gears are preferred to be in the same materials. Hand calculation needs to be submitted.

Due 11/11 (Updated)

Propeller shaft design. The purpose is to find the diameter of the shaft.

HINT: Assume a material for the shaft. Consider the most severe condition of the vehicle – when the engine is making its peak power. If it is the case, you have the load coming from the first gear (transmission) and you have a load coming from the final gear (differential). Do not forget to use the designated fatigue criterion. Use safety factor of 1.5 if necessary

Choose the length of the propeller shaft such that it is reasonable for your vehicle. Keep the source so that you can refer to it in the report.

Bearing catalog load rating (C_{10}) should be computed. Appropriate key needs to be designed as well.

Use the final two weeks to work on your report, CAD, and presentations. No photo of your hand calculation is allowed in your report. Properly write it in word or neatly take screenshots from any source (needs citation).

I will NOT answer to your project **calculation related questions** after 11/11. Seek for help before it gets too late.

Above Average

Prairie View A&M University

Mechanical Engineering



MCEG 4043 – Z0X

Machine Design II

Fall 2020

FINAL REPORT

Transmission Design

Submitted by

Amna Jangda	<i>Amna Jangda</i>
Ximena Cifuentes	<i>Ximena Cifuentes</i>
Jeremy Hunter	<i>Jeremy Hunter</i>
David Trevino	<i>David Trevino</i>
Abdullah EL Aker	<i>Abdullah EL Aker</i>

to

Dr. Jaejong Park

By signing above, we declare that the contents within this report were generated solely by us. We understand the consequences of academic misconduct.

Letter of Transmittal

November 23, 2020

Dr. Jaejong Park
Prairie View A&M University

Dear Dr. Park:

Enclosed within this document is our report entitled Gear and Driveshaft Design for Automotive Powertrain System. This report outlines the specific design and analysis conditions, machine design elements used to base the designs and representation of the analytical procedures used. Throughout this project, many iterations were needed to make sure all calculations were precise and accurate showing us that when engineering, the best solutions are not evident the first time.

Should you have any questions or concerns, please feel free to contact Aamna Jangda via e-mail at ajangda@student.pvamu.edu. We greatly appreciate your time in reviewing this report.

Sincerely,

Aamna Jangda
Ximena Cifuentes
Jeremy Hunter
David Trevino
Abdullah EL Aker

Enclosed
Final Report

ABSTRACT



The purpose of the “Gear and Driveshaft Design for Automotive Powertrain System is to design a multi-speed gearbox based on a dyno graph by performing detailed analysis and design of the shafts, gears, and rolling element bearings. The project first needed for students to pick a car design which was illustrated in the project description, in our case, it was a mid-size sedan-2018 honda accord. This specific design included values and analysis conditions which include the vehicle weight, tire size, final drive, velocity of the highest gear and the rpm of highest and lowest gear. Additionally, a graph was provided that compares the engine rpm vs the power output of the car. The calculations were first completed by hand and then translated to MATLAB in order to provide accurate results and allow for multiple iterations if calculations needed to be changed. Fusion 360 was used to design and 3D model the vital components which included the gearbox and driveshaft. Additionally, the simulation software ANSYS was utilized to analyze the stress and deflection of the gears and shafts.

Using the information provided in the project description, multiple iterations had to be made due feedback provided on initial calculations. With the use of MATLAB, the values simply had to be changed in the coding in order to provide new and accurate values. Changing one value would drastically change the outcome that was desired so it took a few attempts to get it right. Once the desired values were obtained, the group was able to begin designing all the machine elements in the project.

In conclusion, to achieve the desired design considering the given constraints and safety requirements a multiple set of calculations had to be made. The highest gear ratio was calculated to be 0.684 and lowest gear ratio was 2.661. Frictional forces were taken into account. Gear Analysis yielded fairly reasonable values for the gear teeth, face width, etc.. Finally to determine the propeller shaft diameter and its components we assumed the length to be 35 in. All hand calculations were followed by MATLAB calculations tests to verify the accuracy of the final values.

Summary of Comments on group1_2410_843293_Machine Design II Final Report - Group 1.pdf

Page: 3



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Author: japark

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Chapter 1: Introduction

This chapter discusses the background of the project. In the next few sections, the problem statement will be clearly identified along with the strategy used to approach it. In addition, the last section includes the different tasks within the project along with a plan that ensures completion of each task in a timely manner.

1.1 Background

When it comes to traveling by land, nothing comes closer than a vehicle. Within a vehicle, we have a transmission that transfers the power created by the engine to the wheels via a drive shaft and axles. Within this transmission, different gears are utilized and allow for different levels of torque to be applied to the wheels depending on the speed of the vehicle. In order to increase or decrease the torque, the gears in the transmission need to be shifted in either manual or automatic. Before we had automatic cars, only manual cars existed. Manual vehicles were first invented in 1894 by French inventors Louis-Rene Panhard and Emile Lavassor. They developed and demonstrated the world's first three-speed transmission and slowly improved upon their design. By the beginning of the 20th century, car manufacturing became common and utilized the non-synchronized manual transmission based on the design of Panhard and Lavassor. However, the next major innovation happened in 1928 by Cadillac when they introduced the very first synchronized manual transmission which reduced the gear grinding and made it easier to shift between different levels of torque. The very first automatic transmission was invented in 1904, however did not take off until the invention of the Buick Dynaflo transmission in 1948.



1.2 Problem Definition

The scope of the project is to design a gearbox that provides different speeds based on a dyno graph. Successful completion of the design required the team to assume that torque converter, final drive, axle shafts and their components operate at a 100% efficiency and contain infinite life. Finally, the coefficient of friction of tires was assumed to be 0.5.

1.3 Strategy

It is common to transmit power from an engine or motor to an output action through gear reducers which provides an efficient means for the transmission of power through the rotary motion of a shaft or shafts supported by bearings. The goal of this project is to design a 6 gear gearbox with an idler gear such as that shown below by performing a detailed analysis and design of the shafts (Chapter 7), gears (Chapters 13 and 14) and rolling element bearings (Chapter 11) of this relatively simple gear train. The design process described in the power transmission case study in Chapter 18 serves as guidance for this design exercise.

Page: 9

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Reference?			

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Interesting information in the beginning of the report.			

1.4 Project Planning

This project was split into seven phases. In the first phase, the hand calculation of ratio for the highest gear was created by considering all parameters provided in the problem statement such as the speed, final drive, tire size and diameter. Similar to the last phase, in the second phase, the hand calculation of ratio for the lowest gear was created by considering the parameters such as hill grade, vehicle weight, tire size, coefficient of friction, final drive and initial constraints. Utilizing the values from the first two phases, in phase three, we began the gear analysis of our 6-gear gearbox. The free body diagrams were generated for the 6 gears and the number of teeth for each gear was calculated using values gathered from the lowest and highest gear ratio. From here, the bending stress, dynamic factor, transmitted load, diametral pitch and Lewis bending factor were calculated in order to find the face width of the gears. In phase four, the propellor shaft was designed and values were calculated by assuming the length of the shaft and finding the diameter utilizing DE-Goodman equation. Once all the values were gathered from all the previous phases, modeling was started utilizing Fusion 360 software. Within Fusion, the gears were generated as well as the key, shaft, and ball bearings in phase five. In the sixth phase ANSYS software was used to do stress and deflection analysis for the gears and shafts by setting material properties, boundary conditions, mesh, stress, and displacement plots. Finally, in the seventh phase, the final report was created by combining all previous phases

Chapter 2: Executive Summary

This chapter summarizes the design specifications of the gearbox, propeller shaft, bearings, and key components of the transmission design. In addition, the chapter contains all the formulas needed to draw the appropriate measurement conclusions along with the computational values that were assumed or given by the problem statement. Finally, the calculated specifications can be observed in Table 2.1-2.4.

2.1 Gearbox

In this section the gearbox specifications are displayed on Table 2.1. The table contains the number of teeth, diameter, face width, and maximum bending stress for each gear designed. In addition, from the table it can be concluded that the material and safety factor for all gears is constant. Moreover, this section includes the formulas used to calculate the values displayed on Table 2.1.

Table 2.1 Summary of Final Gear Design Specifications

Gear number		N	d	F	Max $\sigma_{bending}$	Material	Safety factor
Pinion		22	2.75 in	2.3230 in	37245 psi	AISI 1095 HR	2.5
Forward gears	1	59	7.37 in	2.3227 in	37245 psi	AISI 1095 HR	2.5
	2	50	6.25 in	2.3227 in	37245 psi	AISI 1095 HR	2.5
	3	41	5.12 in	2.3228 in	37245 psi	AISI 1095 HR	2.5
	4	32	4 in	2.3229 in	37245 psi	AISI 1095 HR	2.5
	5	23	2.8 in	2.3230 in	37245 psi	AISI 1095 HR	2.5
Reverse gear		22	2.75 in	2.3230 in	37245 psi	AISI 1095 HR	2.5

To calculate the number of teeth the following equation was used:

$$N_1 = m (N_p)$$

where,

m is the module

N_p is the number of teeth of the pinion

N_1 is the number of teeth to calculate

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Same face width led to the same safety factors? I will need to see the calculation for that!

 Number: 2 Author: japark Subject: Sticky Note Date: 12/2/2020 10:53:22 AM
module is irrelevant as we are working with US customary unit system.

 Number: 3 Author: japark Subject: Highlight Date: 12/2/2020 10:52:53 AM

To calculate the diameter of the gear the following equation was used:

$$d = N/P$$

Where,

N is the number of teeth

P is the pitch diameter

To calculate the face width the following formula was used:

$$F = (K_v \times W_t \times P) / (\sigma \times Y)$$

where,

K_v is dynamic factor

W_t is the transmitted load

σ is the bending stress

Y is the Lewis form factor

To calculate the maximum bending stress the following formula was used:

$$\sigma_{max} = \sqrt{(M_{xy}^2 + M_{xz}^2)}$$

Where M is the maximum bending stress occurring at the critical point at the surface of x-y and x-z

2.2 Propellor Shaft

In this section the propeller shaft crucial data is shown in Table 2.2, it includes the shaft diameter, maximum torque applied, maximum shear stress, failure criteria, material used, and safety factor. Across all the components used for this design it has a constant safety factor with the same material. Table 2.2 is followed by the equations used to calculate each variable.

Table 2.2 Summary of Final Shaft Design Specifications

Shaft diameter	Max torque	Material	Max $\tau_{torsion}$	Failure criteria	Safety factor
1.0454 in	7291.3 ft.lb	AISI 1095 HR Steel	34585 lbf.in	24.9744 kpsi	2.5

Maximum torque

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This value seems to be absurdly high. the ratio of first gear was less than 3. This means the torque will get amplified by less than 3 times in the propeller shaft. Accord engine is making less than 300 ftlb of torque at the crank.

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$$T_{max} = W_t \times (D/2)$$

where D is the diameter

Maximum torsion was calculated using the highest force value times the distance

Failure criteria

$$S_e = K_a \times K_b \times K_c \times K_d \times S_e'$$

where K_a is the surface factor

K_b is the size factor

K_c is the loading factor

K_d is the temperature modification factor

S_e' is the endurance limit

2.3 Shaft Components

In this section a summary of the two major components calculated values for the shaft key and bearing is represented in Table 2.3 and 2.4. After determining the shaft critical values with the maximum bending stress we were able to pick the key size of the same material and consider a fixed safety factor. Using the standard size of the key, we were able to calculate the load (force and torque) applied on the key. Afterwards a similar approach was used to determine the bearing load.

2.3.1 Key

Table 2.3 displays the key specifications based on the calculated shaft diameter. Some of the items listed on the table include but are not limited to the key sizing, the loading, and material specifications.

² Table 2.3 Summary of Final Key Design specifications

Shaft diameter	Key size			Loads		Material	Safety factor
	w	h	l	Force	Torque		
1 to 1 ¼ in	¼ in	3/16 in	3/32 in	2845.56 lb	3557.3 lb.in	AISI 1095 HR Steel	2.5



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Table could have been made to look better.



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The shaft key standard dimensions were found from the table.

To calculate the force

$$F = T/r$$

where T is the torque

r is the radius

Torque

$$T = (63025 \times \text{HP}) / n$$

where HP is the horsepower

n is the rotational speed

2.3.2 Bearing

Table 2.4 displays the bearing specifications. The table contains the bearing, radial load, axial load, catalog load rating, catalog, and the specific item number.

Table 2.4 Summary of Final Bearing Design Specifications

Bearing	Radial load	Axial load	Catalog load rating	Catalog	Item number
½ in	342.01 lbf	0	4932 lbf	62mm OD	26.553

The bearing standard dimensions were found from the table. On the other hand, force was given

in the dyno graph. To calculate the catalog load rating C_{10}

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

Where F_D is the desired radial load

L_D is the desired life, hours

n_D is the desired speed

L_r is the rating life in hours

n_R is the rating speed



Chapter 3: Design Methodology

3.1 Gear Design Methodology

When designing and analyzing the gears, various equations and steps were necessary to successfully obtain the design elements to complete analysis. The first step taken was to create the free body diagrams for each gear. When creating the free body diagrams the group was able to see the forces and loads that would be acting on each gear. Additionally, the group was able to analyze the direction of rotation that the gears would be moving in. Considering all driving conditions that were given in the dyno graph such as the hill grade, vehicle weight, tire size, coefficient of friction, final drive and initial constraints, the gear ratio of the highest and lowest gear were calculated. Starting off with the gear ratio of the highest gear, it was calculated to be 0.6384 by having the highest gear rpm be divided by the transmission shaft rpm. Utilizing values from the problem statement, the gear ratio for the lowest gear was calculated to be 2.661. Using values obtained from the lowest and highest gear ratio, the gear ratio for all six gears were generated. Using this information, the number of teeth were found by using the initial rpm provided and the gear ratios that were calculated. Once the number of teeth for each gear was found, the diametral pitch, speed, and pitch line velocities were found for each gear. Once the pitch line velocities were obtained, the group was able to calculate the transmitted, axial, and radial loads using the power input given as well as the torque for each gear. Finally, the face width was calculated using the values such as the bending stress, dynamic factor, transmitted load, diametral pitch and Lewis bending factor.

Table 3.1 Gear Design Equations

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	Equation
Compound Reverted Gear Train Ratio	$N_2 + N_3 = N_4 + 2N_5 + N_6 ; N_4 = N_5$
Tangential Pressure Angle	$\Phi_t = \tan^{-1} \frac{\tan \Phi_n}{\cos \Psi}$
Helical Interference	$N_p = \frac{2k \cos \Psi}{(1+2m) \sin^2 \Phi_t} [m + \sqrt{m^2 + (1+2m) \sin^2 \Phi_t}]$
Pitch Diameter	$d = \frac{N}{P}$
Speed	$n_{now} = n_{previous} \frac{N_{previous}}{N_{now}}$
Pitch Line Velocity	$v = \frac{\pi d n}{12}$
Transmitted Load	$W_t = 33000 \frac{H}{V}$
Radial Load	$W_r = W_t \tan \Phi_t$
Axial Load	$W_a = W_t \tan \Psi$
Torque	$T = \frac{d}{2} W_t$



3.1.1 Gear Ratios

Using values given in the project statement, the highest and lowest gear ratios were calculated for a mid-size sedan-2018 Honda Accord. Additionally, a dyno graph was provided to the group to provide additional information.

Vehicle #1 (Mid-size sedan-2018 Honda Accord 2.0t)



Figure 3.1 Dyno graph for Honda Accord



You have lots of parameters here. You should include a paragraph that explains every single one of them unless you have defined it earlier in the report.

To find the highest gear ratio, the group considered all the following parameters

$$d = \text{Diameter} = 25.9 \text{ in}$$

$$N_{in} = \text{Rotational speed} = 2500 \text{ rpm}$$

$$V_x = \text{Expected speed} = 85 \text{ mph}$$

$$f.d = \text{Final drive ratio} = 3.55$$

Where

$$\text{Distance moved in 1 revolution} = \pi d = \pi(25.9) = 81.36 \text{ in/rev}$$

$$\text{Mph converted to rpm} = \frac{85 \text{ miles}}{1 \text{ hour}} * \frac{5280 \text{ ft}}{1 \text{ mile}} * \frac{12 \text{ in}}{1 \text{ ft}} * \frac{1 \text{ hr}}{60 \text{ min}} = 89,760 \text{ in/min}$$

$$\frac{\text{Distance}}{\text{Rpm}} = \frac{89,760 \text{ in/min}}{81.36 \text{ in/rev}} = 1103.11 \text{ rpm}$$

In order to find the rpm of the transmission shaft we used the equation

*Tire rpm * final drive ratio* where $1103.11 \text{ rpm} * 3.55 = 3,916.04 \text{ rpm}$. Using this value, we are able to calculate for the highest gear ratio using the equation

$$\text{Gear Ratio(Highest)} = \frac{N_{in}}{N_{out}} = \frac{2500 \text{ rpm}}{3,916.04 \text{ rpm}} = 0.6384$$

To find the lowest gear ratio, the group considered all the following parameters

$$\mu = \text{Coefficient of friction} = 0.5$$

$$d = \text{Diameter} = 25.9 \text{ in}$$

$$f.d = \text{Final drive ratio} = 3.55$$

$$\theta = \text{Grade hill incline} = 40$$

Where



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$$\text{Pressure Angle} = \tan^{-1}(\theta) = \tan^{-1}(0.40) = 21.8^\circ$$

$$\text{Force of friction} = \mu (mg \cos(\theta)) = (0.5)(3300 \text{ lb} \cos(21.8^\circ)) = 1531.991 \text{ lbf}$$

$$\text{Force in x direction} = F_x = F_f + mg \sin(\theta) = 1531.911 + 3300 \sin(21.8^\circ) = 2757.558$$

$$\text{Transmitted Load} = W_t = F_x = 2757.558$$

$$\text{Tire radius} = \frac{d}{2} = \frac{25.9}{2} = 12.95$$

$$\text{Applied torque} = T = r \cdot W_t = 12.95 \text{ in}(2757.558) = (35710.38 \text{ lb} \cdot \text{in}) \left(\frac{1 \text{ ft}}{12} \right) = 2975.865 \text{ lb} \cdot \text{ft}$$

$$\text{Shaft transmission} = \frac{T}{F_{\text{final Drive}}} = \frac{2975.865 \text{ lb} \cdot \text{ft}}{3.55} = 838.272 \text{ lb} \cdot \text{ft}$$

$$\text{Gear Ratio(Lowest)} = \frac{N_{in}}{N_{out}} = \frac{838.272 \text{ lb} \cdot \text{ft}}{315 \text{ lb} \cdot \text{ft}} = 2.661$$



3.1.2 Gear Size and Force Analysis

The force analysis for each gear was generated by hand and analysis on gear size was performed by using values gathered from the lowest and highest gear ratio. This analysis can be shown in chapter 3.1.2.4 gear sizes in the report.

Page: 18

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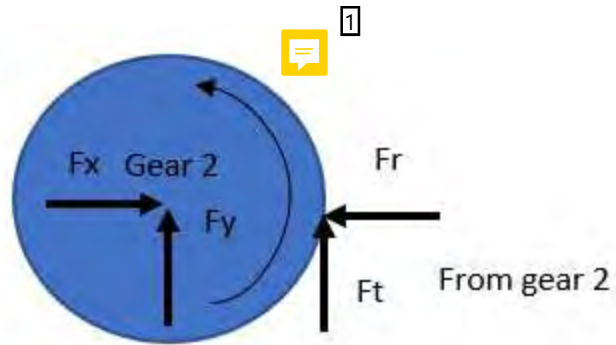


Figure 3.1.1 Gear 1 Force Analysis

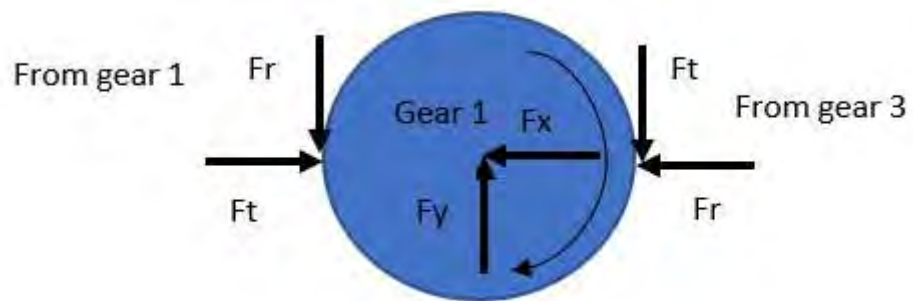


Figure 3.1.2 Gear 2 Force Analysis

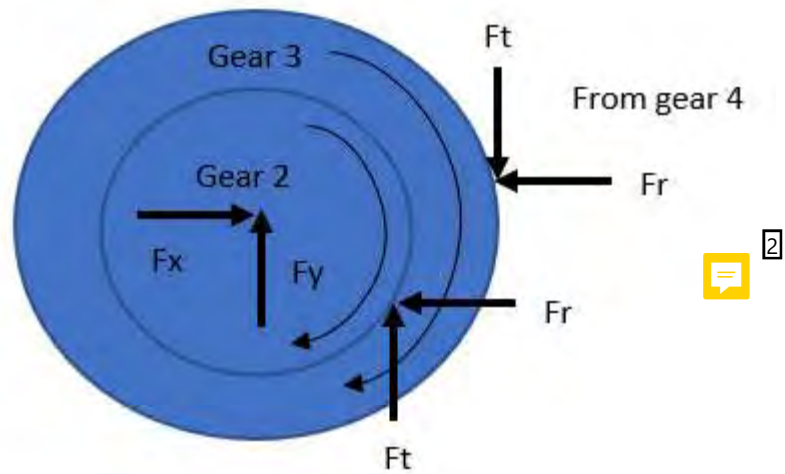


Figure 3.1.3 Gear 3 Force Analysis



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Curved arrow, does this represent torque or rotation direction?



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Figure caption and content are not consistent. Figure 3.1.3 says gear 3 but the figure itself says gear 4?
Also, why is this gear compound?



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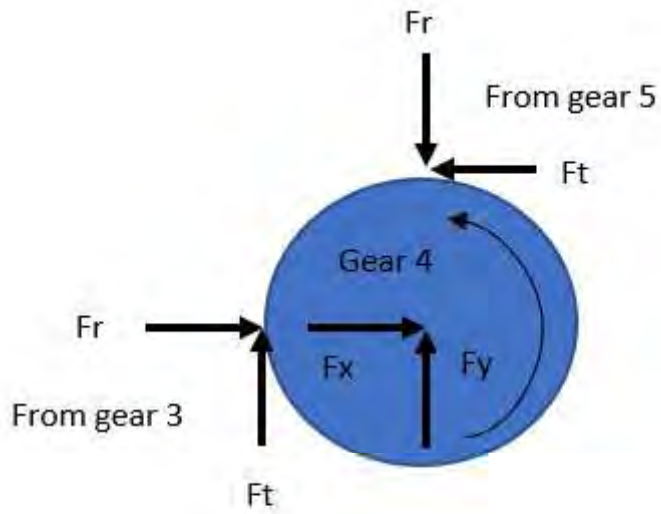


Figure 3.1.4 Gear 4 Force Analysis

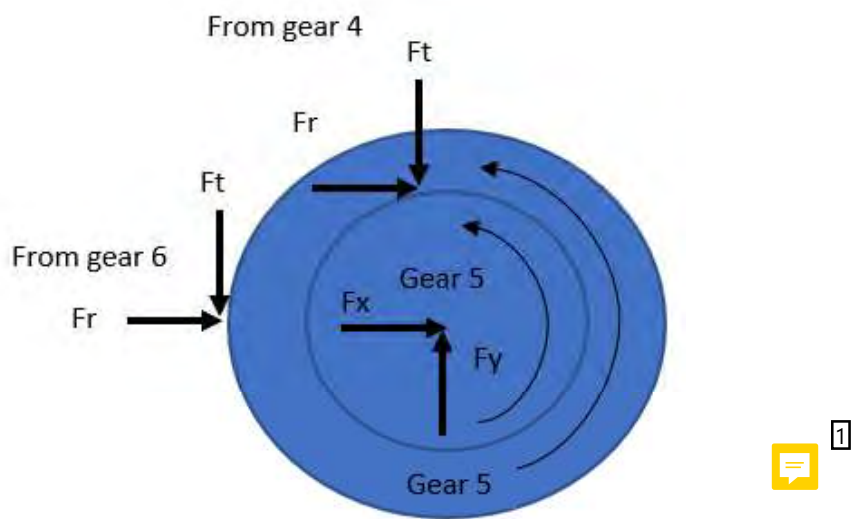


Figure 3.1.5 Gear 5 Force Analysis

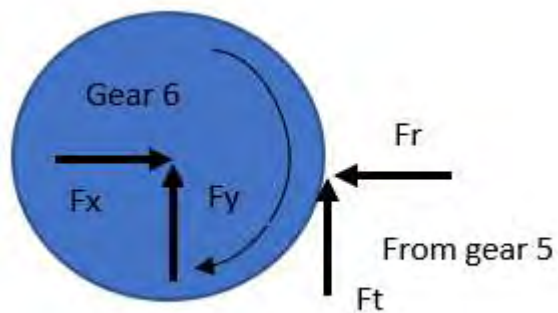


Figure 3.1.6 Gear 6 Force Analysis

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Another compound gear?

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3.1.2.1 Interference Check

If $m < 1$, the given equation for interference cannot be used, instead the N_p is derived from a table.



$$N_p = \frac{2k \cos(\phi)}{(1+2m) \sin^2(\theta)} \left[m + \sqrt{m^2 + (1+2m) \sin^2(\theta)} \right] \quad \text{[2]}$$

Minimum N_p	Max N_G	Integer Max N_G	Max Gear Ratio $m_G = N_G/N_p$
13	16.45	16	1.23
14	26.12	26	1.86
15	45.49	45	3
16	101.07	101	6.31
17	1309.86	1309	77

[4] Figure 3.1.7: Image of Minimum N_p Table [3]

As Figure 3.1.7 shows, 14 was used as the minimum N_p , since the max gear ratio is closest to 1.86. Spur gears were used since the interference equation was not used for the N_p .

3.1.2.2 First Iteration – Face Width

Once all of the equations for the gear analyses were found, it was necessary to calculate both the bending stress for each shaft. In order to calculate the bending stress, the Lewis Form Factor and the dynamic factor were needed. The Lewis Form Factor was found on Table 14-2 based on the number of teeth of the gear per shaft. Using equation (14-7) from page 371, we can manipulate and rearrange the equation in order to find the face width.

Where

$$\text{Bending stress} = \sigma = 165000 \quad \text{[5]}$$

$$\text{Dynamic factor} = K_v = 4352.6085$$

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$$\text{Transmitted load} = W^t = 1.943 \quad \text{[1] [2]}$$

$$\text{Diametral pitch} = P = 6$$

$$\text{Face width} = F = 2.3230 \text{ in}$$

$$\text{Lewis bending factor} = Y = 0.311 @ 22 \text{ Teeth}$$

$$F = \frac{K_{vs} W^t}{\sigma Y} = \frac{(1.943)(4352.61)(6)}{165000(0.331)} = 2.520 \text{ in} \quad \text{[3] [4]}$$

3.1.2.3 Subsequent Iterations – Face Width

For this section, a few different iterations using different materials and altering diametral pitch was performed in MATLAB. The best one among the 3 interactions performed was chosen. The determining factor was the iteration that resulted in a small, realistic diameter and a small face width, either equal to or less than the gear diameter.

- Assumptions made for Iteration 1:
 - P=8, diametral pitch (teeth/in)
 - AISI 1060 HR Steel; Sy=54kpsi, Sut=98 kpsi

Iteration 1 resulted in smaller diameters across all gears, but the face width was too high.

- Assumptions made for Iteration 2:
 - P=4, diametral pitch (teeth/in)
 - AISI 4130 Q&T Steel; Sy=212 kpsi, Sut=236 kpsi

Iteration 2 resulted in small face widths, but the diameters were resulting in one gear being more than 1.5 ft, which was too large.

- Assumptions made for Iteration 3:
 - P=8, diamteral pitch (teeth/in)
 - AISI 1095 HR Steel; Sy=66 kpsi, Sut=120 kpsi

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units are missing.				
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description as to why you need subsequent iteration is necessary would have been nice here.				
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should this be iteration 3?				
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Iteration 3 was the chosen one because it yielded in realistic, small diameters and face widths.

3.1.2.4 Gear sizes

Where

$$\text{Lowest gear ratio} = 2.661$$

$$\text{Highest gear ratio} = 0.6384$$

$$\text{Safety factor} = 2.5$$

$$\text{Power} = 307.5$$

$$\text{Revolution per minute} = 5.44 \text{ rpm}$$

$$\text{AISI 1095 HR}$$

$$\text{Lowest gear ratio} - \text{Highest gear ratio} = 2.661 - 0.6384 = 2.0226$$

Utilizing the value obtained, we can calculate the average gear ratio when we use the formula

$$\frac{2.026}{5} = 0.40452$$

We can add this value to the lowest gear ratio to get the ratio for gear 2 and the rest of the gears.
Where

$$m_6 = 0.6384$$

$$m_5 = 0.6384 + 0.40452 = 1.04292$$

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How about material? Is the 1095 HR steel reasonable for the type of application?				
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What is this and why and how are you using it? I can see it in the calculation but have description explaining it. Just like I said, this report is targeting a person who does not have solid ME knowledge.				
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$$m_4 = 1.04292 + 0.40452 = 1.44744$$

$$m_3 = 1.44744 + 0.40452 = 1.85196$$

$$m_2 = 1.85196 + 0.40452 = 2.25648$$

$$m_1 = 2.25648 + 0.40452 = 2.661$$



Table 3.2 Gear ratios

Gears		Gear ratio	Diameter	Max torque at the wheel	Max speed at the wheel
Forward Gears	1	2.661	2.75 in		2028.5 rpm
	2	2.25648	7.37 in	1819.4	2393.6 rpm
	3	1.85196	6.25 in	1819.4	2919 rpm
	4	1.44744	5.12 in	15766	3740 rpm
	5	1.04292	4 in	0	5203 rpm
	6	0.6384	2.8 in	141900	5440 rpm
Reverse gear		2.661	2.75 in		2028.5 rpm

3.2 Shaft Design Methodology

The first step taken was creating a preliminary shaft layout for each shaft including dimensions such as the length of the shaft. Once the layout was developed, the free body diagrams for each shaft were created incorporating the loads previously calculated. After the free body diagrams were completed, shear and moment diagrams were generated for both the XY and XZ planes.

3.2.1 Shaft Shear, Moment and Torque Diagram

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unit is missing, and your 6th gear is giving the highest torque? Does not seem right.
Gear 5 gives 0 torque at the wheel?

 Number: 3 Author: japark Subject: Sticky Note Date: 12/2/2020 11:31:23 AM
What about the ground speed?

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Figure 3.2.1 through figure 3.2.5 are all of the free body diagrams with their shear and moment diagrams for each shaft on both the XY plane and the XZ plane. All of the moment diagrams show what that max bending moment is on all of the shafts.

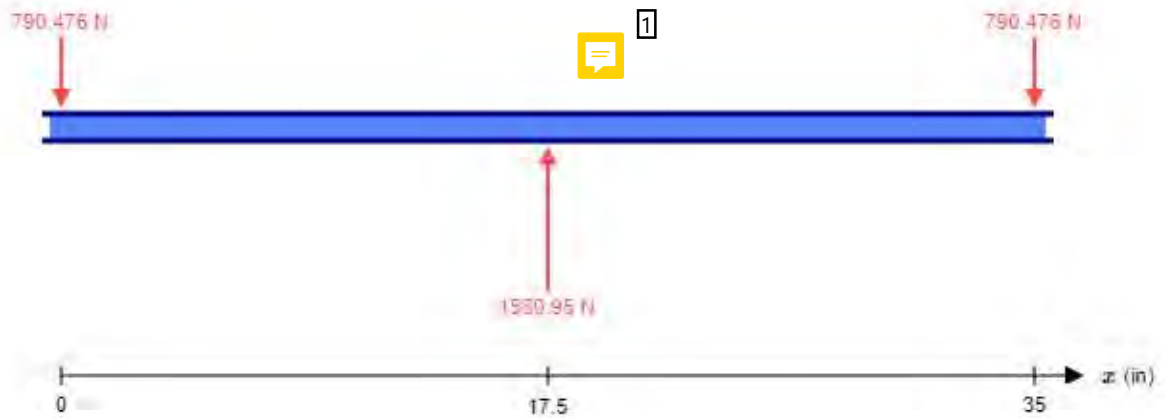
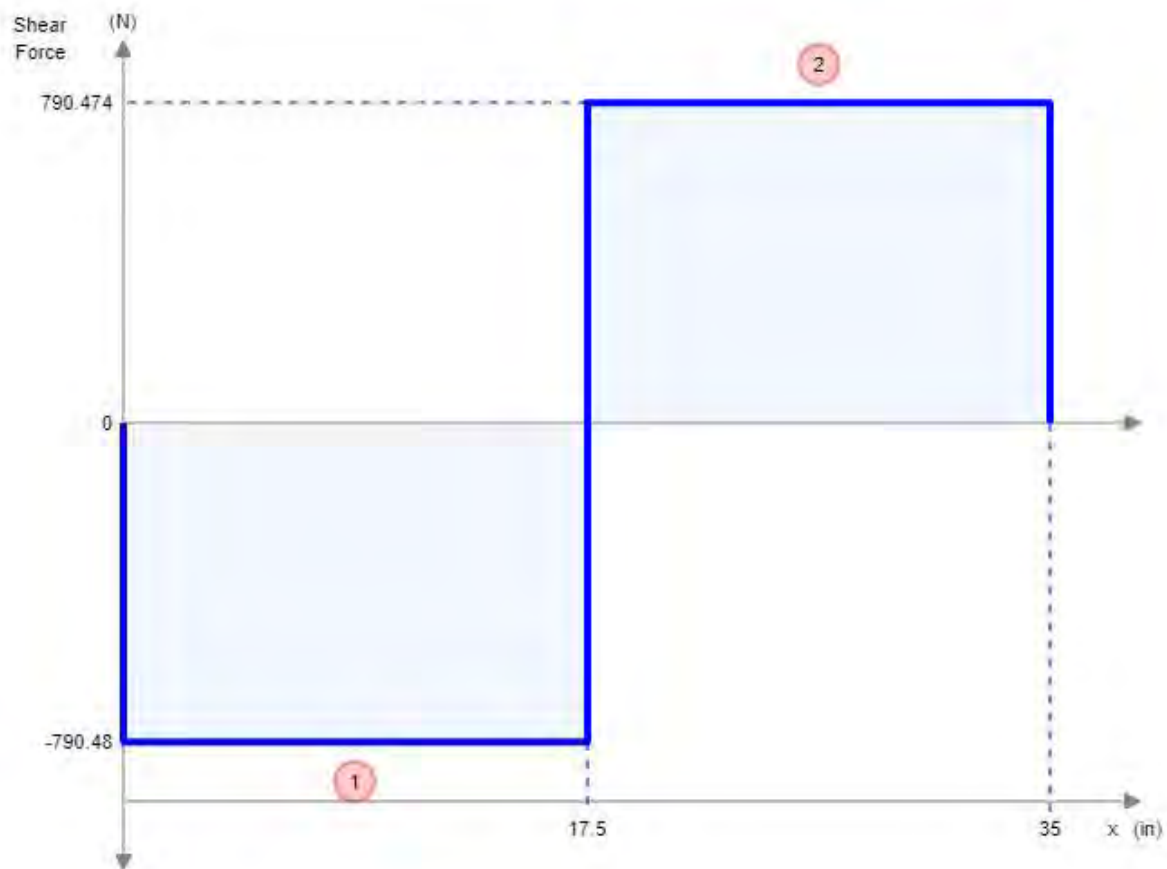


Figure 3.2.1 Shaft A Free Body Diagram on XY plane



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So you only have one support. This will lead to unstable solution.

Figure 3.2.2 Shaft A Shear Diagram on XY plane

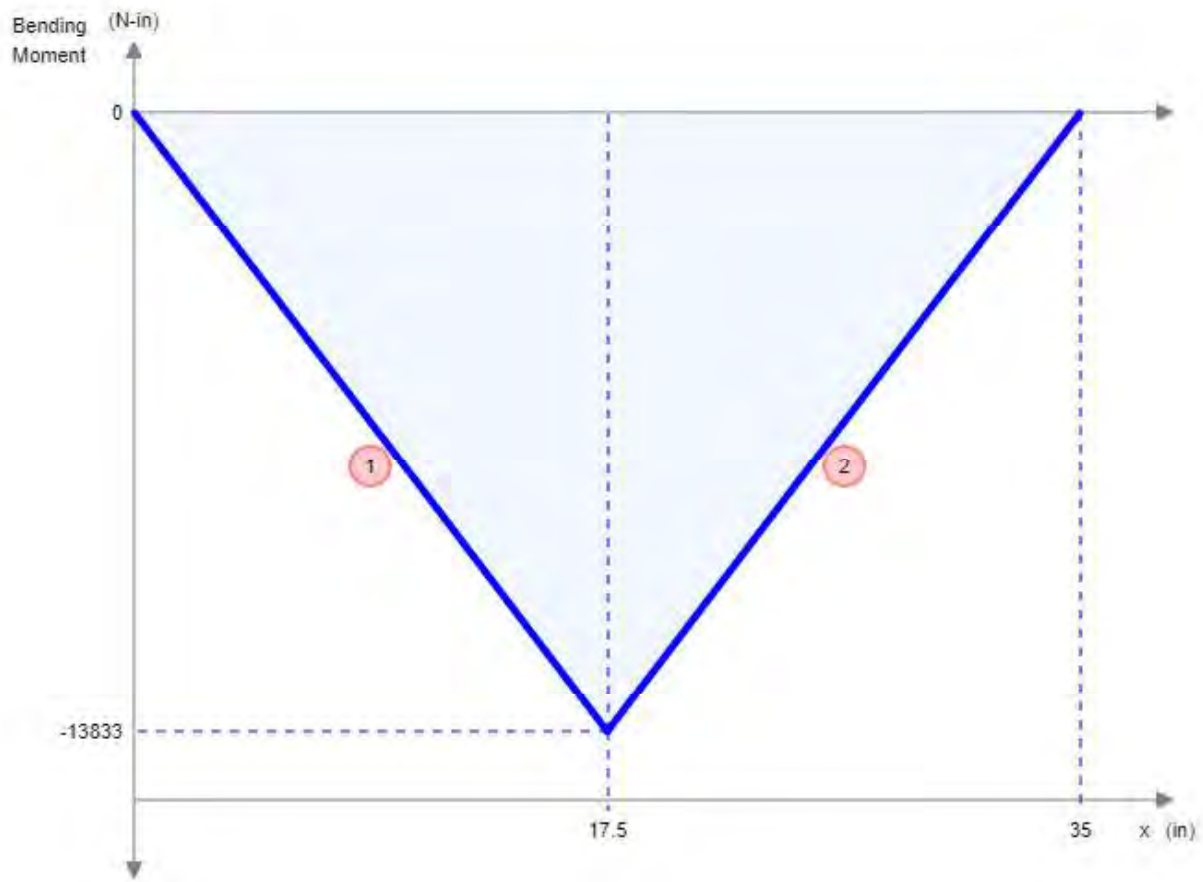


Figure 3.2.3 Shaft A Moment Diagram on XY plane

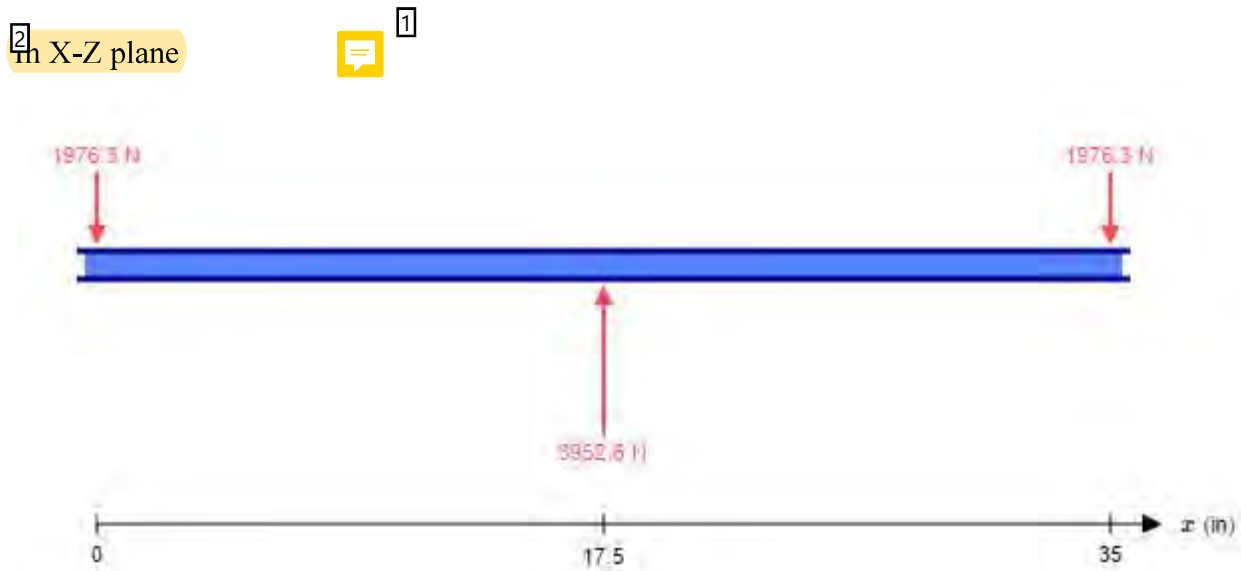


Figure 3.2.4 Shaft A Free Body Diagram on XZ plane



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The transmitted loads from gears at both ends should point to other direction, otherwise, you will have nonzero net torque at the shaft resulting in angular acceleration.

This is why you will have stability issues if you decide to have only one support.



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Figure 3.2.5 Shaft A Shear Diagram on XZ plane

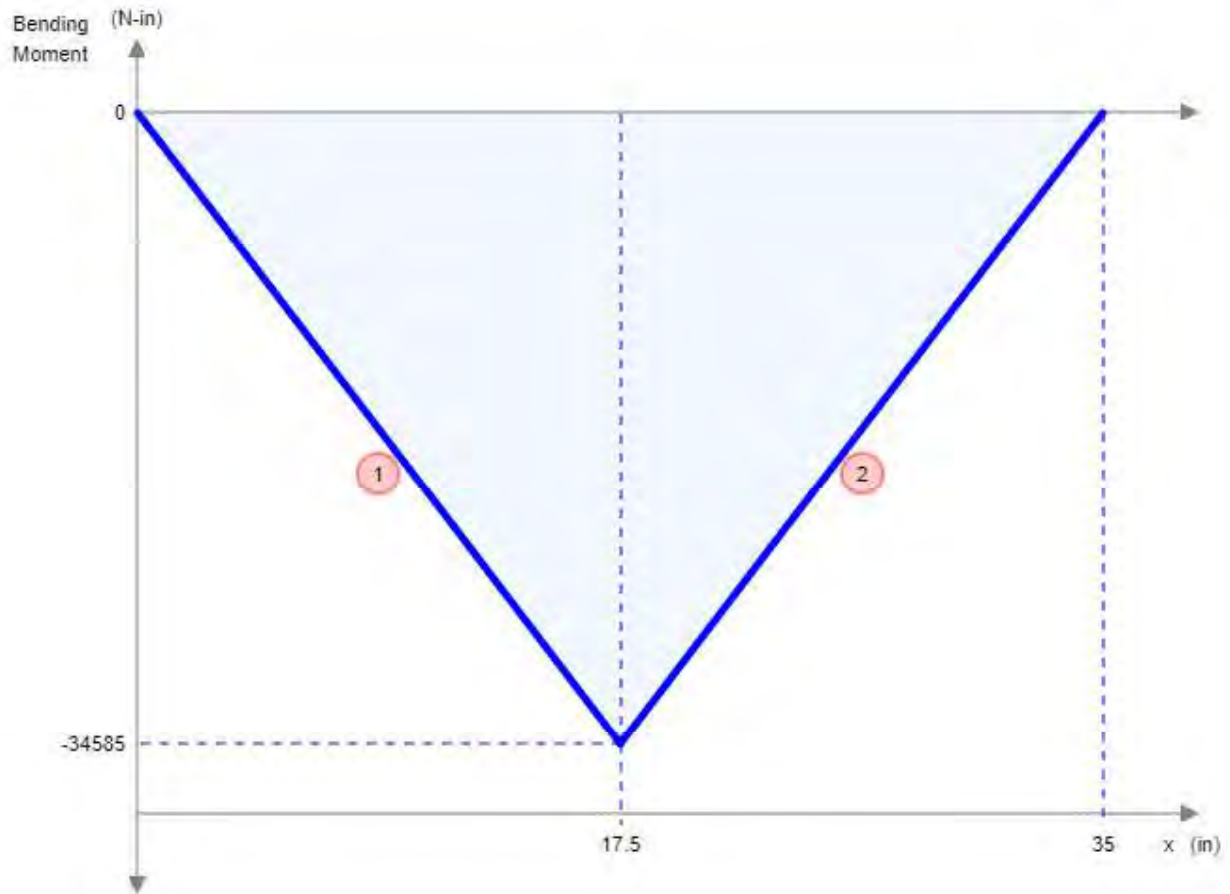


Figure 3.2.6 Shaft A Moment Diagram on XZ plane

3.2.2 Critical Locations ²

Next, the S_e equation was used to calculate the endurance limit at the critical location of a machine part in the geometry and condition of use.

$$S_e = k_a k_b k_c S'_e$$

k_a = Surface condition modification factor

k_b = Size modification factor

k_c = Load modification factor

S'_e = Rotary – beam test specimen endurance limit

Given

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Where is it?				

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$$\text{Shaft Material} = \text{AISI 1095 HR } S_{ut} = 120 \text{ kpsi } S_y = 66 \text{ kpsi } S_e = 0.5 (120) = 60 \text{ kpsi}$$

Using Table 6-2 and equation (6-19) from page 295

$$k_a = aS_{ut}^b, \text{ where } a = 14.4 \text{ kpsi}, b = -0.718$$

$$k_a = (14.4)S_{ut}^{(-0.718)} = 0.4897$$

Using equation (6-20) from page 296

$$k_b = 0.85$$

Using equation (6-26) from page 298

$$k_c = 1$$

Additionally, we assume

$$k_d = 1, k_e = 1$$

Using equation (6-8) from page 290

$$S'_e = 0.5 (120) = 60 \text{ kpsi}$$

Using all gathered values, we calculated for S_e by using the following equation from equation (6-18) from page 295

$$S_e = k_a k_b k_c = S'$$

$$S_e = (0.4897)(0.85)(1)(1) (60) = 24.9744$$

3.2.3 Fatigue Analysis

The fatigue analysis was performed finding k_t and k_{ts} values from the book Table 7.1 in page 365 or Table 3.2.1 of this report. The K_t and K_{ts} values were taken at the Shoulder fillet and were used to solve for k_f and k_{fs} .

Table 7-1

First Iteration Estimates for Stress-Concentration Factors K_t and K_{ts} .

Warning: These factors are only estimates for use when actual dimensions are not yet determined. Do *not* use these once actual dimensions are available.

	Bending	Torsional	Axial
Shoulder fillet—sharp ($r/d = 0.02$)	2.7	2.2	3.0
Shoulder fillet—well rounded ($r/d = 0.1$)	1.7	1.5	1.9
End-mill keyseat ($r/d = 0.02$)	2.14	3.0	—
Sled runner keyseat	1.7	—	—
Retaining ring groove	5.0	3.0	5.0

Missing values in the table are not readily available.

Table 3.2.1 K_t and K_{ts} values

Using the table above, our $k_t = 2.7$ and $k_{ts} = 2.2$. Now that we have our k_t and k_{ts} values, we assume the a notch radius to be 0.1 and used figure 6-20 in page 303 to find our q and q_s values. Utilizing the figure, the $q = 0.85$ and $q_s = 0.9$

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.)

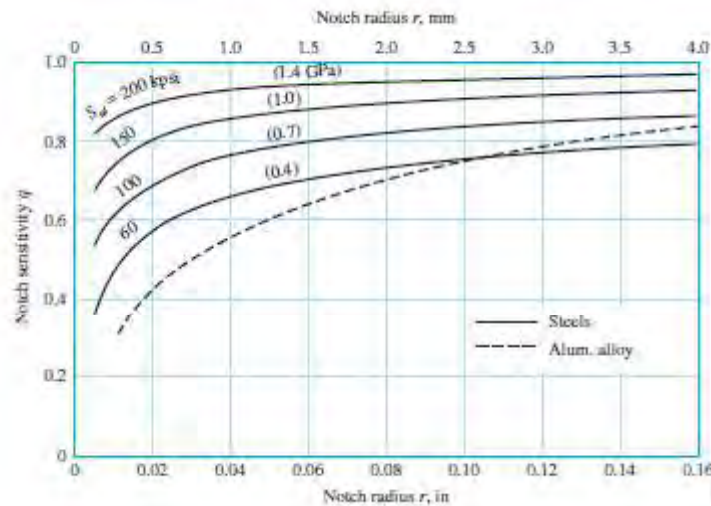


Figure 3.2.7 Notch sensitivity

Using equation (6-32) from page 303

$$k_f = 1 + q(k_t - 1) = 1 + 0.85(2.7 - 1) = 2.4450$$

$$k_{fs} = 1 + q_s(k_{ts} - 1) = 1 + 0.9(2.2 - 1) = 2.0800$$

Now that all the values have been found, we can use equation (7-8) in page 360 to find the diameter of the shaft utilizing DE-Goodman equation.

$$d = \left(\frac{16}{\pi} \left\{ \frac{1}{S_e} [4 (K_f M_a)^2 + 3 (K_{fs} T_a)^2] + \frac{1}{S_{ut}} [4 (K_f M_m)^2 + 3 (K_{fs} T_m)^2] \right\} \right)^{\frac{1}{3}} = 1.05 \text{ in}$$

3.3 Key Methodology

When considering what bearing to select, different parameters have to be considered such as the contact stress that will be occurring on the inner ring, rolling element, and the outer ring along with the overall bearing life and loads. When considering the bearing life common measures are the number of revolutions of the inner ring until fatigue becomes evident and how long the bearing will be at use at its specific speed until fatigue becomes evident. Additionally, the key was chosen from table 7-6 in page 383.

Shaft Diameter		Key Size		Keyway Depth
Over	To (Incl.)	w	h	
$\frac{5}{16}$	$\frac{7}{16}$	$\frac{3}{32}$	$\frac{3}{32}$	$\frac{3}{64}$
$\frac{7}{16}$	$\frac{9}{16}$	$\frac{1}{8}$	$\frac{3}{32}$	$\frac{3}{64}$
		$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{16}$
$\frac{9}{16}$	$\frac{7}{8}$	$\frac{3}{16}$	$\frac{1}{8}$	$\frac{1}{16}$
		$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{32}$
$\frac{7}{8}$	$1\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{3}{32}$
		$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{8}$

Figure 3.3.1 Key Selection

All bearings selected for this project were rolling bearings. In addition, the bearing catalog load rating was calculated using equation (11-3) from page 367 and using values given in the dynograph.

$$C_{10} = F D \left(\frac{L_D n_D 60}{L_R n_R 60} \right)^{1/a} \text{ Where } a = 3, F_D = 342.01 \text{ lbf}, L_D = 20,000, n_D = 2500 \text{ rpm},$$

$$C_{10} = 342.01 \left(\frac{20,000(2500)60}{10^6} \right)^{1/3} = 4932 \text{ lbf}$$

 Number: 1 Author: japark Subject: Sticky Note Date: 12/2/2020 11:47:06 AM
This is bearing analysis not key.

 Number: 2 Author: japark Subject: Highlight Date: 12/2/2020 11:46:55 AM

 Number: 3 Author: japark Subject: Sticky Note Date: 12/2/2020 11:46:31 AM
How did you find this value? Referring to your V&M diagram, shaft experiences 1580 lb on XY and 3952 lb on XZ plane.

 Number: 4 Author: japark Subject: Highlight Date: 12/2/2020 11:44:57 AM

3.3.1 Key Design

The key specifications were chosen after determining the shaft length and diameter. Using the standard table we were able to pick a proper key size to meet the safety requirements for the shaft. After picking the material, AISI 1095 HR, same material as the shaft we calculated the loads on the key (Torque and Force). Using the standard dimensions of the shaft key size we generated a sketch of the key.

Chapter 4: CAD Models

In this chapter the CAD models the output transmission shaft components are displayed. The components included in the following sections include six forward gears, one reverse gear, the output shaft, driving shaft, keys, and bearings. The models are displayed along with their respective engineering drawings. In addition, the engineering drawings contain all of their critical dimensions.

4.1 Pinion

FFigure 4.1 is the 3D model of pinion gear. The pinion is attached to shaft 2 by Key 1. The number of teeth selected for the pinion was based on the pressure angle 20° . There is also a groove in the bore of the gear, which is the keyway to help connect to the propeller shaft.

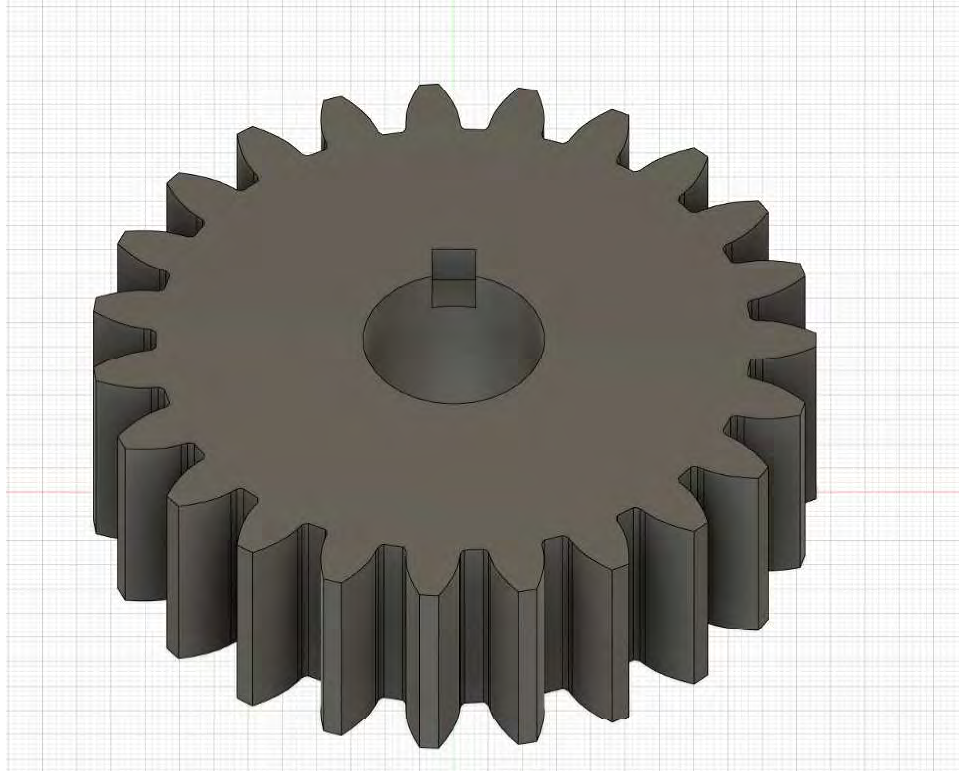


Figure 4.1 Pinion Gear 3D Model

Figure 4.2 is the 2D model drawing of the Pinion Gear. The model was done in Fusion based on the 3D model.

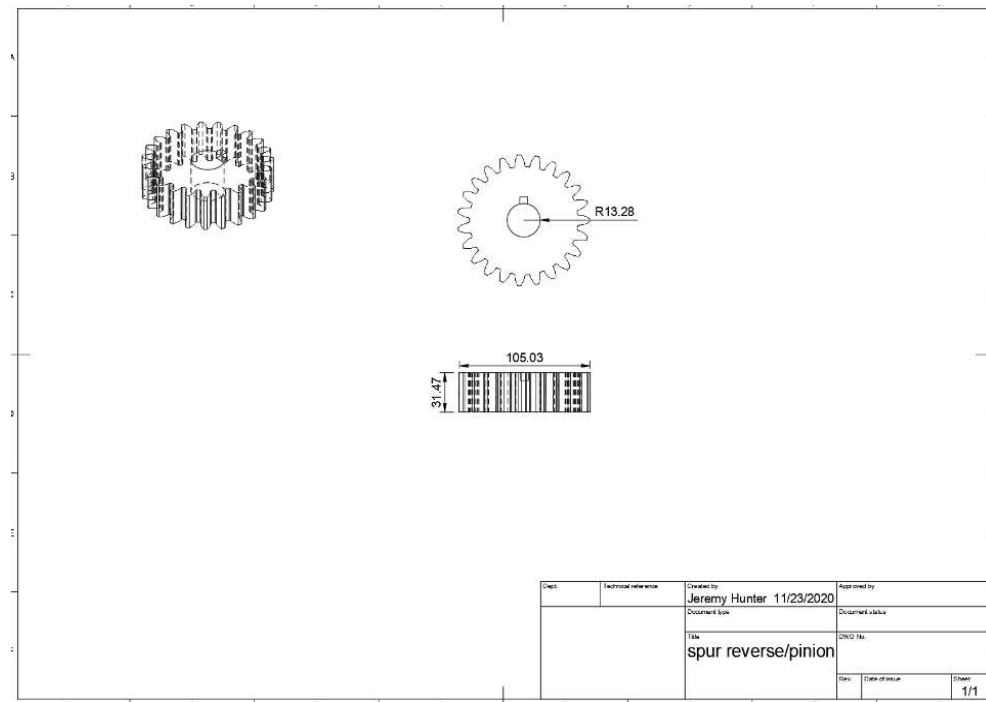


Figure 4.2 Pinion Gear 2D Model



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The engineering drawing should contain all necessary information for anyone to replicate your part. Number of teeth, pressure angle and dimensions of the pitch circle, key, etc. are missing.

4.2 1st Gear

Figure 4.2 through 4.14 show the 3D models and their 2D drawings of gear 1 through 6. All of these gears are on the propeller shaft. They are attached by Key 1 to the shaft. The pressure angle of 21.8° was used on all the gears and it is a spur gear. The groove in the bore of the gear helps keyway connect to the propeller shaft.

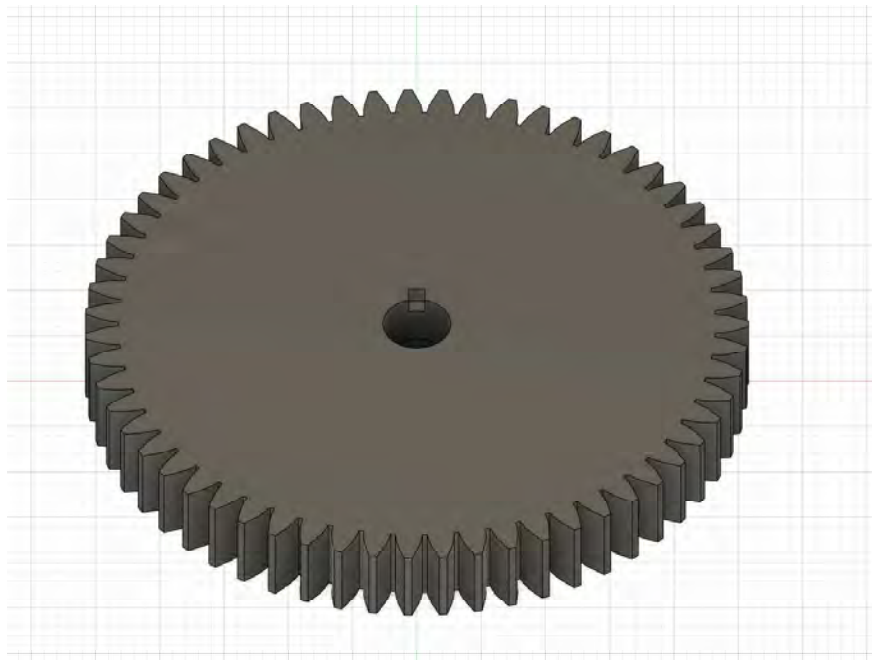
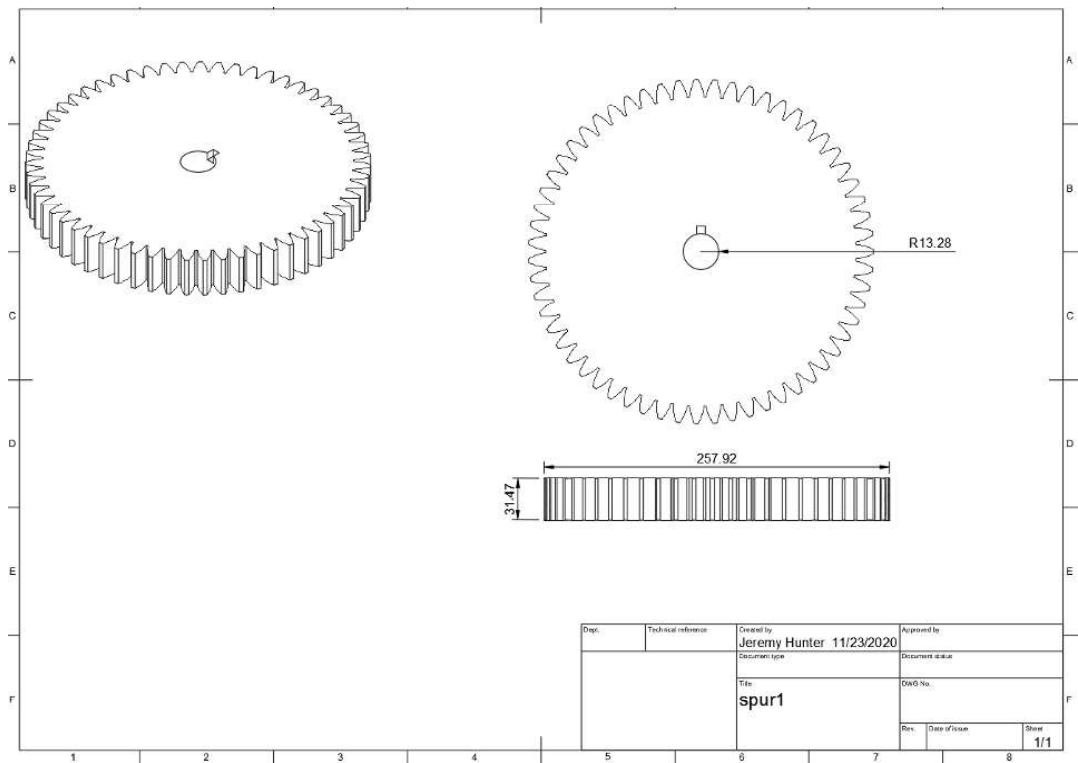


Figure 4.3 First Gear 3D Model



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The same thing happens here.

4.3 2nd Gear

Figure 4.4 First Gear 2D Model

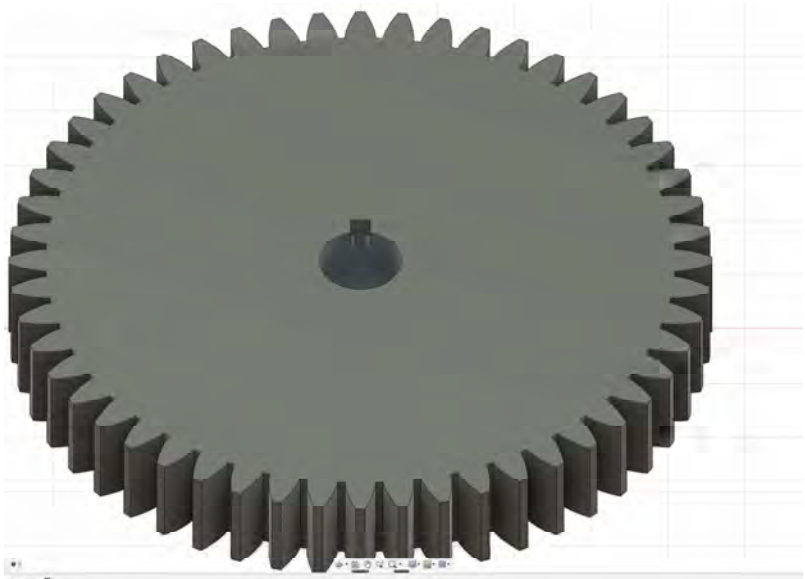


Figure 4.5 Second Gear 3D Model

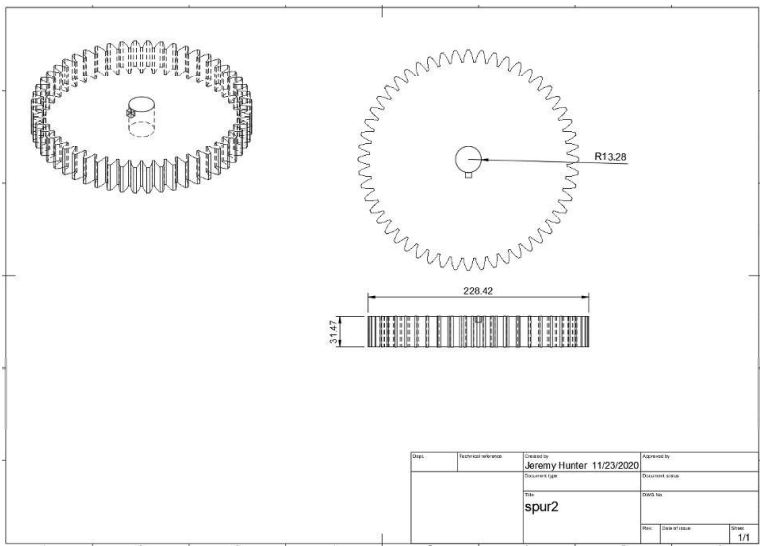


Figure 4.6 Second Gear 2D Model

4.4 3rd Gear

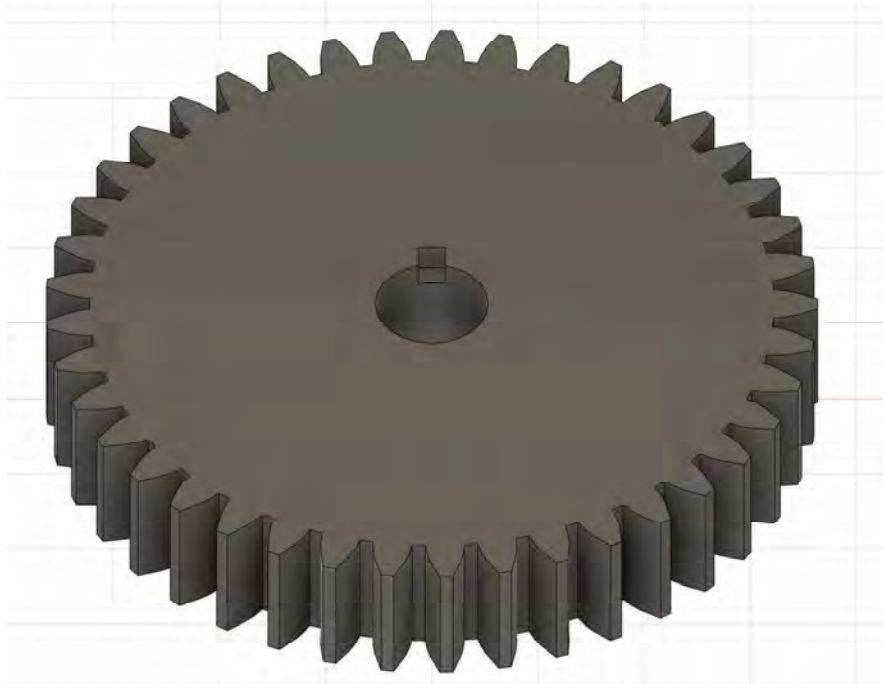


Figure 4.7 Third Gear 3D Model

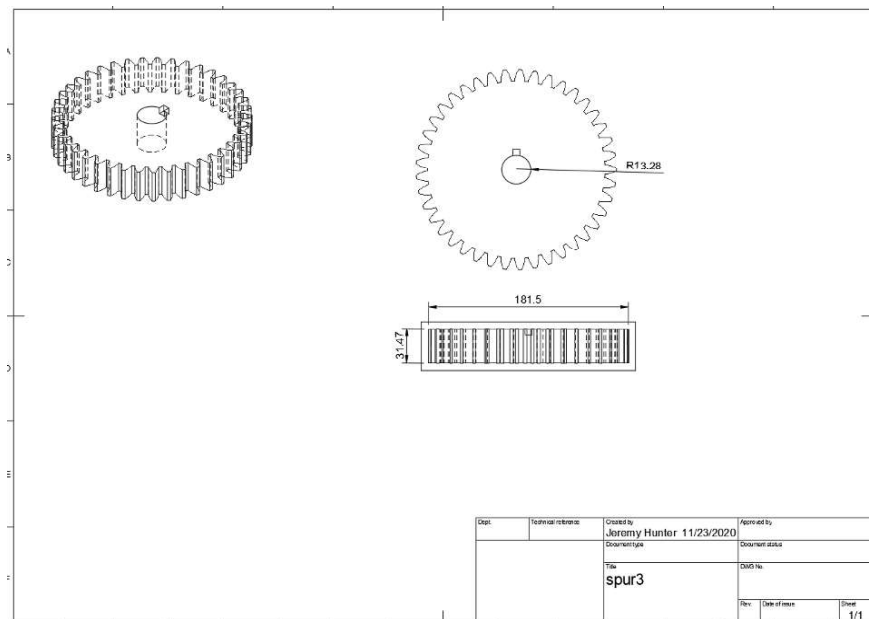


Figure 4.8 Third Gear 2D Model

4.5 4th Gear

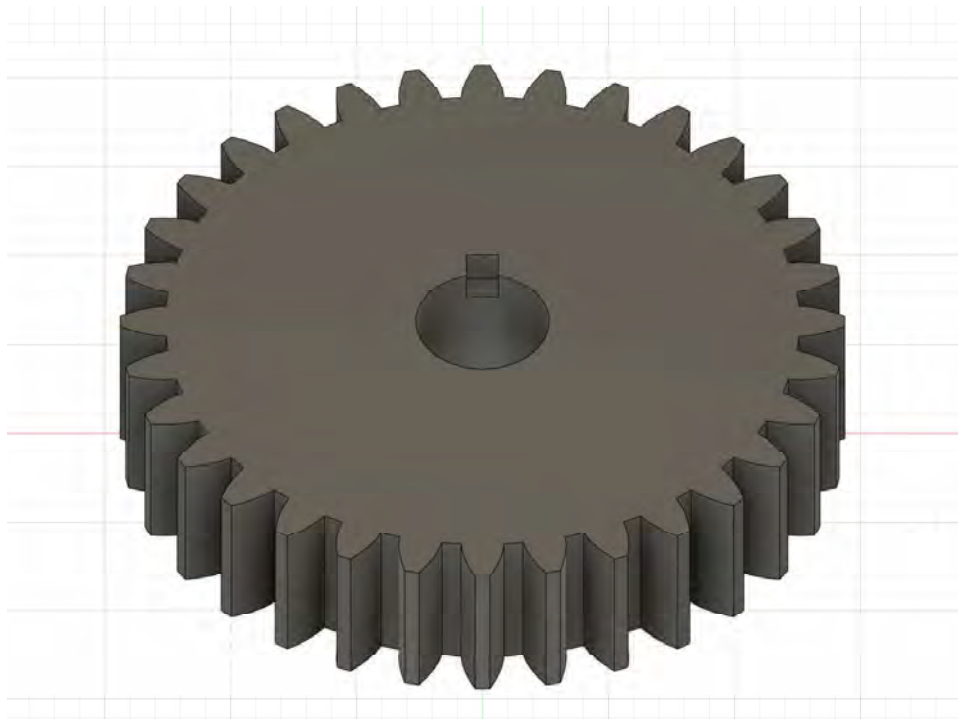


Figure 4.9 Fourth Gear 3D Model

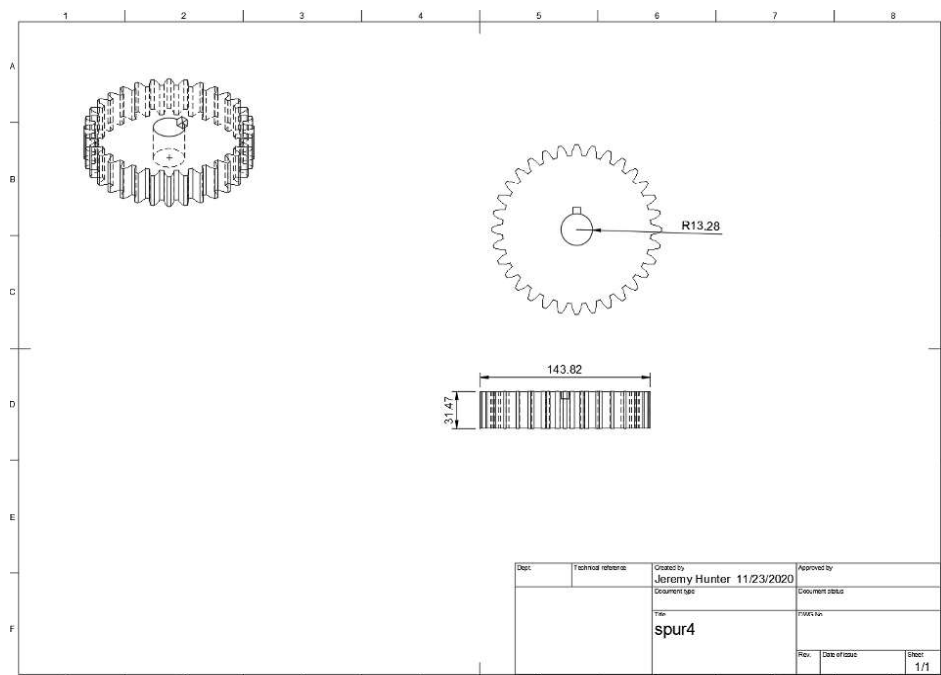


Figure 4.10 Fourth Gear 2D Model

4.6 5th Gear

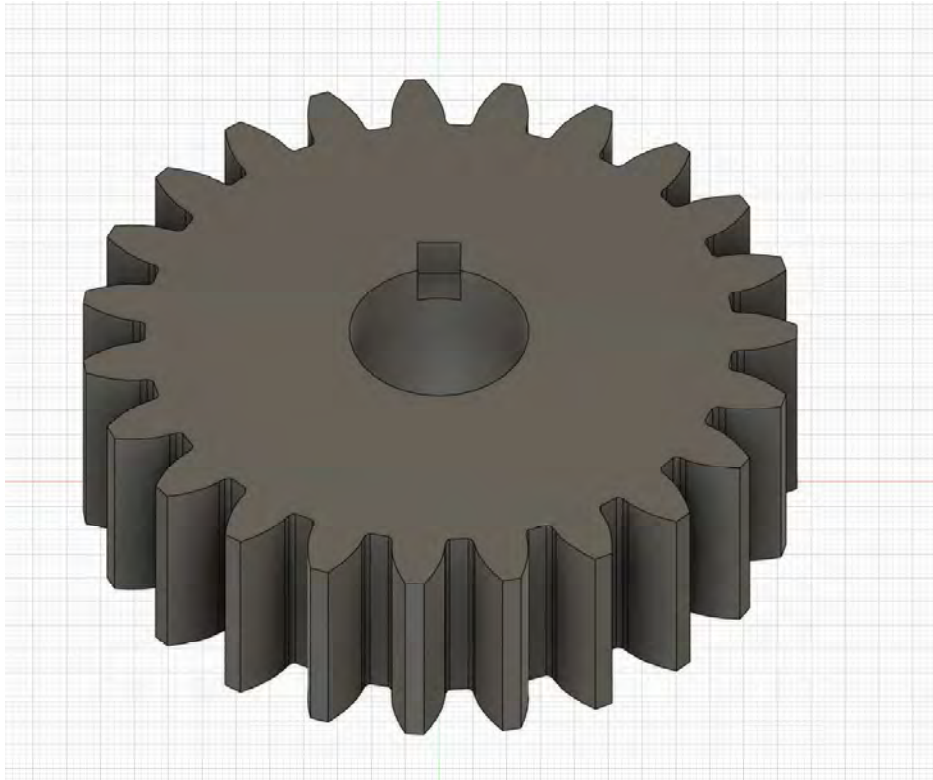


Figure 4.11 Fifth Gear 3D Model

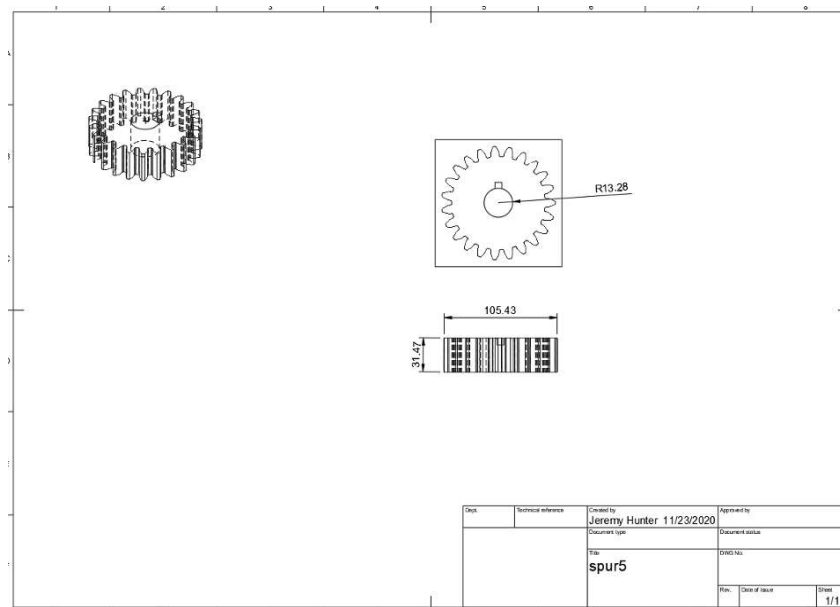


Figure 4.12 Fifth Gear 2D Model

4.7 6th Gear

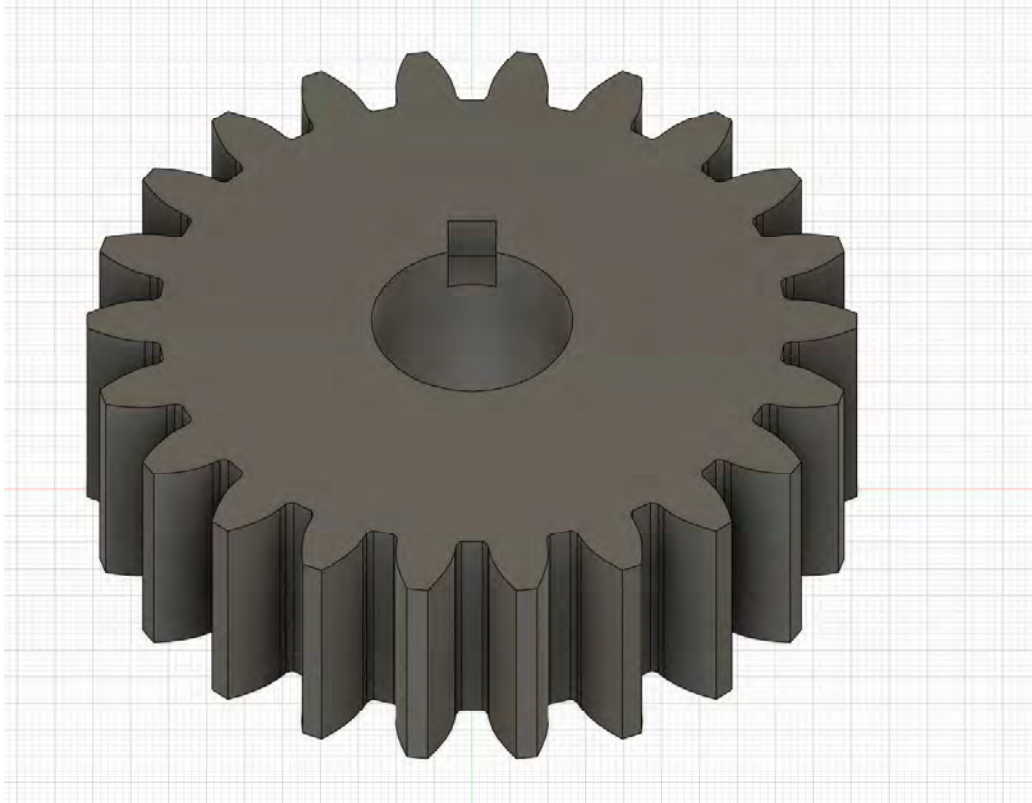


Figure 4.13 Six Gear 3D Model

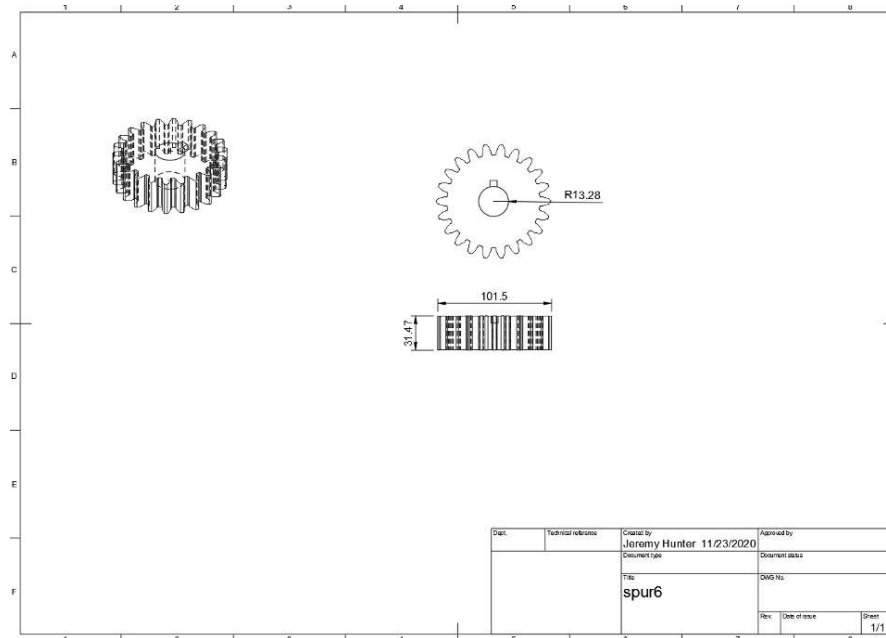


Figure 4.14 Six Gear 2D Model

4.8 Reverse Gear

Figure 4.15 & 4.16 is the 3D model of reverse gear and the 2D drawing model. The reverse gear has the same parameters as the pinion. It does not alter the speed or the torque, but it changes the direction of the wheels by using an extra gear to change shaft rotation. It is not connected to either shaft.

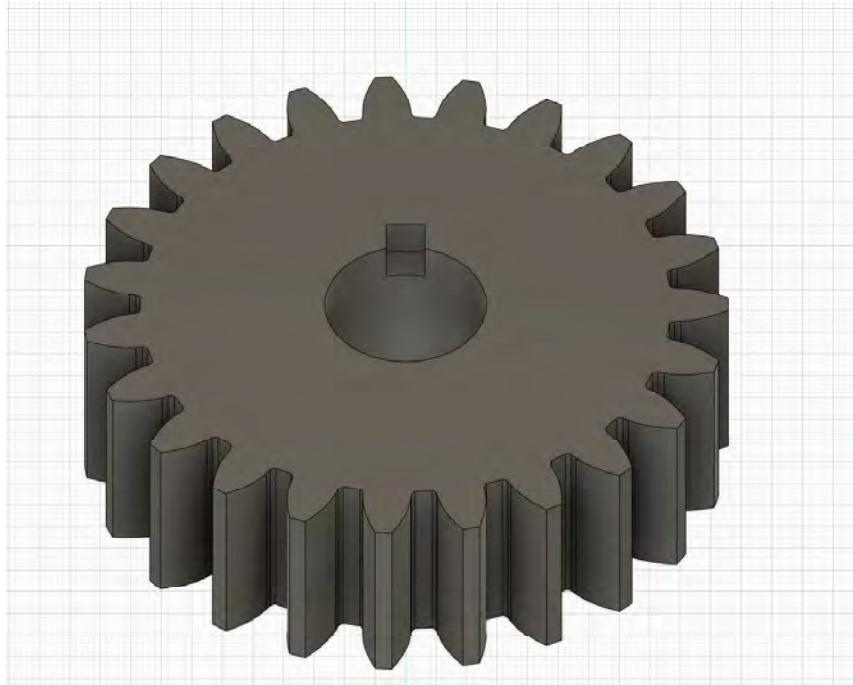


Figure 4.15 Reversed Gear 3D Model

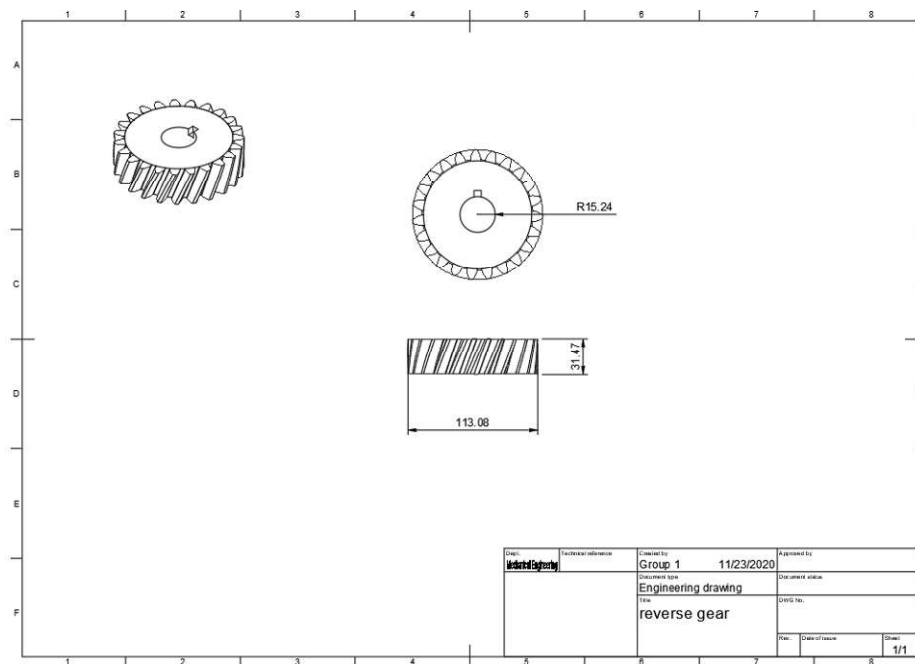


Figure 4.16 Reversed Gear 2D Model

4.9 Shaft and its Components (Keys and Bearings)

In this section the propeller and driver shaft 3D and 2D models can be found. The Models were developed in the Fusion 360 software and are displayed in Figures 4.16-4.19.

Propeller Shaft

The propeller shaft 3D model is displayed in Figure 4.16. The shaft is 35 in long and has a diameter of 1.05 in. In addition, on Figure 4.17 the 2D model of the propeller shaft is shown along with the mentioned dimensions.



Figure 4.17 Propeller Shaft 3D Model

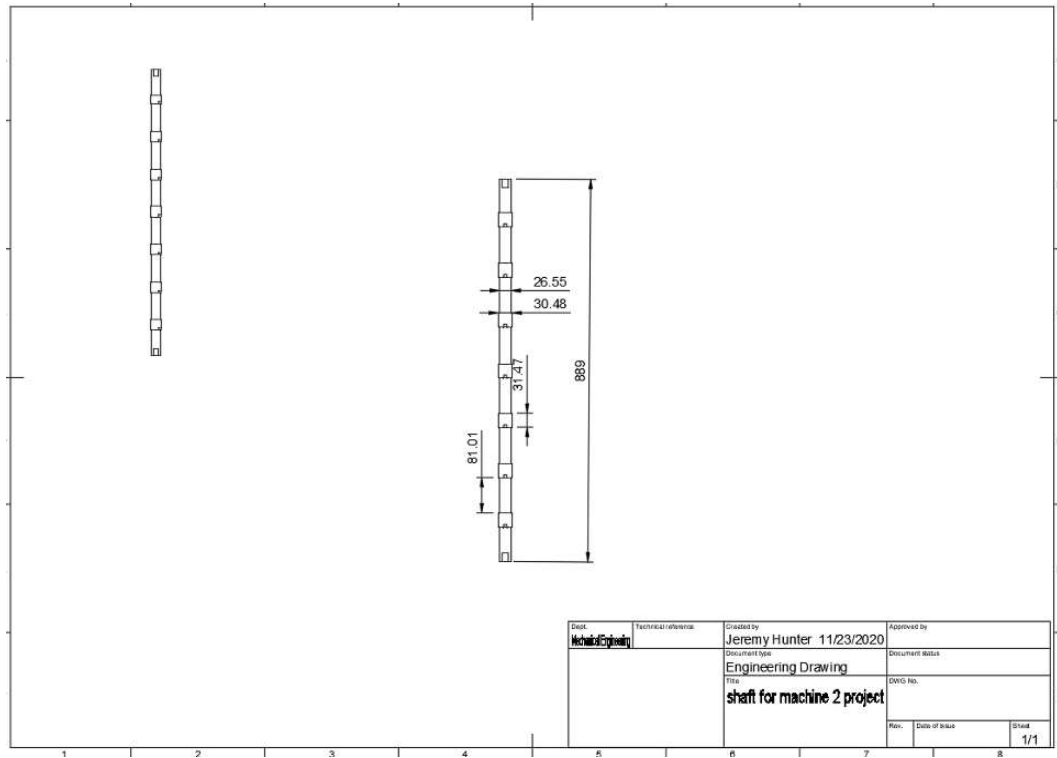


Figure 4.18 Propeller Shaft 2D Model

Driver Shaft

In figure 4.18 the 3D model of the driver shaft can be found. The driver shaft only holds space for 1 pinion who serves as the driving gear for the proposed design. In addition, the 2D model is provided in Figure 4.19.



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What is the driver shaft? The description says pinion will be mounted on this, does this mean this is part of crankshaft?

Technical drawing of a shaft for machine 2 project. The drawing shows a vertical shaft with various dimensions. The total length is 889. The diameter is 26.55. The distance from the top to the first step is 30.48. The distance between the two steps is 31.47. The distance from the bottom to the second step is 81.01. The drawing is labeled 'shaft for machine 2 project'.

Dept.	Technical reference	Created by	Approved by
Technical Engineering		Jeremy Hunter 11/23/2020	
		Document type	Document status
		Engineering Drawing	
		Title	DWG No.
		shaft for machine 2 project	
		Rev.	Date of issue
			Sheet
			1/1

Shaft Keys

43

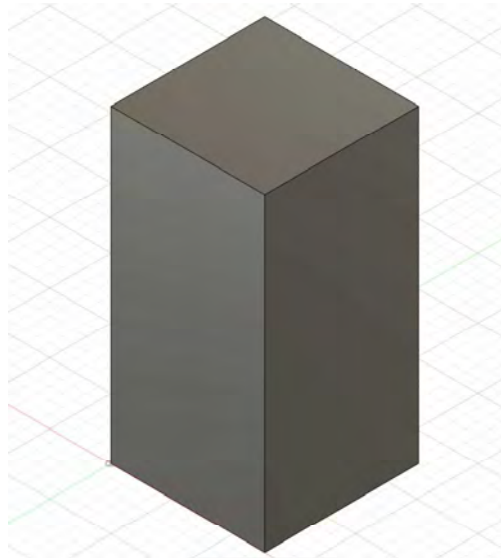


Figure 4.21 Shaft Key 3D Model

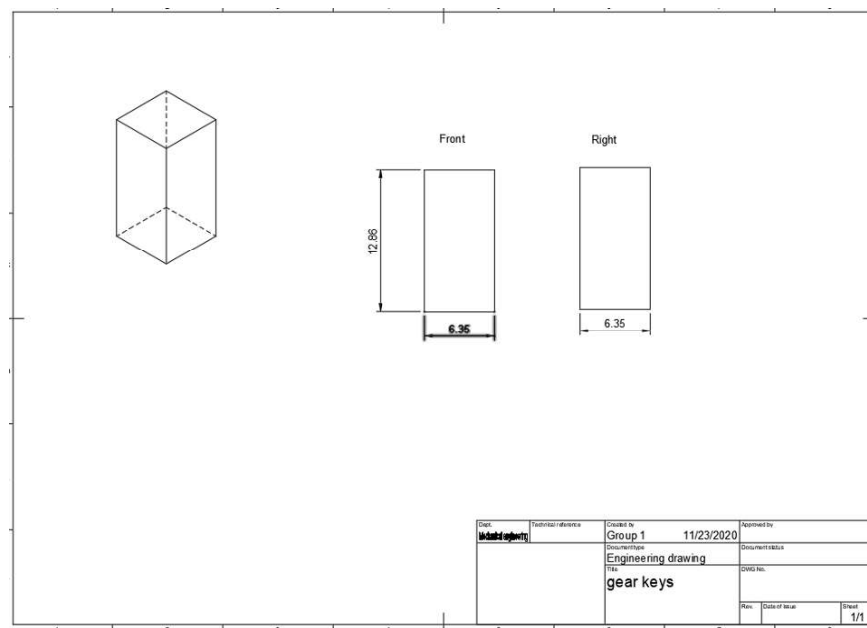


Figure 4.22 Shaft Key 2D Model

Bearing

Figure 4.22 & 4.23 shows the 3D model and 2D drawing of the Radial Ball Bearing which are the most widely used type of rolling element bearing in the world today due to their versatility and

overall performance. These bearings are produced with high quality materials and exacting tolerances. Figure 4.22 represents the bearings on the propeller shaft.

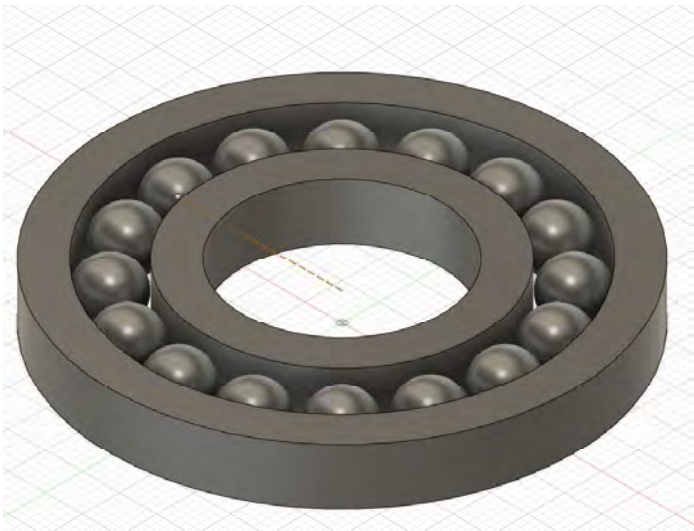


Figure 4.23 Bearing 3D Model

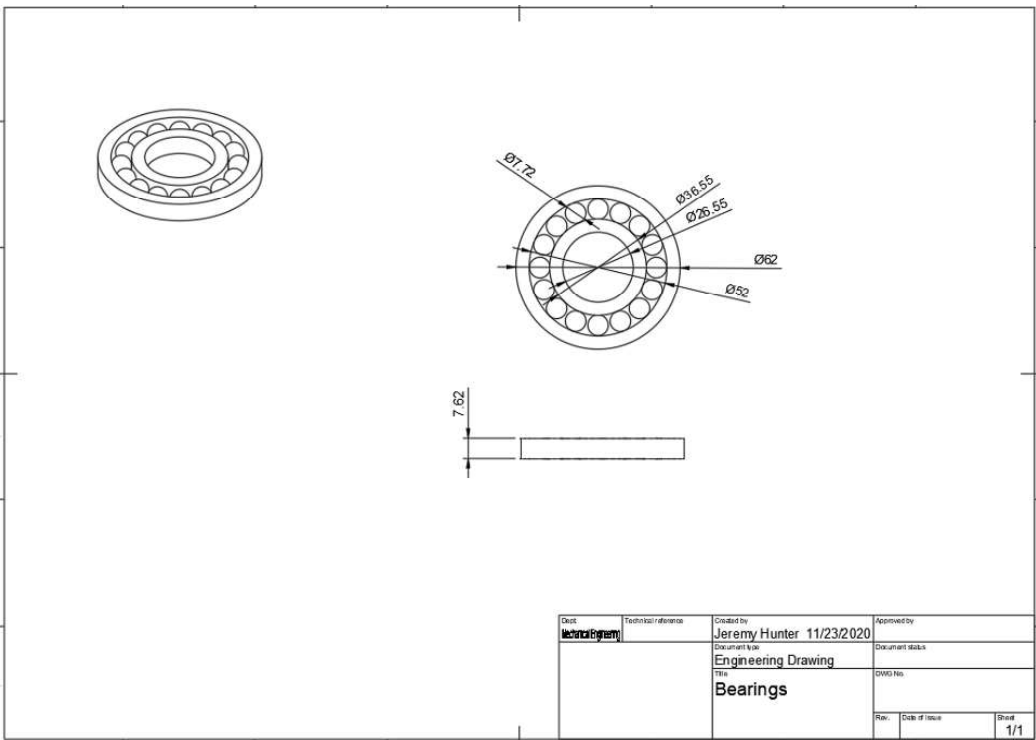


Figure 4.24 Bearing 2D Model

Figures 4.25 and 4.26 display the assembled model of the proposed transmission design. Figure 4.25 contains the 3D model of the combination of all the gears 6 forward, one reversed, and the

pinion. In addition, figure 4.26 contains the 2D drawing of all the components assembled together along with the general measurements of the system.

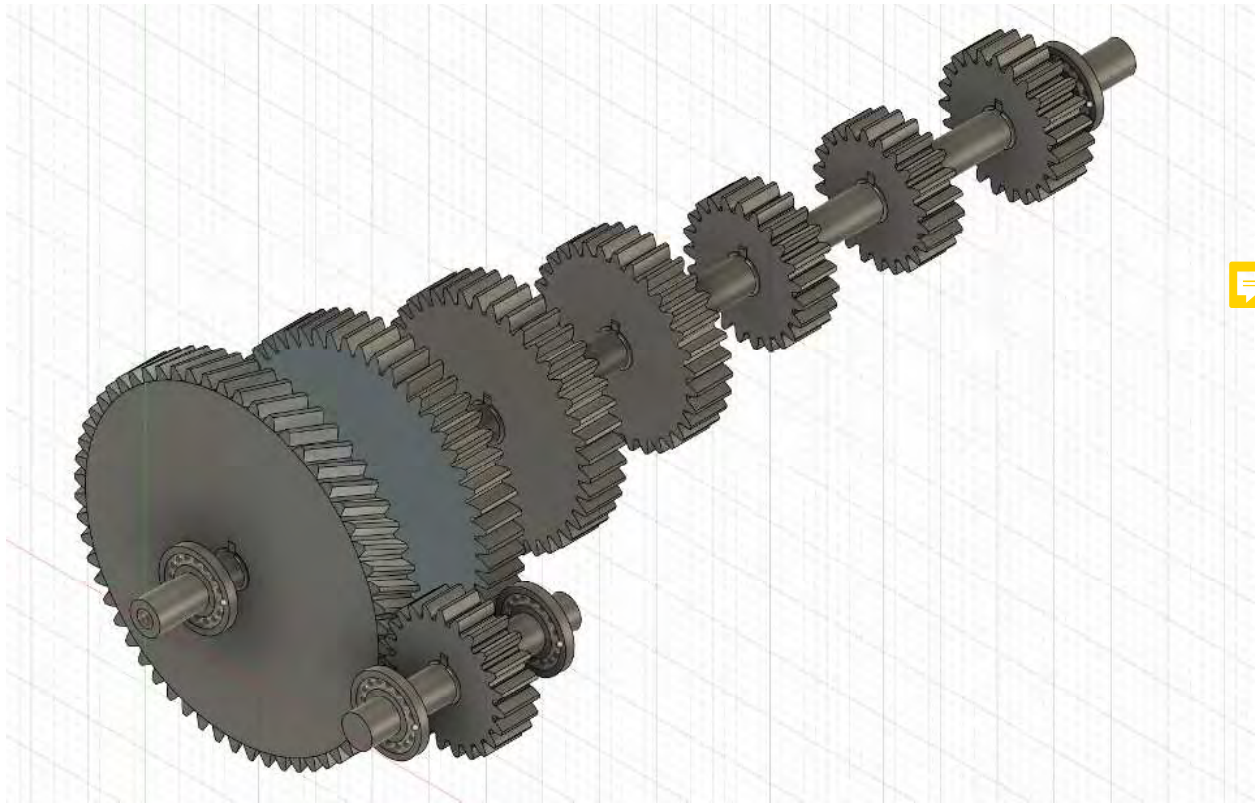


Figure 4.25 Fully Assembled Gears

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The gears will not be assembled like this but it is ok. Your first gear is helical.

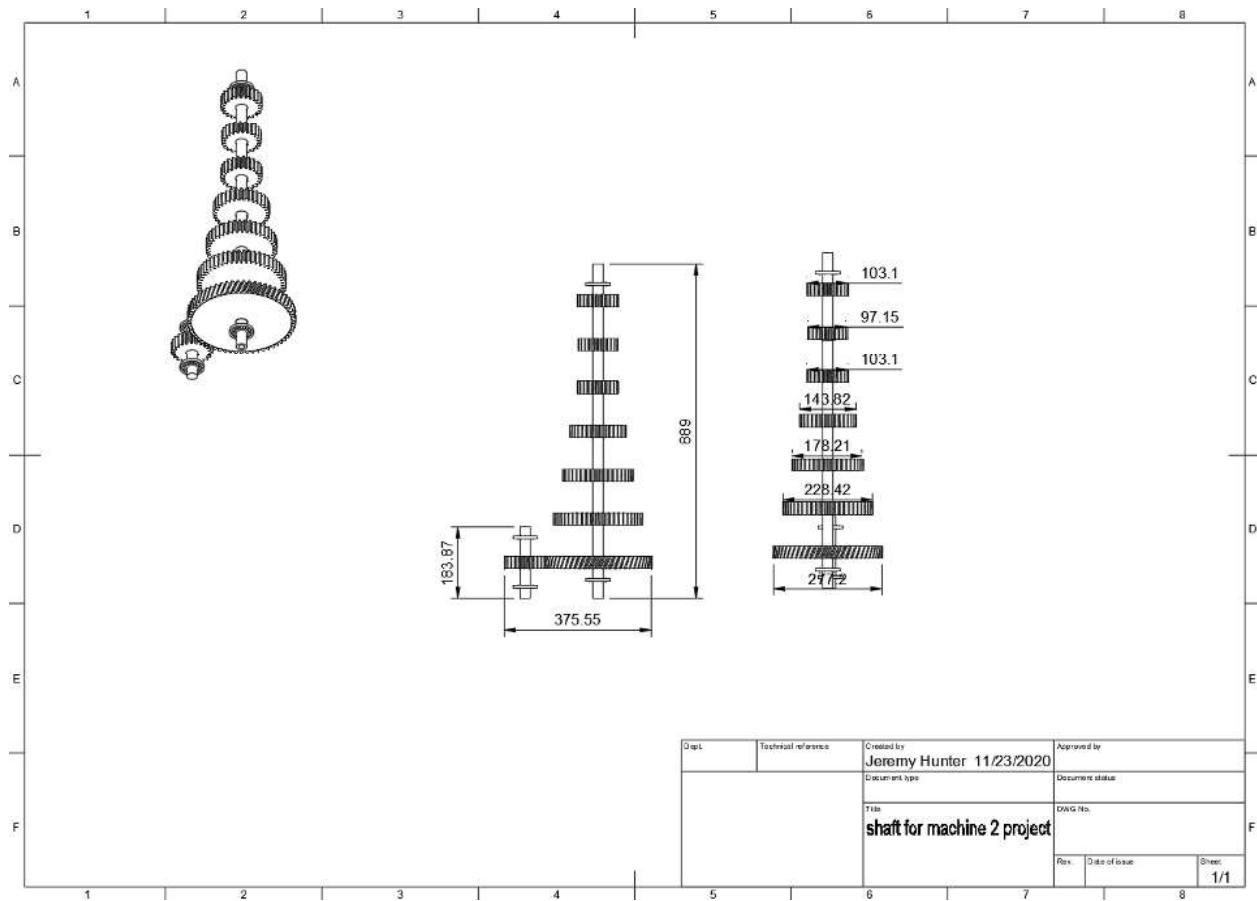


Figure 4.26 Fully assembled 2D model

Chapter 5: ANSYS Simulation



Stress analysis of the propeller shaft was done using ANSYS. The shaft is fixed and one end is selected, forces are given at chosen points. The reactant forces act in opposite directions. Torque acts at two points in opposite directions. The reactant forces and bending moments were initially calculated. Then the maximum shear stress, normal stress were calculated. Those calculated values were then used to create ANSYS model.

5.1 Propeller Shaft Analysis

In figure 5.1 the total deformation of the propeller shaft is displayed. As part of the simulation, each gear transmitted load was applied in the same direction. In addition, the ends of the shaft were marked as fixed cylindrical supports and an acceleration was added to the shaft. Moreover, in figure 5.1 the equivalent stress simulation is displayed. Finally, the same process was followed to perform the simulation analysis of the driver shaft. However, the force applied to the pinion was in the opposite direction to the ones on the propeller shaft.

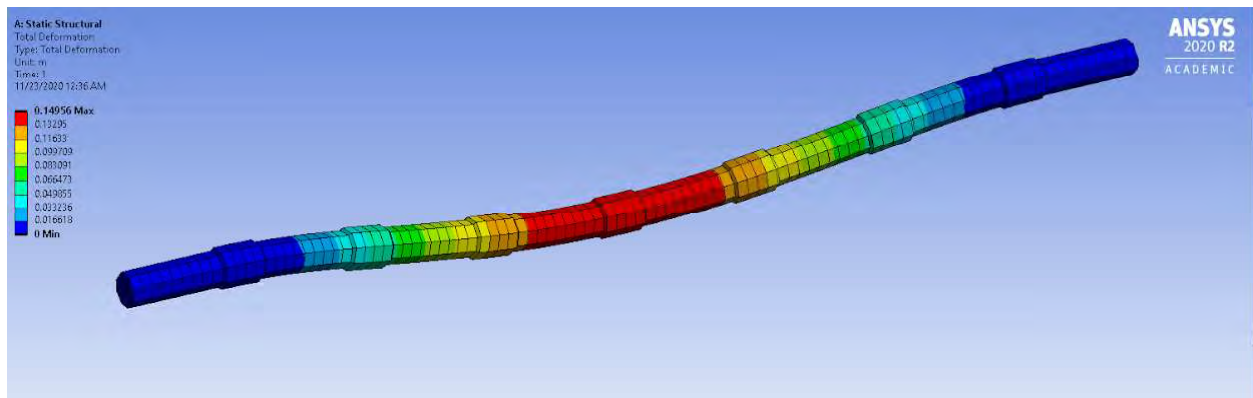


Figure 5.1 Propeller Shaft Total Deformation

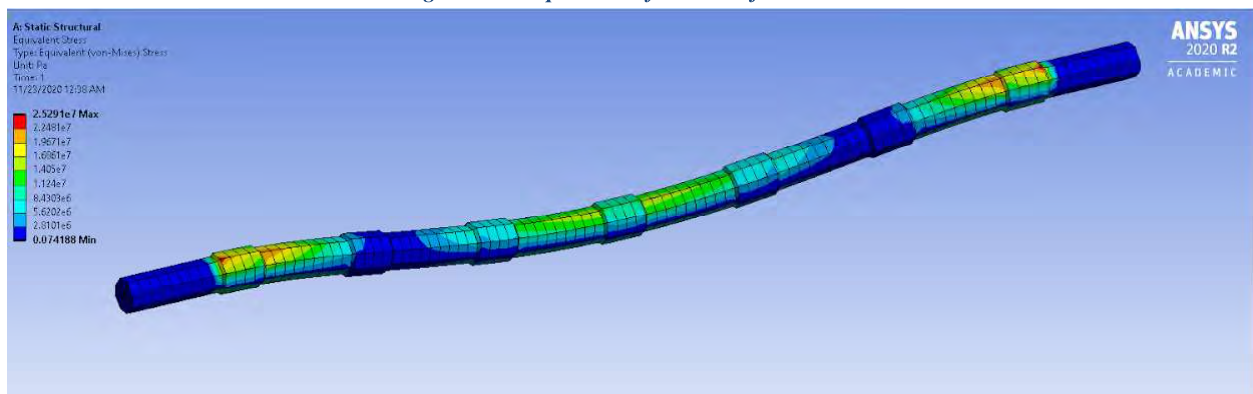


Figure 5.2 Propeller Shaft Equivalent Stress



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Whenever you are showing FEA results, there are things that must go with it.

- 1) Geometry - it is provided in the previous chapter
- 2) Boundary conditions - although your description partially describes it, you need to provide a figure that how your model is setup.
- 3) Material properties
- 4) Result contours - the color legends should be clearly legible.

Without any one of these, the FEA result does not give the reader any usable information.

5.2 Driver Shaft Analysis

Figure 5.2 shows the Driver shaft total deformation. It was created in ANSYS using the calculated loads. The red areas represent the highest area of twist due to torque causing extreme stress in the area. That could be due to the pinion being at that location.

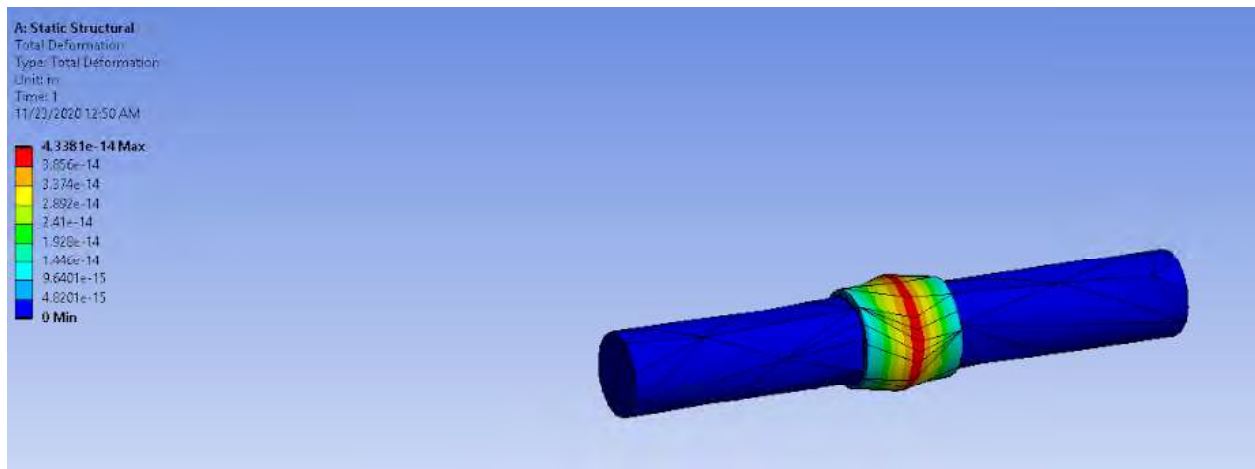


Figure 5.3 Driver Shaft Total Deformation

In Figure 5.4, the equivalent stress across the drive shaft is displayed. This was done in ANSYS by using the calculated bending and shear stress. The figure shows evenly distributed stress and no extreme high points of stress which could lead to material failure.

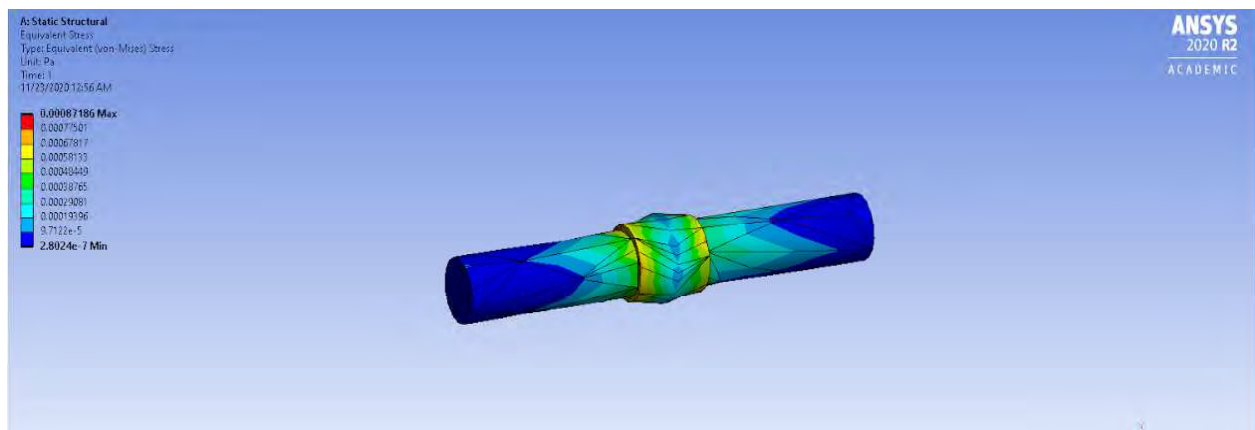


Figure 5.4 Driver Shaft Equivalent Stress

Chapter 6: Conclusion

In this project, a variety of analysis were carried out in order to achieve an optimal gearbox design for 2018 Honda Accord. The manual calculations were carried out, three different iterations of values were investigated. First, the gear ratios were calculated, which ranged from 0.6384 to 2.661. Using the values for the ratios, the teeth of the pinion was determined using a table. After the pinion teeth were found the calculations for the rest of the gears were done by using equations from the book. A variety of parameters were found like diameter, face width, etc.. After the manual calculations were completed, the code was created in MATLAB. The code enabled a more flexible and reliable analysis during the design process, the numbers were easy to change and outputted more precise results. The gearbox design was then constructed in Fusion 360. All the gears and shafts were constructed by hand. Then a simulation was run in ANSYS for the shafts using the calculated loads.

Furthermore, the results of this project match the standard Honda Accord parameters. For starters, the values of the gear ratio range from 0.566 to 2.652 which is extremely close to what was calculated. In addition, in order to find the length of the shaft, Honda was called and asked to provide an accurate shaft length for their vehicle which was 35 in. Using data provided in the problem statement and data gathered from honda, the group was able to accurately calculate the required values and demonstrate that our calculations are not far off from real world tests.

Throughout this project, the group encountered multiple problems. For starters, the shaft analysis had multiple iterations and fixes due to it not being correctly done. EES allowed us to minimize the effect of having to hand calculate everything, however it was still a chore. Additionally multiple iterations had to be done when choosing a material that would fit our standards to ensure that the values gathered were accurate resulting in the group having to backtrack.

Overall, the group obtained valuable learning experience when tasked with completing this project. Reinforcing what we learned in class, this project allowed the students to be able to apply their knowledge to a potential real world scenario. Additionally, it demonstrated that we won't always get the right answer in our first run and sometimes multiple if not dozen attempts must be made.

5.2 Recommendations

In order to design an effective transmission gear box, students have to make sure to have a good understanding of the gear and shaft nomenclature. In addition, the student should develop skills in CAD drawing, ANSYS analysis, and MATLAB code before working on a project of this nature. Finally, several iterations need to be made to be able to select the best transmission design.

References

Budynas, R. G., Nisbett, J. K., & Shigley, J. E. (2015). *Shigley's mechanical engineering design*. New York, NY: McGraw-Hill Education.

SkyCiv. (n.d.). Free Online Beam Calculator for Cantilever or Simply Supported Beams. Retrieved November 25, 2019, from <https://skyciv.com/free-beam-calculator/>.

Appendix

MATLAB Code for Lowest Gear Calculations :

MATLAB Code Input

"friction coefficient"

$u=.5$

"incline 40% $\tan^{-1}(.40)$ "

$oo=21.801$

"tire diameter"

$td=25.9$

"weight"

$w=3300$

"friction force"

$Ff=u*(w*.928)$

"Force in x"

$Fx=Ff+(w*.3714)$

"tire radius"

$Tr=25.9/2$

"Torque"

$T=(Tr)*(Fx)$

"applied torque into feet"

$Ta=T/12$

"Final drive"

$fd=3.55$

"shaft transmission"

$St=Ta/fd$

"graph torque at 2500 RPM"

$gt=315$

"Lowest gear ratio"

$LGR=St/gt$

Machine2projectpt2

MATLAB Code Output

ans =

"friction coefficient"

u =

0.5000

ans =

"incline 40% tan-1(.40)"

oo =

21.8010

ans =

"tire diameter"

td =

25.9000

ans =

"weight"

w =

3300

ans =

"friction force"

Ff =

1532

ans =

"Force in x"

Fx =

2.755e+03

ans =

"tire radius"

Tr =

12.9500

ans =

"Torque"

T =

2.7575e+04

ans =

"applied torque into feet"

Ta =

2.9756e+03

ans =

"Final drive"

fd =

3.5500

ans =

"shaft transmission"

St =

831.8613

ans =

"graph torque at 2500 RPM"

gt =

315

ans =

"Lowest gear ratio"

LGR =

2.6515

Hand Calculation: 2.661

MATLAB Solution: 2.6515

MATLAB Code for Highest Gear calculations:

MATLAB Input

"Diameter in Inches"

$$D = 25.9$$

"Rotational Speed in RPM"

$$N_{in} = 2500$$

"Expected Speed MPH"

$$V_x = 85$$

$$\text{"Pi"} = 3.14$$

$$\text{Pi} = 3.14$$

"Distanced moved in 1 revolution in Inches"

$$\text{Dis} = \text{Pi} * D$$

"MPH to in/in"

"Miles/hour->feet/miles->inches/feet->hour/min"

"Mile in feet"

$$M = 5280$$

"Feet in inches"

$$f = 12$$

"Hour in minutes"

$$h = 60$$

"Distance traveled per minute at 85 MPH in inches per minute"

$$dm = ((V_x) * (M) * (f)) / h$$

"Tire RPM"

$$\text{Trpm} = dm / \text{Dis}$$

"Final Drive Ratio"

$$fd = 3.55$$

"RPM of transmission"

$$T = \text{Trpm} * fd$$

"Highest gear ratio is GR"

$$GR = N_{in} / T$$

MATLAB Output

```
> Machine2projectpt1
```

```
ans =
```

```
"Diameter in Inches"
```

```
D =
```

```
25.9000
```

```
ans =
```

```
"Rotational Speed in RPM"
```

```
Nin =
```

```
2500
```

```
ans =
```

```
"Expected Speed MPH"
```

```
Vx =
```

```
85
```

```
ans =
```

```
"Pi = 3.14"
```

```
Pi =
```

```
3.1400
```

```
ans =
```

```
"Distanced moved in 1 revolution in Inches"
```

```
Dis =
```

```
81.3260
```

```
ans =
```

```
"MPH to in/in"
```

```
ans =
```

```
"Miles/hour->feet/miles->inches/feet->hour/min"
```

```
ans =
```

```
"Mile in feet"
```

```
M =
```

5280

ans =

"Feet in inches"

f =

12

ans =

"Hour in minutes"

h =

60

ans =

"Distance traveled per minute at 85 MPH in inches per minute"

dm =

89760

ans =

"Tire RPM"

Trpm =

1.1037e+03

ans =

"Final Drive Ratio"

fd =

3.5500

ans =

"RPM of transmission"

T =

3.9182e+03

ans =

"Highest gear ratio is GR"

GR =

0.6381

```
>> Machine2projectpt1
```

```
ans =
```

```
"Diameter in Inches"
```

```
D =
```

```
25.9000
```

```
ans =
```

```
"Rotational Speed in RPM"
```

```
Nin =
```

```
2500
```

```
ans =
```

```
"Expected Speed MPH"
```

```
Vx =
```

```
85
```

```
ans =
```

```
"Pi = 3.14"
```

```
Pi =
```

```
3.1400
```

```
ans =
```

```
"Distanced moved in 1 revolution in Inches"
```

```
Dis =
```

```
81.3260
```

```
ans =
```

```
"MPH to in/in"
```

```
ans =
```

```
"Miles/hour->feet/miles->inches/feet->hour/min"
```

```
ans =
```

```
"Mile in feet"
```

```
M =
```

```
5280
```

ans =

"Feet in inches"

f =

12

ans =

"Hour in minutes"

h =

60

ans =

"Distance traveled per minute at 85 MPH in inches per minute"

dm =

89760

ans =

"Tire RPM"

Trpm =

1.1037e+03

ans =

"Final Drive Ratio"

fd =

3.5500

ans =

"RPM of transmission"

T =

3.9182e+03

ans =

"Highest gear ratio is GR"

GR =

0.6381

Hand Calculation: 0.6384

MATLAB Calculation: 0.6381

MATLAB Code for Gear Analysis

```
2 - LGR= 2.661
3
4 - HGR= .6384
5 - "safety factor"
6 - SF= 2.5
7 - "rpm"
8 - n= 5440
9 - "Horsepower"
10 - H= 307.5
11 - "LGR-HGR"
12 - diff=LGR-HGR
13 - "# of gears in between"
14 - m0=diff/5
15 - m6=.6384
16 - m5=0.6384+m0
17 - m4=1.04292+m0
18 - m3=1.44744+m0
19 - m2=1.8196+m0
20 - m1=2.25648+m0
21 - "1/m=Ng/Np"
22 - N6= 14
23 - Np=(1/0.6384)*14
24
25 - Ng1=m1*Np
26 - Ng2=m2*Np
27 - Ng3=m3*Np
28 - Ng4=m4*Np
29 - Ng5=m5*Np
30
31 - "round up to next tooth number"
32
33 - Ng1r=59
34 - Ng2r=50
35 - Ng3r=41
36 - Ng4r=32
37 - Ng5r=23
38 - Ng6r=22
39
40 - "Diametral Pitch (dp)"
```



```

53 - dp2=Ng2r/P
54 - dp1=Ng1r/P
55 -
56 - "n is 3.14"
57 - pi=3.14
58 - V1i=(pi*dp1*n)/12
59 - V2i=(pi*dp2*n)/12
60 - V3i=(pi*dp3*n)/12
61 - V4i=(pi*dp4*n)/12
62 - V5i=(pi*dp5*n)/12
63 - V6i=(pi*dp6*n)/12
64 -
65 - V1=(V1i*1000)
66 - V2=(V2i*1000)
67 - V3=(V3i*1000)
68 - V4=(V4i*1000)
69 - V5=(V5i*1000)
70 - V6=(V6i*1000)
71 -
72 -
73 - sig=(Sy/SF)
74 -
75 - Wt1=33000*(H/V1)
76 - Wt2=33000*(H/V2)
77 - Wt3=33000*(H/V3)
78 - Wt4=33000*(H/V4)
79 - Wt5=33000*(H/V5)
80 - Wt6=33000*(H/V6)
81 -
82 - Kv1=(1200+V1)/1200
83 - Kv2=(1200+V2)/1200
84 - Kv3=(1200+V3)/1200
85 - Kv4=(1200+V4)/1200
86 - Kv5=(1200+V5)/1200
87 - Kv6=(1200+V6)/1200
88 -
89 - "face width"
90 -
91 - F1=(Kv1*Wt1*P)/(sig*y)
92 - F2=(Kv2*Wt2*P)/(sig*y)
93 - F3=(Kv3*Wt3*P)/(sig*y)
94 - F4=(Kv4*Wt4*P)/(sig*y)
95 - F5=(Kv5*Wt5*P)/(sig*y)
96 - F6=(Kv6*Wt6*P)/(sig*y)
97 -

```

```
>> Machine2projectpt3copy
```

```
LGR =
```

```
2.6610
```

```
HGR =
```

```
0.6384
```

```
ans =
```

```
"safety factor"
```

```
SF =
```

```
1.5000
```

```
ans =
```

```
"rpm"
```

```
n =
```

```
5440
```

```
ans =
```

```
"Horsepower"
```

```
H =
```

```
307.5000
```

```
ans =
```

```
"LGR-HGR"
```

```
diff =
```

```
2.0226
```

```
ans =
```

```
"# of gears in between"
```

```
 $f_{z\ m0}$  =
```

```

m0 =
    0.4045

m6 =
    0.6354

m5 =
    1.0429

m4 =
    1.4474

m3 =
    1.8520

m2 =
    2.2241

m1 =
    2.6610

ans =
    "1/m=Nq/Np"

N6 =
    14

Np =
    21.9298

Nq1 =
    58.3553

Nq2 =
    45.7746

```

```

Ng2 =
    48.7746

Ng3 =
    40.6132

Ng4 =
    31.7421

Ng5 =
    22.8714

ans =

    "round up to next tooth number"

Ng1r =
    59

Ng2r =
    50

Ng3r =
    41

Ng4r =
    32

Ng5r =
    25

Ng6r =
    22

ans =

    "Diametral Pitch (dp)"

```

```
Command Window
ans =

    "Diametral Pitch (dp)"

P =

    6

ans =

    "For material AISI-1141QNT (Psi)"

Sy =

    165000

y =

    0.3310

dp6 =

    3.6667

dp5 =

    3.8333

dp4 =

    5.3333

dp3 =

    6.8333

dp2 =

    8.3333

dp1 =

    9.8333

ans =

    "n is 3.14"
```



```
Command Window

V4 =
    7.5918e+06

V5 =
    5.4566e+06

V6 =
    5.2194e+06

sig =
    66000

Wt1 =
    0.7250

Wt2 =
    0.8554

Wt3 =
    1.0432

Wt4 =
    1.3366

Wt5 =
    1.8597

Wt6 =
    1.9442

Ev1 =
    1.1666e+04

Ev2 =
    5.8862e+03
```

```
Command Window

9.8862e+03

Kv3 =

8.1069e+03

Kv4 =

6.3275e+03

Kv5 =

4.5482e+03

Kv6 =

4.3505e+03

ans =

"face width"

F1 =

2.3227

F2 =

2.3227

F3 =

2.3228

F4 =

2.3229

F5 =

2.3230

F6 =

2.3230

fx >> |
```

MATLAB Code for Shaft Design

"Material Description AISI 1095 HR Steel"

"Sut=120 Sy=66"

dp=3.6

n=5440

hp=307.05

"diameter assumptions in inches"

D=2.95

L=35

SF=2.5

pi=3.14

$V_f = (\pi * dp * n) / 12$

$W_t = (33000 * hp) / V_f$

"p=tan21.8=.3999"

$F_r = W_t * .3999$

"magnitude"

$F_m = ((W_t^2) + (F_r^2))^{(1/2)}$

"torque (T1=Tm)"

P=8

NG1=59

$D_1 = NG1 / 8$

$T_1 = W_t * (D_1 / 2)$

"Moments"

"XY Plane"

$F_y = (F_r * L) / (L / 2)$

$M_{b1} = F_r * (L / 2)$

"XZ Plane"

$M_{a1} = 0$

$M_{a2} = (W_t * L) / (L / 2)$

$$M_{b2} = (W_t * L) / 2$$

$$M_{c1} = 0$$

$$M_{c2} = 0$$

"Resultant moment $M_B = M_a$ Because critical location is B"

$$M_A = ((M_{a1}^2) + (M_{a2}^2))^{(1/2)}$$

$$M_B = ((M_{b1}^2) + (M_{b2}^2))^{(1/2)}$$

$$M_C = ((M_{c1}^2) + (M_{c2}^2))^{(1/2)}$$

"Calculations for Endurance Limit"

$$S_{ut} = 120$$

$$S_y = 66$$

$$S_{Ep} = 0.5 * S_{ut}$$

$$a = 11$$

$$b = -0.650$$

$$K_a = a * (S_{ut}^b)$$

$$K_b = .85$$

$$K_c = 1$$

$$K_d = 1$$

$$K_e = 1$$

$$S_e = K_a * K_b * K_c * K_d * K_e * S_{Ep}$$

" K_f and K_{fs} "

$$r = D / 2$$

"Notch radius assumption"

$$N_r = .1$$

"Figure 6-20/6-21"

$$q = 0.85$$

$$q_s = 0.9$$

"Table 7.1"

$$K_t = 2.7 \quad K_{ts} = 2.2 \quad K_f = 1 + (q * (K_t - 1))$$

$$K_{fs} = 1 + (q_s * (K_{ts} - 1))$$

"Because shaft has constant rotation"

$$M_m = 0$$

$$T_a = 0$$

$$M_a = M_B$$

$$T_m = T_1$$

"Diameter of shaft"

$$A_A = ((4 * (K_f * M_a)^2) + (3 * (K_{fs} * T_a)^2))^{1/2}$$

$$B_B = ((4 * (K_f * M_m)^2) + (3 * (K_{fs} * T_m)^2))^{1/2}$$

$$N = ((\pi * (D_1^3)) / 16) * ((A_A / S_e) + (B_B / S_{ut}))^{-1}$$

$$D_s = (((16 * N) / \pi)) * ((A_A / S_e) + (B_B / S_{ut}))^{1/3}$$

"Bearing Life hrs"

$$L_d = 20000$$

$$n_d = 2500$$

$$F_d = 342.01$$

$$C_{10} = F_d * (((L_d * n_d * 60) / 10^6)^{1/3}) \gg \text{Shaftmachinedesign}$$

ans =

"Material Description AISI 1095 HR Steel"

ans =

$$S_{ut} = 120 \quad S_y = 66$$

$$d_p = 3.6000$$

$$n = 5440$$

$$h_p = 307.0500$$

ans =

"diameter assumptions in inches"

$$D = 2.9500$$

$$L = 35$$

$$SF = 1.5000$$

$$\pi = 3.1400$$

$$V_f = 5.1245e+03$$

$$W_t = 1.9773e+03$$

ans =

$$p = \tan 21.8 = .3999$$

$$F_r = 790.7235$$

ans =

"magnitude"

$$F_m = 2.1295e+03$$

ans =

"torque (T1=Tm)"

$$P = 8$$

$$NG_1 = 59$$

$$D_1 = 7.3750$$

$$T_1 = 7.2913e+03$$

ans =

"Moments"

ans =

"XY Plane"

$$F_y = 1.5814e+03$$

$$M_{b1} = 1.3838e+04$$

ans =

"XZ Plane"

$$M_{a1} = 0$$

$$M_{a2} = 3.9546e+03$$

$$M_{b2} = 3.4603e+04$$

$$M_{c1} = 0$$

$$M_{c2} = 0$$

ans =

"Resultant moment MB=Ma Because critical location is B"

$$M_A = 3.9546e+03$$

$$MB = 3.7267e+04$$

$$MC = 0$$

ans =

"Calculations for Endurance Limit"

$$S_{ut} = 120$$

$$S_y = 66$$

$$SE_p = 60$$

$$a = 11$$

$$b = -0.6500$$

$$K_a = 0.4897$$

$$K_b = 0.8500$$

$$K_c = 1$$

$$K_d = 1$$

$$K_e = 1$$

$$S_e = 24.9744$$

ans =

"Kf and Kfs"

$$r = 3.6875$$

ans =

"Notch radius assumption"

$$N_r = 0.1000$$

ans =

"Figure 6-20/6-21"

$$q = 0.8500$$

$$q_s = 0.9000$$

ans =

"Table 7.1"

$$K_t = 2.7000$$

$$K_{ts} = 2.2000$$

$$K_f = 2.4450$$

$$K_{fs} = 2.0800$$

ans =

"Because shaft has constant rotation"

$$M_m = 0$$

$$T_a = 0$$

$$M_a = 3.7267e+04$$

$$T_m = 7.2913e+03$$

ans =

"Diameter of shaft"

$$A_A = 1.8224e+05$$

$$B_B = 2.6268e+04$$

$$N = 0.0105$$

$$D_s = 1.0454$$

ans =

"Bearing Life hrs"

$$L_d = 20000$$

$$n_d = 2500$$

$$F_d = 342.0100$$

$$C_{10} = 4.9326e+03$$





The report contains all necessary values 10/10

All calculation steps follow correct procedure 10/10

The report contains design iteration 8/10

The values are reasonable and feasible 8/10

Formatting look professional 8/10

CAD drawing of all components are included 10/10

2D engineering drawings are included 8/10

FEA is included 5/10

Enough description is provided 7/10

Participation 15/15

89/95

Average

**MECHANICAL ENGINEERING DEPARTMENT
PRAIRIE VIEW A&M UNIVERSITY**



**MCEG 4043-Y02 MACHINE DESIGN 2
FALL 2020**

Report on
**“Gear and Driveshaft Design for automotive powertrain
system”**

Submitted by
Tyler Marcel, Habeeb Ayangbade, and Asher Carson

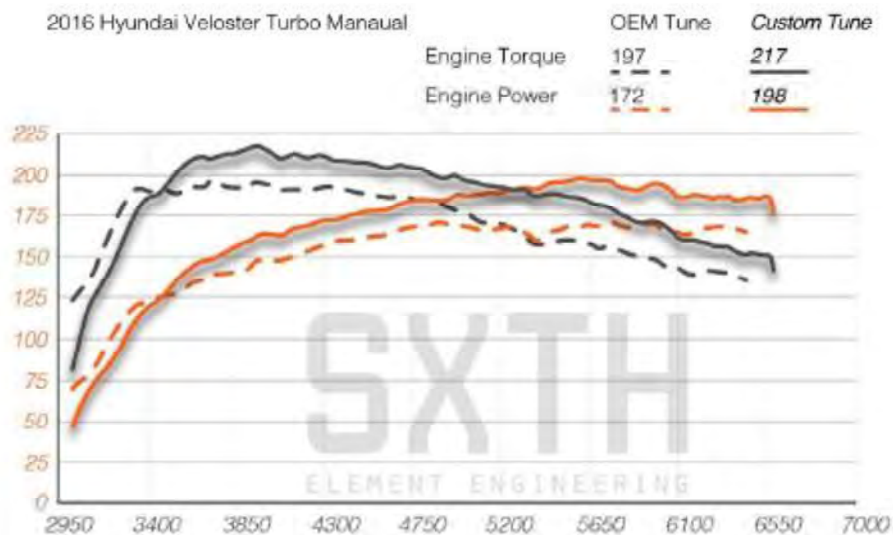
On
November 23, 2020
To
Dr. Jaejong Park
Machine Design 2 Instructor

Asher Carson	Asher Carson
Habeeb Ayangbade	Habeeb Ayangbade
Tyler Marcel	Tyler Marcel

By signing above, we declare that the contents within this report were generated solely by us. We understand the consequences of academic misconduct.

Abstract

In this report, the group will be detailing the process of calculating and designing the geartrain and components of the 2016 Hyundai Veloster. The group received the custom values of the vehicle, where the calculations will be based off. The project is separated into several assignments, starting with calculating the highest and lowest gear ratio. After that, the group then calculated the sizes of the gear using force analysis and lastly calculating the propeller shaft diameter and length. After calculating these, each component is designed and compiled together in an assembly. This project created an in-depth process on how to design the geartrain system. Throughout this project, the group had to manufacture a viable design based on the numbers they calculated, which included many assumptions like the size of components and those materials. MATLAB is the software that was used to do the calculations, and NX was used to design the individual components. MATLAB was used to check the hand calculation or supplement it in some situations. This report shows the groups process overall and our final designs for the gear assembly.



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Summary of Comments on group6_LATE_2564_843514_MCEG 4043 Project Final Report-1.pdf

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1. Introduction.

1.1 Background

Assigned a 2016 Hyundai Veloster Turbo, a brief description of the manual transmission's background is provided. A manual transmission is where the driver of a multi-speed vehicle is required to manually select the gears by maneuvering a clutch and gear stick. In the late 19th to early 20th century many automobiles used a non-synchronous design where the driver had the responsibility to control the timing and throttle when shifting in order to get the gears rotating at the same speed. This was paramount because the gears wouldn't mesh if done incorrectly, this task was somewhat difficult to do so many drivers would encounter issues and hear the gears making a rough, grinding sound. Over the next few decades most cars would switch over to the synchronous design and by the early 1950's it was widespread across the engineering world.

The function of the transmission is to transfer engine power to the propeller shaft and rear wheels. Depending on the engine speed and torque the gears inside the transmission alter the drive-wheel torque and speed. Vehicles with lower gear ratios serve as a torque multiplier and enable the engine to acquire enough power to accelerate from a halt. The design of a system to transmit power requires many steps that need to be addressed such as, the vehicles power and torque requirements, gear specification, shaft layout, force analysis, material selection, shaft design for stress and deflection, bearing selection, and key selection.

1.2 Problem definition

The overall objective of this report is to show detail about the design of a multi-speed gearbox including components such as the shafts, gears, and rolling element bearings. A Hyundai Veloster with three forward and one reverse speed is the focus of this project. Vehicle characteristics are peak engine power is 198 Hp, peak engine torque is 217 lb-ft, vehicle weight is 2900 lb, tire size; 225/40/R18, final drive: 4.64. Constraints to take into consideration are 100 mph should take place at 3500 RPM (highest gear) and initial effort to climb 40% grade hill is made at 3000 RPM, shafts need to have infinite life, bearings are to have at least 20,000 hours of life, and minimum safety factor is 2.5 for all components being designed.

1.3 Strategy

This is a long process that should be planned out in order to strategize the procedure. First to determine overall sizing demands for the system one must determine the power and torque requirements. Now the selection of suitable gears can be completed to find gear ratios. Now the overall layout of the shaft must be specified, then the force analysis, shear and moment diagrams can be produced for the shafts. Taking fatigue into consideration the material selection process can be next. With the overall layout

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Interesting information. Better have a reference though.

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incomplete sentence.

determined now a deeper design for the shafts can be made taking stress and deflection into concern. With shaft diameters now being estimated a bearing catalog can be used for selection. Finally appropriate selection for keys can be specified.

1.4 Project planning

The project can be broken up into four sections, each one consisting of numerous steps from the strategy section above. First a hand calculation of ratio for the highest gear. Given the parameters (engine speed, final drive, tire size). Also writing a computer code in MATLAB that correlates the engine speed and ground speed based on our hand calculation.

After finding the ratio for the highest gear another hand calculation for the ratio for the lowest gear is completed. The project calls for driving conditions (hill grade, vehicle weight, tire size) when doing this computation. Another MATLAB code was written that correlates the engine speed and the torque on the propeller shaft.

Next the size analysis on all the gears which is based on an assumed diametral pitch. The number of teeth, pitch radii was also computed. Another MATLAB code was written that provides diameter and radii given gear ratio and diametral pitch. Also force analysis on all of the gears, then use those force values to choose the face width and an adequate material for the gears.

Last is finding the diameter of the propeller shaft. First choose a material for the shaft and choose the length of the propeller shaft. Bearing catalog load rating (C10) is then calculated and a suitable key will be designed last. CAD models of all the components will be designed as well.

2. Executive Summary

2.1 Gearbox

First we calculate the number of teeth on the pinion and gears using the following equations:

$$N_p = \frac{2k}{3\sin^2\phi} \left[1 + \sqrt{1 + 3\sin^2\phi} \right] N_{G1,G2,G3} = (N_p) \text{ (Corresponding gear ratio)}$$

To calculate the d (pitch diameter) of each gear the following equation is used:

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

To calculate the F (Face width) of each gear the following equation is used:

$$F = \frac{K_v W_t P}{\sigma_Y} \text{ where } W_t = 33,000 \frac{hp}{V}, K_v = \frac{1200+V}{1200}, V = \frac{\pi d n}{12}$$

To find the σ (Max Bending) the following equation is used:

$$\sigma = \frac{K_v W_t P}{F Y}$$

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The purpose of this chapter is to showcase your final numbers. The procedure you took to get the values should be in chapter 3.

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where does this parenthesis close?

Gear number		N	d		Max $\sigma_{bending}$	Material	Safety factor
Pinion		25	1.79 in	2.99 in	94.14 Mpa	AISI 1030	3.62
Forward gears	1	97	6.93 in	1.76 in	93.99 Mpa	AISI 1030	3.63
	2	55	3.93 in	2.04 in	93.96 Mpa	AISI 1030	3.63
	3	14	1 in	4.61 in	93.89 Mpa	AISI 1030	3.64

The results from the table above are fulfilled from the equations stated. Due to the project requiring a design constraint of 2.5 safety factor all the gears are well within range with the lowest being a factor of 3.62. Choosing a material with a high enough yield strength allows for enough flexibility to withstand the stresses induced from the engine.

2.2 Propeller Shaft

Here the following equation is used to find the shaft diameter:

$$d = \left\{ \frac{16n}{\pi} \left(\frac{2(K_f M_a)}{S_e} + \frac{[3(K_{fs} T_m)^2]^{1/2}}{S_{ut}} \right) \right\}^{1/3}$$

To find the torque in the shaft use equation:

$$T = W_t (d_{G1}/2)$$

For the Max $\tau_{Torsion}$ use the equation:

$$\tau = Tr/J$$

Shaft diameter	Max torque	Material	Max $\tau_{torsion}$	2 failure criteria	Safety factor
3.086 in	35.61 lb-ft	AISI 4340	144.77 psi	4 bending 5 moment on shaft x-z plane	1.5

Since gear 1 had the largest radius then it will apply the greatest amount of torque and force onto the shaft. With that in mind the forces from other gears don't need to be taken into consideration. However after calculating the reaction forces on the shaft the shear and moment diagram can be formulated in order to find the critical point, which in turn is the mid-section where the bearing is on the shaft. This since the bending moment is largest at this point.

2.3 Shaft components

2.3.1 Key

Shaft diameter	Key size			Material	Safety factor
	w	h	l		
3.086	3/4 in	3/4 in	2 in	AISI 1045 HR	1.5

2.3.2 Bearing

Bearing	8 radial load	Axial load	Catalog load rating	Catalog	Item number
Ball Bearing			4,067 lb	AST	R22

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The face should be the same for any meshing gear pair.				
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There is no such failure criteria.				
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This value seems to be way too small.				
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I do not see your calculation steps for key and bearing.				
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	Number: 8	Author: japark	Subject: Highlight	Date: 12/2/2020 4:41:54 PM
values are missing.				

3. Design Methodology

3.1 Gear design methodology

3.1.1 Gear ratios

Here we begin to calculate the highest and lowest gear ratio

Highest gear ratio (lowest gear)

Given: vehicle weight: 2900 lb, Tire size: 225/40/R18, Final drive: 4.64, 100 mph occurs at 3500 RPM

Tire circumference Aspect ratio = 40

Side wall length $\frac{225mm \times 40}{100} = 90mm = 3.54 \text{ in}$

Tire diameter $(3.54 \text{ in} \times 2) + 18 \text{ in} = 25.08 \text{ in}$

Tire circumference $\pi \times 25.08 \text{ in} = 78.75 \text{ in}$

RPM of Drive Axle

$$\frac{\text{rev}}{\text{min}} \times 25.08 \text{ in} \times \frac{1 \text{ mile}}{5280 \text{ ft}} \times \frac{1 \text{ ft}}{12 \text{ in}} \times 60 = 100 \text{ mph}$$

$$\frac{\text{rev}}{\text{min}} = \frac{100 \text{ mph} \times 5280 \text{ ft} \times 12 \text{ in}}{60 \times 78.75 \text{ in}} = 1340.95 \text{ rev/min}$$

RPM Propeller Shaft

$$4.64 \times 1340.95 \text{ rev/min} = 6222.008 \text{ rev/min}$$

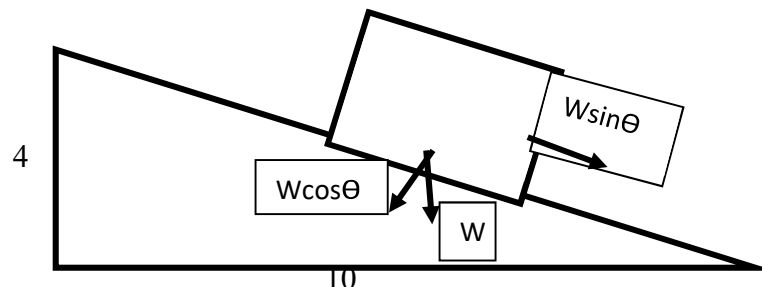
Highest Gear Ratio

$$\frac{3500 \text{ RPM}}{6222.008 \text{ RPM}} = 0.562$$

Lowest gear ratio (highest gear)

Given: Initial effort to climb 40% grade hill occurs at 2000 RPM

Coefficient of friction of tires can be assumed to be 0.5



$$\tan \Theta = 4/10 \quad \tan \Theta = 21.8^\circ$$

Vehicle weight = 2900 lb

$$F - W \sin \Theta - 0.5F = 0$$

$$0.5F = 2900 \text{ lb} \times \sin(21.8) \quad 0.5F = 1076.97$$

$$F = 2153.94 \text{ lb}$$

$$T = F \times R$$

$$R = 25.08/2 = 12.54 \text{ in}$$

$$T = 2153.94 (12.54/12) = 2250.867 \text{ lb}\cdot\text{ft}$$

$$T_p = T_G \times \frac{1}{\text{Final Drive}} \quad T_p = 2250.867 \times \frac{1}{4.64} = 485.101 \text{ lb}\cdot\text{ft}$$

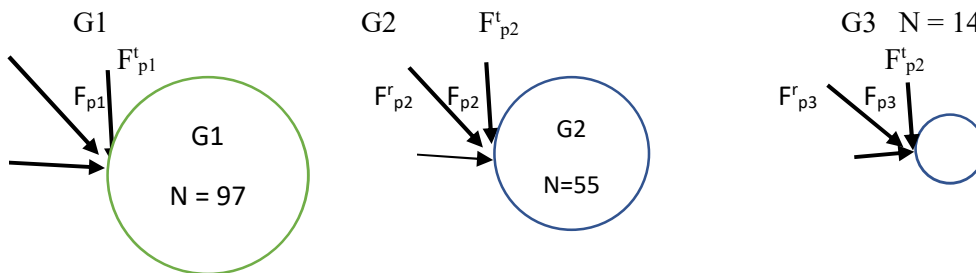
$$m = \frac{T_G}{T_p} \quad \text{For engine RPM} = 3000 \quad \text{Torque} = 125 \text{ lb}\cdot\text{ft}$$

$$m = \frac{485.101 \text{ lb}\cdot\text{ft}}{125 \text{ lb}\cdot\text{ft}} = 3.88$$

3.1.2 Gear size and force analysis

Here we are calculating the face width and diameter of each of the gears.

Free-body diagram of the gears



To find the forces on the gears the transmitted load W_t is found with this equation:

$$W_t = 33,000 \frac{H}{V} \quad \text{where, } V = \pi d n / 12$$

$$W_t = \frac{33,000(198 \text{ hp})}{\pi(10.9373 \text{ in})(5550 \text{ rpm})} = 34.26 \text{ lb} \cdot \text{ft} \quad \text{This is also equal to the tangential load } F^t$$

For the radial loads the equation $F^r = F^t \tan 20^\circ$

For the normal loads the equation $F^n = F^t / \cos 20^\circ$

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What is this?

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Clearly these gears are not in static equilibrium.

3.1.2.4 Gear sizes

In order to calculate the diameter of each gear, the group selected a pitch diameter that then multiplied it by the number of teeth to find the diameter

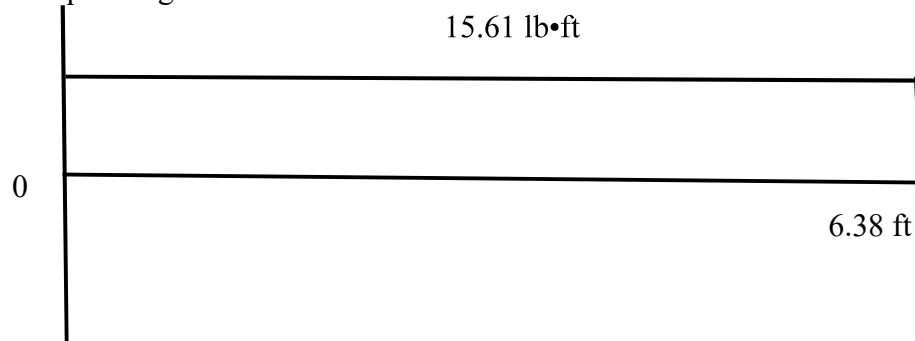
Gears		Gear ratio	Diameter
Forward gears	1	3.88	6.93 in
	2	2.20	3.93 in
	3	0.562	1 in

3.2 Shaft design methodology

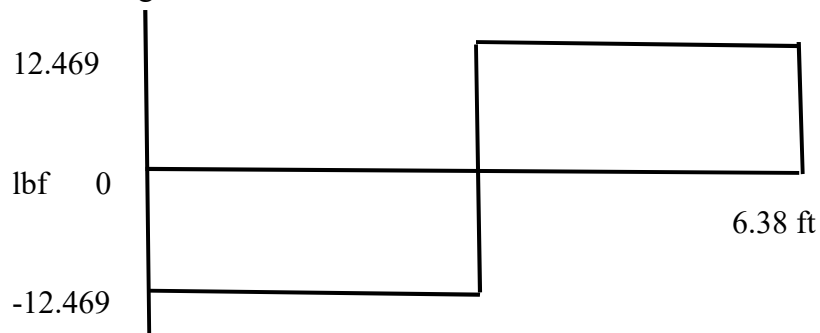
When designing the shaft the primary load must be taken into consideration. Since the propeller shaft rotates, torsion will be a main focus. Since torque is force x distance in our case the gear with the largest radius (1st gear) will have the longest distance thus providing the most torque on the shaft.

3.2.1 Shaft shear, moment & torque diagram (If necessary)

Torque Diagram



Shear Diagram X-Y Plane





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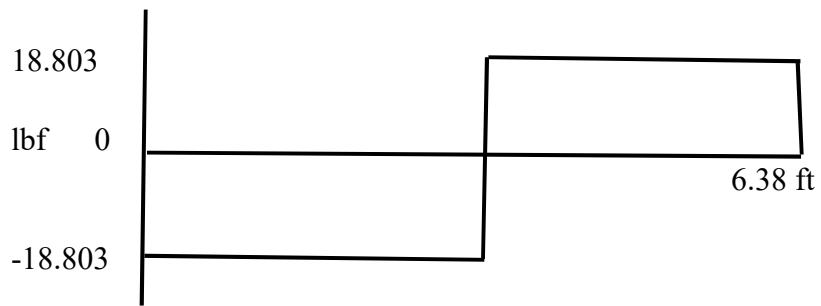
so you decided to go with 1 bearing setup. This will result in a unstable solution since the transmitted loads will go in the opposite direction which will result in bending issues.



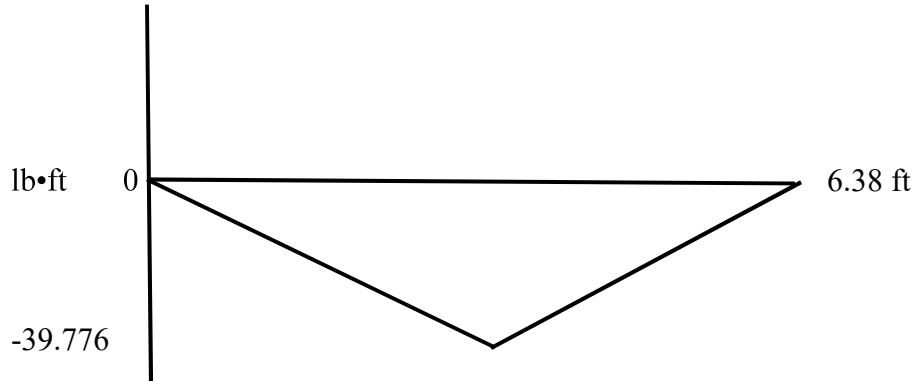
Number: 2 Author: japark Subject: Sticky Note Date: 12/2/2020 4:48:42 PM

Where did your calculation for S_e , K_f , and K_{fs} ? which failure criteria are you using to compute your diameter?

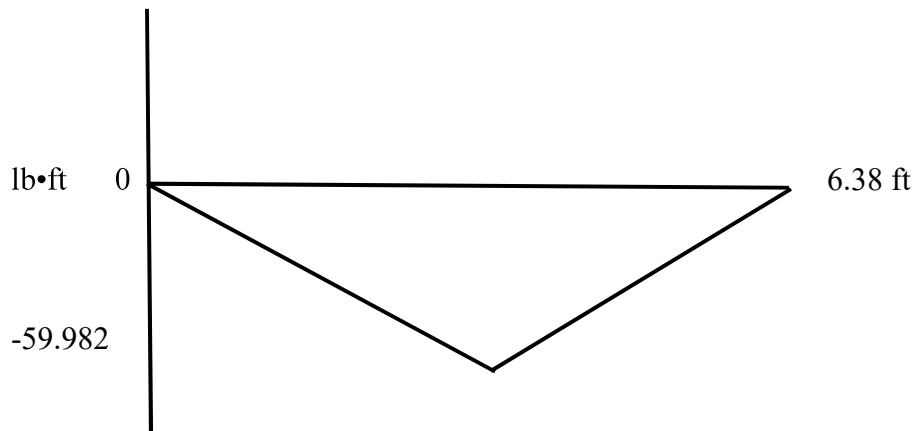
Shear Diagram X-Z Plane



Moment Diagram X-Y Plane

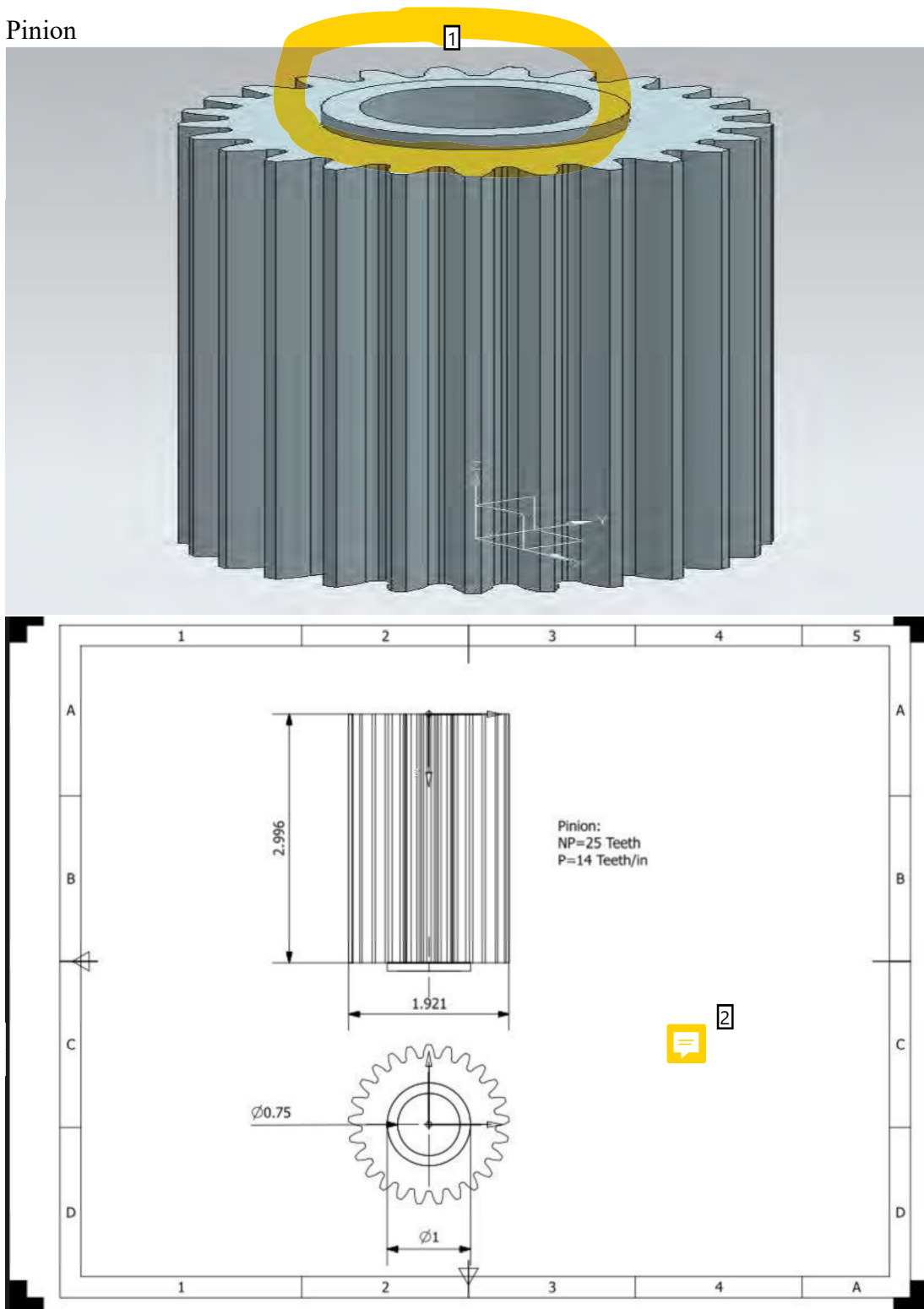


Moment Diagram X-Z Plane



CAD models

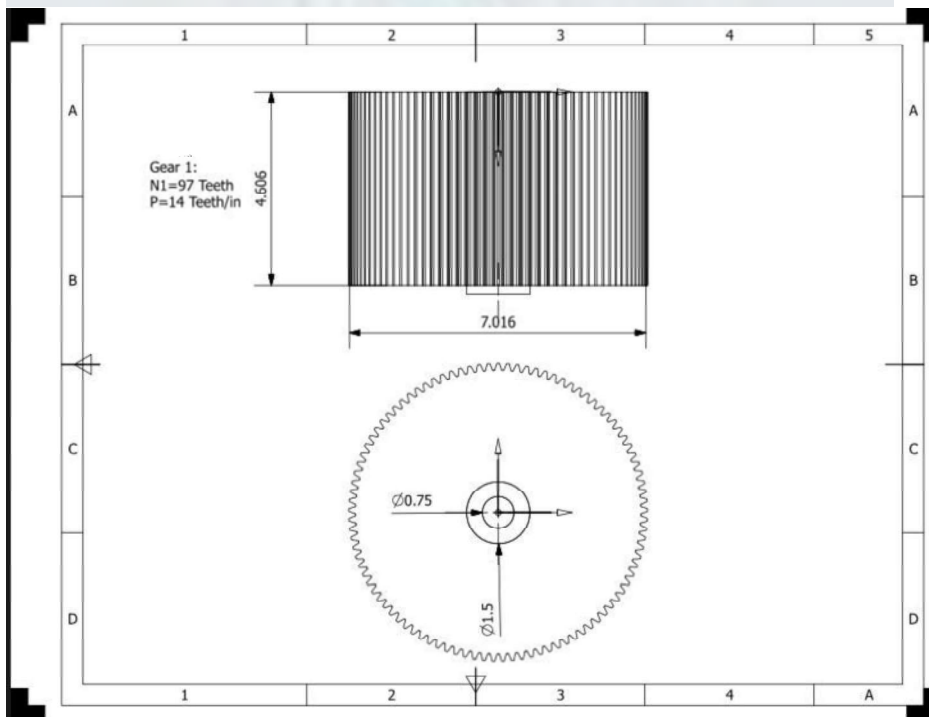
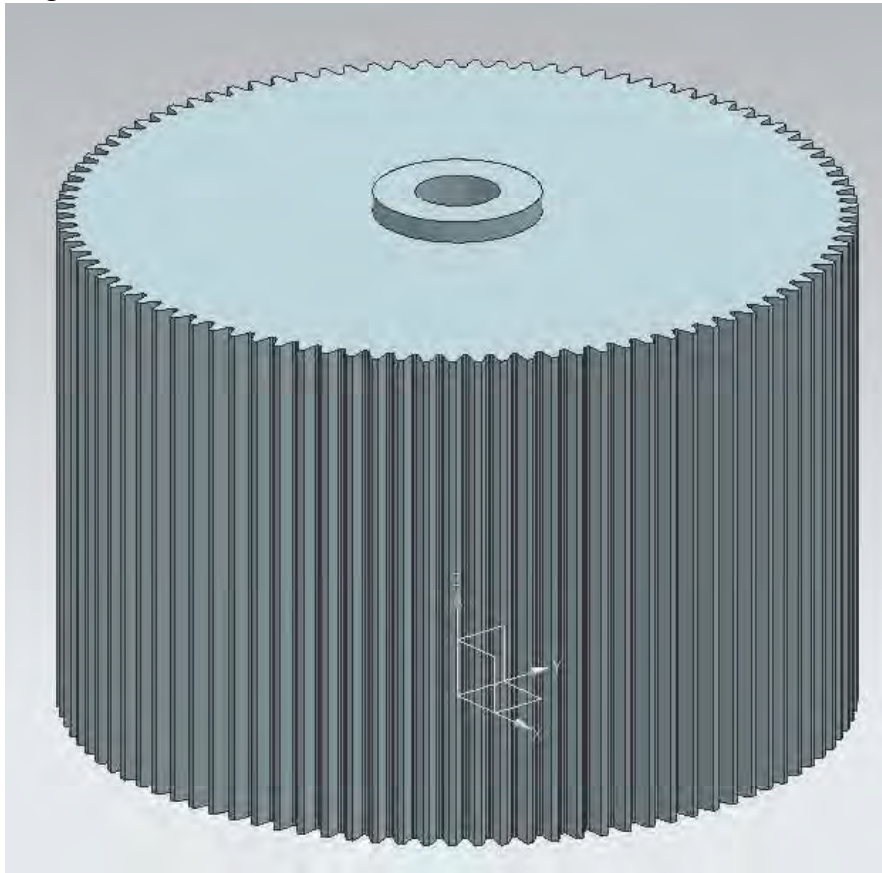
3.3 Pinion



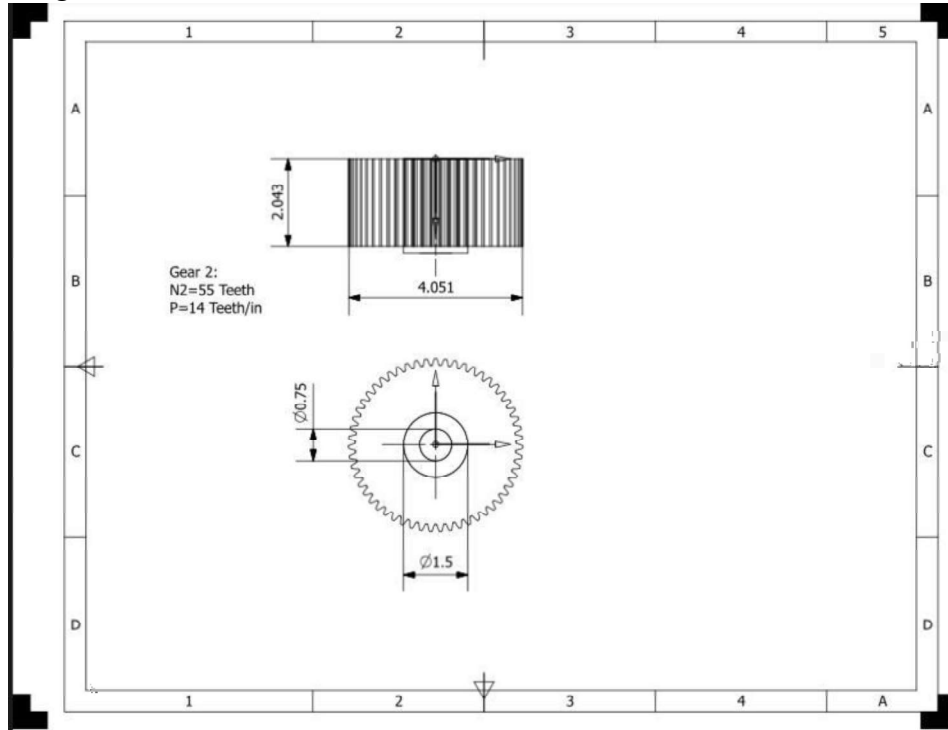
 Number: 1 Author: japark Subject: Highlight Date: 12/2/2020 4:45:30 PM
Why this is not flush with the rest of the gear?

 Number: 2 Author: japark Subject: Sticky Note Date: 12/2/2020 4:45:46 PM
What about dimensions for key?

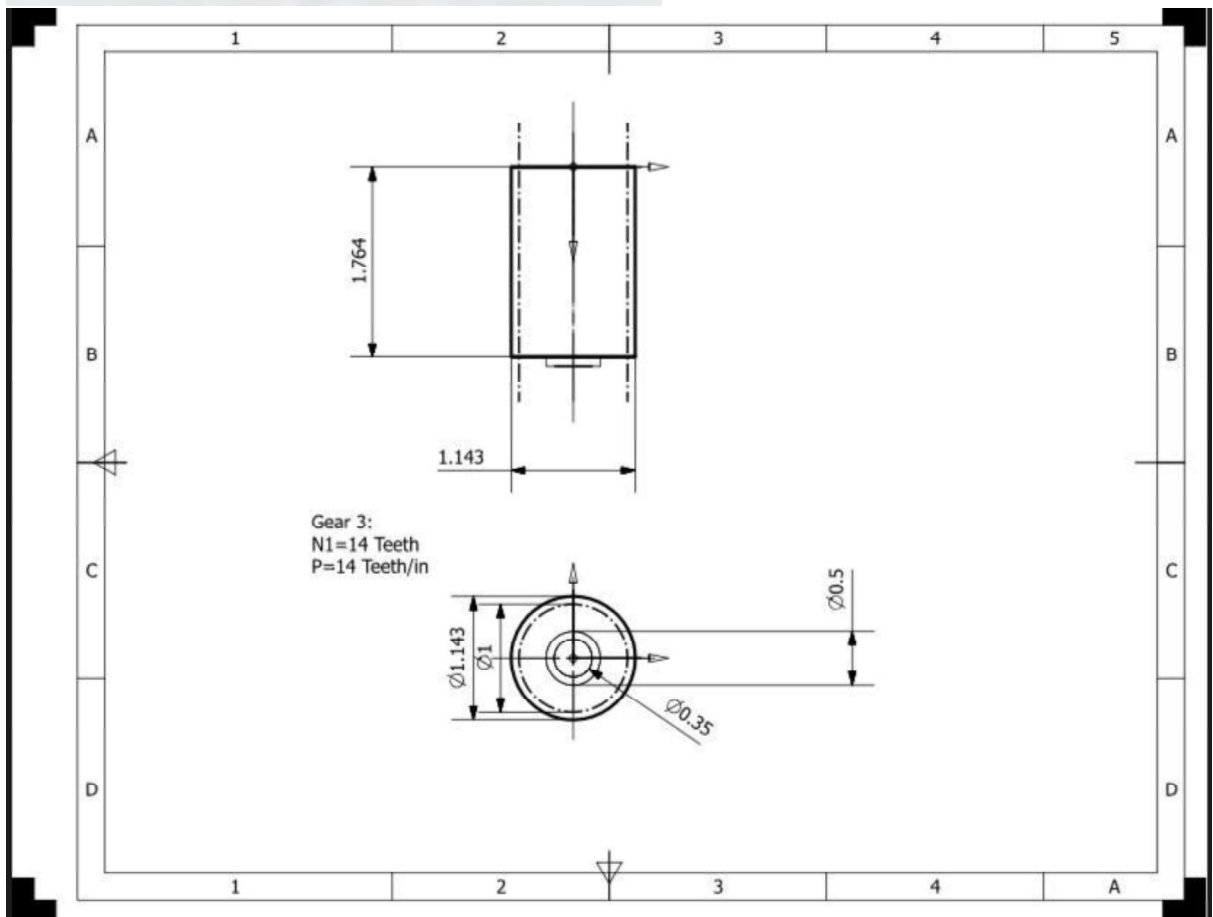
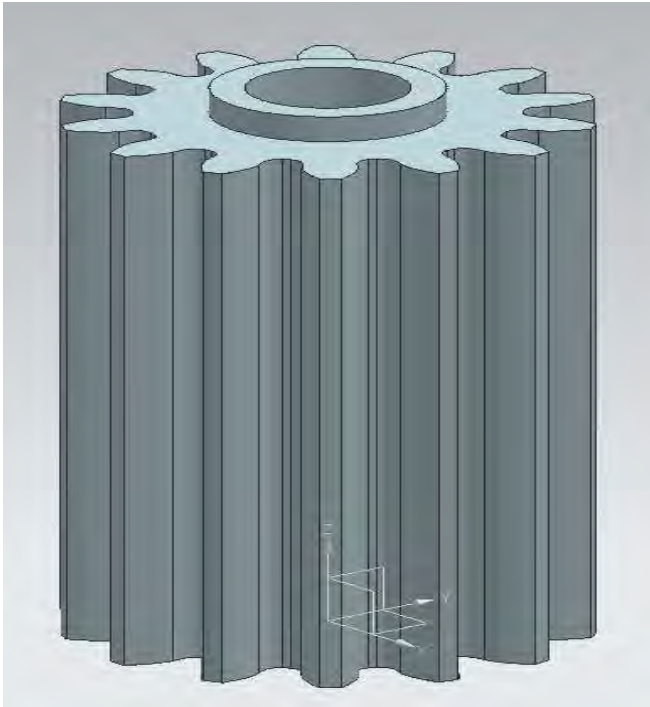
3.4 1st gear



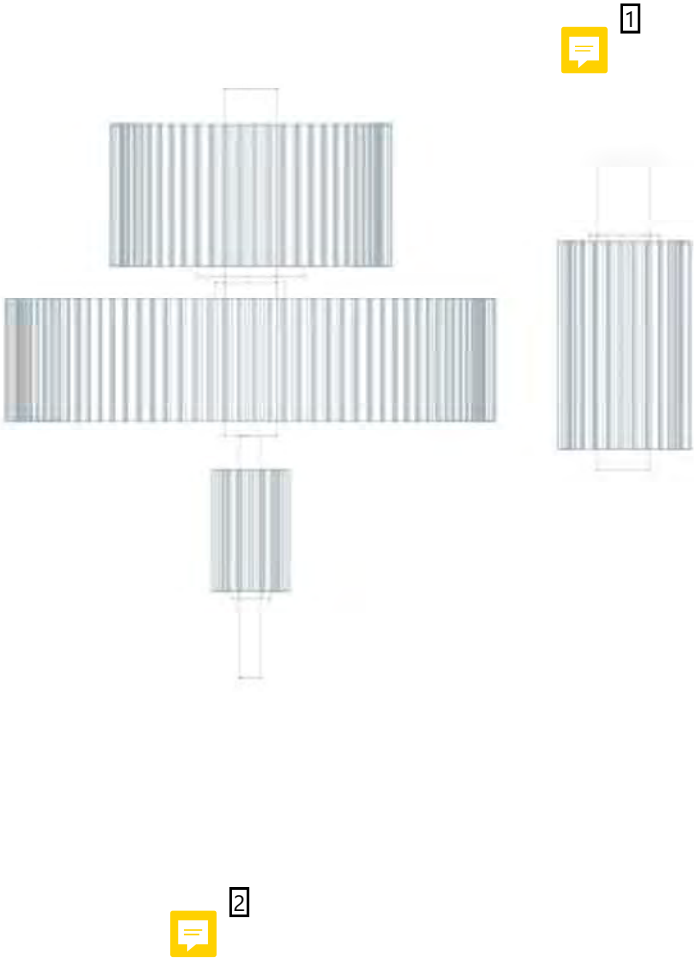
3.5 2nd gear



3.6 3rd gear



CAD Assembly



 Number: 1 Author: japark Subject: Sticky Note Date: 12/2/2020 4:46:52 PM
Here you see the problem associated with gears with different face widths

 Number: 2 Author: japark Subject: Sticky Note Date: 12/2/2020 4:47:06 PM
Why empty space here?

4. Conclusion

This report went through a detailed process on the calculation and designing of a geartrain for a 2016 Hyundai Veloster. Throughout the project our group was intrigued by the assignments and tasks required. This real world example provided valuable design experience including hand calculations, coding, CAD modeling, and it touched on all facets of the design process. However there were some complexities with certain tasks. Our group spent a lot of time on the MATLAB codes, this was due to our unfamiliarity with the software. But having the ability to supplement MATLAB for the calculations was a very useful tool in most cases of this project, thus becoming an important ability for future assignments or jobs. The NX portions for the individual components brought forth the task of creativity especially with the assembly process. Overall this project provided our group with the ability to utilize the knowledge acquired in this our MCEG 4043 course and we gained significant design insight.

References

- 1) Budynas, R. G., Nisbett, J. K., & Shigley, J. E. (2016). Shigley's Mechanical Engineering Design. New York: McGraw Hill.
- 2) Kohara Gears. (n.d.) *Calculation of Gear Dimensions*. KHK Stock Gears.
https://khkgears.net/new/gear_knowledge/gear_technical_reference/calculation_gear_dimensions.html

- 3) CASY-Center for Research on Complex Automated Systems. (2006, November 10). *Modelling of an automated manual transmission system*. Science Direct.
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- 4) American Roller Bearings. (n.d.) *Bearing Selection, Load, & Life*. AM Roll.
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- 5) AST Bearings. (n.d.) *R Series Ball Bearing*. AST Ball Bearings.
https://www.astbearings.com/catalog/precision_r_series/R22

Appendix

Engine and ground speed correlation using MATLAB

```
11/22/20 2:27 PM C:\U...\HighestGearCalculationRevised.m 1 of 1

%% Highest Gear Calculations
% Given
VehicleWeight=3900;
FinalDrive=4.64;
HighestGearRatio=0.562;
% The given tire size is 225/40/R18
RimSize=18;
AspectRatio=40;
%% Calculations
%Calculate tire specifications
TireWidth_mm=(225*AspectRatio)/100;
TireWidth_in=TireWidth_mm/25.4;
% Converting the Tire width from mm to in
TireDiameter_in=(TireWidth_in*2)+RimSize;
TireCircumference_in=TireDiameter_in*pi;
%Calculate Speed
RPM_speed=input('Enter engine RPMs: ');
% Allows the user to input the rpms of the engine
PropellerShaft_RPM=RPM_speed/HighestGearRatio;
DriveShaft_RPM=PropellerShaft_RPM/FinalDrive;
GroundSpeed_RPM=DriveShaft_RPM*TireCircumference_in;
GroundSpeed_MPH=(GroundSpeed_RPM*60)*(1/5280)*(1/12);
```

Engine speed and the torque on propeller shaft correlation using MATLAB

11/22/20 2:28 PM C:\Users\ash...\LowestGearCalculation.m 1 of 1

```
% Lowest Gear Calculations
% Given
W_lb=2900;
Final_Drive=4.64;
% The given tire size is 225/40/R18
% 40 grade hill means you have a 4-10 right triangle
theta=atand(4/10);
%% Force Calculation
%The force pulling the car back down the hill would be the Y-component of
%the weight
F_x_lb=(W_lb)*(cosd(theta));
F_y_lb=(W_lb)*(sind(theta));
%Coefficient of friction is assumed to be 0.5
F_lb=(F_y_lb)/0.5;
%% Torque Calculation
%Tire diameter was found was using tiresize.com/calculator/
%Torque is calculated by multiplying the force and the radius
d_T_in=25.08;
r_T_ft=(d_T_in)/2;
T_g=(F_lb)*(r_T_ft)*(1/12);
%T_g torque of the gear
%% Gear Ratio Calculation
%T_p=Torque of the pinion
%At the given engine speed of 3000 rpms is 125 lb*ft
T_p=T_g*(1/Final_Drive);
T_3000=125;
m=T_p/T_3000;
```

Provides diameter and radii upon given necessary inputs (gear ratio, diametral pitch)


```

%% Gear Size Calculation
%User Input and Assumptions
HGR=input("Enter the Highest Gear Ratio: ");
LGR=input("Enter the Lowest Gear Ratio: ");
theta=input("Enter the assumed theta: ");
%The k value is assumed to be 1
k=1;
%% Calculations for 3rd Gear
% Taking the inverse of the highest gear
m_1=(HGR)^-1;
% ratio in order to find the teeth
N_p0=((2*k)/((1+2*(m_1)*(sind(theta))^2))*(m_1)+sqrt(((m_1)^2)+(1+2*(m_1)*(sind(theta))^2)));
N_g3=round(N_p0);
% N_g3 is the smallest number of teeth for the third gear
N_px=N_g3*m_1;
N_p=round(N_px);
% N_p is the number of teeth for the pinion
%% Calculations for the additional gears
%The gear ratio for the second gear should be between the highest and
slowest gear ratio
m_2=input("Enter the gear ratio for the second gear: ");
N_g1=N_p*LGR;
N_g2=N_p*m_2;
%% Calculate the diameters of each gear
P_d=input("Enter the diametral pitch: ");
d_p=N_p/P_d;
d_1=N_g1/P_d;
d_2=N_g2/P_d;
d_3=N_g3/P_d;
%% Calculate the values for the face width
n=input("What are the engine rpms: ");
H=input("What is the horsepower of the engine at the assumed rpms: ");
V_p=(pi*d_p*n)/12;
V_1=(pi*d_1*n)/12;
V_2=(pi*d_2*n)/12;
V_3=(pi*d_3*n)/12;
W_tp=(33000*((H)/(V_p)))/1000;
W_t1=(33000*((H)/(V_1)))/1000;
W_t2=(33000*((H)/(V_2)))/1000;
W_t3=(33000*((H)/(V_3)))/1000;
K_vp=(1200+V_p)/1200;
K_v1=(1200+V_1)/1200;
K_v2=(1200+V_2)/1200;
K_v3=(1200+V_3)/1200;
%% Calculate the face width of each gear
S=input("Based on the material, what is the yield strength in kpsi: ");
Y=input("What is the Y value for the pinion: ");
F_p=(K_vp*W_tp*P_d)/(S*Y);

```

```

Y_1=input("What is the Y value for gear 1: ");
F_1=(K_v1*W_t1*P_d)/(S*Y_1);
Y_2=input("What is the Y value for gear 2: ");
F_2=(K_v2*W_t2*P_d)/(S*Y_2);
Y_3=input("What is the Y value for gear 3: ");
F_3=(K_v3*W_t3*P_d)/(S*Y_3);

```



1



The report contains all necessary values 8/10

All calculation steps follow correct procedure 8/10

The report contains design iteration 5/10

The values are reasonable and feasible 8/10

Formatting look professional 8/10

CAD drawing of all components are included 10/10

2D engineering drawings are included 8/10

FEA is included 0/10

Enough description is provided 7/10

Participation 15/15

77/95

Below Average

Prairie View A&M University

Mechanical Engineering



MCEG 4043 – Z01K

Machine Design II

Fall 2020

FINAL REPORT

Transmission Design

Submitted by

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Priscille Djomo	<i>Priscille Djomo</i>
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Darryl A	<i>Darryl A</i>

to

Dr. Jaejong Park

Summary of Comments on group3_LATE_3101_843580_Project Final Report Guidelines.pdf

Page: 1



Number: 1
Section X?

Author: japark

Subject: Highlight

Date: 12/2/2020 1:40:42 PM

By signing above, we declare that the contents within this report were generated solely by us. We understand the consequences of academic misconduct.

Abstract

The project was asked to build a transmission six speed gearbox for AUDI A4 using the Junitornic graphs provided by the teacher. The plan along the project is to find all the values that involve the forward gear, the reverse gear, the propeller shaft, the bearing design using the bearing catalog load rating, and the shaft design. We chose 1040 Q&T material that has $S_y = 593\text{KSI}(86\text{MPa})$ and its other properties are found in table 20 of the book. We know that in order to do that, we had to use all the fundamentals equations that were provided to us during the semester. So we have to first find the number of teeth of both the gear and the pinion, then find the gear ratio (m) for both the highest gear and the lowest gear and simultaneously find the interference using the formula. Then we have to find the face width. Also, we had to find the torque using an inclined plane where we supposed that the car is going downhill, we also had to calculate the power in Hp and we will do the forces analysis, we had to write a MATLAB code. We have to provide the CAD of the gears that contributed to realize the project.

 Number: 1 Author: japark Subject: Highlight Date: 12/2/2020 1:24:12 PM
This should be in the first page.

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1. Introduction

1.1 Background:

Throughout this project in order to get a much more thorough understanding of our design components we can refer to data sheet to retrieve our values for power, torque, vehicle weight, final drive ratio, engine characteristics, wheel size, and tire size. In our gear and drive shaft analysis, we were required to create a transmission with a minimum safety factor of 2.5 as well as include 6 forward and 1 reverse speed value while disregarding the Gear surface wear. Throughout free body and design analysis we will be able to analyze our vehicle in a grade hill analysis with climb up a 40% grade hill starting from the specified Rotations per minute. Based on our previous design constraints listed, the bearings for our vehicle are expected to have a minimum life of 20,000 hours and an infinite life based on the failure criterion. Also, our group will provide at least three design inputs in order to improve the entire shaft design.

1.2 Problem definition

Our project for this Machine Design 2 course was to design a gear and driveshaft for an automotive powertrain system for a compact sedan-2017 Audi A4. For ICE powered motor vehicles the Transmission design is one of the most essential components of the vehicle. Throughout this project we have analyzed the gear systems in order to create efficient power delivery between the wheels of the vehicle from shaft motion to the engine. Our group will be using key formulas and concepts we have learned about Shaft design, gear design, and rolling element bearings we have gone into depth research about throughout chapters 7,11, 13,14, as well as chapter 18 for guidance on this design concept. With the initial design constraints at our leisure, we can assume that the torque, final drive, axle shafts,& axle shaft components have a 100 percent efficiency with infinite life.

1.3 Strategy

In order to start this project off we have calculated the ratio for the highest gear ratio by considering various constraints such as the engine speed, final drive, and the tire size. By analyzing a free body diagram of the downgrade hill analysis for our vehicle, we were able to calculate the degree of movement, the Force acting upon the vehicle throughout the downhill motion, Torque of the gear and pinion, as well as the number of teeth for both the gear and pinion. Throughout the Torque and Number of teeth relation we were able to find a gear ratio of 2.96.

Next for our project we were required to find the lowest gear ratio through hand calculation. Considering all the driving conditions provided such as the hill grade, vehicle weight, or tire size, we have analyzed free body diagrams to get these results. In addition to the free body diagram analysis, our group wrote a code by using the MATLab software that defines the torque force on the propeller shaft which connects the transmission and final drive from our hand calculations completed.

 Number: 1 Author: japark Subject: Highlight Date: 12/2/2020 1:27:25 PM
This is more of a design constraint.

 Number: 2 Author: japark Subject: Highlight Date: 12/2/2020 1:29:16 PM
downhill? What do you meant by this?

Another necessary component of our project design is to complete size analysis on the forward and reverse gears based on our diametral pitch value that was assumed. By considering all the design constraints, our group analyzed the number of teeth and pitch radii before submitting hand calculations and computer code that includes diameter and radii inputs. Also by providing a force analysis on all the gears, our group had the ability to choose a face width for our vehicle component. Lastly we will need to analyze the propeller shaft design in order to find the accurate value of the shaft diameter. 

1.4 Project planning

For the Project we will conclude our results by providing hand calculations for system components, ANSYS design analysis, and 3D models of our components in order to show our results properly and effectively. Along with component descriptions our group will provide hand calculations necessary for each component of the system such as the gear ratio, shaft diameter, bending stress, and contact stress. Through the use of ANSYS we had the ability to analyze the deflections and stress analysis of the gears and shafts including properties, conditions, plots of stress, plots of deformation, as well as maximum stress locations. Lastly, our group used the NX software in order to create 3D models of shafts, spur gears, and bearings.

2. Executive Summary

2.1 Gearbox

In this section, we calculate for the number of teeth for the pinions and the respective gears provided in the table below

Gear number		N(in)	d(in)	ϕ (in)	$\sigma_{bending}^{max}(mpa)$	Material	Safety factor
Pinion		77	2.1667	1.2876	3.8093E+04	1040 Q&T	2.5
Forward gears	1	77	6.4167	1.2876	3.8093E+04	1040 Q&T	2.5
	2	73	6.0833	1.2876	3.8093E+04	1040 Q&T	2.5
	3	68	5.667	1.2876	3.8093E+04	1040 Q&T	2.5
	4	63	5.2500	1.2876	3.8093E+04	1040 Q&T	2.5
	5	58	4.833	1.2876	3.8093E+04	1040 Q&T	2.5
Reverse gear		14	1.1667	1.2876	3.8093E+04	1040 Q&T	2.5

The results provided in the table above show the value using the formulae learnt through the semester with some assumptions made to fit into the constraints of our chosen material and gearing requirements.

2.2 Propeller Shaft

For the shaft diameter we used the Goodman equation shown below to calculate the shaft diameter, this value is shown in the table below alongside a list of other variables.

Page: 6

 Number: 1 Author: japark Subject: Sticky Note Date: 12/2/2020 1:29:48 PM
How about key and bearings?

 Number: 2 Author: japark Subject: Highlight Date: 12/2/2020 1:31:36 PM
Why MPa when all other parameters are in US customary units?

 Number: 3 Author: japark Subject: Highlight Date: 12/2/2020 1:31:06 PM
How come all gear have the same face width and safety factor? We shall see in your calculation.

 Number: 4 Author: japark Subject: Highlight Date: 12/2/2020 1:30:41 PM

$$d = \left(\frac{16n}{\pi} \left\{ \frac{1}{S_e} [4(K_f M_a)^2 + 3(K_{fs} T_a)^2]^{1/2} + \frac{1}{S_{ut}} [4(K_f M_m)^2 + 3(K_{fs} T_m)^2]^{1/2} \right\} \right)^{1/3}$$

Shaft diameter	Max torque	Material	Max $\tau_{torsion}$	Failure criteria	Safety factor
5.3969	869.0063	1090 Q&T	7696	1.700	2.5

The table above shows the different values solved for with the use of engineering formulas to depict the effects of the torque on a material of that shaft diameter.

2.3 Shaft components

Here similar to the propeller shaft, use of the goodman equation is implemented to determine the keys shaft.

2.3.1 Key

Shaft diameter	Key size			Loads		Material	Safety factor
	w	h	l	Force	Torque		
5.3969	3		0.153	17.104	7.6994	1040 Q&T	2.5

2.3.2 Bearing

Bearing	Radial load	Axial load	Catalog load rating	Catalog	Item number
138699	14060	10000	10	10	

1.1 Gear design methodology

1.1.1 Gear ratios

Given: vehicle weight: 3550lb, Tire size: 225/50/R50, Final drive: 4.27, 65 mph occurs at 2000RPM, Aspect ratio = 50, radius =17

Tire width in inches = $(225 \times 50) / 100 \times 25.4 = 4.429$

RPM of Drive Axle

Page: 7

	Number: 1	Author: japark	Subject: Sticky Note	Date: 12/2/2020 1:32:00 PM
	This does not look professional at all.			
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	Units are missing.			
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	Unit? Are you still in MPa?			
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	Number: 12	Author: japark	Subject: Highlight	Date: 12/2/2020 1:33:49 PM
	Number: 13	Author: japark	Subject: Highlight	Date: 12/2/2020 1:34:25 PM
	incomplete.			

$$= \frac{15 \times 5280 \times 12}{60} = 68640 \text{ RPM}$$

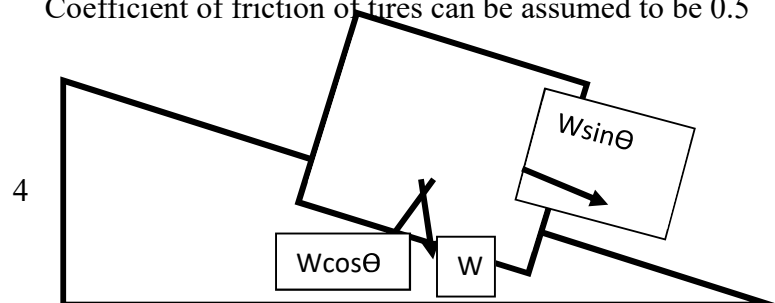
Highest Gear Ratio

$$\frac{2000 \text{ RPM}}{3607.72 \text{ RPM}} = 0.554$$

Lowest gear ratio (highest gear)

Given: Initial effort to climb 40% grade hill occurs at 2000 RPM

Coefficient of friction of tires can be assumed to be 0.5



$$m_l = \frac{6535}{225} = 2.96$$

Using force calculation, we determine the force required to pull the weight up the incline. This value was then taken and multiplied by the radius to determine the torque, which was then related to the angular velocity to find the torque at the set RPM of 2000 to give a value that is then compared to the max torque.

3.1.2.4 Gear sizes

Gears		Gear ratio	⁴ Diameter	⁵ Max torque at the wheel	⁵ Max speed at the wheel(mph)
Forward gears	1	2.96	$d1 = (NG1/P) = (77/12) = 6.42$	7.699	65
	2	2.8	$d2 = (NG2/P) = (73/12) = 6.083$	7.699	65
	3	2.6	$d3 = (NG3/P) = (68/12) = 5.67$	7.699	65

Page: 8

	Number: 1	Author: japark	Subject: Highlight	Date: 12/2/2020 1:35:02 PM
	Include description why you are solving it like this.			
	Number: 2	Author: japark	Subject: Highlight	Date: 12/2/2020 1:35:37 PM
	Where did 68640 RPM go?			
	Number: 3	Author: japark	Subject: Highlight	Date: 12/2/2020 1:36:15 PM
	Where did all these value come from?			
	Number: 4	Author: japark	Subject: Sticky Note	Date: 12/2/2020 1:37:23 PM
	Label your tables and figures.			
	Number: 5	Author: japark	Subject: Sticky Note	Date: 12/2/2020 1:36:58 PM
	If all gears have the same max torque and speed, what is the purpose of having multiple gears?			
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	unit missing.			
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	Number: 8	Author: japark	Subject: Highlight	Date: 12/2/2020 1:36:24 PM
	Number: 9	Author: japark	Subject: Highlight	Date: 12/2/2020 1:36:28 PM

	4	2.4	$d_4 = (NG_4/P) = (63/12) = 5.25$	7.699	65
	5	2.2	$d_5 = (NG_5/P) = (58/12) = 4.833$	7.699	65
	6	0.568	$d_6 = (NG_6/P)$	7.699	65
Reverse gear		2.96	$=d_1 = 6.42$	7.699	65

2.4 Shaft design methodology

The shaft designed should support the transmitted load of the gears, with it having the ability to rotate its torque and torque  have to be capable of moving and shifting through gears as required.



Number: 1

Author: japark

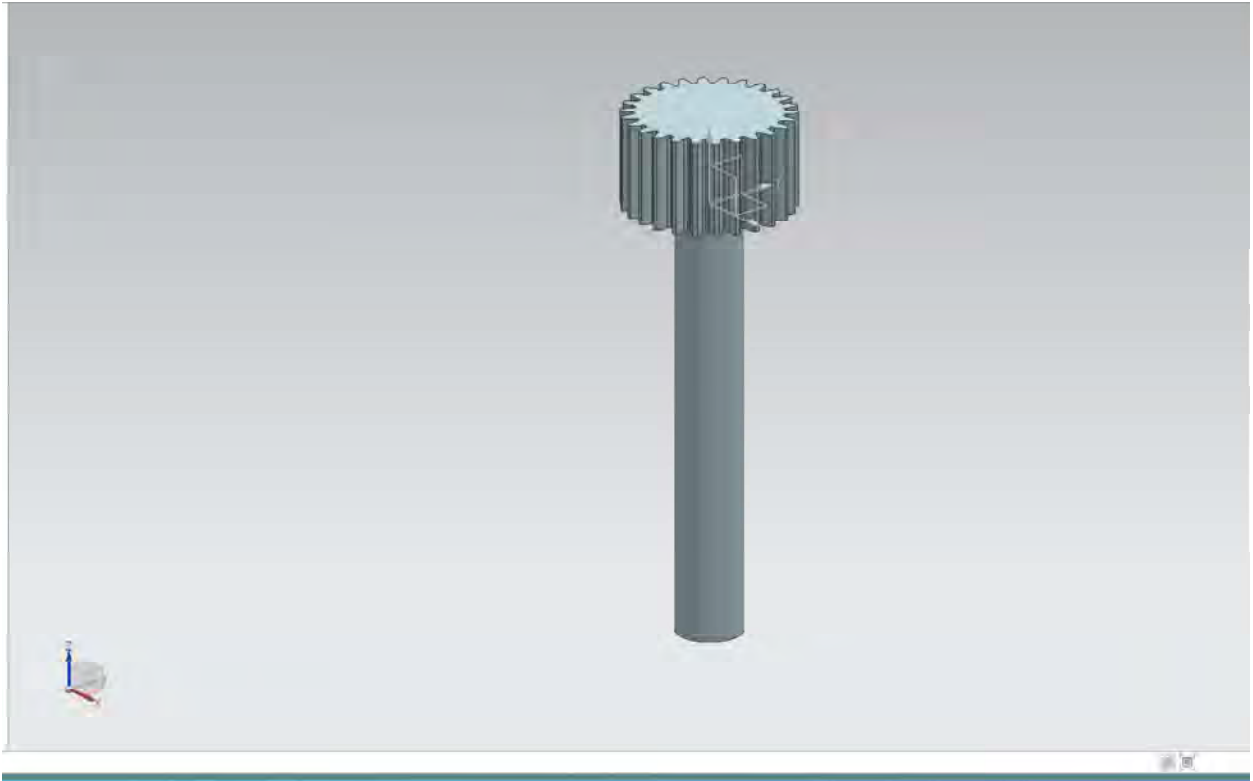
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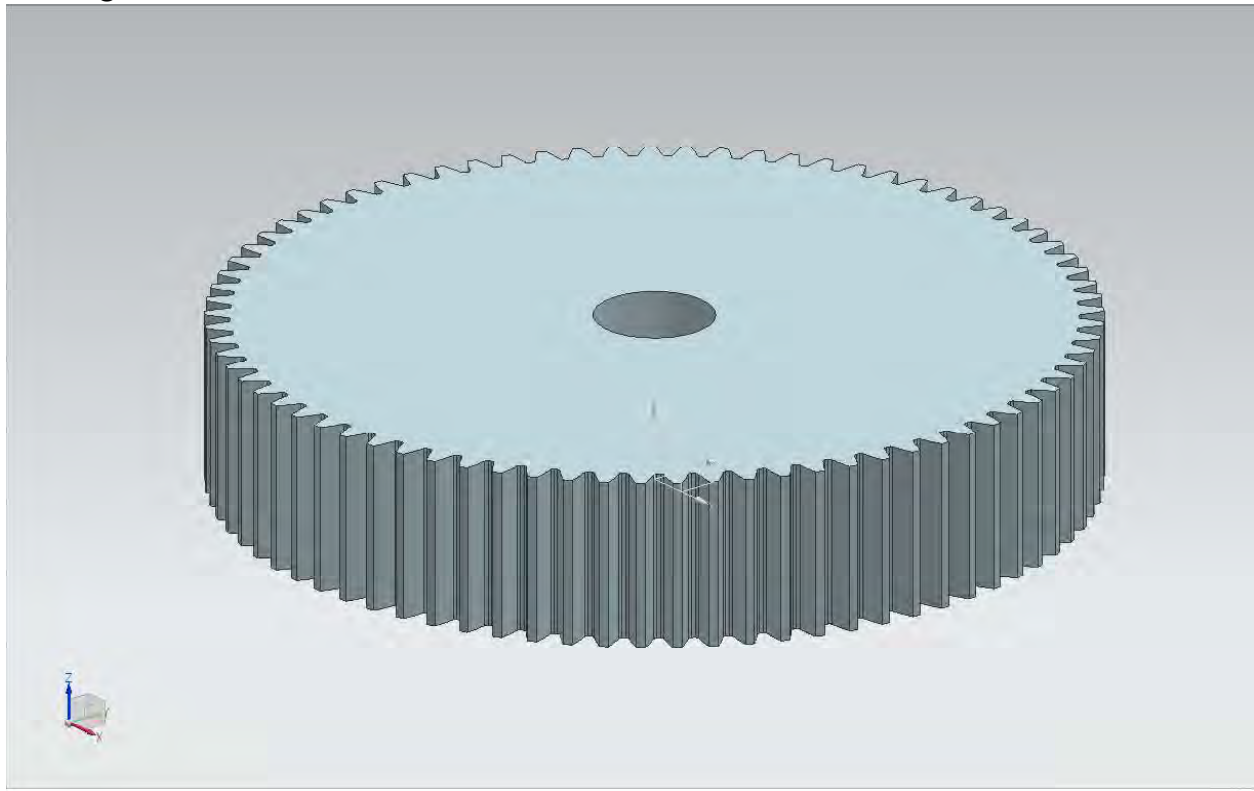
Shaft design is missing completely.

3. CAD models

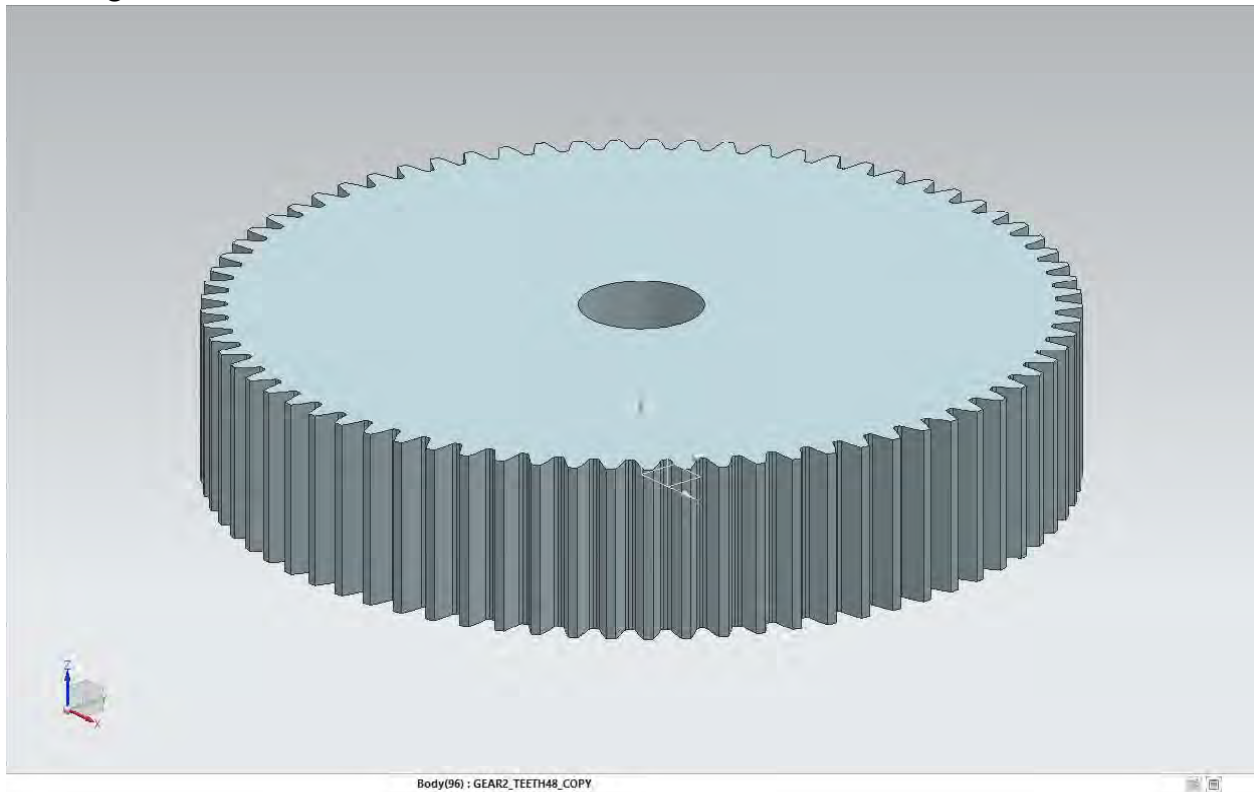
4.1 Pinion



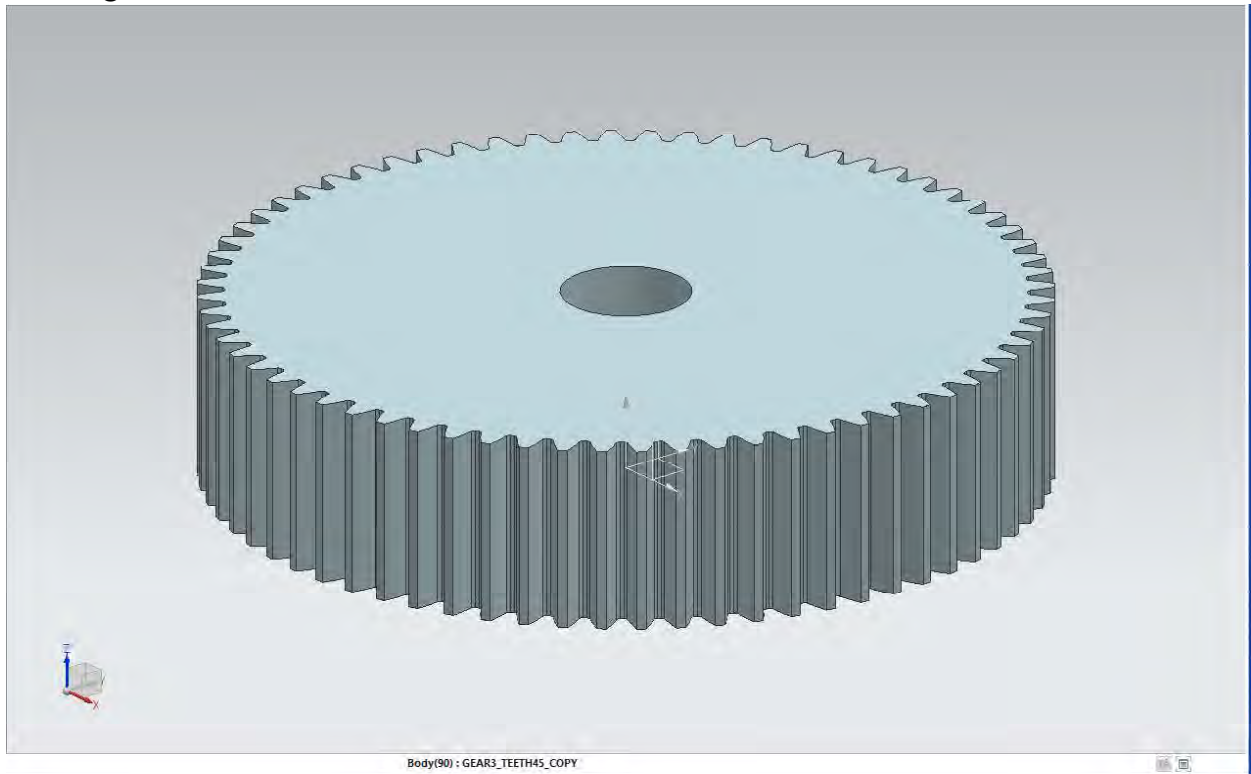
4.2 1st gear



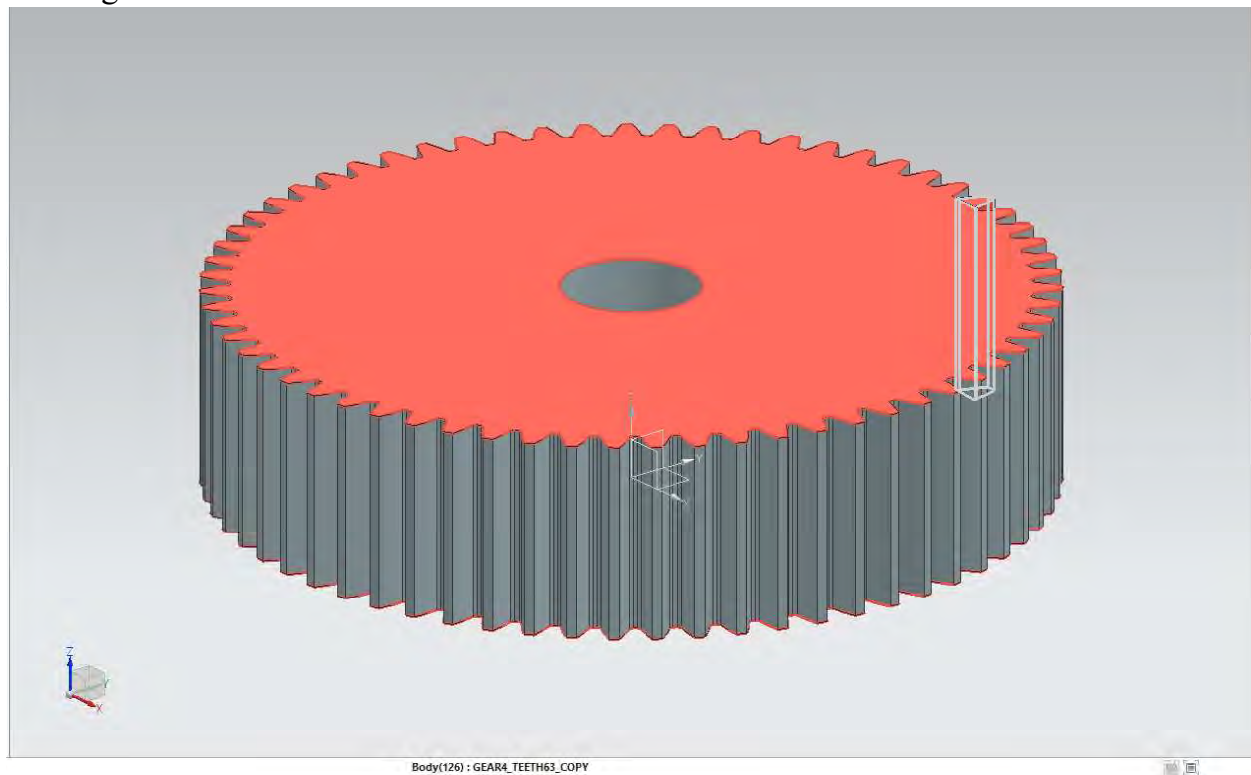
4.3 2nd gear



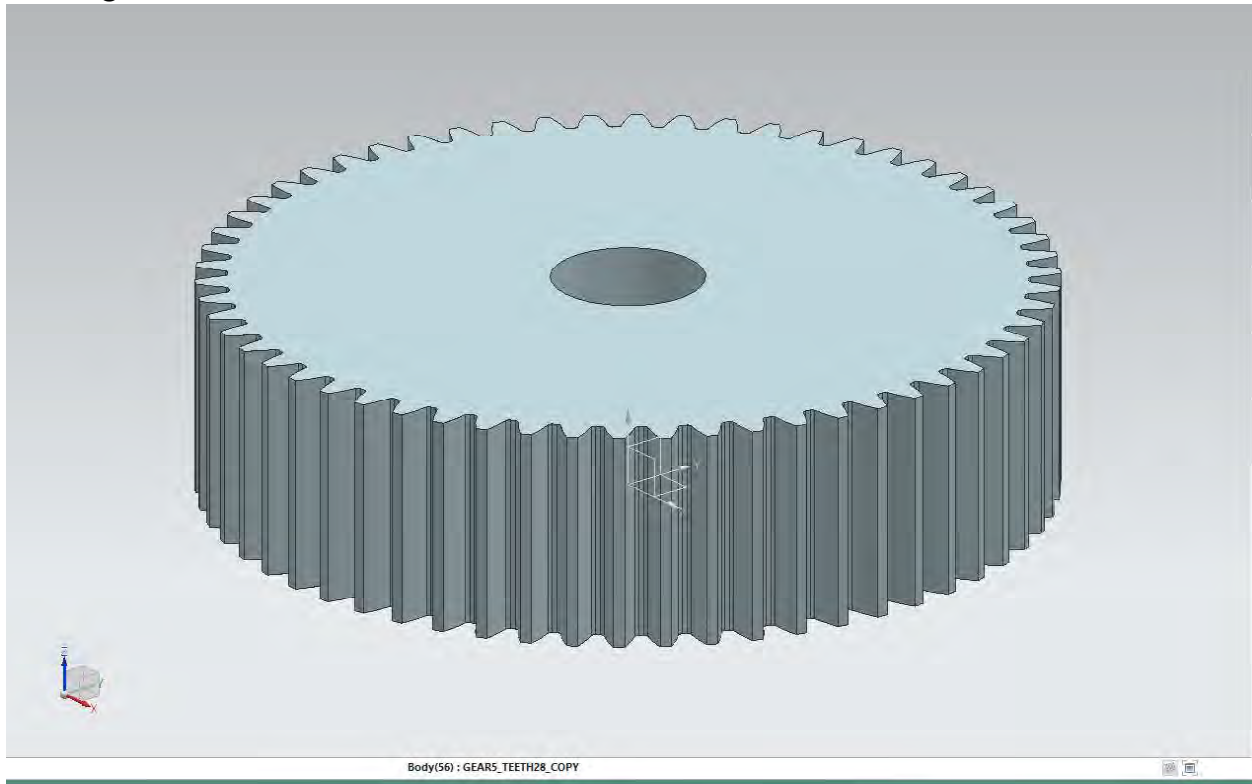
4.5 3rd gear



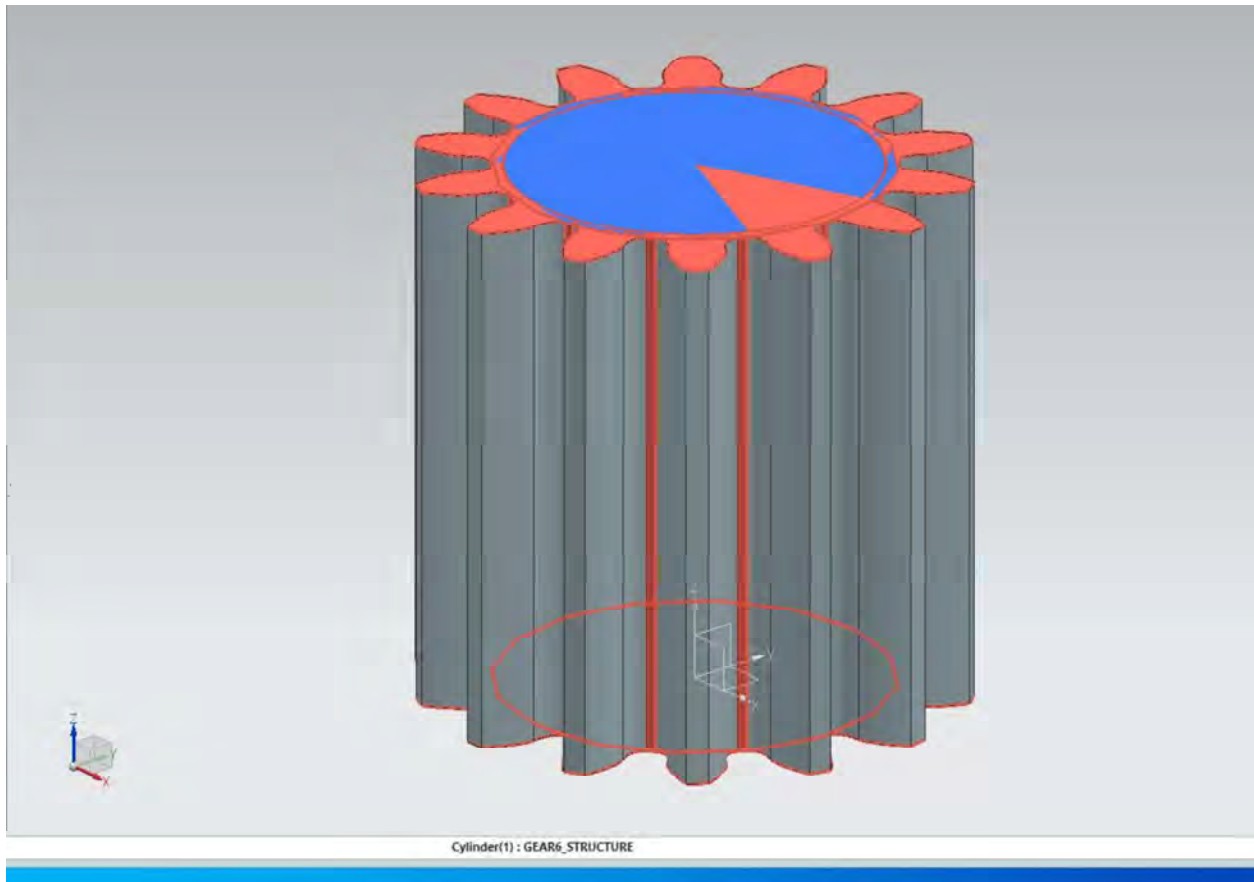
4.5 4th gear



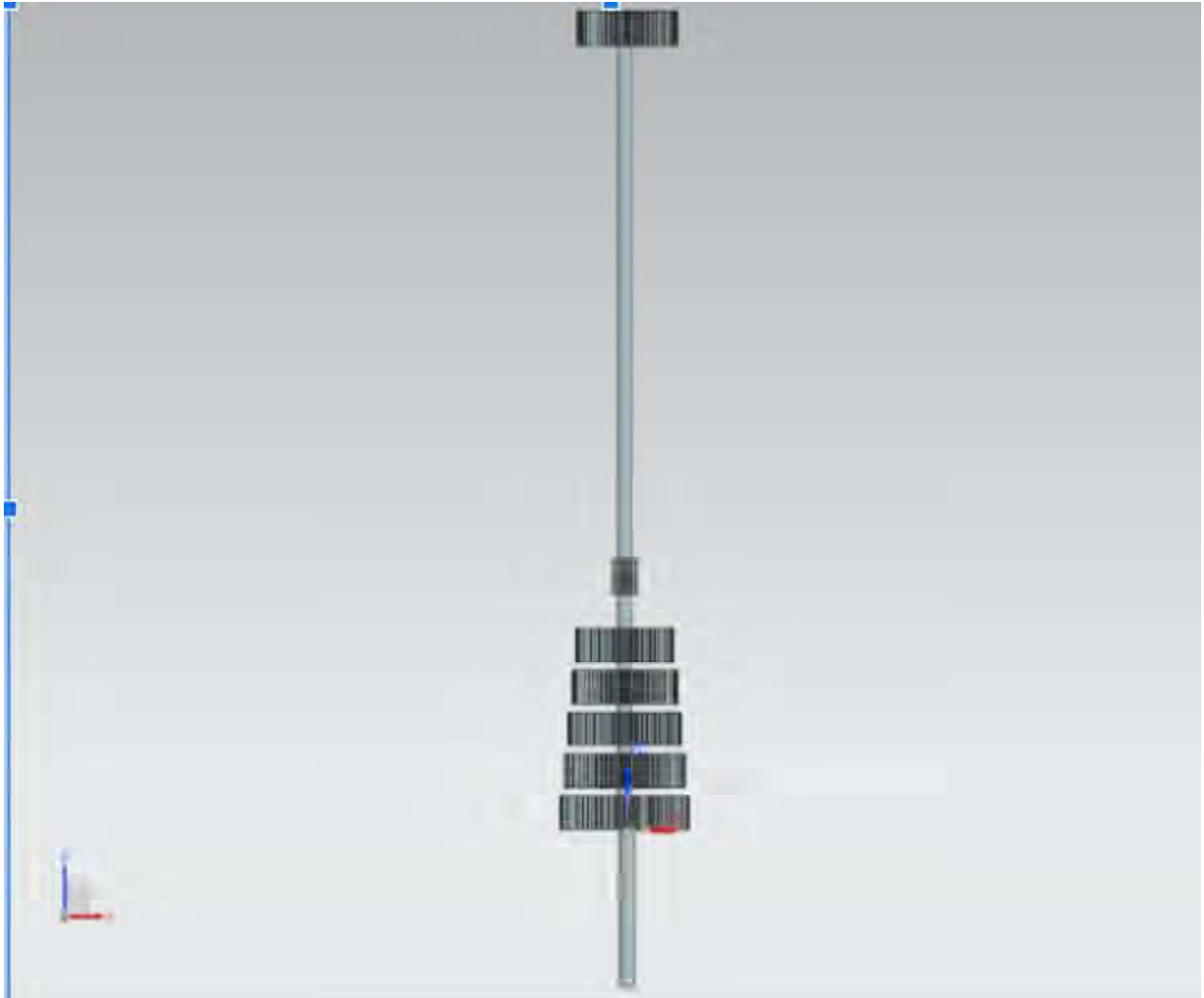
4.6 5th gear

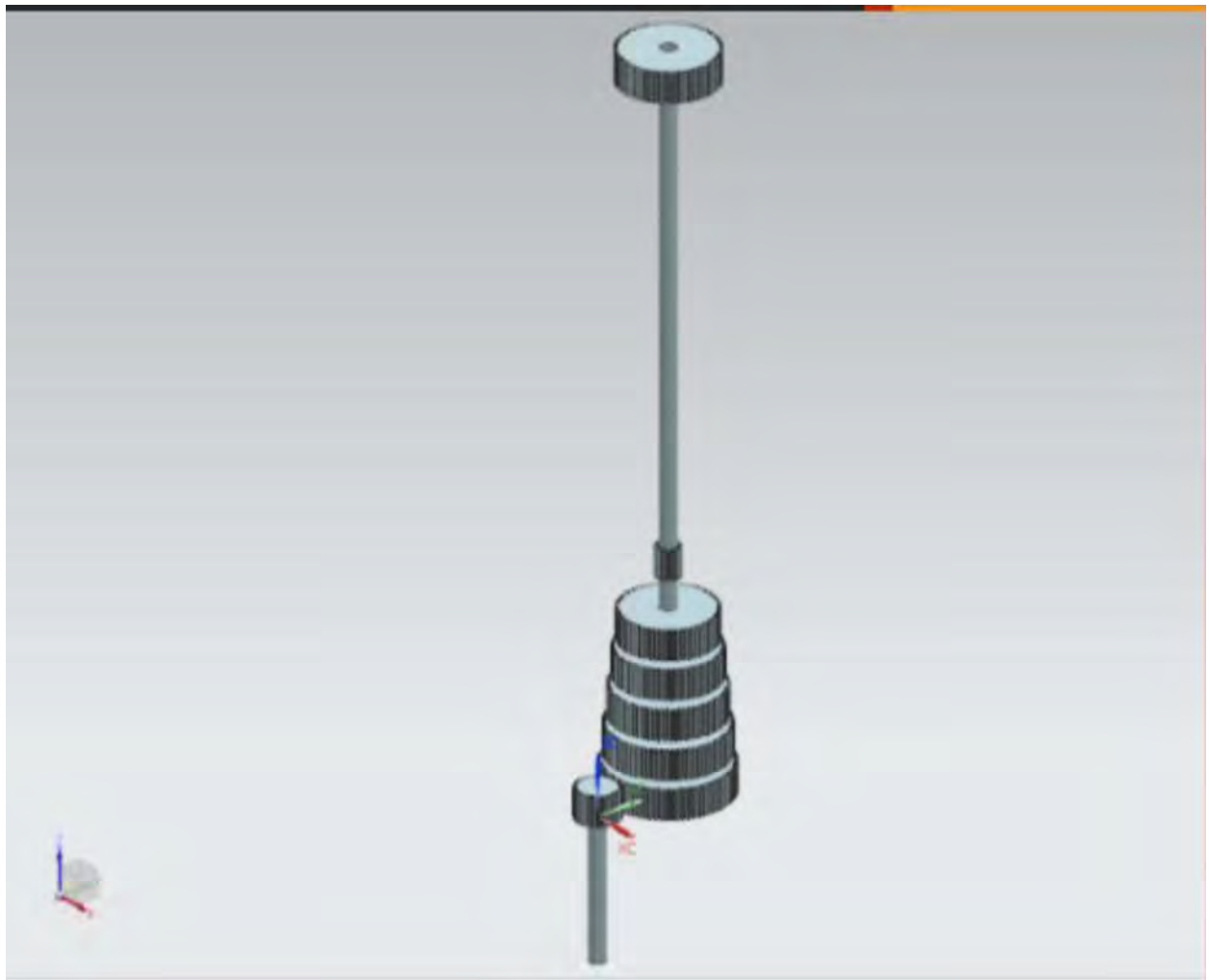


4.7 6th gear



4.9 Shaft and its components (keys and bearings)





4. Conclusion

In conclusion, we were able to design a six speed gearbox of an AUDI A4 using all the information provided above. We noticed that all assumptions were crossed checked with the values plugged in for MATLAB code and it came out almost the same. Therefore, in order to build a gearbox and specially a 6 speed gearbox, we must choose the pattern of calculations that we did above. Forces analysis are very important to build any gearbox.





Number: 1 Author: japark Subject: Sticky Note Date: 12/2/2020 1:40:00 PM

This is a very disappointing report. The report is missing a few very critical components. Shaft analysis, bearings and keys. no 2D engineering drawing is provided. Units are mixed up and not included for large number of parameters.



Number: 2 Author: japark Subject: Sticky Note Date: 12/15/2020 1:09:32 PM



Author: japark Subject: Sticky Note Date: 12/15/2020 1:10:31 PM

The report contains all necessary values 4/10

All calculation steps follow correct procedure 9/10

The report contains design iteration 5/10

The values are reasonable and feasible 8/10

Formatting look professional 5/10

CAD drawing of all components are included 10/10

2D engineering drawings are included 0/10

FEA is included 0/10

Enough description is provided 5/10

Participation 15/15

61/95