MAE 4342 Mechanical Design II

Fall 2023

Hoist Reducer: Gear-Box Design Project

Submitted to

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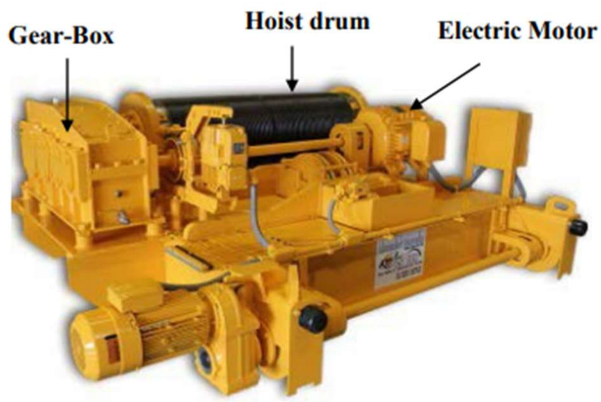
I. Introduction

The goal in the gear box design project is to design a prototype gearbox that will allow the hoist drum of a crane to elevate loads of up to 7.5 T at a maximum speed of 30 ft/min. We know that an electric motor will be used at it is rated at 15HP at 1800 RPM. The theoretical gearbox used will reduce the input speed and increase the torque on the output shaft.



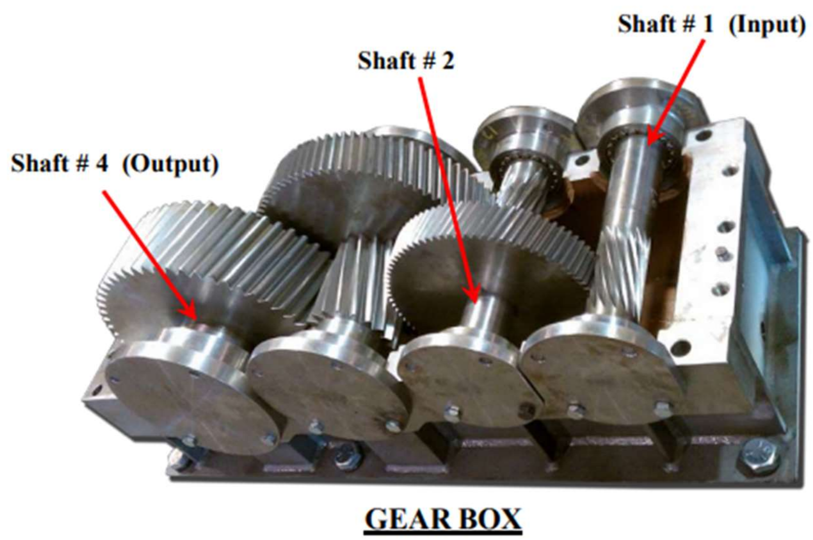


Figure 1: Gear-Box Design

Figure 2 shows the housing dimensions of the gear box in which the theoretical gear box design must fit inside. The gearbox will be designed for 10 years at an average use of 5 hour/day, 200 days/year. Given the specifications listed in section III of the report, the team will need to design all of the pinions, all of the gears, shaft # 2, carrying the stage 1 reduction gear as well as the stage 2 reduction pinion and lastly the key on shaft #4.

II. Design procedure

In gearbox design, the pinion is the primary focus for fatigue failure analysis due to its smaller number of teeth leading to higher stress cycles, greater stress concentration. This makes it more susceptible to fatigue failure.

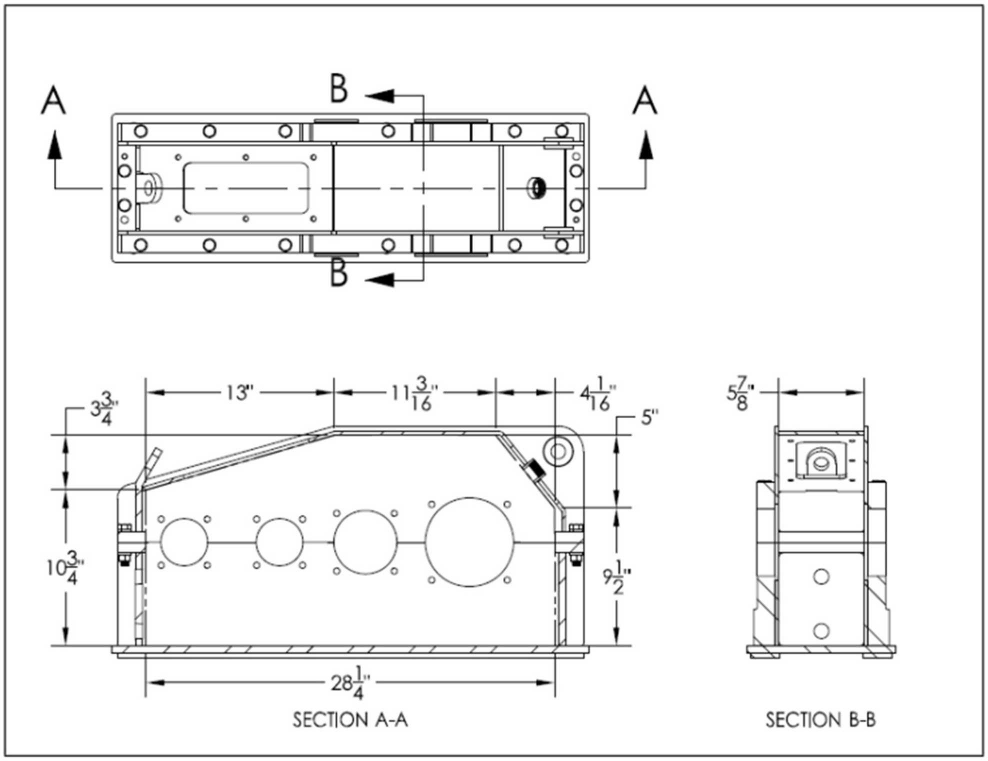


Figure 2: Gearbox Housing Dimensions

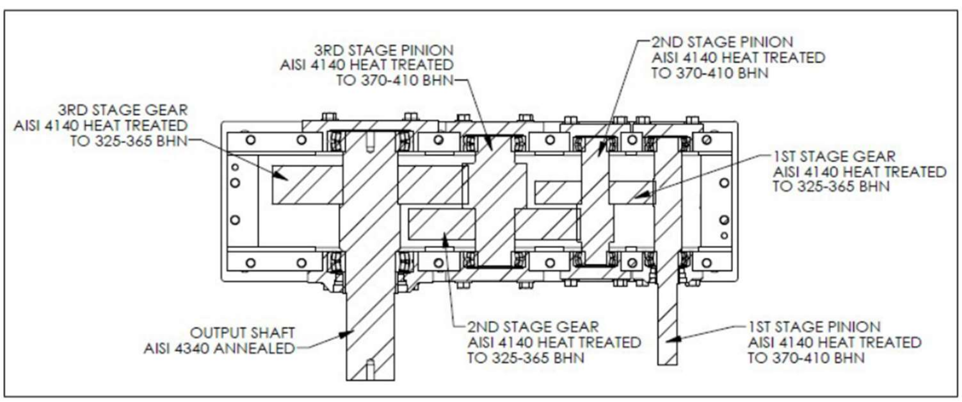


Figure 3: Gear and Shaft Materials

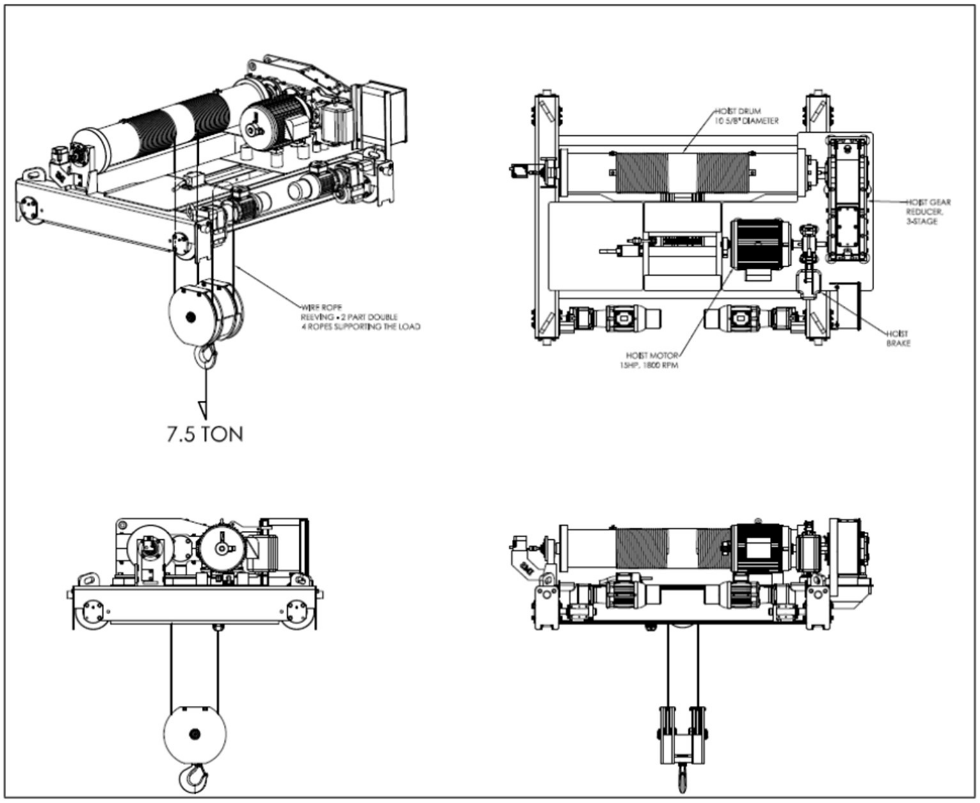


Figure 4: Hoist Model

The gears ratio is limited in range of 3:1 to 10:1, and the pressure angle is 25 degrees 2 set of gear teeth and pinion are initially chosen for the gearbox design $[\mathrm{N_p} = 17, \mathrm{m_1} = \mathrm{m_2} = \mathrm{m_3} = 8]$ and $[\mathrm{N_p} = 14, \mathrm{m_1} = \mathrm{m_2} = \mathrm{m_3} = 4]$ to reduce interpolation of bending factor J process. Due to size housing constraints set of $[\mathrm{N_p} = 14, \mathrm{m_1} = \mathrm{m_2} = \mathrm{m_3} = 4]$ is chosen.

Shaft 2 design consists of torsion, bending analysis, with the given material and ASME equation, diameter of shaft can be decided. Key is designed with lower strength material than output shaft which will be failed first to prevent failure of whole gears system. Pinions are designed based on surface and bending fatigue analysis with assumptions, the size of pinions and gear must meet housing dimensions constraints.

III. Assumption & Specifications

Factors of safety: fatigue safety factors are 1.2 surface, 1.5 bending

<u>Key Dimensions:</u>

- Shape: Square
- Width (W): 1 inch

- Height (H): 1 inch
- Material: Steel AISI 4140 heat treated (Annealed @ 1450F)

Table A-10 Mechanical Properties for Some Alloy and Tool Steels
Data from Various Sources.* Approximate Values. Consult Material Manufacturers for More Accurate Information

SAE / AISI Number	Condition	Tensile Yield Strength (0.2% offset)		Ultimate Tensile Strength		Elongation over 2 in	Brinell or Rockwell Hardness
		kpsi	MPa	kpsi	MPa	%	
4140	annealed @ 1450°F	61	421	95	655	26	197HB
	normalized @ 1650°F	95	655	148	1020	18	302HB
	quench & temper @ 1200°F	95	655	110	758	22	230HB
	quench & temper @ 800°F	165	1138	181	1248	13	370HB
	quench & temper @ 400°F	238	1641	257	1772	8	510HB
4340	quench & temper @ 1200°F	124	855	140	965	19	280HB
	quench & temper @ 1000°F	156	1076	170	1172	13	360HB
	quench & temper @ 800°F	198	1365	213	1469	10	430HB
	quench & temper @ 600°F	230	1586	250	1724	10	486HB

Gear Ratios: Pressure angle is 25-deg

- Range: Minimum ratio 3:1, Maximum ratio 10:1

Gear Specifications & Assumption(-):

1. Shaft 1: 1.875-in OD, AISI 4140 t
   - Pinion Teeth (N_{P1}): 14
   - Gear Teeth on Shaft II (N_{g1}): 56
   - Module (m_1): 4
2. Shaft 2: Unknown OD, AISI 4140 Heat treated

3. Shaft 3: 3.00 -in OD, AISI 4140 Heat treated
   - Pinion Teeth (N_{P2}): 14
   - Gear Teeth on Shaft III (N_{g2}): 56
   - Module (m_2): 4

4 Shaft 4: 4.00 -in OD, AISI 4340 Annealed
   - Pinion Teeth (N_{P3}): 14
   - Gear Teeth on Shaft IV (Ng3): 56
   - Module (m_3): 4

IV. Calculations

Nomenclature (Dakito):

F = Face width

J = Bending strength geometric factor

K_v = Dynamic factor

K_a = Application factor

K_s = Size factor

W_t = Tangential force

W_r = Radial force

W = Magnitude of force

P = power

ω = Angular velocity

S_y = Yield strength

S_{ut} = Ultimate strength

d_{equiv} = Equivalent diameter

C_{temp} = Temperature factor

C_{surf} = Surface factor

C_{size} = size factor

C_{load} = Load factor

S_e = Corrected endurance factor

S_f = Corrected fatigue strength

K_f = Bending fatigue strength stress concentration factor for alternating comp.

K_{fm} = Bending fatigue strength stress concentration factor for mean components

d = Diameter of the shaft

M_a = Bending moment

N_f = Safety factor

T_m = Mean torque

$S_{fb}: Fatique\ strength$

A. **Transmitted torque calculation** (Phat)

$OD_4 = 4\ in;\ \ P = 15\ hp\ at\ \omega = 1800\ rpm$

$AISI$ Steel AISI 4140 Annealed @ 1450F => $S_y = 61\ ksi\ \ S_{ut} = 95Ksi$

$$T_{p1} = \frac{63000\ Power}{w} = \frac{63000\ (15)}{1800} = 525\ lb - in$$

$$m_1 = \frac{N_{g1}}{N_{p1}} = \frac{\omega_{p1}}{\omega_{g1}},\ T_{p1} = \frac{P}{\omega_{p1}} => \omega_{p1} = \frac{P}{T_{p1}}, \omega_{g1} = \frac{P}{T_{g1}} => \ \ m_1 = \frac{T_{g1}}{T_{p1}}$$

$$T_{g1} = m_1 T_{p1} = (525)(4) = 2100\ lb - in$$

$$T_{g1} = T_{p2} = \ 2100\ lb - in$$

$$m_2 = \frac{T_{g2}}{T_{p2}} => T_{g2} = m_2 * T_{p2} = (2100\ ) * (4) = 8400\ lb - in$$

$$T_{g2} = T_{p3} = \ 8400\ lb$$

$$\mathrm{m_3 = \frac{T_{g3}}{T_{p3}} =>\ \ T_{g3} = m_3 * T_{p3} = (8400\ ) * (4) =\ 33600\ lb - in}$$

Maximum Torque (Tmax): 33600 lb-in

Minimum Torque (Tmin): 0 lb-in

B. <u>**Gear and pinions:**</u>

<u>*** Bending Fatigue Pinion**: (Phat)</u>

$$S_{fb} = \frac{K_L}{K_T K_R} S_{fb}{}'$$

Fatigue Bending Gear Stress based on Brinell hardness

1. $S'_{fp} = 34 \rightarrow 40 \times\ 10^3\ \ psi\ (table\ 12 - 20)$

Table 12-20 Bending-Fatigue Strengths S_{fb}' for a Selection of Gear Materials*†

Material	Class	Material Designation	Heat Treatment	Minimum Surface Hardness	Bending-Fatigue Strength psi x 10^3	Bending-Fatigue Strength MPa
Steel	A1—A5		Through hardened	≤180 HB	25—33	170—230
			Through hardened	240 HB	31—41	210—280
			Through hardened	300 HB	36—47	250—325
			Through hardened	360 HB	40—52	280—360
			Through hardened	400 HB	42—56	290—390
			Flame or induction hardened	Type A pattern 50—54 HRC	45—55	310—380
			Flame or induction hardened	Type B pattern	22	150
			Carburized and case hardened	55—64 HRC	55—75	380—520
		AISI 4140	Nitrided	84.6 HR15N	34—40	230—310
		AISI 4340	Nitrided	83.5 HR15N	36—47	250—325
		Nitralloy 135M	Nitrided	90.0 HR15N	38—48	260—330
		Nitralloy N	Nitrided	90.0 HR15N	40—50	280—345
		2.5% Chrome	Nitrided	87.5—90.0 15N	55—65	380—450
Cast iron	20	Class 20	As cast		5	34
	30	Class 30	As cast	175 HB	8.5	59
	40	Class 40	As cast	200 HB	13	90
Nodular (ductile) iron	A-7-a	60-40-18	Annealed	140 HB	22—33	152—228
	A-7-c	80-55-06	Quenched and tempered	179 HB	22—33	152—228
	A-7-d	100-70-03	Quenched and tempered	229 HB	27—40	186—276
	A-7-e	120-90-02	Quenched and tempered	269 HB	31—44	213—303
Malleable iron (pearlitic)	A-8-c	45007		165 HB	10	70
	A-8-e	50005		180 HB	13	90
	A-8-f	53007		195 HB	16	110
	A-8-i	80002		240 HB	21	145
Bronze	Bronze 2	AGMA 2C	Sand cast	40 ksi min tensile strength	5.7	40
	Al/Br 3	ASTM B-148 alloy 954	Heat treated	90 ksi min tensile strength	23.6	160

Table 12-19
AGMA Factor K_R

Reliability %	K_R
90	0.85
99	1.00
99.9	1.25
99.99	1.50

2. Temperature Factor $K_T = \frac{460+T_F}{620} = 1$ (for steel material in oil temp up to 250F)

3. Reliability Factor $K_R = 0.85\ (table\ 12-19)$
4. Modifies strength K_L based

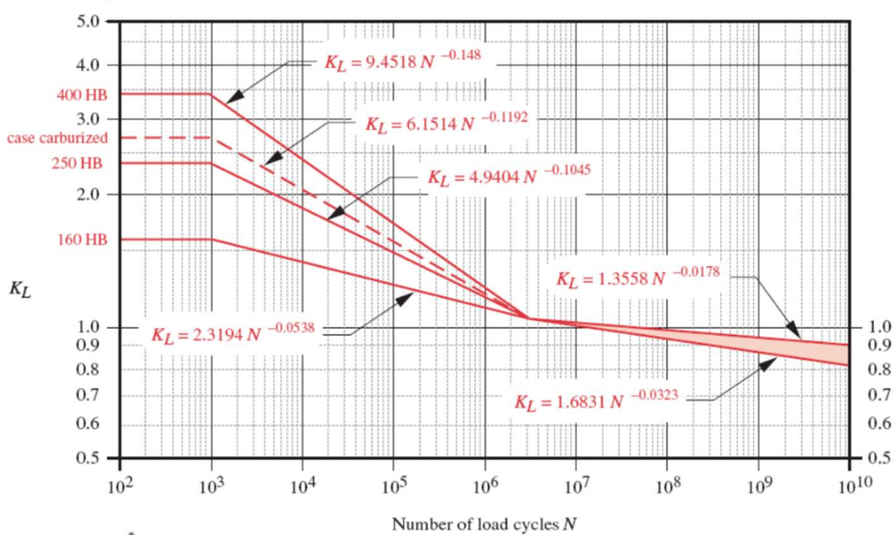


Figure 5. Modifies strength based on cycle life

$$10\ years.5\frac{hour}{day}200\frac{days}{year} = 600000\ min$$

$$=> N = 600000min\frac{2\pi}{rev}\frac{1800rev}{1min} \approx 6.785 \times 10^9 cycle$$

$K_L = 1.3558 \times N^{-0.0178}$=0.91

$$S_{fp} = 40 \times 10^3 \frac{0.91}{(1)0.85} = 42823\, psi$$

⇨ Define σ_b

$$N_{fb} = \frac{S_{fb}}{\sigma_b} => \sigma_b = \frac{S_{fb}}{N_{fb}}$$

5. Define face width based on AGMA stress equation

AGMA bending stress equation:

$$\sigma_b = \frac{W_t p_d}{FJ}\frac{K_a K_m}{K_v}K_s K_B K_I$$ U.S. specifications

$$\sigma_b = \frac{W_t}{FmJ}\frac{K_a K_m}{K_v}K_s K_B K_I$$ SI specifications - m is the metric module

- F is face width (minimum face width is 3 to 5 times p_c)
- J is bending strength geometric factor (AGMA; Tables 12-8 through 12-15)
- K_v is dynamic factor (depends on tangential velocity -- Figure 12-22)
- K_m is load distribution factor (to account for misalignments -- Table 12-16)
- K_a is application factor (application dependent; "Shocks" -- Table 12-17)
- K_s is size factor (similar concept as Equation 6.7b -- AGMA recommends value of 1)
- K_B is rim thickness factor (for gears made of rims and spokes-- Figure 12-23)
- K_I is idler factor (set to 1.42 for an idler gear and 1 for a non-idler gear)

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Figure 6. AGMA stress equation guidance

$$F = \frac{W_t p_d K_a K_m}{J\, K_v} K_s K_B K_I \frac{N_{fb}}{S_{fb}}$$

6. Find J

Table 12-13 AGMA Bending Geometry Factor J for 25°, Full-Depth Teeth with HPSTC Loading

	Pinion teeth															
Gear teeth	12		14		17		21		26		35		55		135	
	P	G	P	G	P	G	P	G	P	G	P	G	P	G	P	G
12	U	U														
14	U	U	0.33	0.33												
17	U	U	0.33	0.36	0.36	0.36										
21	U	U	0.33	0.39	0.36	0.39	0.39	0.39								
26	U	U	0.33	0.41	0.37	0.42	0.40	0.42	0.43	0.43						
35	U	U	0.34	0.44	0.37	0.45	0.40	0.45	0.43	0.46	0.46	0.46				
55	U	U	0.34	0.47	0.38	0.48	0.41	0.49	0.44	0.49	0.47	0.50	0.51	0.51		
135	U	U	0.35	0.51	0.38	0.52	0.42	0.53	0.45	0.53	0.48	0.54	0.53	0.56	0.57	0.57

from table => $J_p = 0.34$,$J_G = 0.47$

7. <u>Find Kv</u>
   Assume $Q_v = 11$
   $B = 0.25(12 - Q_v)^{0.6667} = 0.25$
   $A = 50 + 56(1 - 0.73) = 92$
   $K_V = (\frac{A}{A+\sqrt{V_t}})^B$
   Where
   $V_t = \pi d_p n_p \ (fpm)$

$$W_t = \frac{H}{V_t} \ (lbf)$$

8. Assume: K_m= 1.7 (Table 12-16)

9. Assume $K_a = 1$ (Table 12-17)

10. Assume Ks =1 (AGMA recommends value of 1)

11. Assume K_B = 1

12. K_I =1 (no idler)

13. Face width range
    $\frac{8}{p_d} \leq F \leq \frac{16}{p_d}$

Using MATLAB to find facewidth.

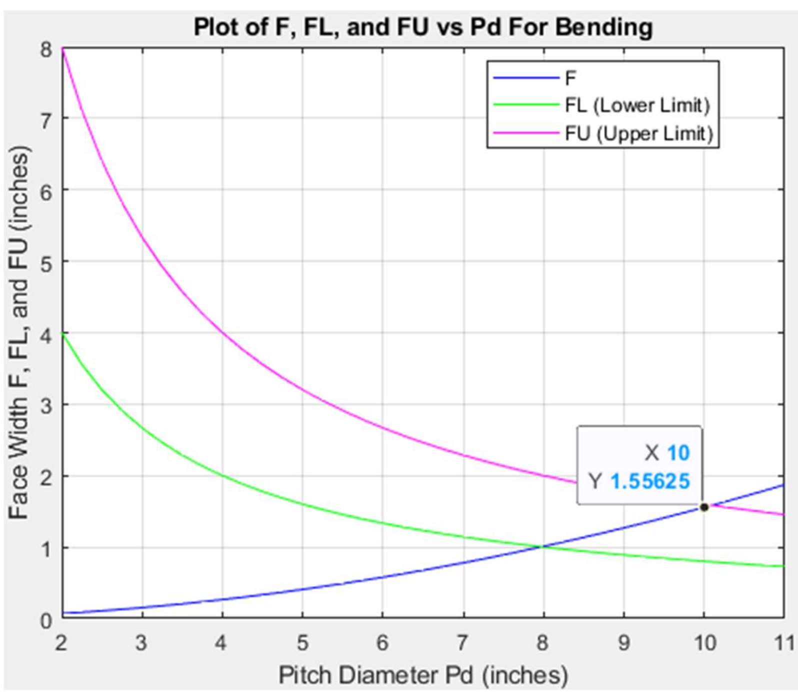


Figure 7. Bending Fatigue Analysis Pinion 1 & Gear 1

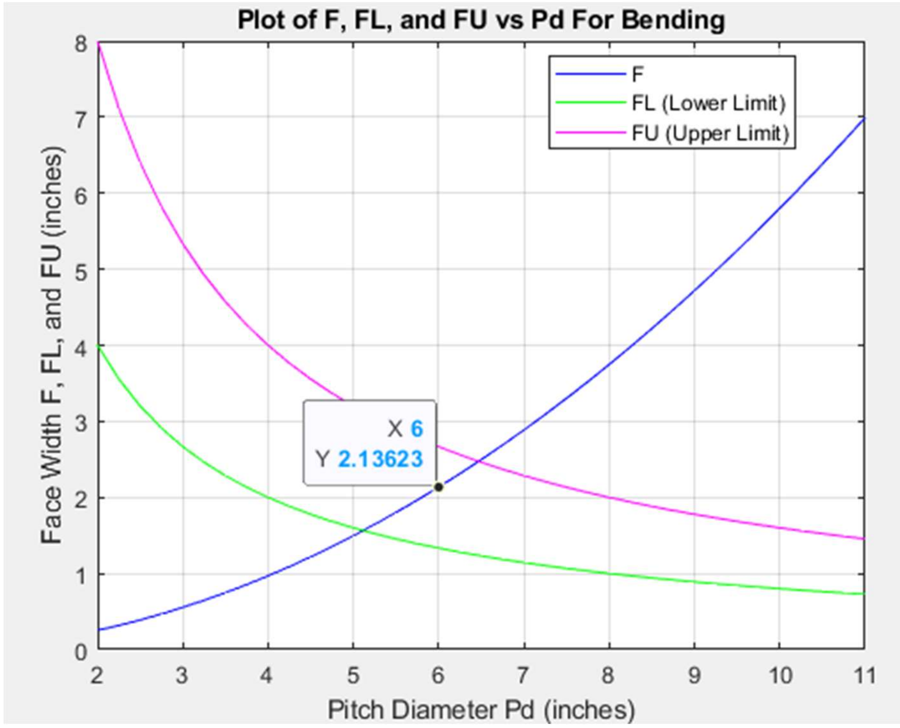


Figure 8. Bending Fatigue Analysis Pinion 2 & Gear 2

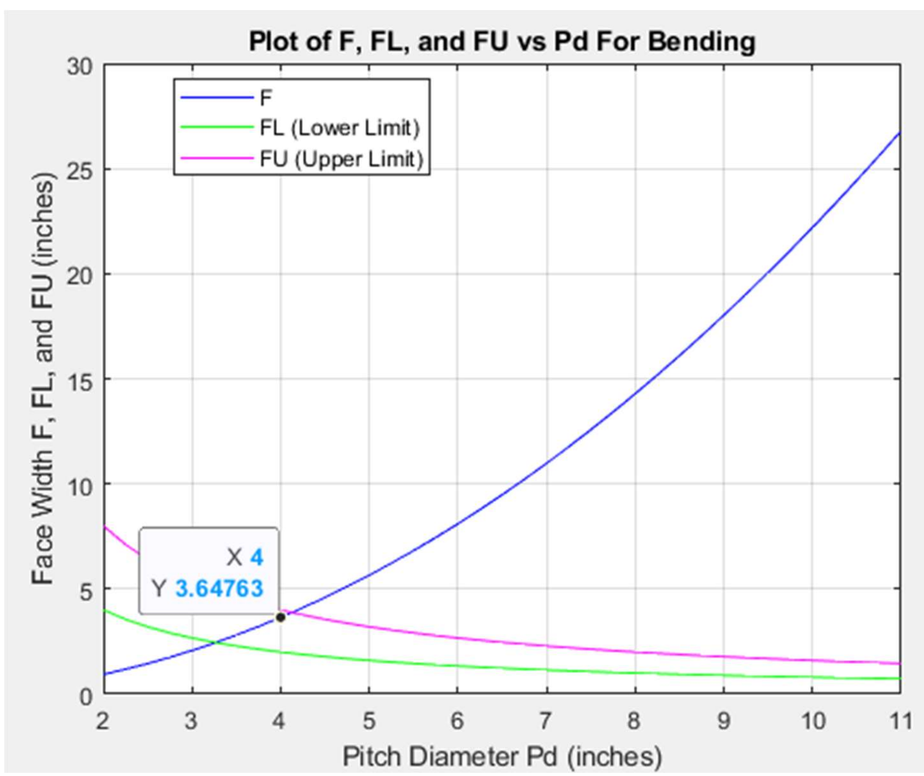


Figure 9. Bending Fatigue Analysis Pinion 3 & Gear 3

*** Surface Fatigue of Pinion**:(Phat)

$$\sigma_c = C_P \sqrt{\frac{W_t C_a C_m}{F I d C_v} C_s C_f}$$

1. Elastic Coefficient
$C_p = 2300psi^{0.5}(table\ 12 - 18)$

2. C_a, C_m, C_v, C_s *are equal respectively to* K_a, K_m, K_v, K_s

$$C_a = C_s = 1$$

$$C_m = K_m = 1.7$$

$$C_v = K_v$$

3. Surface finish factor $C_f = 1$

4. Surface geometry factor I:
$$I = \left(\frac{sin\emptyset cos\emptyset}{2}\right)\left(\frac{N_g}{N_p + N_g}\right) = \frac{(sin(25^o)\cos(25^o))}{2}\frac{56}{56+14} = 0.1532$$

5. Upper limit

$$F \le \frac{16}{p_d}$$

6. Lower limit

$$\frac{8}{p_d} \le F$$

7. Safety factor

$$N_{fs} = \left(\frac{S_{fc}}{\sigma_c}\right)^2$$

8. Face width

$$F = \frac{C_a W_t C_m}{C_V d I}\left(\frac{C_p}{S_{fcp}}\right)^2 N_{fs}$$

9. EnduRace strength:

$$S_{fcp} = \frac{C_L C_H}{C_T C_R} S'_{fc}$$

$$S'_{fc} = 175000\, psi \; (\, table\; 12 - 21)$$

$$C_T = K_T = 1$$

$$C_R = K_R = 0.85$$

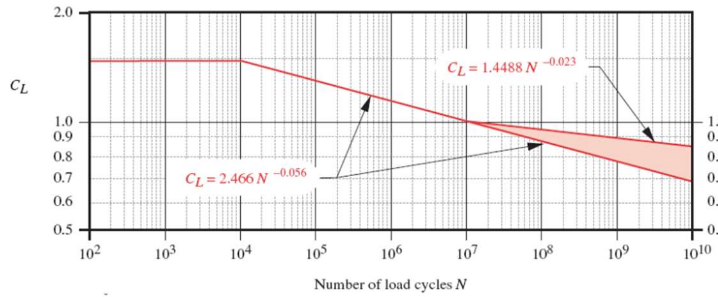


Figure 10. Life factor CL

$$C_L = 1.4488\,(6.785 \times 10^9)^{-0.023} = 0.86$$
$$C_H = 1\; (only\; applied\; to\; gear\; tooth\; strenght)$$
$$S_{fcp} = \frac{C_L C_H}{C_T C_R} S'_{fc} = \frac{0.86}{0.85}(175000 psi) = 172000\, psi$$

Using MATLAB to find facewidth.

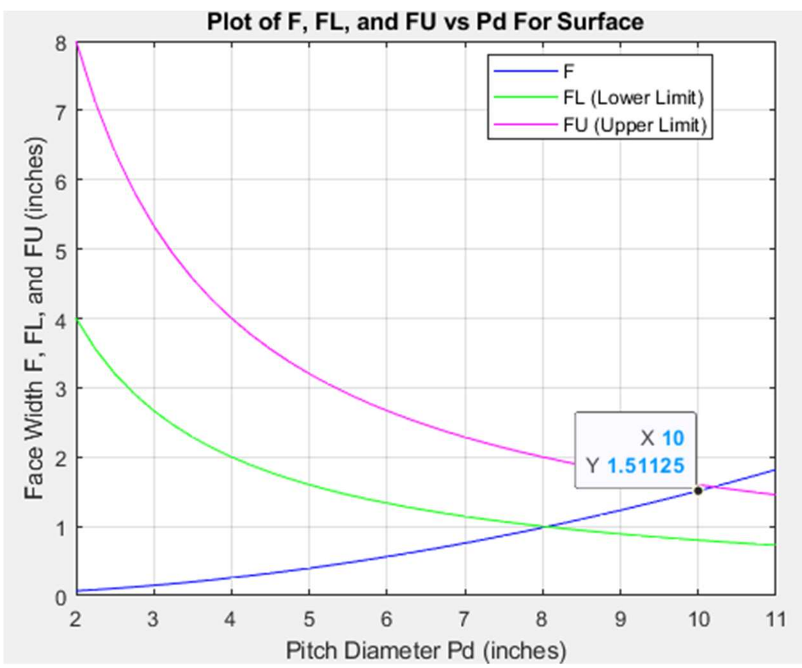


Figure 12. Surface Fatigue Analysis Pinion 1 & Gear 1:

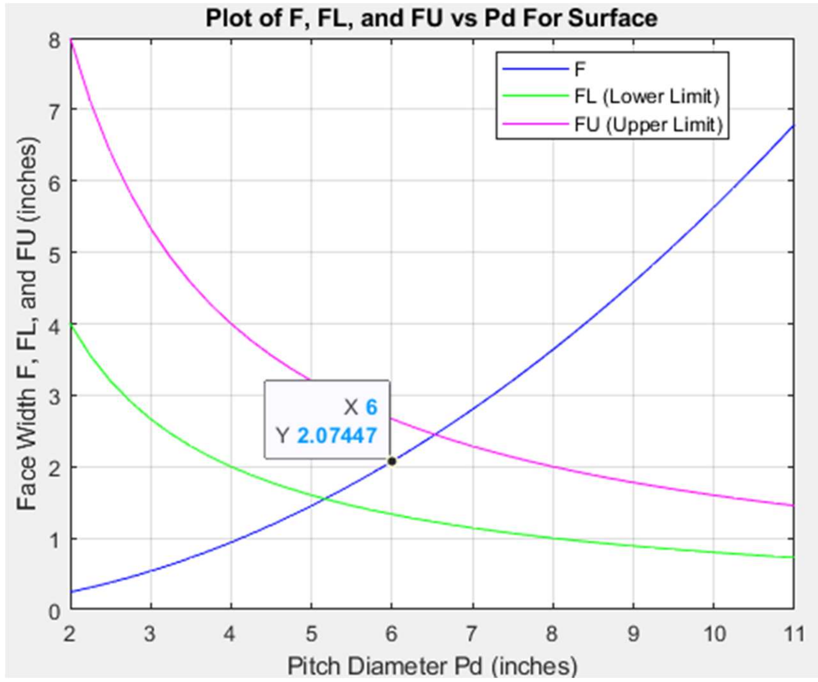


Figure 11. Surface Fatigue Analysis Pinion 2 & Gear

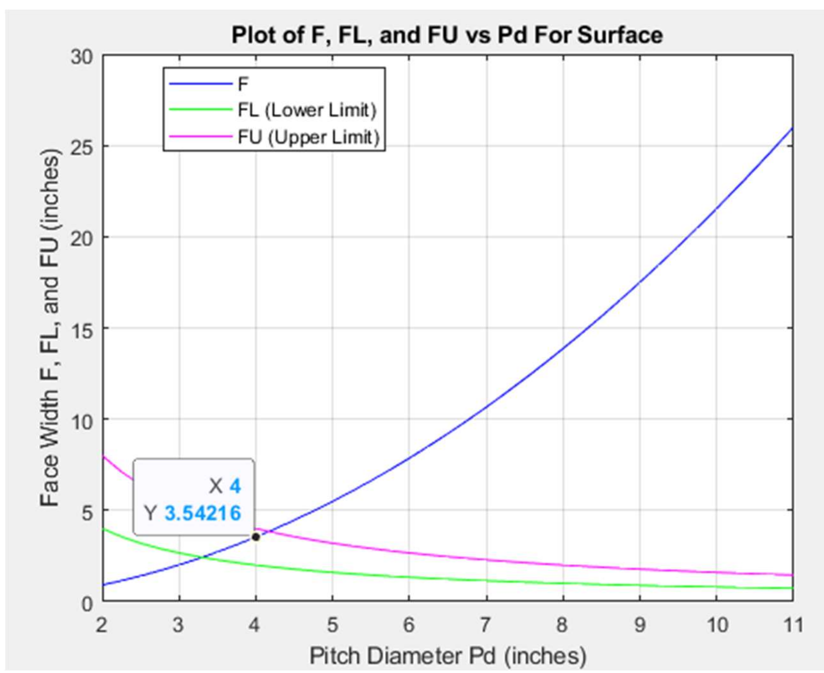


Figure 13. Surface Fatigue Analysis Pinion 3 & Gear 3

C. **<u>Gears & pinions facewidth decision</u>**

Final facewidths are determined to satisfy housing dimensions, stresses, and requirements:

$$F_{set\,1} = 1.6\ in\ and\ P_d = 10\ in^{-1}$$

$$F_{set\,2} = 2.15\ in\ and\ P_d = 6\ in^{-1}$$

$$F_{set\,3} = 3.65\ in\ and\ P_d = 4\ in^{-1}$$

D. Diameter of shaft 2: (Dakito)

Torque at shaft 1 = 525 lb-in, Sy = 61 ksi, Sut = 95 ksi, Nf = 1.5, $m_{g1} = 4$

$$d = \left\{\frac{32N_f}{\pi}\left[\left(K_f\frac{M_a}{S_e}\right)^2 + \frac{3}{4}\left(K_{fsm}\frac{T_m}{S_y}\right)^2\right]^{1/2}\right\}^{1/3}$$

1. Define endurance strength Se $= c_{load}c_{size}c_{surf}c_{temp}c_{rel}Se'$

Se' = 0.5 Sut = (0.5)(95 ksi) = 47.5 ksi

$c_{load} = 1$

Assume diameter range 0.3 in≤d ≤10 in => $c_{size} = 0.869d^{-0.097}$

Machine => $c_{surf} = 0.80773$

Room temperature => $c_{temp} = 1$

90% reliability=> $c_{relia} = 0.897$;

⇨ Se $= 29.907\ d^{-0.097}$

2. Define torque

Tm = 8400 lb.in

3. Define maximum moment

Ma – is found using singularity and the weight of gear #2

$$D_{pinion1} = \frac{14}{p_d} = 1.4$$

$$D_{pinio} \quad = \frac{14}{pd} = 2.33$$

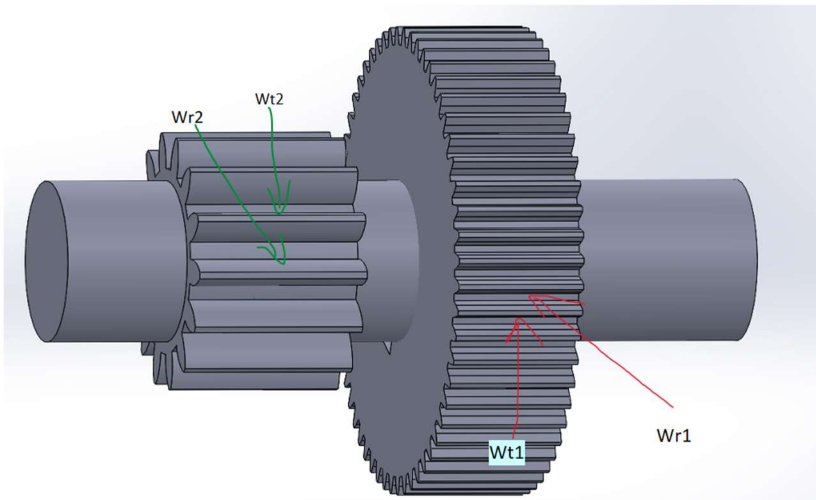
$$W_{t\,set1} = \frac{2T_{in}}{D_{pinion}} = \frac{2\,(525)}{1.4} = 750\ lb$$

$$W_{r\,set1} = W_{t\,set\,1}\tan 2\,5^0 = 350\ lb$$

$$W_{t\ set2} = \frac{2T_{in}}{D_{pinion}} = \frac{2\ (2100)}{2.33} = 1802.6\ lb$$

$$W_{r\ set2} = W_{t\ set\ 2} \tan 25^0 = 840.6\ lb$$

Weight of gear 1 is obtained from Solid Works = 9.72 lb and shaft 2 weight = 9.46 lb. However, weight is negligible since bending forces are acting on z-axis.



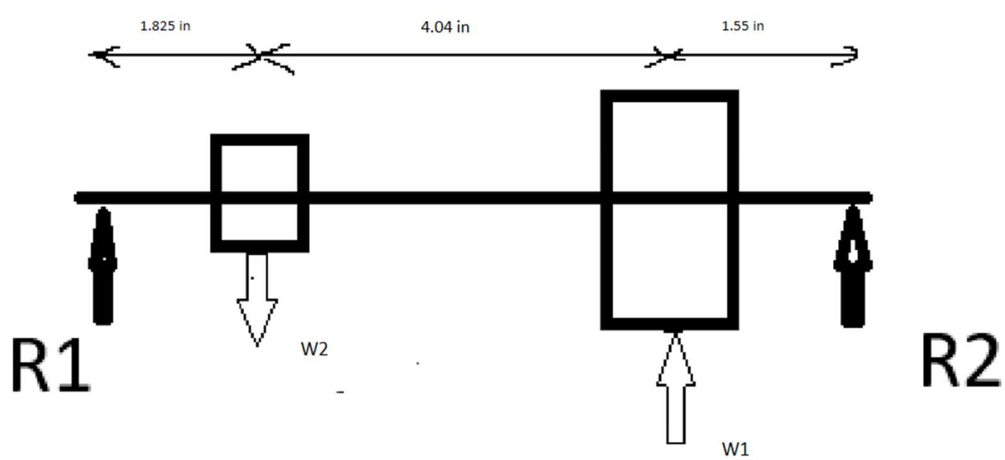


Figure 14. Shaft 2 force diagram.

$$q = R_1 < a - 0 >^{-1} - W_2 < a - 1.825 >^{-1} + W_1 < a - 4.71 >^{-1} + R_2 < a - 7.415 >^{-1}$$

$$V = R_1 < a - 0 >^{0} - W_2 < a - 1.825 >^{0} + W_1 < a - 4.71 >^{0} + R_2 < a - 7.415 >^{0}$$

$$M = R_1 < a - 0 >^{1} - W_2 < a - 1.825 >^{1} + W_1 < a - 4.71 >^{1} + R_2 < a - 7.415 >^{1}$$

$$At\ a = L^+, V = M = 0 => R_2 = W_2 - W_1 - R_1$$

$$=> \ R_1(7.415) - W_2(7.415 - 1.825) + W_1(7.415 - 4.71) = 0 => R_1 = \frac{5.59W_2 - 2.705W_1}{(7.415)}$$

a is replaced by x and W_2 is replaced by W_{r2} for radial x-direction (bending analysis)

$$R_{1x} = \frac{5.59(840.6) - 2.705(350)}{(7.415)} = 506\ lb$$

$$R_{2x} = W_{r2} - W_{r1} - R_{1x} = 840.6 - 350 - 506 = -15.4lb$$

a is replaced by y and W_2 is replaced by W_{t2} for tangential y-direction (torsion analysis)

$$R_{1y} = \frac{5.59(1802.6) - 2.705(750)}{(7.415)} = 1085.3\ lb$$

$$R_{2y} = 1802.6 - 750 - 1085.3 = -32.7\ lb$$

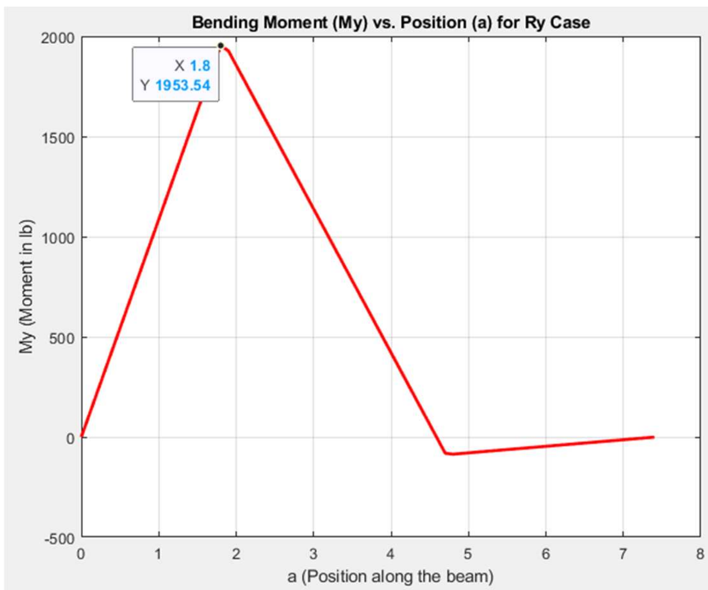


Figure 15 Bending moment plot

- Ma = 1953.53 lb in
  Ta= 998 lb in

$$Kf = 1 + 0.5 * (K_t - 1)$$
$$Kfs = 1 + 0.57 * (K_{ts} - 1)$$

Since the shaft is uniform => Kf=Kfs=1

Using MATLAB obtain **d= 1.522 inch => d=1.6 inch.**

E. <u>Key Gear 4:</u> (Phat)
AISI 4140 Heat treated

From Table 10-2 for $3.75 < d \leq 4.5$

$=> w = 1\ in$

$$\mathrm{F_a} = \frac{\mathrm{T_{max} - T_{min}}}{2(0.5\mathrm{d})} = \frac{33600\mathrm{lb}}{2*(0.5)*(4\ \mathrm{in})} = 8400\ \mathrm{lb}$$

$$\mathrm{F_m} = \frac{\mathrm{T_{max} + T_{min}}}{2(0.5\mathrm{d})} = \frac{33600\ \mathrm{lbin}}{2*(0.5)*(4\ \mathrm{in})} = 8400\ \mathrm{lb}$$

7.5 T = 16534.7 lb

Fmax = 16800lb => able to lift 7.5 T

Find $S_e = C_{load} * C_{size} * C_{surf} * C_{lemp} * C_{reb} * S'_e$

Bending → $\mathrm{C_{load}} = 1$

$$\mathrm{d_{eqv}} = \sqrt{\frac{\mathrm{A_{95}}}{0.0766}}$$

Shaft 4 diameter = 4 in => $0.3\ in \leq d_{eqv} \leq 10\ in,\ \ w = 1\ in\ (table\ 10-2)$

$C_{size} = 0.869 * d_{eqv}^{-0.097}$

Machined → $A = 2.7, b = -0.265$

$C_{surf} = 2.7(S_{ut}\ ksi)^{-0.265} = 2.7(95\ ksi)^{-0.265} = 0.8077$

$C_{lemp} = 1\ (room\ temp)$

$C_{reb} = 0.897\ (90\%\ reliability)$

Steel key => $\mathrm{S'_e} = 0.5\mathrm{S_{ut}} = 0.5(95\ \mathrm{ksi}) = 47.5\ \mathrm{ksi}$

$$S_e = \left(0.869\left(\sqrt{\frac{wL}{0.0766}}\right)^{-0.097}\right)(0.8077)(1)(0.897)(47.5\ ksi)$$

$where\ w = 1\ in$

$$L = \frac{\sqrt{3}N_d(S_{ut}F_a + S_eF_m)}{wS_{ut}S_e}$$

Use MATLAB find L

$\mathbf{L = 1.058\ in}$

$\left(using\ N_p = 17\ m_1 = m_2 = m_3 = 8, obtained\ L = 9.2\ inches\ which\ is\ longer\ than\ housing\ width\right)$

V. **Drawings** (Phat)

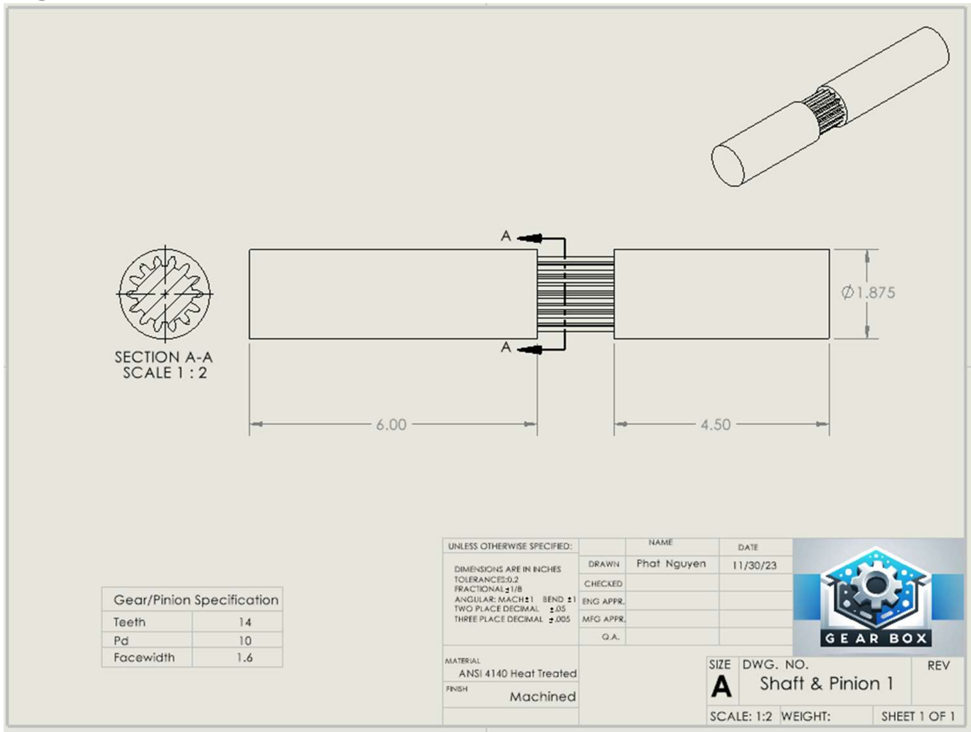


Figure 16. Shaft 1- Pinion 1

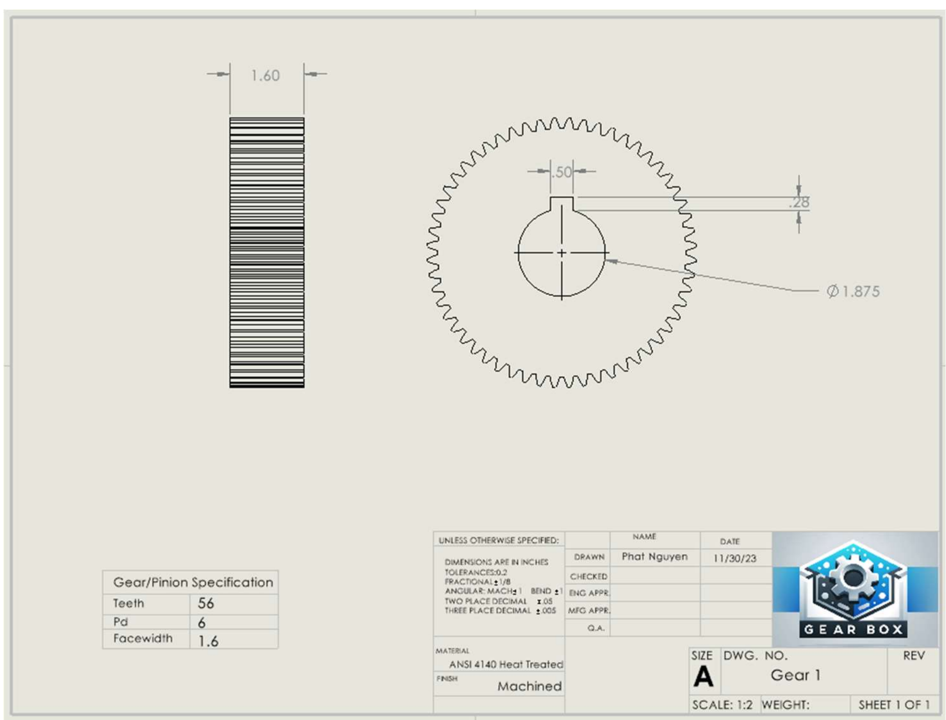


Figure 17. Gear 1

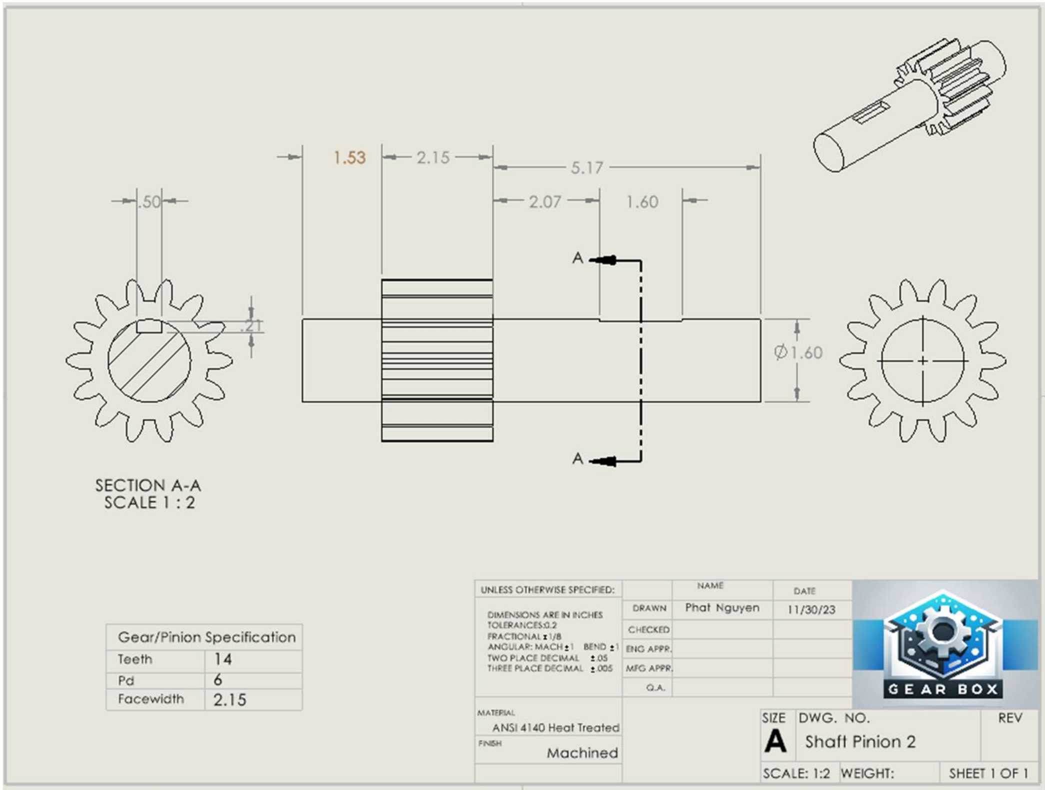


Figure 18. Shaft 2- Pinion 2

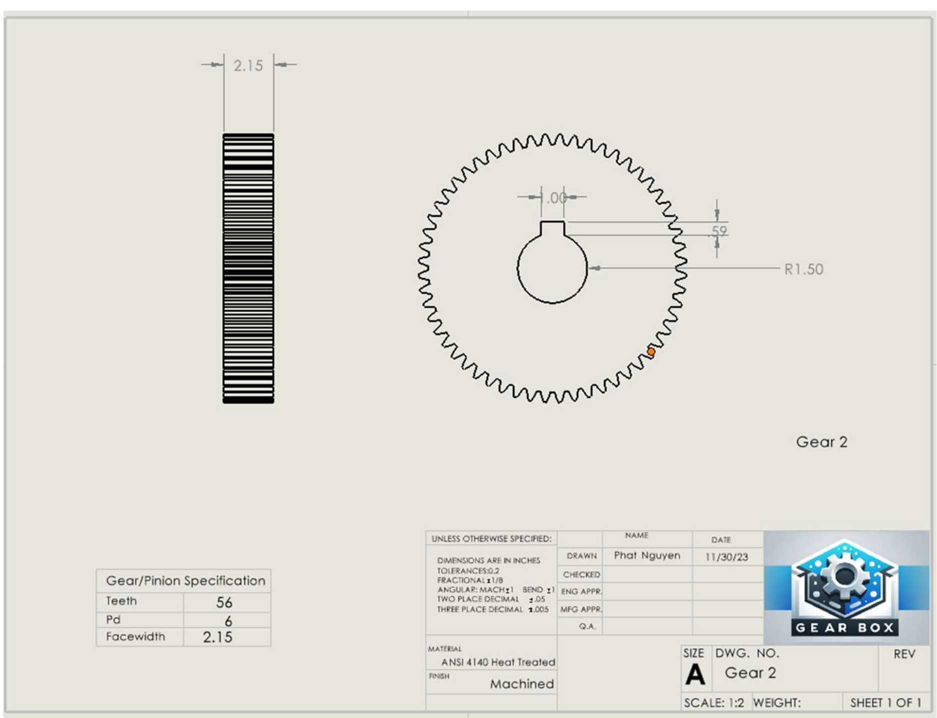


Figure 19. Gear 2

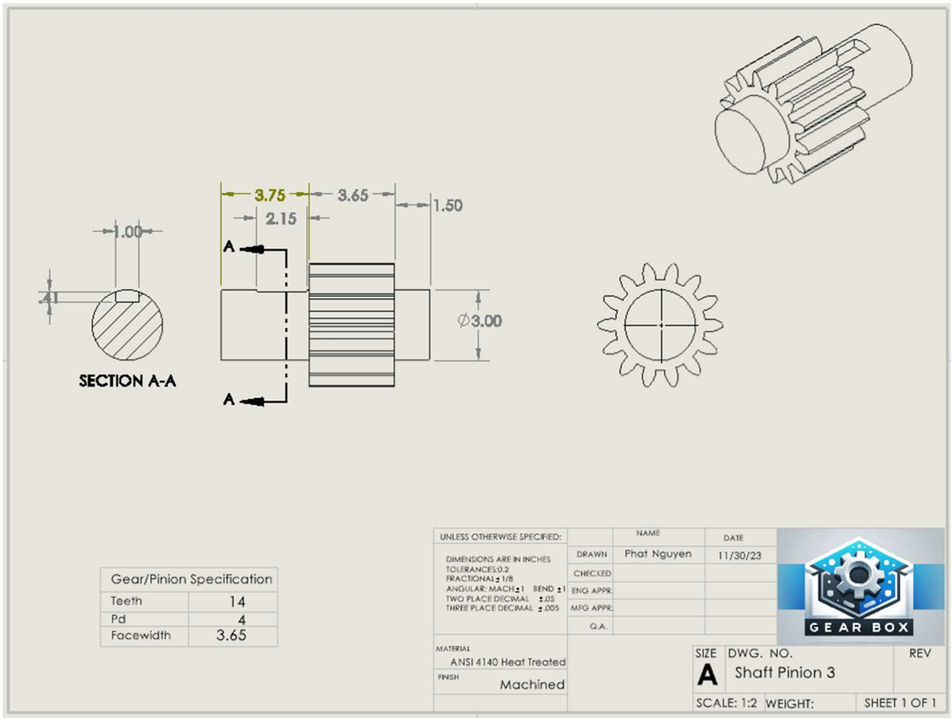


Figure 20. Shaft 3- Pinion 3

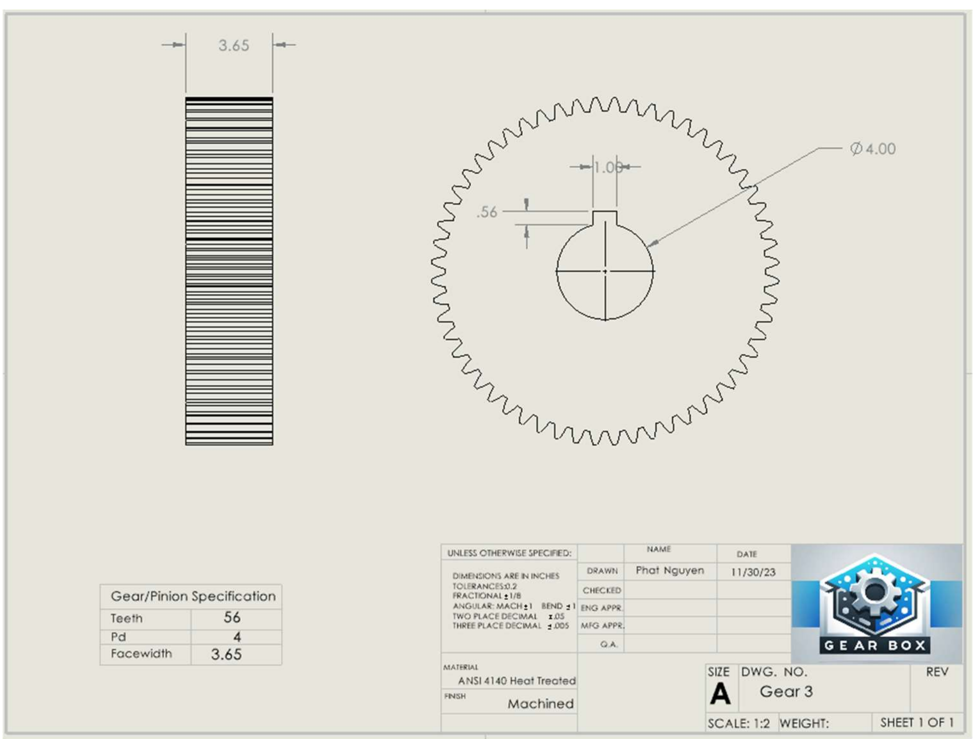


Figure 21. Gear 3

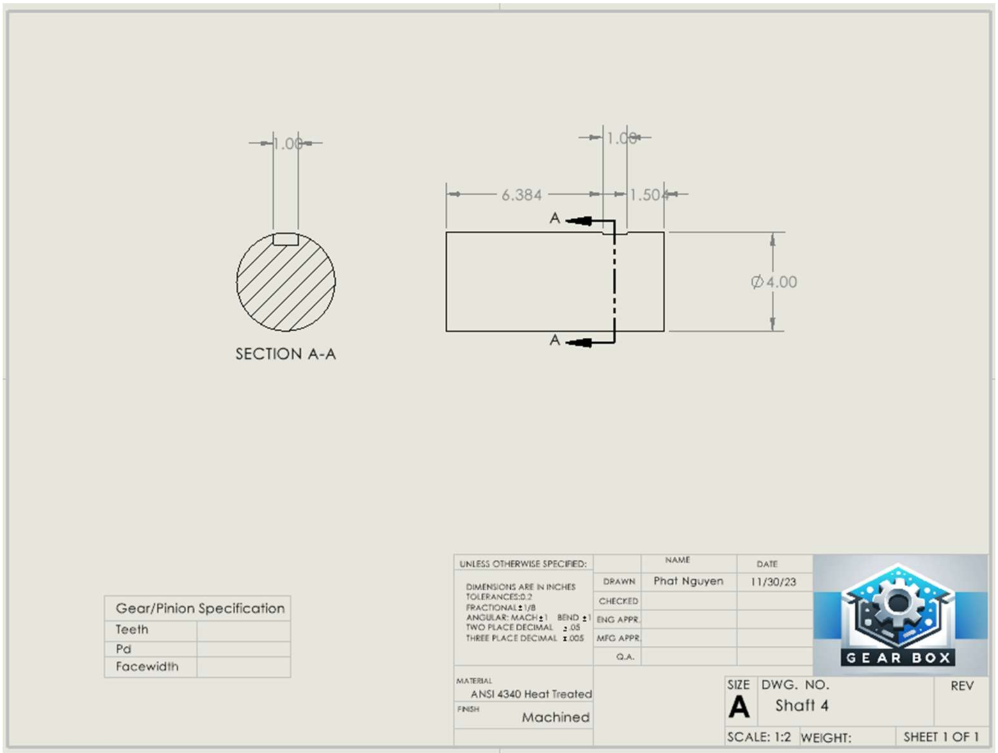


Figure 22. Shaft 4

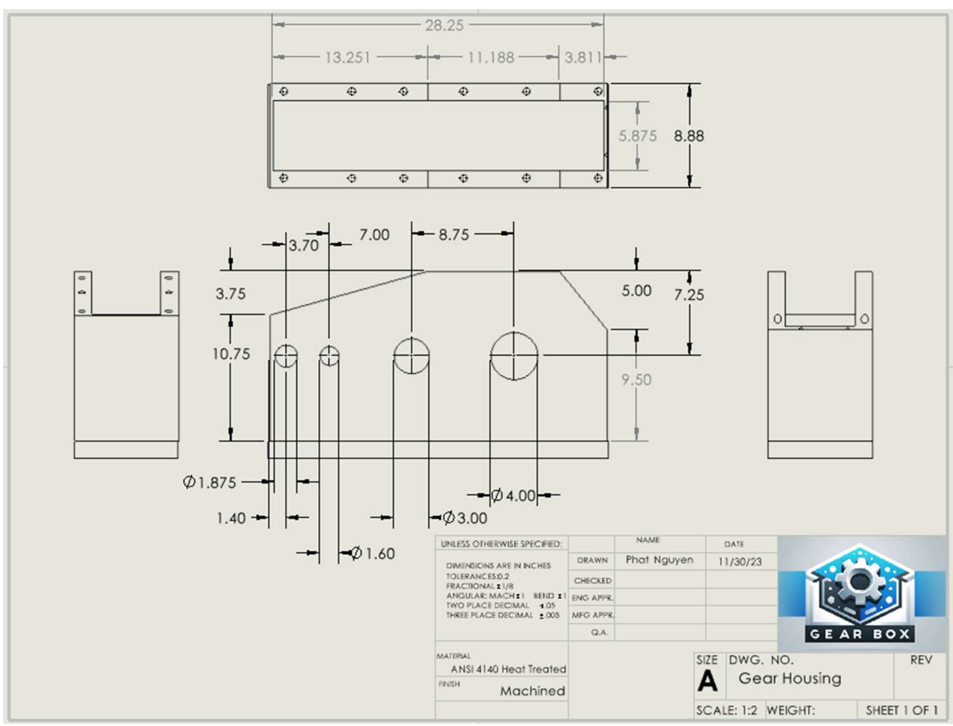


Figure 23. Housing Drawing

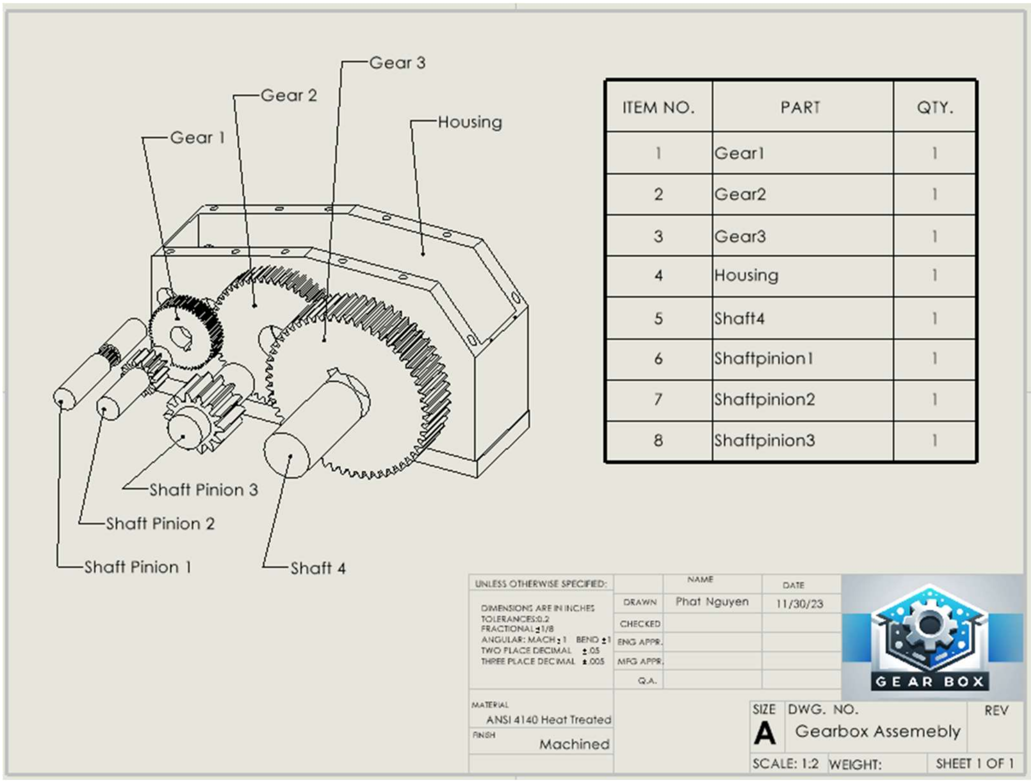


Figure 24. Gear Box Assembly

VI. Conclusion (Phat)

The Gearbox Project is a reversed engineer of a gearbox to power a hoist drum on an overhead crane, achieving the goal of lifting 7.5T at a maximum speed of 30 ft/min, as driven by a 15HP electric motor with 1800 rpm. The design specifications, the project developed a 3-stage speed reduction gearbox which increases the power of the motor up to 64 times, from 525 lb-in to torque to 33600 lb-in. A safety key placed in gear 4 and shaft 4 to ensure that the key connecting the gear to the output shaft is the weak link to fail first upon accidental overloads, thus providing a fail-safe mechanism.

The gear box is engineered for a 10-year lifespan with a 90% reliability factor, the gearbox is optimized for infinite shaft life, incorporating safety factors of 1.2 for surface and 1.5 for bending to ensure the key on the output shaft acts as a fail-safe. With iterative processes of face width choices, all gears, pinions, shafts fit into a given housing dimensions and meet all specific output requirements. The documentation presented in the final report reflects the design procedure, assumptions, calculations, and drawings. This project is a capstone of mechanical design which requires knowledge of pinion, gear, and shafts. This project is a capstone of mechanical design which requires knowledge of pinion, gear, shafts, and analysis skills such as MATLAB and SolidWorks.

Appendix

Bending fatigue MATLAB:

```
N_fb = 1.5; % Safety Factor, unitless
Phi = 25; % Pressure Angle, degrees
Qv = 11; % Gear Quality Index, unitless
Ng = 56; % Number of Teeth of Gear, teeth
Np = 14; % Number of Teeth of Pinion, teeth
np = 1800; % Revolution per minute, rpm
H = 15; % Power, hp
J = 0.34; % Geometry Factor, unitless
K_m = 1.7; % Load-Distribution Factor, unitless
K_a = 1; % Application Factor, unitless
K_s = 1; % Size factor, unitless
K_B = 1; % Rim thickness factor, unitless
K_I = 1; % Idler factor, unitless (set to 1.42 for an idler gear and 1 for a non-idler gear)
B = .25*(12-Qv)^.6667; % Coefficient for calculation, unitless
A = 50+56*(1-B); % Coefficient for calculation, unitless
Pd = 2:0.25:11; % Range of Pitch Diameters
dp = Np ./ Pd; % Pitch Diameter of Pinion
Vt = pi * dp * np; % Tangential Velocity
K_v = (A ./ (A + sqrt(Vt))).^B; % Dynamic Factor
W_t = H * 6600 ./ (Vt / 60); % Transmitted Load
S_fb = 42823 ; % Bending-Endurance Strength, psi
FU = 16 ./ Pd; % Upper Limit for F
FL = 8 ./ Pd; % Lower Limit for F

% Calculation of F, the Face Width
F = (W_t .* Pd .* K_a .* K_m) ./ (J .* K_v) .* K_s .* K_B .* K_I .* N_fb ./ S_fb;

% Plotting F, FL, FU vs Pd
figure;
plot(Pd, F, 'b', Pd, FL, 'g', Pd, FU, 'm');
title('Plot of F, FL, and FU vs Pd For Bending');
xlabel('Pitch Diameter Pd (inches)');
ylabel('Face Width F, FL, and FU (inches)');
legend({'F', 'FL (Lower Limit)', 'FU (Upper Limit)'}, 'Location', 'best');
grid on;
```

Surface fatigue MATLAB:

```
Nfs = 1.2; % Safety Factor, unitless
Phi = 25; % Pressure Angle, degrees
Qv = 11; % Gear Quality Index, unitless
Ng = 56; % Number of Teeth of Gear, teeth
Np = 14; % Number of Teeth of Pinion, teeth
np = 1800; % Revolution per minute, rpm
H = 15; % Power, hp
I = 0.1532; % Geometry Factor, unitless
C_m = 1.7; % Load-Distribution Factor, unitless
C_a = 1; % Application Factor, unitless
C_s = 1; % Size factor, unitless
C_f =1; %Surface finish factor
C_p=2300;
K_B = 1; % Rim thickness factor, unitless
K_I = 1; % Idler factor, unitless (set to 1.42 for an idler gear and 1 for a non-idler gear)
B = .25*(12-Qv)^.6667; % Coefficient for calculation, unitless
A = 50+56*(1-B); % Coefficient for calculation, unitless
Pd = 2:0.25:11; % Range of Pitch Diameters
dp = Np ./ Pd; % Pitch Diameter of Pinion
Vt = pi * dp * np; % Tangential Velocity
C_v = (A ./ (A + sqrt(Vt))).^B; % Dynamic Factor
W_t = H * 6600 ./ (Vt / 60); % Transmitted Load
Sfcp = 172000 ; % Surface-Endurance Strength, psi
FU = 16 ./ Pd; % Upper Limit for F
FL = 8 ./ Pd; % Lower Limit for F
% Calculation of F, the Face Width
F = (C_a.*W_t.*C_m)./(C_v.*dp.*I) .*(C_p./Sfcp).^2 *Nfs;
% Plotting F, FL, FU vs Pd
figure;
plot(Pd, F, 'b', Pd, FL, 'g', Pd, FU, 'm');
title('Plot of F, FL, and FU vs Pd For Surface');
xlabel('Pitch Diameter Pd (inches)');
ylabel('Face Width F, FL, and FU (inches)');
legend({'F', 'FL (Lower Limit)', 'FU (Upper Limit)'}, 'Location', 'best');
grid on;
```

Weight of Gear 1

Gear1.SLDPRT

Options...

Override Mass Properties... Recalculate

- [x] Include hidden bodies/components
- [ ] Create Center of Mass feature
- [ ] Show weld bead mass

Report coordinate values relative to: -- default --

```
Mass properties of Gear1
    Configuration: Default
    Coordinate system: -- default --

Density = 0.28 pounds per cubic inch

Mass = 9.72 pounds

Volume = 34.27 cubic inches

Surface area = 109.48 square inches

Center of mass: ( inches )
    X = 0.00
    Y = -0.01
    Z = 0.80

Principal axes of inertia and principal moments of inertia: ( pounds * square
Taken at the center of mass.
    Ix = ( 1.00, 0.00, 0.00)     Px = 23.17
    Iy = ( 0.00, 1.00, 0.00)     Py = 23.23
    Iz = ( 0.00, 0.00, 1.00)     Pz = 42.25

Moments of inertia: ( pounds * square inches )
Taken at the center of mass and aligned with the output coordinate system.
    Lxx = 23.17      Lxy = 0.00      Lxz = 0.00
    Lyx = 0.00       Lyy = 23.23     Lyz = 0.00
    Lzx = 0.00       Lzy = 0.00      Lzz = 42.25

Moments of inertia: ( pounds * square inches )
Taken at the output coordinate system. (Using positive tensor notation.)
    Ixx = 29.39      Ixy = 0.00      Ixz = 0.00
    Iyx = 0.00       Iyy = 29.45     Iyz = -0.05
    Izx = 0.00       Izy = -0.05     Izz = 42.25
```

Shaft moment

```matlab
clc
clear all
close all
R1x = 506 ;% lb
R2x = -15.4;%lb
Wr1 = 350;%radial force
Wr2 = 840.6;%radial force
% Define constants for Ry case
R1y = 1085.3; % lb
R2y = -32.7;
Wt1 = 750;%tangential force
Wt2 = 1802.6;%tangential force
% Define the range for a
a = 0:0.1:7.415;
% Calculate M for Rx case
Mx = R2x * (a - 0) - Wr2 * (a - 1.825).*(a>1.825) + Wr1*(a-4.71).*(a>4.71) +R1x *(a - 7.415).*(a>7.415);
% Calculate M for Ry case
My = R1y * (a - 0) - Wt2 * (a - 1.825).*(a>1.825) + Wt1 * (a - 4.71).*(a>4.71) + R2y * (a - 7.415).*(a>7.415);
% Plotting
plot(a, Mx, 'b', 'LineWidth', 2); % Plot Mx (moment along x-axis) in blue
xlabel('a (Position along the beam)');
ylabel('Mx (Moment in lb)');
title('Bending Moment (Mx) vs. Position (a) for Rx Case');
grid on; % Add grid for better readability
% Plotting the results for Ry case
figure;
plot(a, My, 'r', 'LineWidth', 2); % Plot My (moment along y-axis) in red
xlabel('a (Position along the beam)');
ylabel('My (Moment in lb)');
title('Bending Moment (My) vs. Position (a) for Ry Case');
grid on; % Add grid for better readability
```

Shaft diameter

```
syms d
N = 1.5;
Tm =8400;
Sut = 46.58E3;
Ma = 1953.53;
Se = 29.9*(d^-0.097)*1E3;
Ta = 998;
Mm = 0.6125;
Kf = 1;
Kfs = 1;
eq = d == (32*N/pi*((sqrt((Kf*Ma)^2 + (3/4)*(Kfs*Ta)^2)/Se) + (sqrt((Kf*Mm)^2 + (3/4)*(Kfs*Tm)^2)/Sut)))^(1/3);
d = vpasolve(eq, d)
```