[GEAR PUMP DESIGN PROJECT]

WRITERS
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Contents

1. Modified items

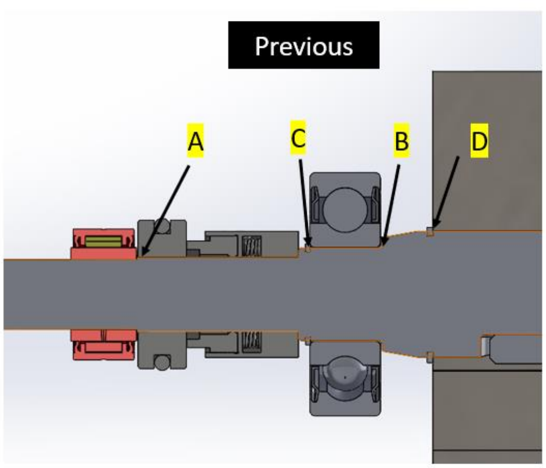
1.1 Summary of Safety Factor Change

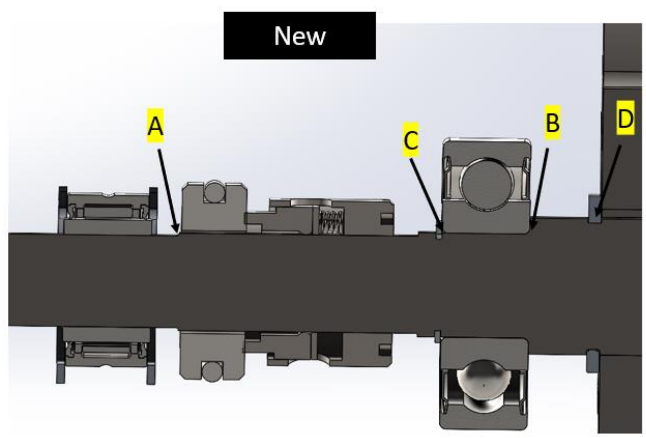
Component	Material (Before)	Material (New)	Key feature (Before)	Key Feature (New)	Stress Type	Safety factor (Before)	Safety factor (New)
Primary shaft (Center)	AISI 1040 CD	AISI 1030 Q&T	Ø 1.0625"	Ø 0.875"	Fatigue	6.80	5.44
Secondary shaft (Center)	AISI 1040 CD	AISI 1045 CD	Ø 1.0625"	Ø 0.875"	Fatigue	6.10	3.9
Key	AISI 1015 HR	AISI 1010 CD	1/4" x 1/4" x 3/4"	1/8" x 3/16" x 3/4"	Shear	3.76	3.52
					Bending	3.26	2.15

The main changes to the primary shaft design were the material, which was updated from AISI 1040 CD to AISI 1030 Q&T, and the center diameter, which was reduced from 1.0625 inches to 0.875 inches. As a result, the safety factor at the center of the shaft decreased from 6.8 to 5.44.

The material for the second shaft was changed from AISI 1040 CD to AISI 1045 CD, and the center diameter was adjusted to match that of the primary shaft. As a result, the safety factor decreased from 6.1 to 3.9.

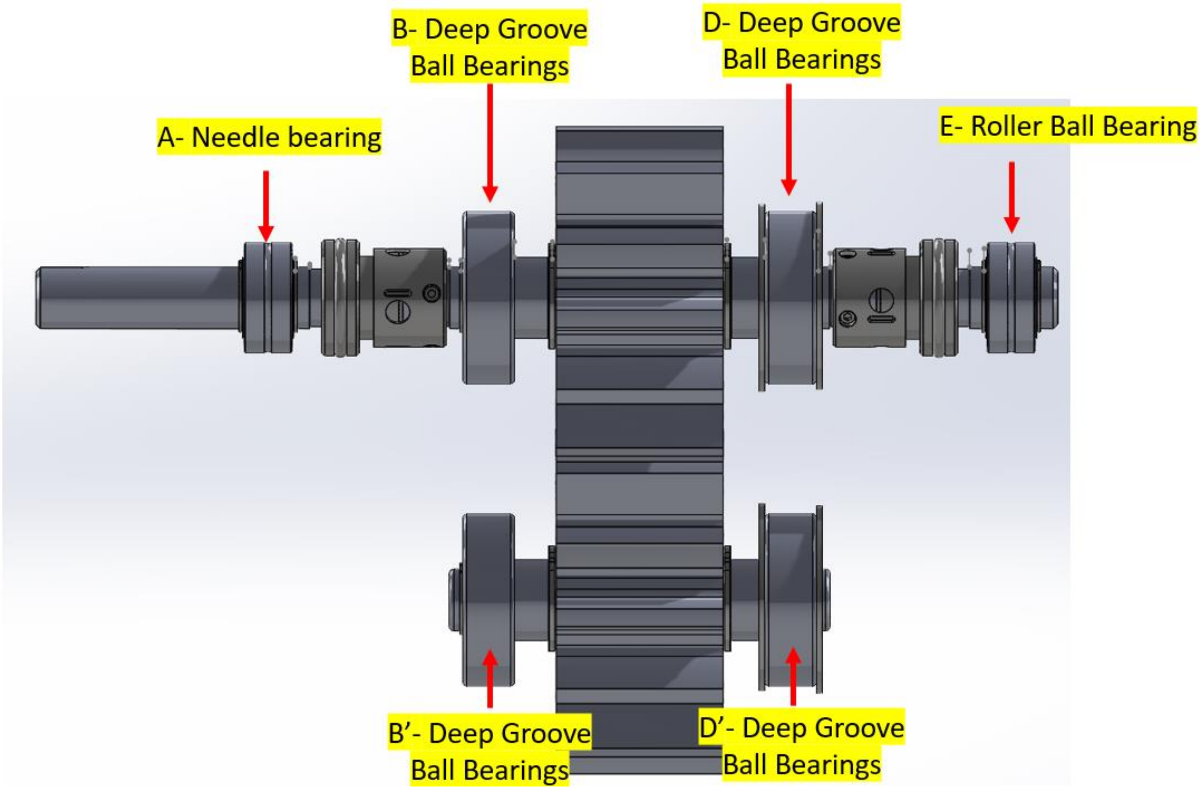
Additionally, the key material was changed from AISI 1015 HR to AISI 1010 CD, and its size was reduced from 1/4” x 1/4” to 1/8” x 3/16”. These changes lowered the shear safety factor from 3.76 to 3.52 and the bending safety factor from 3.26 to 2.15.





Location	Safety factor (Before)	Safety factor (New)
A	2.8	3.89
B	5.29	4.81
C	3.64	2.59
D	4.25	7.14

Since the interim report analyzed only four sections, the same sections in the new design were selected for comparison. In the new design, the reduced diameter caused the safety factor at location C to decrease. However, the safety factor at location D increased due to a wider groove and larger radius, which were made possible by using wider retaining rings for the gear and a change in material.



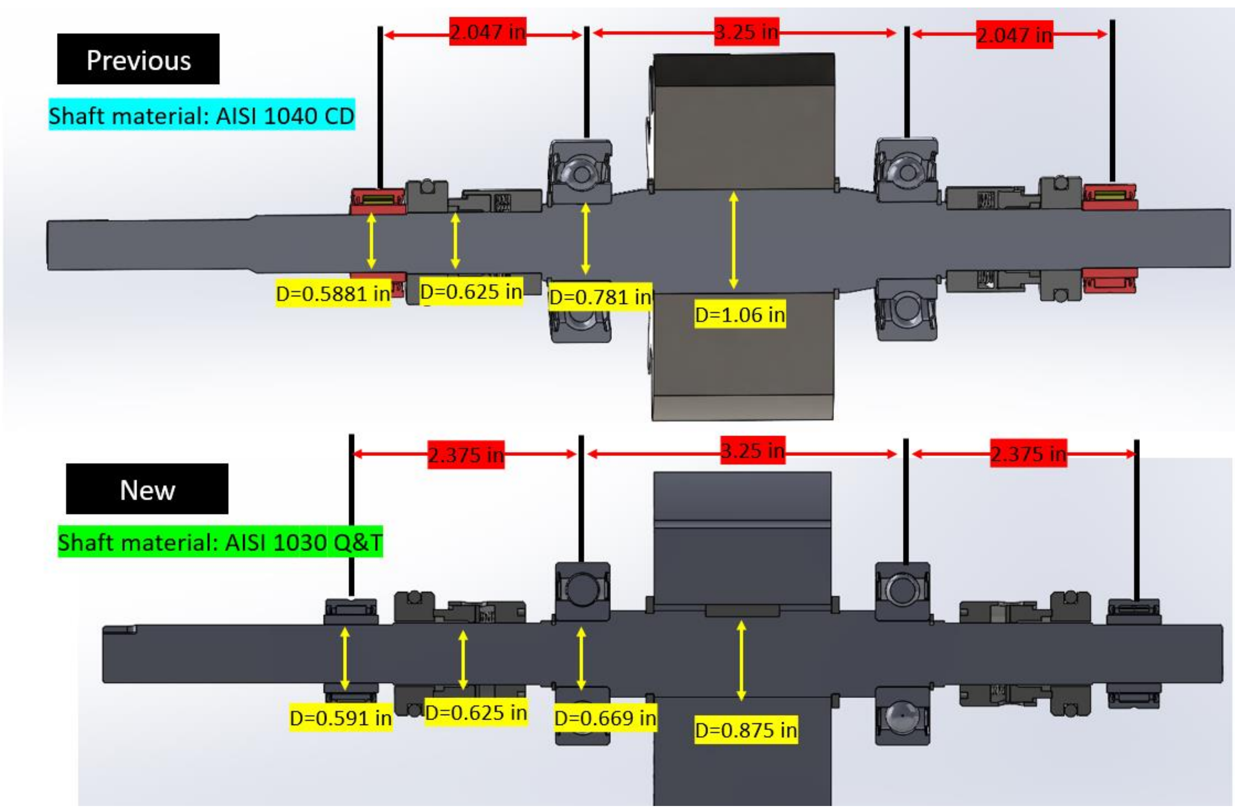
The bearing changes were driven by the shaft diameter adjustments. For the deep groove ball bearings, a smaller size was used, which resulted in an almost half reduced rating life compared to the previous bearings. The needle bearing remained unchanged.

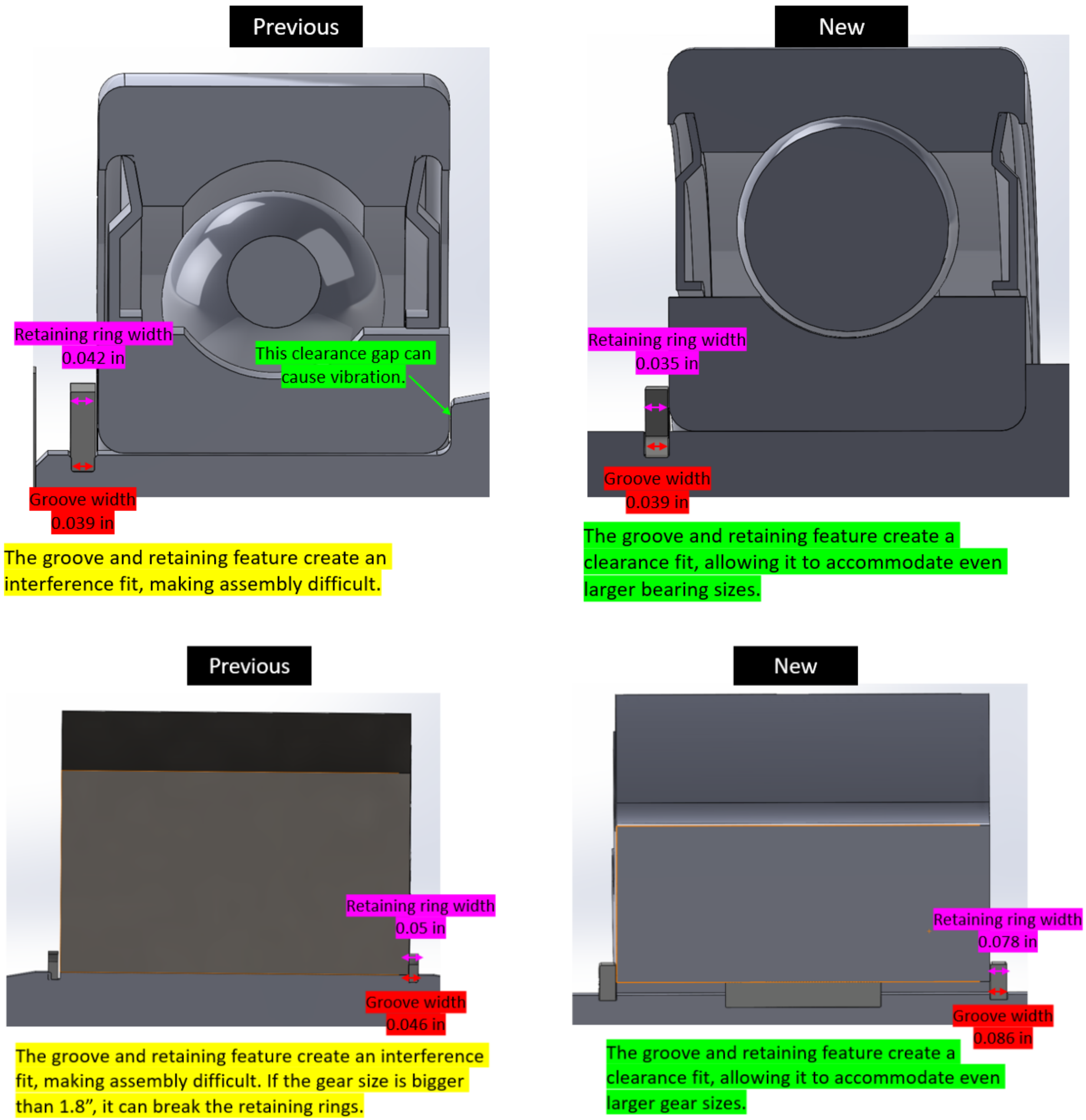
Location	Type	Model (Before)	Model (New)	Rating Life(hr) (Before)	Rating Life(hr) (New)
B & D	Deep groove ball bearing	6304-2Z	6303-2Z	102,507	54,157
B' & D'	Deep groove ball bearing	6304-2Z	6303-2Z	102,507	54,157

1.2 Primary Shaft

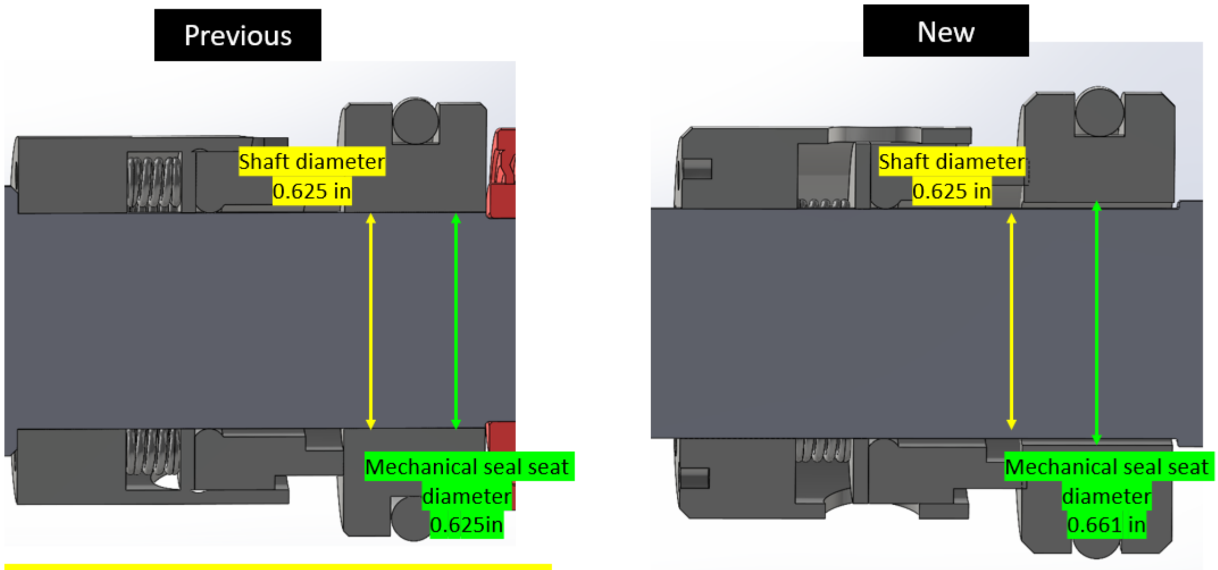
The material of the primary shaft was changed from 1040 CD to 1045 CD because the safety factor of the groove near the needle bearing was below 2.5, making it prone to failure before the key. Additionally, after further analysis, we discovered that the safety factor in the key area at the end of the shaft was below 2.5. As a result, we had to change the material again, this time from 1045 CD to 1030 Q&T, which offers higher yield and tensile strength.

Additionally, there was an error in setting the distance between the needle bearing and the deep groove ball bearings, which ended up being 2.047 inches. This has now been corrected to 2.375 inches as specified in the input. Another major change involved the shaft diameter.





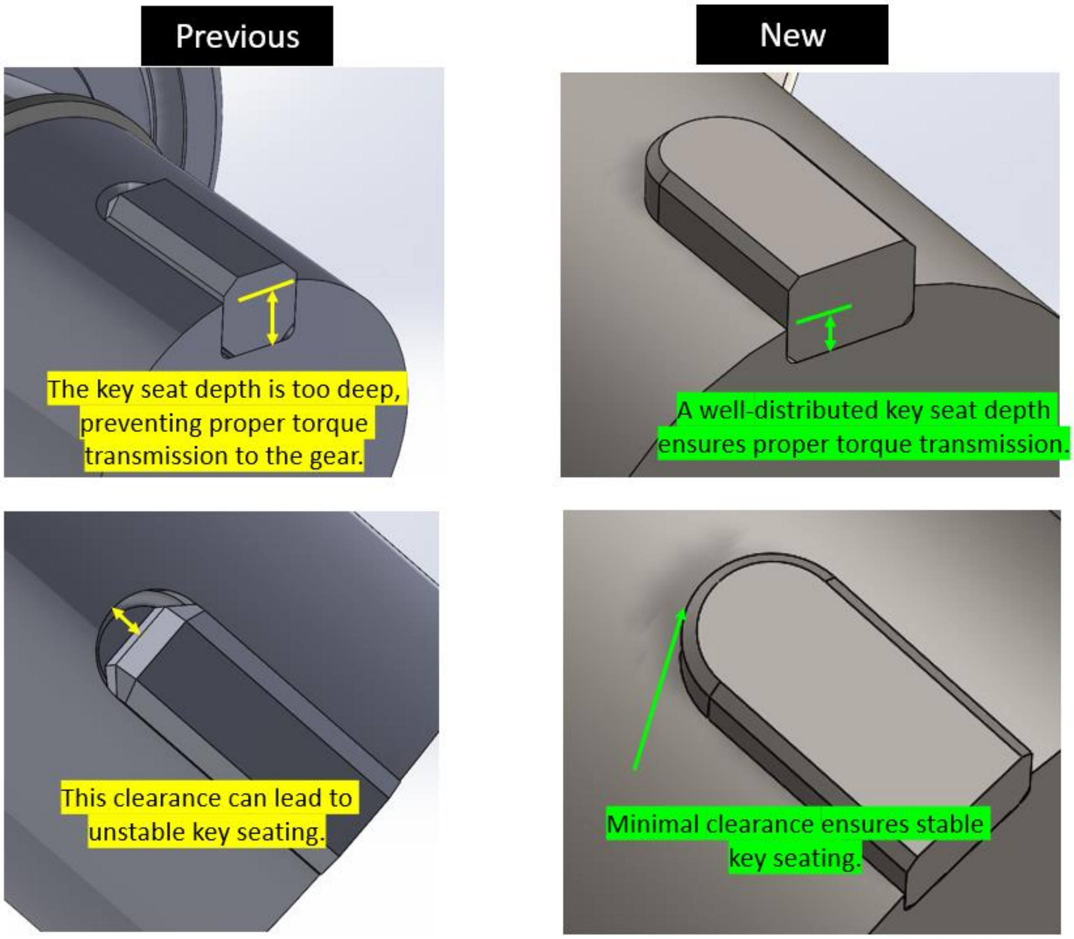
The grooves near both the gear and the deep groove ball bearings had interference issues with the selected retaining rings. In the worst-case scenario, this could have led to the retaining ring breaking and causing instability. To address this, we added clearance to ensure no damage would occur, even in the worst case.



The mechanical seal seat moves along the shaft, preventing it from properly sealing the oil.

The mechanical seal seat will rest on the housing, ensuring proper oil sealing.

This isn’t directly related to the primary shaft design changes. However, we selected a mechanical seal seat larger than the shaft diameter and mounted it on the housing to ensure proper oil sealing.



The key seat depth was too deep, which made it unlikely for the key to transmit torque properly and could have caused high stress at the contact area with the gear. We adjusted the key seat to ensure the contact area is large enough to withstand the applied stress. Additionally, we eliminated the too much clearance between the shaft and the key to prevent rattling during rotation.

1.3 Secondary Shaft

The secondary shaft underwent the same changes as the primary shaft. The material was changed from AISI 1040 to AISI 1045, and the diameter was adjusted similarly to the primary shaft.

1.4 Bearing

The deep groove ball bearings needed to be changed due to the shaft diameter adjustment. The bearings were updated from SKF 6304-2Z to SKF 6303-2Z to accommodate the diameter reduction from 0.781 inches to 0.669 inches. The needle bearings weren't changed.

1.5 Key

The initial safety factor for the key was 3.76 for shear and 3.26 for bending, which was higher than that of certain sections in the shaft. To ensure the key would fail before the shaft in case of overload, we needed to lower the safety factor. The original material, AISI 1015 HR, was also not ideal for machining. Therefore, we changed the material to AISI 1010 CD and reduced the key size from 1/4" x 1/4" to 1/8" x 3/16" as part of the adjustment.

1.6 Gear housing

The changes to the shaft material and diameter also impacted the design of the gear housing. The initial shaft material, AISI 1040 CD, had a yield strength of 71 ksi, while the new material, AISI 1045 CD, has a yield strength of 77 ksi. Additionally, the shaft diameter was reduced from 1.06 inches to 0.875 inches. These changes affected the outlet diameter, which was adjusted from 0.856 inches to 0.847 inches, and the required thickness of the housing between the outside of the groove and any edge or hole, which increased from 0.19 inches to 0.206 inches.

2. Executive Summary

2.1 Initial Design Parameters

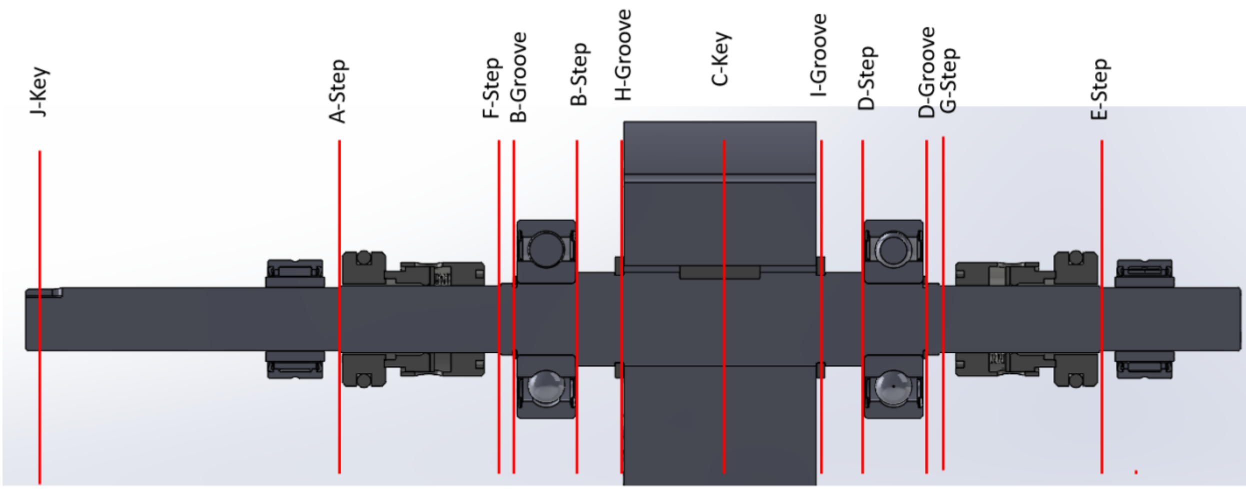
The following list represents the initial design parameters. When design changes become necessary, these parameters are kept the same as much as possible. In real-life scenarios, when customers specify certain requirements, engineers are expected to find alternative solutions that meet these specifications without altering the original requirements.

- Horsepower (HP): 10
- Revolutions per minute (RPM): 1500
- Distance between Bearings B and C: 3.25 inches

- Distance between Bearings A and B: 2.375 inches
- Gear Housing Efficiency: 0.7
- Gear Housing Service Factor: 1
- Gear Housing Inlet Diameter: 1 inch
- Number of Gear Teeth: 13
- Diametral Pitch: 4 teeth per inch
- Gear Width (b): 1.8 inches

2.2 Summary of Key Design

Primary Shaft Summary

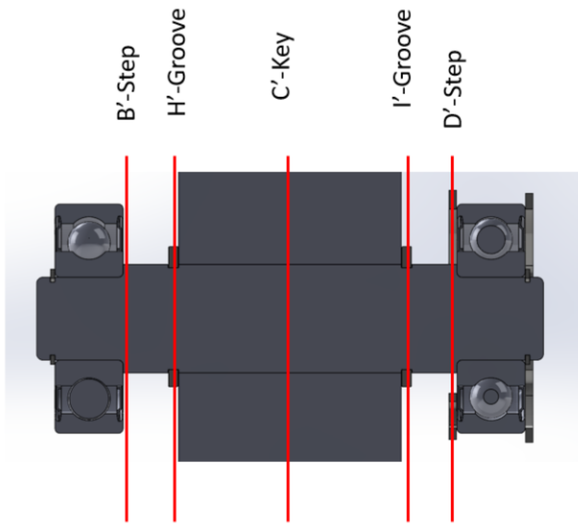


Section	Diameter (in)	X-location	Torque (lb-ft)	Shear Force (lb)	Bending Moment (lb-in)	Radius (in)	Thickness (in)	Width (in)	q	qs	D/d	r/d	r/t	a/t	Kt	Kts	Safety Factor
A (Step)	0.591	0.4513	35.01	-22.44	-10.13	0.017	0.017		0.700	0.7	1.058	0.029			2.1	1.4	3.89
B (Groove)	0.669	2.08	35.01	-22.44	-46.68	0.005	0.041	0.039	0.600	0.75			0.123	0.963	5.1	3	2.59
B (Step)	0.669	2.6505	35.01	129.28	-17.68	0.039	0.103		0.780	0.82	1.308	0.058			1.9	1.6	4.81
C (Key)	0.875	4	35.01	129.28	156.79	0.016	0.063	0.188	0.700	0.7					2.158	2.235	5.44
D (Step)	0.669	5.3495	0.00	-129.28	-17.68	0.039	0.103		0.780	0.82	1.308	0.058			1.9	1.6	30.17
D (Groove)	0.669	5.92	0.00	22.44	-46.68	0.005	0.041	0.039	0.600	0.75			0.123	0.963	5.1	3	5.65
E (Step)	0.591	7.5487	0.00	22.44	-10.13	0.017	0.017		0.700	0.7	1.058	0.029			2.1	1.4	35.38
F (Step)	0.625	1.9502	35.01	-22.44	-43.76	0.005	0.022		0.600	0.75	1.070	0.008			2.65	1.9	2.92
G(Step)	0.625	6.0498	0.00	22.44	-43.76	0.005	0.022		0.600	0.75	1.070	0.008			2.65	1.9	7.62
H (Groove)	0.875	3.057	35.01	129.28	34.87	0.005	0.027	0.086	0.600	0.75			0.185	3.185	4.1	2.5	7.14
I (Groove)	0.875	4.943	0.00	-129.28	34.87	0.005	0.027	0.086	0.600	0.75			0.185	3.185	4.1	2.5	19.79
J (Key)	0.591	-	35.01	0.00	0.00	0.016	0.063	0.188	0.700	0.7					2.158	2.235	2.69

The weakest point of the primary shaft is the groove area near the needle bearing. This is primarily due to the high Kt and Kts values resulting from the groove shape. Additionally, the small radius reduces notch sensitivity and increases stress concentration, leading to a safety factor of 2.59.

The area where the highest torque and bending moment are applied is location C, where the key seat is located. Nevertheless, due to the larger diameter in this area, the safety factor remains at 5.44.

Secondary Shaft Summary



Section	Diameter (in)	X-location	Torque (lb-ft)	Shear Force (lb)	Bending Moment (lb-in)	Radius (in)	Thickness (in)	Width (in)	q	qs	D/d	r/d	r/t	a/t	Kt	Kts	Safety Factor
B' (Step)	0.669	2.08	0	129.28	35.62	0.039	0.103		0.7	0.73	1.31	0.06			1.92	1.62	12.82
C' (Key)	0.875	1.625	35.01	129.28	210.08	0.0156	0.615		0.62	0.65					2.158	2.235	3.9
D' (Step)	0.669	1.17	0	-129.28	35.62	0.039	0.103		0.7	0.73	1.31	0.06			1.92	1.62	12.82
H'(Groove)	0.875	0.682	0	129.28	88.17	0.005	0.027	0.086	0.5	0.55			0.185	3.185	4.1	2.5	7.26
I' (Groove)	0.875	2.568	0	-129.28	88.17	0.005	0.027	0.086	0.5	0.55			0.185	3.185	4.1	2.5	7.26

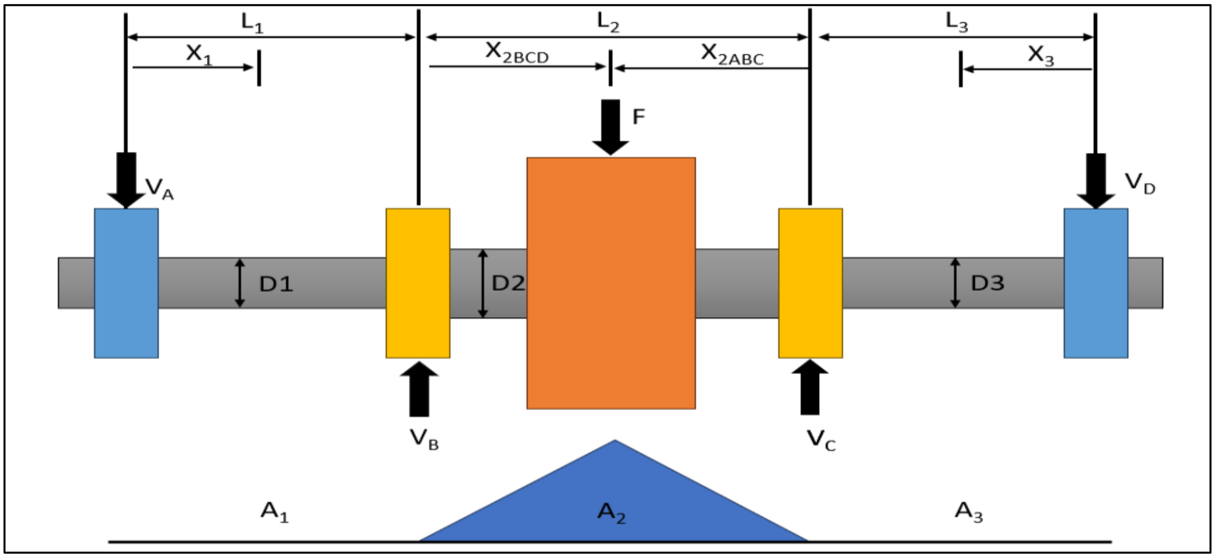
The C' area of the secondary shaft, where the key seat is located, has the lowest safety factor, even lower than that of the primary shaft. This is attributed to the assumption that 100% of the torque is transmitted through the center of the secondary shaft, coupled with a higher bending moment compared to the primary shaft.

3. Force Analysis

3.1 Primary Shaft

The force applied to the primary shaft will be analyzed using the diagram below. Needle bearings are located at points A and D, while roller ball bearings are at points B and C.

The load is applied at the center of the diagram, where the gear rotates.



3.1.1 Torque Analysis

The initial torque is transmitted from the electric motor to the shaft end, maintaining a stable torque level until it reaches the dissipation point, where the gear is located. Beyond this point, the torque dissipates entirely, resulting in zero torque.

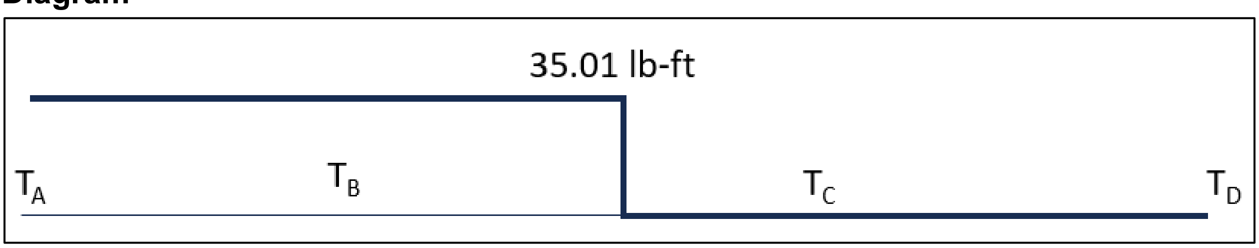
Calculation

$$T = \frac{H}{n} \times 550 \times 60 \times \frac{1}{2\pi} = \frac{10}{1500} \times 550 \times 60 \times \frac{1}{2\pi} = 35.01\ lb - ft$$

Results

- T_A = 35.01 lb-ft
- T_B = 35.01 lb-ft
- $T_{B\text{-}C}$ = 35.01 lb-ft
- T_C = 0
- T_D = 0

Diagram



3.1.2 Bending Moment Analysis

3.1.2.1 First approach

Given that there are four bearings and a single transmitted load, calculating the reaction forces at locations A, B, C, and D directly is not feasible. Therefore, we assumed that the lengths between A-B and C-D are identical, with the load applied precisely at the center of each bearing. Under these assumptions, the reaction forces at A and D, as well as at B and C, are considered identical, making it possible to obtain preliminary results.

However, this approach does not account for shaft deflection, which can impact the accuracy of the reaction force distribution. To address this, an additional method that incorporates shaft deflection analysis should be applied for more precise results.

3.1.2.2 Second approach

The following 3-moment equation was applied to account for shaft deflection, considering the varying diameters across each section. Since the sections are designed symmetrically, we assume that the bending moments at locations A and D are identical, as well as those at B and C. Additionally, since A and D are located at the ends of the supports, their bending moments are assumed to be zero. This simplifies the analysis while allowing for an accurate reflection of the shaft's deflection behavior across its length.

$$\frac{L_1 M_A}{EI_1} + 2M_B\left(\frac{L_1}{EI_1} + \frac{L_2}{EI_2}\right) + \frac{M_C L_2}{EI_2} = -\frac{6A_1\bar{\bar{x}}_1}{EI_1 L_1} - \frac{6A_2\bar{\bar{x}}_{2ABC}}{EI_2 L_2}$$

Calculation

$$I_1 = I_3 = \frac{\pi}{64}(D_1)^4 = \frac{\pi}{64}(0.669)^4 = 0.0098in^4$$

$$I_2 = \frac{\pi}{64}(D_2)^4 = \frac{\pi}{64}(0.875)^4 = 0.0288in^4$$

$$M_A = M_D = 0$$

$$M_B = M_C$$

Equation for span "ABC"

$$\frac{L_1 M_A}{EI_1} + 2M_B\left(\frac{L_1}{EI_1} + \frac{L_2}{EI_2}\right) + \frac{M_C L_2}{EI_2} = -\frac{6A_1\bar{\bar{x}}_1}{EI_1 L_1} - \frac{6A_2\bar{\bar{x}}_{2ABC}}{EI_2 L_2}$$

$$\cancel{\frac{L_1 M_A}{EI_1}} + 2M_B\left(\frac{L_1}{\cancel{E}I_1} + \frac{L_2}{\cancel{E}I_2}\right) + \frac{M_B L_2}{\cancel{E}I_2} = -\cancel{\frac{6A_1\bar{\bar{x}}_1}{EI_1 L_1}} - \frac{6A_2\bar{\bar{x}}_{2ABC}}{\cancel{E}I_2 L_2}$$

$$2M_B\left(\frac{2.375}{0.0098}+\frac{3.25}{0.0288}\right)+M_B\left(\frac{3.25}{0.0288}\right)=-\frac{6(420.17)(1.625)}{0.0288(3.25)}$$

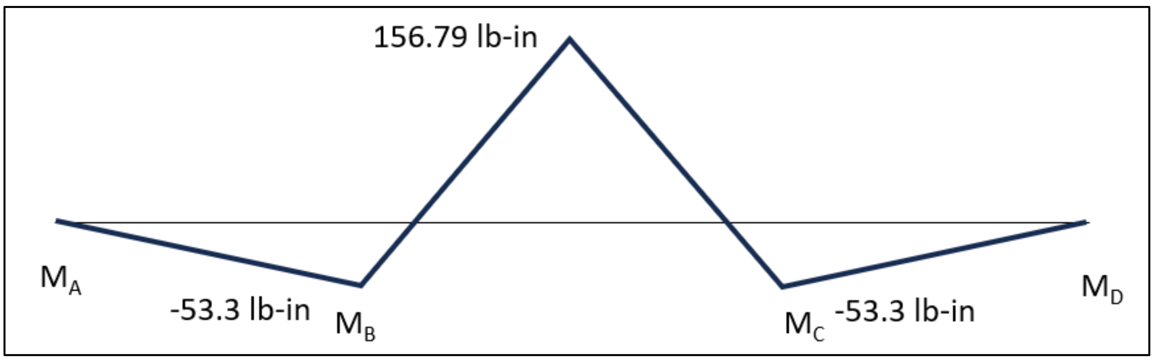
$$M_B=-53.3\ in-lb$$

$$M_C=-53.3\ in-lb$$

Results

- $M_A=0$
- $M_B=-53.3\ in\ -lb$
- $M_C=-53.3\ in-lb$
- $M_D=0$
- $M_{B-C}=M_B+X_{2BCD}\times V_{B2}=-53.3+1.625(129.3)=156.8\ in-lb$

Diagram

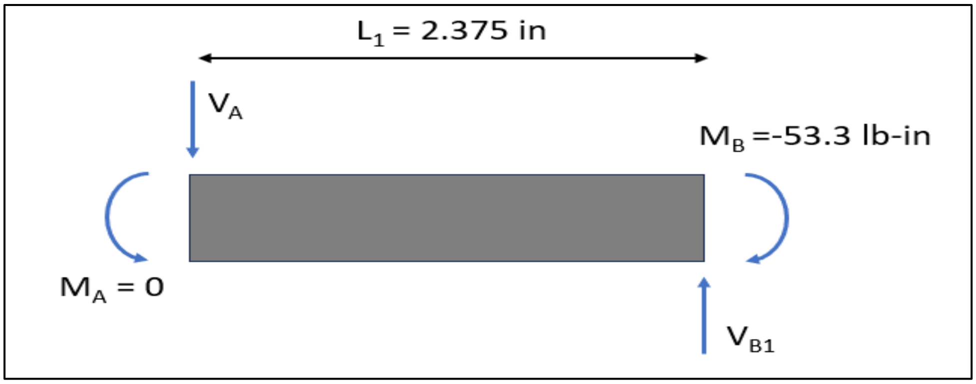


3.1.3 Shear Force Analysis

Shear force was calculated by sectioning the shaft into segments A-B, B-C, and C-D. Since points B and C appear in two sections, the forces at these locations were summed to determine the total force at each point. This approach ensures that the combined effects of all applied forces at B and C are accurately accounted for in the shear force analysis.

Calculation

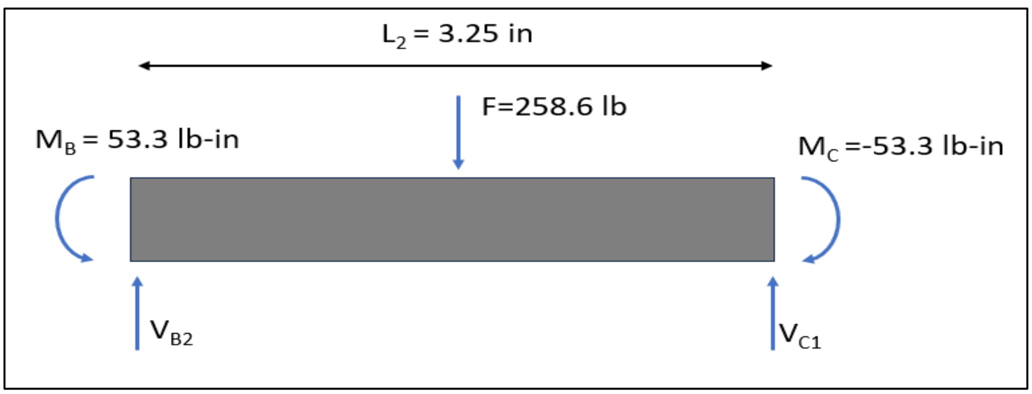
Location A-B



$$+\circlearrowright \sum M_A = 0,\ L_1 V_{B1} - M_B = 0$$
$$2.375 V_{B1} = 53.3$$
$$V_{B1} = 22.44\ lb$$

$$+\uparrow \sum F_Y = 0,\ V_A + V_{B1} = 0$$
$$V_A = -22.44\ lb$$

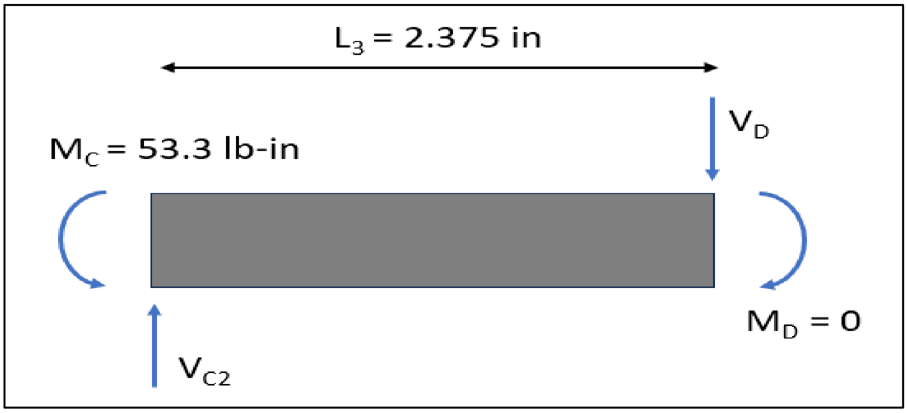
Location B-C



$$+\circlearrowright \sum M_B = 0,\ M_B + M_C - \frac{FL_2}{2} + L_2 V_{C1} = 0$$
$$53.3 - 53.3 - \frac{258.6(3.25)}{2} + 3.25 V_{C1} = 0$$
$$V_{C1} = 129.3\ lb$$

$$+\uparrow \sum F_Y = 0,\ V_{B2} + V_{C1} + F = 0$$
$$V_{B2} + 129.3 - 258.6 = 0$$
$$V_{B2} = 129.3\ lb$$

Location C-D



$$+\circlearrowright \sum M_D = 0,\ L_3 V_{C2} - M_C = 0$$

$$2.375 V_{C2} = 53.3$$

$$V_{C2} = 22.44\ lb$$

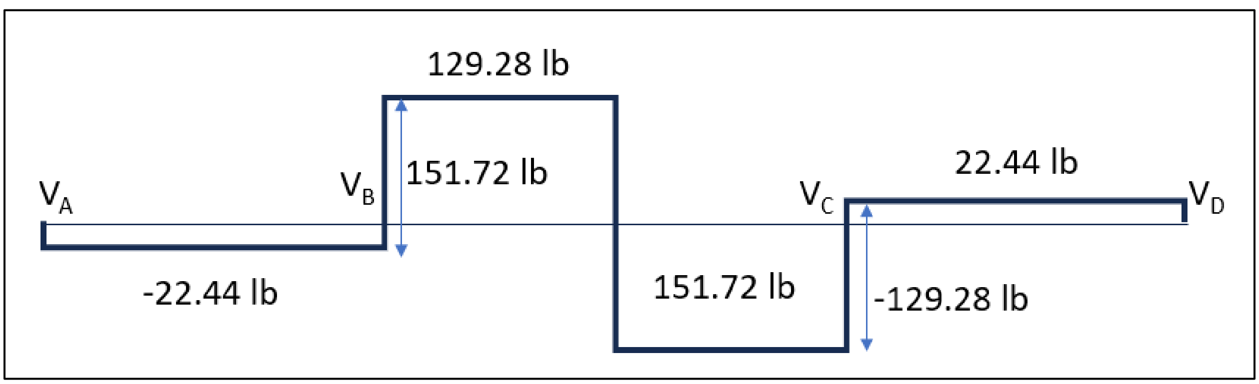
$$+\uparrow \sum F_Y = 0,\ V_D + V_{C2} = 0$$

$$V_D = -22.44\ lb$$

Results

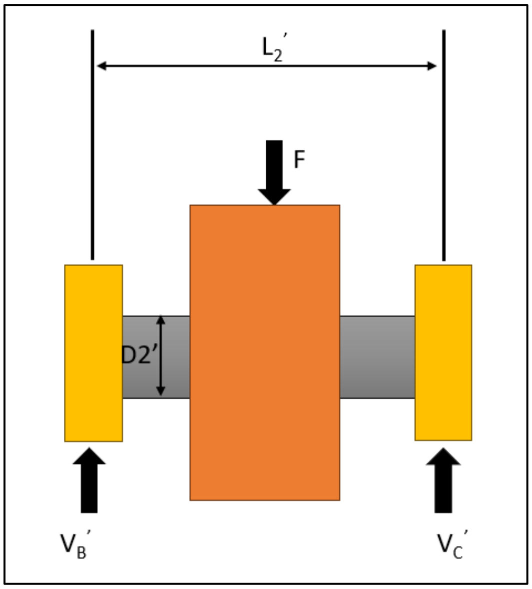
- $V_A = -22.44\ lb$
- $V_B = V_{B1} + V_{B2} = 22.44 + 129.3 = 151.72\ lb$
- $V_C = V_{C1} + V_{C2} = 129.3 + 22.44 = 151.72\ lb$
- $V_D = -22.44\ lb$

Diagram



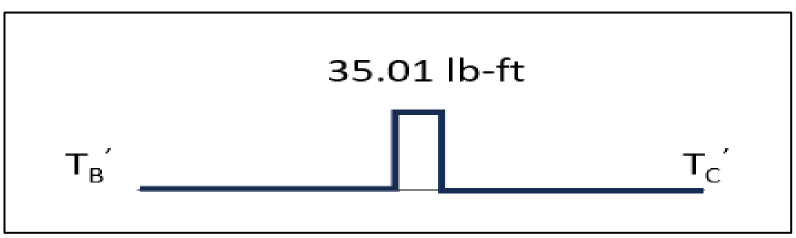
3.2 Secondary Shaft

The force applied to the secondary shaft will be analyzed using the diagram below. The roller ball bearings are at points B' and C'.



3.2.1 Torque Analysis

In the interim report, we assumed that no torque was applied to the second shaft, as the ideal gear dissipates torque through kinetic energy. However, in this final report, we assume that all the torque applied at the center of the gear, due to the fluid, resists the rotational motion. Although some energy dissipation may occur, we assume that 100% of the torque is transmitted to the ideal gear and the secondary shaft. This analysis is therefore based on the worst-case scenario.



3.2.2 Bending Moment Analysis

Since the second shaft is a simple beam structure, M_B' and M_C' have zero bending moments, while the center experiences the highest bending moment.

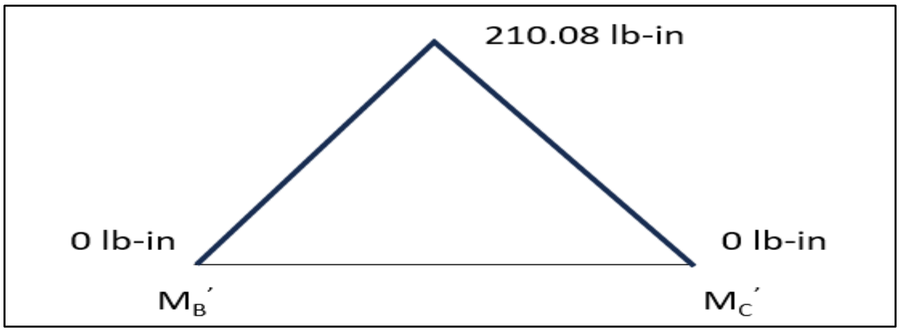
Calculation

- $M'_{B-C} = \frac{L_2}{2} \times V'_B = \frac{3.25}{2} \times 129.28 = 210.08\ in - lb$

Results

- $M'_{B-C} = 210.08\ in - lb$
- $M'_{B} = 0$
- $M'_{C} = 0$

Diagram



3.2.3 Shear Force Analysis

Since the second shaft is a simple beam structure, points B' and C' each support half of the vertical force applied at the center. Additionally, it's assumed that the pinion gear transfers all power to the ideal gear, meaning the efficiency is 100%.

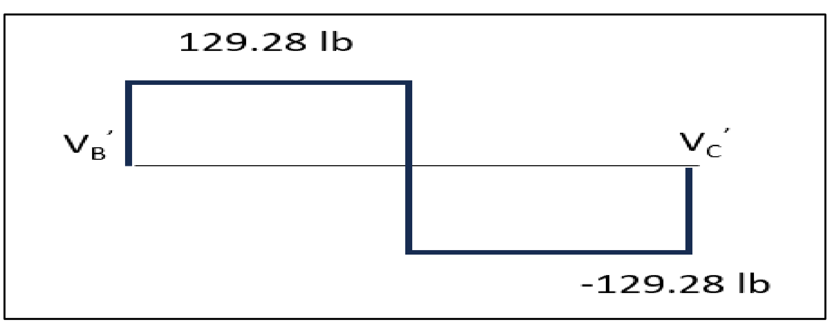
Calculation

$$V'_B = V'_C = \frac{F}{2} = \frac{258.6}{2} = 129.3\ lb$$

Results

- $V'_{B-C} = 258.6\ in - lb$
- $V'_{B} = 129.3\ lb$
- $V'_{C} = 129.3\ lb$

Diagram



4. Design and Analysis of Major Components

4.1 Gear

4.1.1 Bending Stress Analysis

The gear is made of carburized grade 1 steel, with a bending strength of 55,000 psi.

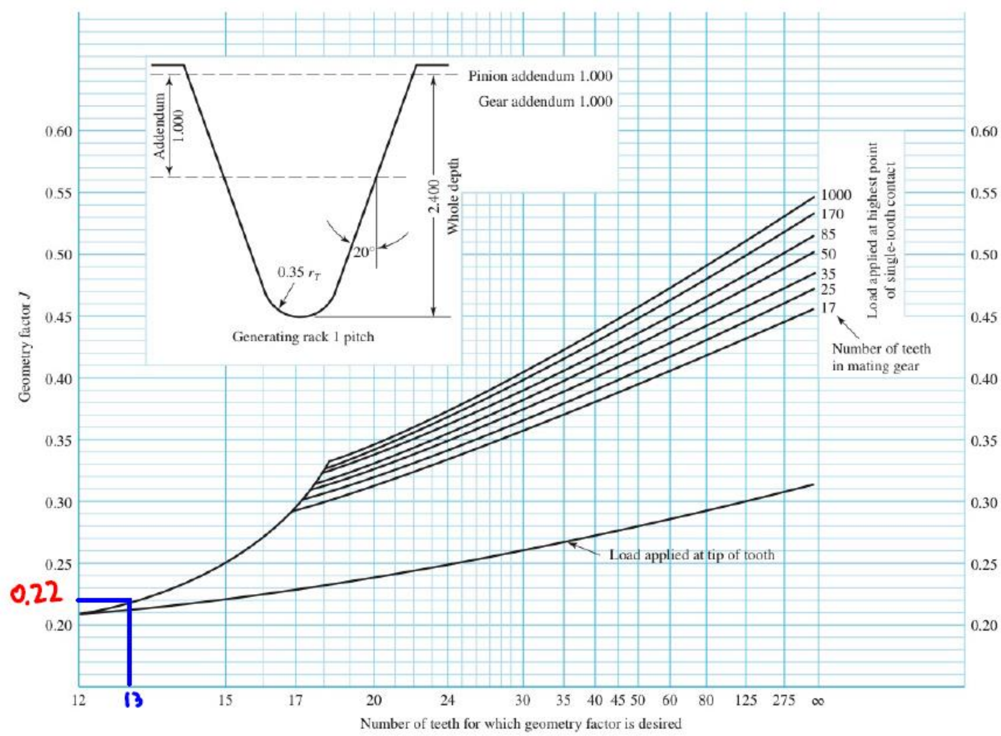
The overload factor K_O is set to 1, assuming that both the driven machine and power source operate uniformly.

The dynamic factor K_V is calculated as 1.15, based on a quality number Q_V of10, a quality factor A of 0.4, and a speed factor B of 83.78.

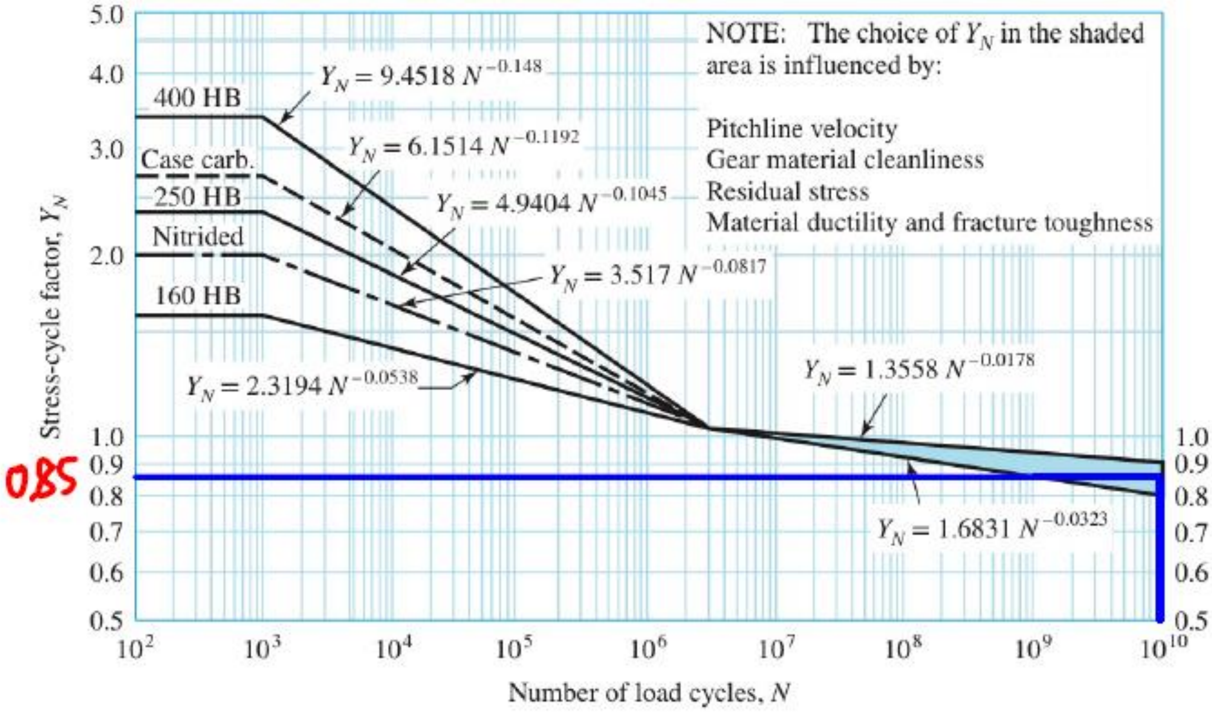
The load-distribution factor K_m is set to 1 to represent uniform load distribution along the line of contact, as the gear is located midspan between two bearings at the point of zero slope.

The rim-thickness factor K_B is also set to 1 because the backup ratio m_B is 1. This value was calculated based on a tooth height of 0.6 inches and a rim thickness of 0.84 inches.

The geometry factor J is set to 0.22, based on both the pinion and gear having 13 teeth.



The stress-cycle factor for bending strength Y_N is set to 0.85, assuming an infinite number of load cycles.



The temperature factor K_T is set to 1, assuming that the oil flowing through the gear will absorb heat and maintain the gear temperature below 250°F.

The reliability factor K_R is set to 1, based on a reliability of 0.99.

Based on all these inputs, the bending stress on the gear is 3,007 psi, resulting in a safety factor of 15.55. Therefore, the gear is safe from bending stress.

Calculation

$$d_P = \frac{N_p}{P_d} = \frac{13}{4} = 3.25$$

$$V = \frac{\pi n d_p}{12} = \frac{\pi(1500)(3.25)}{12} = 1276.27\ ft/min$$

$$T = \frac{H}{n} \times 550 \times 60 \times \frac{1}{2\pi} = \frac{10}{1500} \times 550 \times 60 \times \frac{1}{2\pi} = 35.01\ lb - ft$$

$$W^t = \frac{H}{V} \times 33000 = \frac{10}{1276.27} \times 33000 = 258.566\ lb$$

$$\mathrm{B} = 0.25(12 - \mathrm{Q_V})^{\frac{2}{3}} = 0.25(12 - 10)^{\frac{2}{3}} = 0.39685$$

$$A = 50 + 56(1 - B) = 50 + 56(1 - 0.39685) = 83.776$$

$$\mathrm{K_V} = \left(\frac{\mathrm{A} + \sqrt{\mathrm{V}}}{\mathrm{A}}\right)^{\mathrm{B}} = \left(\frac{83.776 + \sqrt{1276.27}}{83.776}\right)^{0.39685} = 1.1514$$

$$\sigma = \frac{W_t K_0 K_V K_S P_d K_m K_B}{FJ} = \frac{258.6(1)(1.15)(1)(4)(1)(1)}{1.8(0.22)} = 3007\ psi$$

$$S_F = \frac{S_t Y_N}{K_T K_R \sigma} = \frac{55000(0.85)}{1(1)(3007)} = 15.55$$

Symbol	Description	Value	Unit	Assumption	Equation
N_p	Number of teeth	13	teeth		
P_d	Diametral pitch	4	teeth/inch		
d_p	Pitch diameter	3.25	inch		$d_P = \frac{N_p}{P_d}$
H	Power	10	hp		
n	Gear Speed	1500	rpm		
V	Pitch-line velocities	1276.3	ft/min		$V = \frac{\pi n d_p}{12}$
T	Torque	35.01	lbf-ft		$T = \frac{H}{n} \times 550 \times 60 \times \frac{1}{2\pi}$
W^t	Transmitted load	258.6	lbf		$W^t = \frac{H}{V} \times 33000$

Symbol	Description	Value	Unit	Assumption	Equation
K_0	Overload Factor	1		Power source and Driven machine are uniform	
K_v	Dynamic Factor	1.15			$K_V = \left(\frac{A+\sqrt{V}}{A}\right)^B$
Q_v	Quality Number	10			
B	Quality Factor	0.40			$B = 0.25(12 - Q_V)^{\frac{2}{3}}$
A	Speed Factor	83.78			$A = 50 + 56(1 - B)$
K_s	Size Factor	1			
K_m	Load-distribution Factor	1		Assumed uniform load	
K_B	Rim-thickness Factor	1		Back up ratio (m_B) > 1.2	
F	Face width	1.8			
J	Geometry Factor	0.22			
σ	Bending Stress	3007.12	psi		$\sigma = \frac{W_t K_0 K_V K_S P_d K_m K_B}{FJ}$
S_t	Gear Bending Strength	55000	psi	Carburized, Grade 1	
Y_N	Stress-cycle factor for bending strength	0.85			
K_T	Temperature factor	1			
K_R	Reliability factor	1		Reliability 0.99	
S_F	Safety factor bending	15.55			$S_F = S_t Y_N / K_T K_R \sigma$

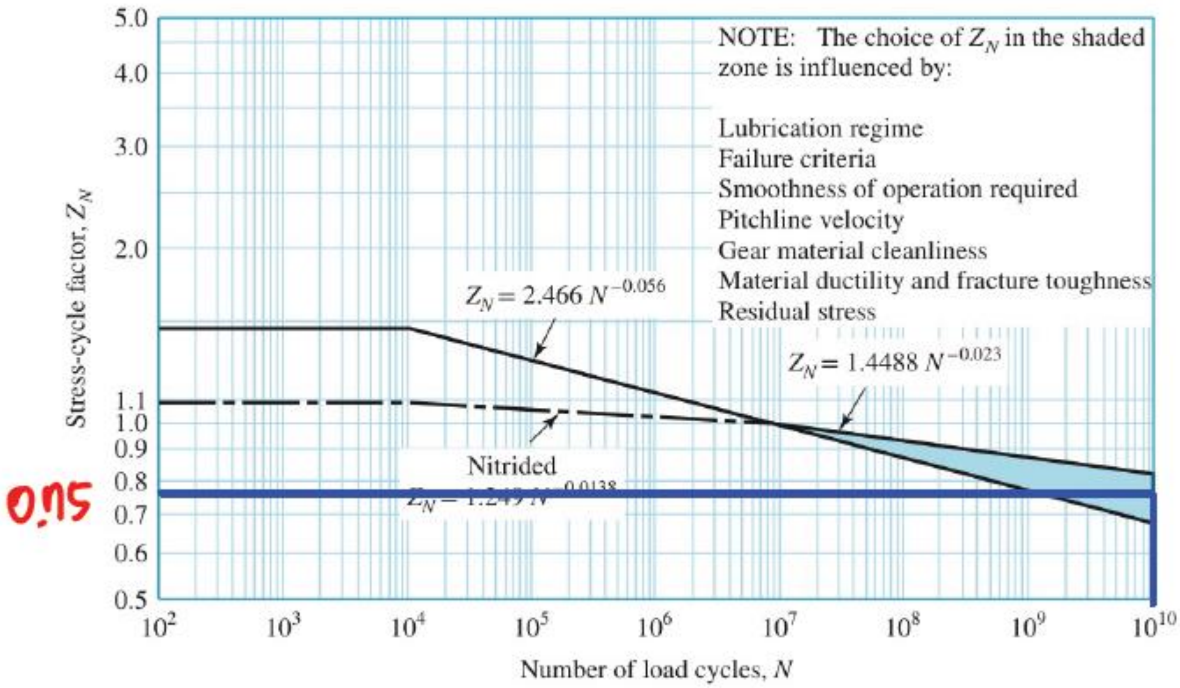
4.1.2 Wear Analysis

The selected gear material is carburized grade 1 steel, which has a surface endurance strength of 150,000 psi.

The elastic coefficient C_p is 2,300 $\sqrt{psi}$, assuming both gears are made of steel.

The load-sharing ratio m_N is 1 because spur gears are used. Similarly, the speed ratio m_G is 1, as both gears have the same number of teeth. Based on these values and a 20-degree pressure angle, the geometry factor I is calculated as 0.0803.

The stress-cycle factor, Z_N is 0.75, assuming an infinite number of load cycles.



Using all the inputs above, the contact stress is calculated as 57,883 psi, resulting in a wear safety factor of 2.33.

Calculation

$$\mathrm{I} = \frac{\cos\emptyset_t \sin\emptyset_t}{2\mathrm{m_N}} \times \frac{\mathrm{m_G}}{\mathrm{m_G}+1} = \frac{\cos 20° \sin 20°}{2(1)} \times \frac{1}{1+1} = 0.0803$$

$$\sigma_C = C_P\left(\frac{W^t K_0 K_V K_S K_m C_f}{d_P F I}\right)^{\frac{1}{2}} = 2300\left\{\frac{258.6(1)(1.15)(1)(1)(1)}{3.25(1.8)(0.0803)}\right\}^{0.5} = 57883\, psi$$

$$S_H = \frac{S_C Z_N C_H}{K_T K_R \sigma_C} = \frac{180000(0.75)(1)}{1(1)(57883)} = 2.332$$

Symbol	Description	Value	Unit	Assumption	Equation
C_p	Elastic Coefficient	2300	psi^0.5	Gear and Pinion Material: Steel	
C_f	Form factor	1			
I	Geometry factor	0.0803			$I = \frac{cos\emptyset_t sin\emptyset_t}{2m_N} \times \frac{m_G}{m_G + 1}$
Ø	Pressure angle	20	° (degree)		
N_G	Number of gear teeth	13			
N_P	Number of pinion teeth	13			
m_G	Speed ratio	1			$m_G = N_G/N_P$
m_N	Load-sharing ratio	1			
σ_c	Gear contact stress	57883.4	psi		$\sigma_C = C_P\left(\frac{W^t K_0 K_V K_S K_m C_f}{d_P F I}\right)^{\frac{1}{2}}$
S_c	Surface endurance strength	180000	psi	Carburized, Grade 1	
Z_N	Stress-cycle factor for pitting resistance	0.75		Infinite load cycle	
C_H	Hardness-ratio factor	1			$c_H = 1.0 + A'(m_G - 1.0)$
S_H	Safety factor wear	2.332			$S_H = \frac{S_C Z_N C_H}{K_T K_R \sigma_C}$

4.1.3 Bearing Analysis

When the shaft rotates, the key rotates along with it, applying force on the keyway in the gear and causing stress. The cross-sectional area of the keyway is 0.046875 in², and the force applied is 960.4 lb. Combining these values, the bearing stress is calculated to be 20,488 psi. Assuming the material used is 1045 CD carburized steel, with a yield strength on the hole side of 77 ksi, the safety factor for bearing stress is 2,64.

$$A_b = 0.5 \times h \times L_{key} = 0.5 \times \frac{1}{8} \times \frac{3}{4} = 0.046875\ in^2$$

$$F = \frac{T}{d/2} = \frac{420.17}{\frac{0.875}{2}} = 960.4\ lbs$$

$$\sigma_b = \frac{F}{A_b} = \frac{960.4}{0.046875} = 20488\ psi$$

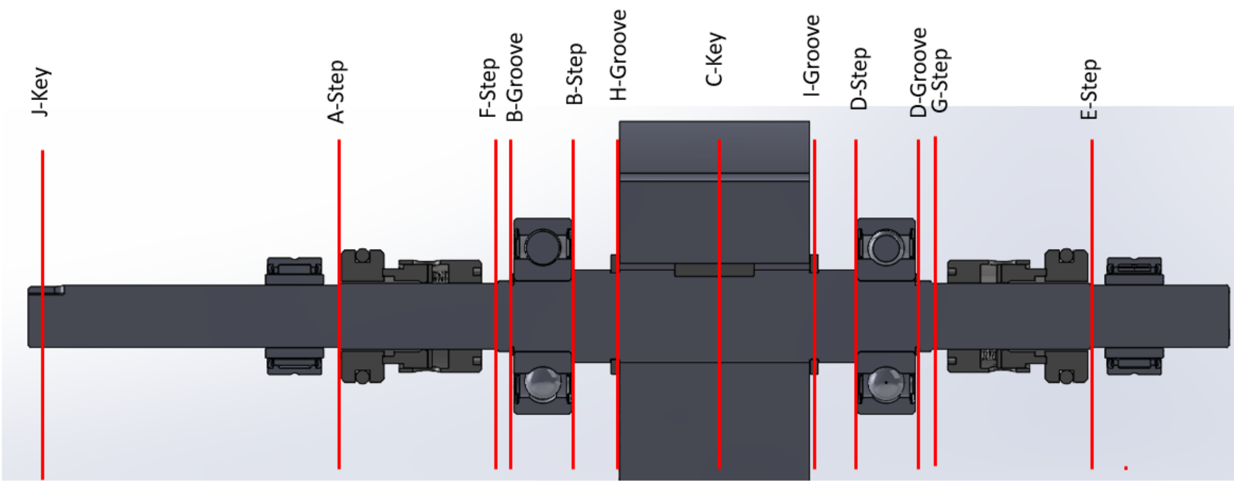
$$n_{bearning} = \frac{S_{yt}(Gear)}{\sigma_b} = \frac{77000}{20488} = 3.76$$

4.1.4 Comment on Gear

The gear has a bending safety factor of 15.55, a wear safety factor of 2.33, and a bearing safety factor of 3.76. Since S_H^2 ($2.33^2 = 5.43$) is smaller than S_F (15.55), wear poses a threat to its functionality. The required safety factor depends on the application, the number of intended load cycles, and the designer's judgment. There are several ways to increase the safety factors, such as using alternative materials like Grade 2 steel. Modifying the size can also help, for example, increasing the keyway size from 1/8" x 3/16" to 1/4" x 1/4" to increase the cross-sectional area and reduce bearing stress. Other methods include carburizing the keyway area or using multiple keys to distribute the load. A zero-backlash design can also improve load distribution and reduce stress concentrations, further enhancing the safety factors.

4.2 Shaft

4.2.1 Primary Shaft Design and Analysis



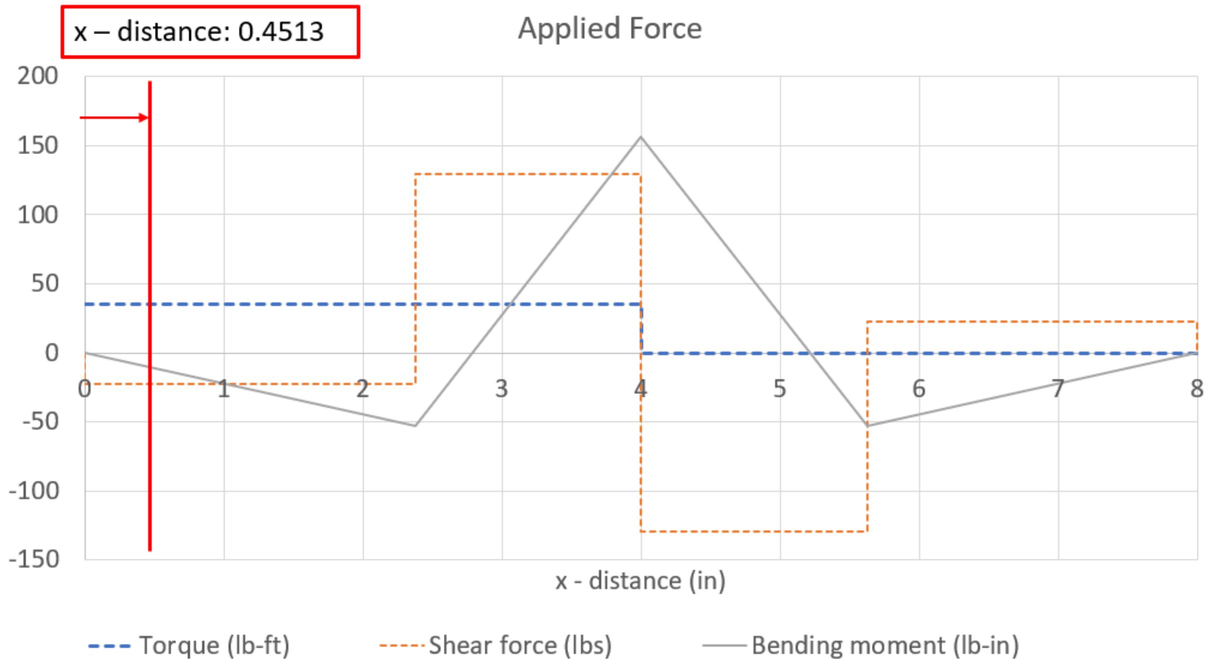
The primary shaft was analyzed in multiple sections where stress concentrations occur. For evaluating the shaft's fatigue safety, the DE-ASME equation was used. The shaft is made of AISI 1030 Q&T steel, with an ultimate tensile strength of 116 ksi and a yield strength of 90 ksi.

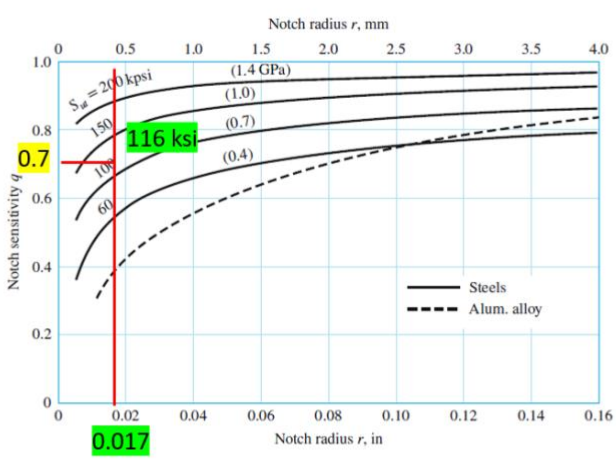
4.2.1.1 A – Step

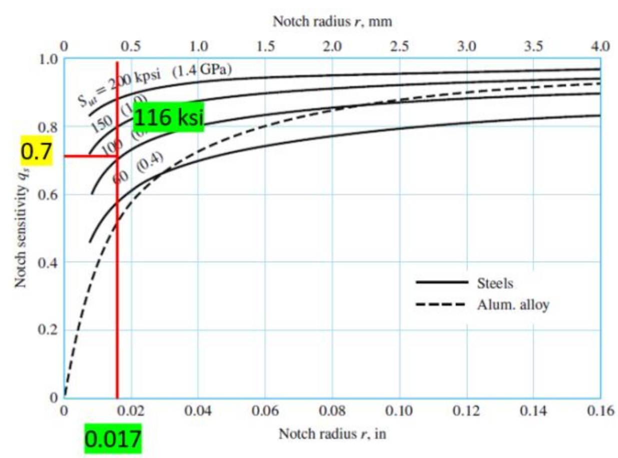
The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

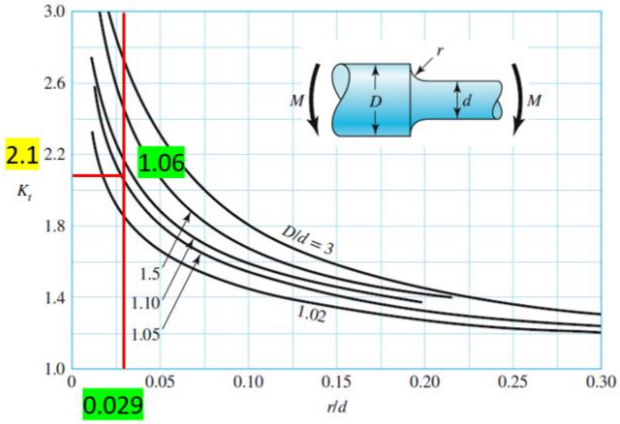
- Torque: 35.01 lb-ft
- Shear force: -22.44 lb
- Bending moment: -10.13 lb-in
- D: 0.591 in
- t: 0.017 in
- q: 0.70
- q_s: 0.70

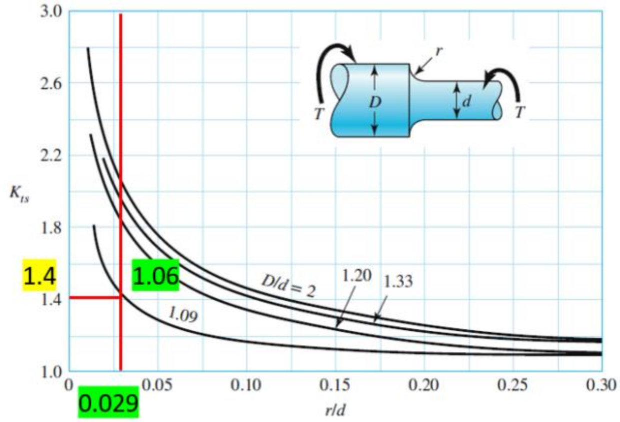
- r: 0.017 in
- Kt: 2.1
- Kts: 1.4











Also, the safety factor was calculated based on these inputs is 3.89.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.7(2.1 - 1) = 1.77$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.7(1.4 - 1) = 1.28$$

$$k_a = aS_{ut}^b = 2 \times 116^{-0.217} = 0.71$$

$$k_b = 0.879d^{-0.107} = 0.879(0.591)^{-0.107} = 0.93$$

$$Se' = 0.5S_{ut} = 0.5 \times 116000 = 58{,}000psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S_e'} = 0.71(0.93)(1)(1)(0.814)(58{,}000) = 31{,}299\ psi$$

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

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

$$= \frac{16}{\pi 0.591^3}\left[4\left(\frac{1.77 \times 10.13}{31{,}299}\right)^2 + 3\left(\frac{1.28 \times 420.17}{90{,}000}\right)^2\right]^{\frac{1}{2}} = 0.257$$

$$n = \frac{1}{0.257} = 3.89$$

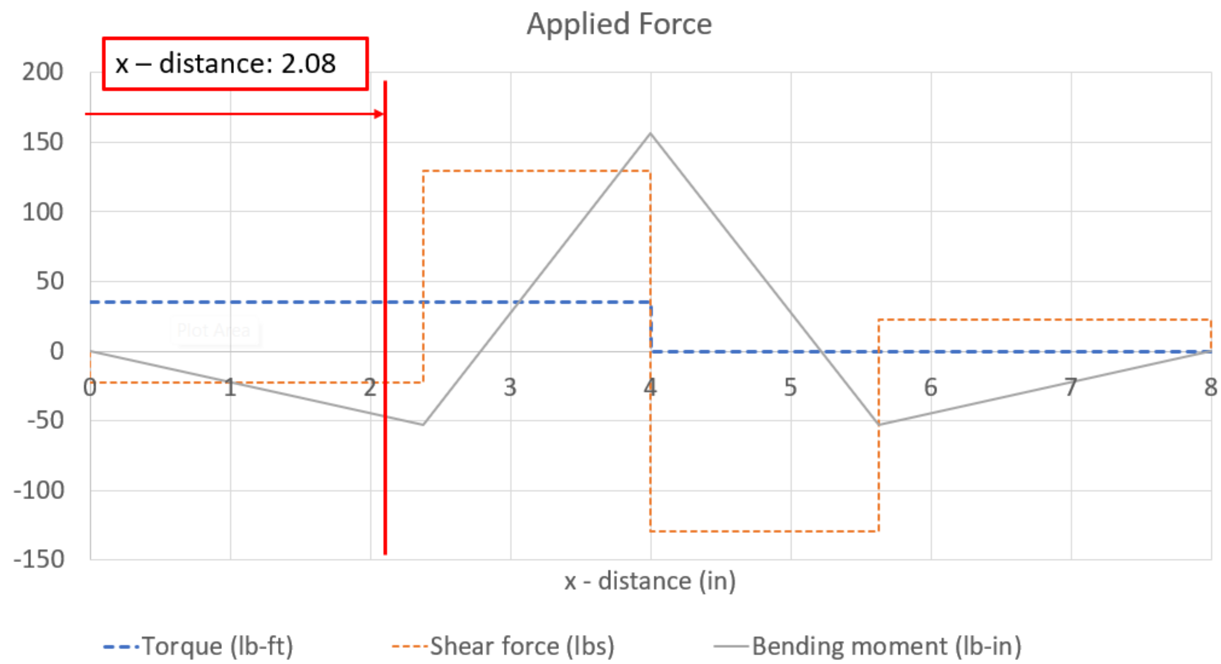
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	35.01	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	420.17	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert \frac{T_{max} - T_{min}}{2} \right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	10.13	lbf-in		$M_a = \left\lvert \frac{M_{max} - M_{min}}{2} \right\rvert$
S_{ut}	Ultimate Tensile Strength	116000	psi	Shaft material: AISI 1030 Q&T	
S_{yt}	Yield Tensile Strength	90000	psi	Shaft material: AISI 1030 Q&T	
q	Notch sensitivity	0.7			
qs	Notch sensitivity	0.7			
r	Notch radius	0.0156	in		
K_t	Stress-Concentration factor	2.1			
K_{ts}	Stress-Concentration factor	1.4			
d_{input}	Shaft diameter (input)	0.591	in		
K_f	Fatigue Stess-Concentration Factor	1.77			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	1.28			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.71			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.930			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	58000	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	31299	psi		$S_e = k_a k_b k_c k_d k_e S_e'$
1/n	Utilization Factor	0.26			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	3.89			

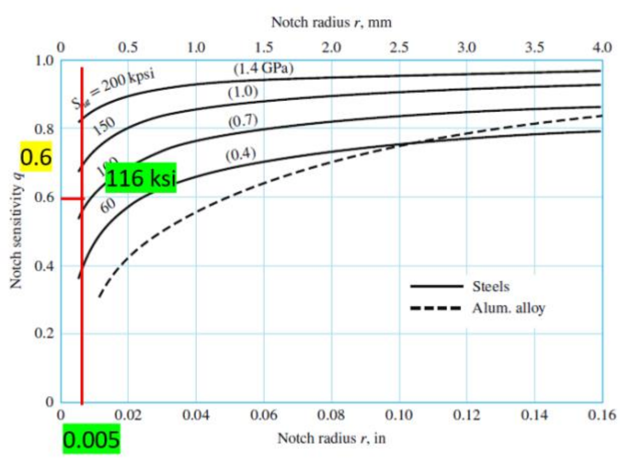
4.2.1.2 B – Groove

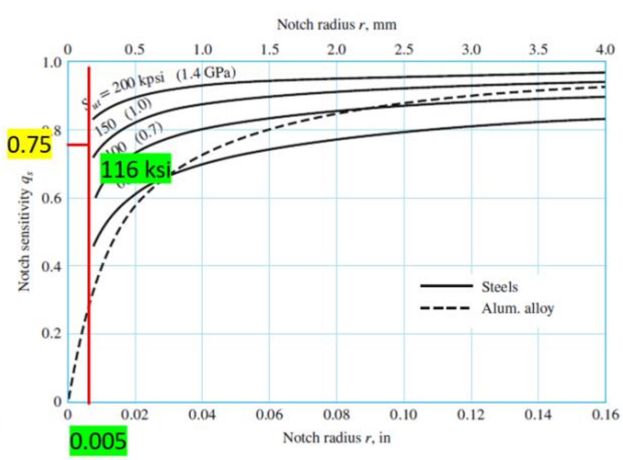
The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

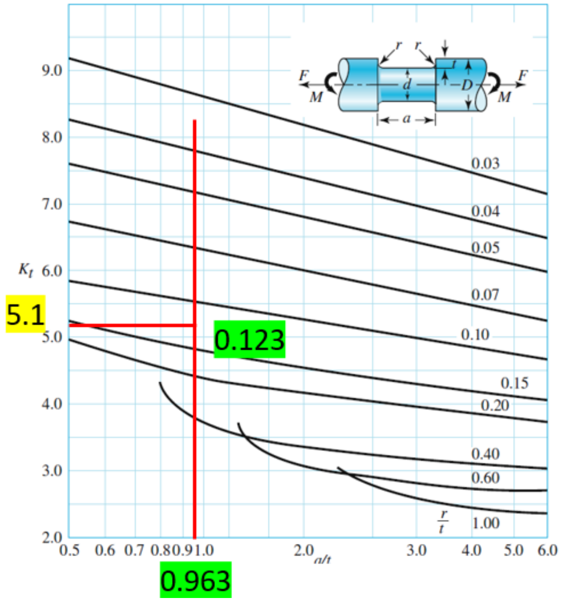
- Torque: 35.01 lb-ft
- Shear force: -22.44 lb
- Bending moment: -46.68 lb-in
- D: 0.669 in
- t: 0.0405 in
- a: 0.039 in
- q: 0.6
- q_s: 0.75

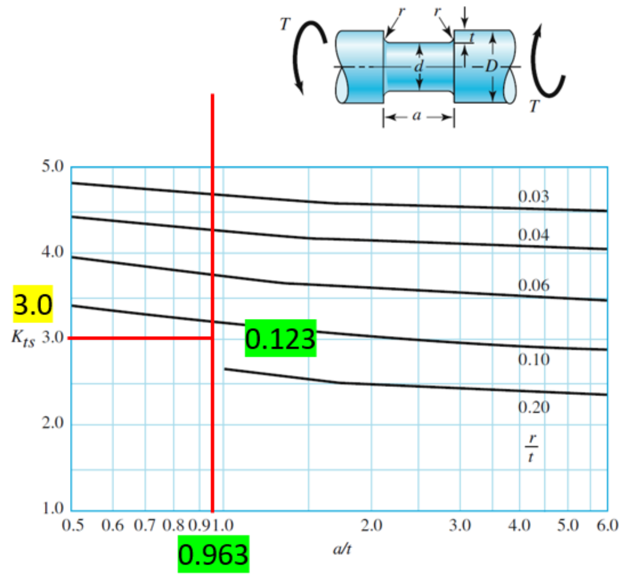
- r: 0.005 in
- Kt: 5.1
- Kts: 3











Also, the safety factor was calculated based on these inputs is 2.59.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.6(5.1 - 1) = 3.46$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.75(3 - 1) = 2.5$$

$$k_a = aS_{ut}^b = 2 \times 116^{-0.217} = 0.71$$

$$k_b = 0.879d^{-0.107} = 0.879(0.669)^{-0.107} = 0.918$$

$$Se' = 0.5S_{ut} = 0.5 \times 116000 = 58{,}000psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S_e'} = 0.71(0.918)(1)(1)(0.814)(58{,}000) = 30{,}886\,psi$$

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

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

$$= \frac{16}{\pi 0.669^3}\left[4\left(\frac{3.46 \times 46.68}{30{,}886}\right)^2 + 3\left(\frac{2.5 \times 420.17}{90{,}000}\right)^2\right]^{\frac{1}{2}} = 0.387$$

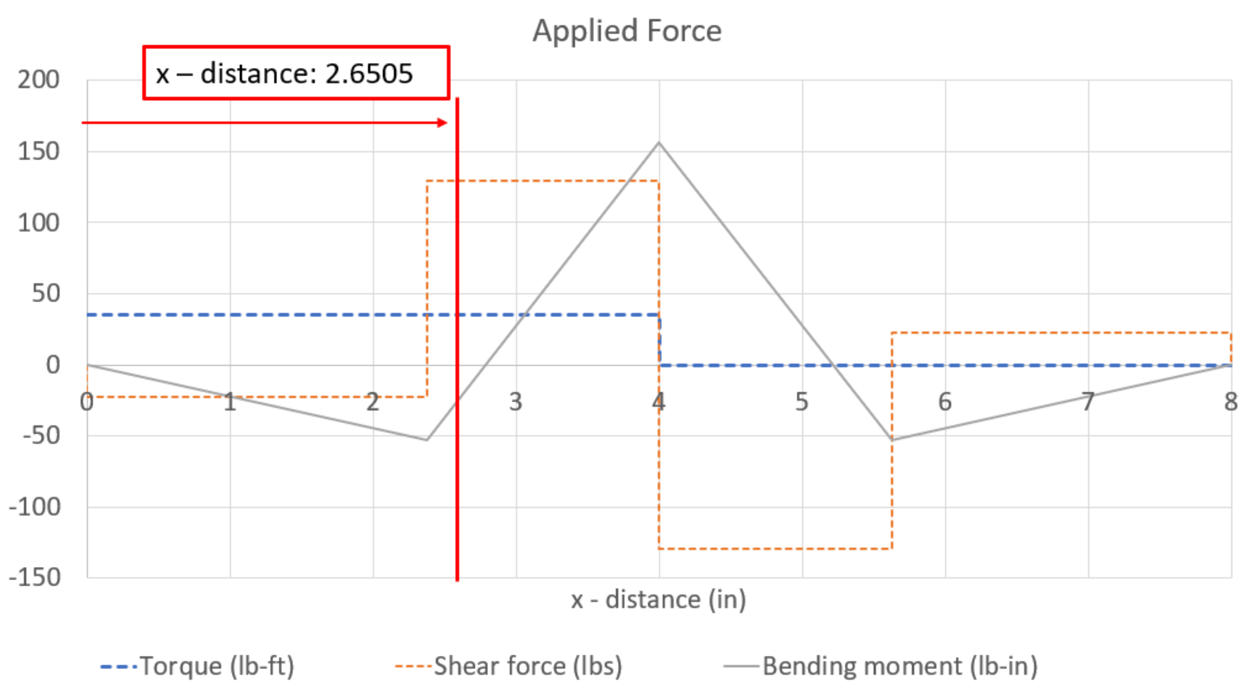
$$n = \frac{1}{0.387} = 2.59$$

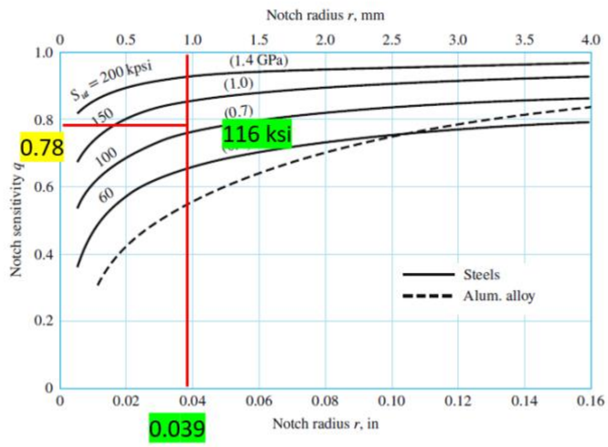
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	35.01	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	420.17	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\|\frac{T_{max} - T_{min}}{2}\right\|$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	46.48	lbf-in		$M_a = \left\|\frac{M_{max} - M_{min}}{2}\right\|$
S_{ut}	Ultimate Tensile Strength	116000	psi	Shaft material: AISI 1030 Q&T	
S_{yt}	Yield Tensile Strength	90000	psi	Shaft material: AISI 1030 Q&T	
q	Notch sensitivity	0.6			
qs	Notch sensitivity	0.75			
r	Notch radius	0.005	in		
K_t	Stress-Concentration factor	5.1			
K_{ts}	Stress-Concentration factor	3			
d_{input}	Shaft diameter (input)	0.669	in		
K_f	Fatigue Stess-Concentration Factor	3.46			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	2.5			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.71			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.918			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	58000	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	30886	psi		$S_e = k_a k_b k_c k_d k_e S'_e$
1/n	Utilization Factor	0.387			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	2.59			

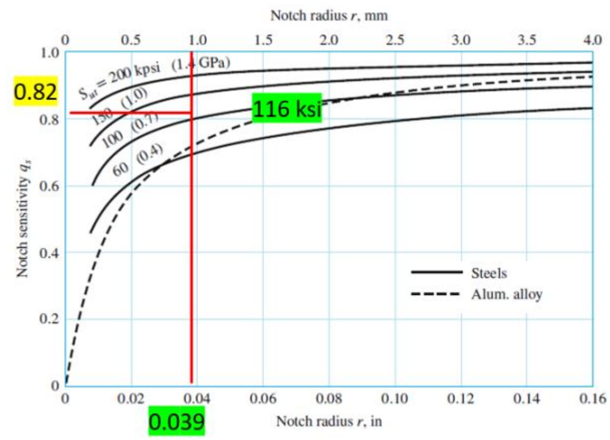
4.2.1.3 B – Step

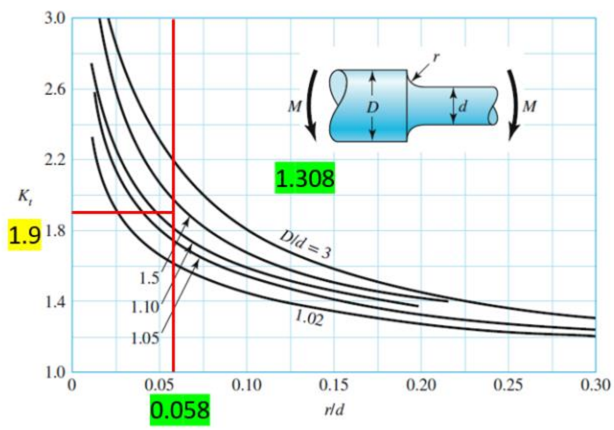
The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

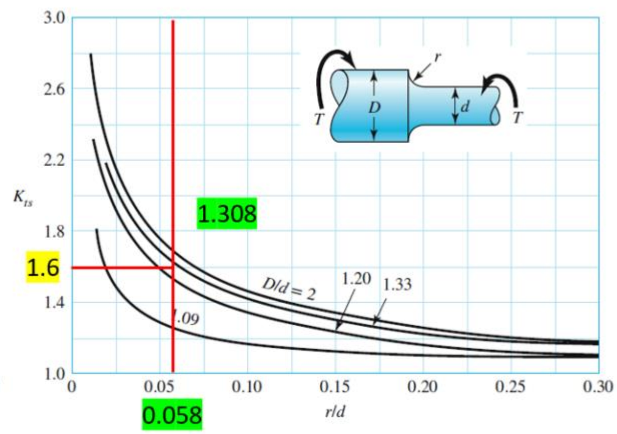
- Torque: 35.01 lb-ft
- Shear force: 129.28 lb
- Bending moment: -17.68 lb-in
- D: 0.669 in
- t: 0.103 in
- q: 0.78
- q_s: 0.82
- r: 0.039 in
- Kt: 1.9
- Kts: 1.6











Also, the safety factor was calculated based on these inputs is 4.81.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.78(1.9 - 1) = 1.702$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.82(1.6 - 1) = 1.492$$

$$k_a = aS_{ut}^b = 2 \times 116^{-0.217} = 0.71$$

$$k_b = 0.879d^{-0.107} = 0.879(0.669)^{-0.107} = 0.918$$

$$Se' = 0.5S_{ut} = 0.5 \times 116000 = 58{,}000psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S_e'} = 0.71(0.918)(1)(1)(0.814)(58{,}000) = 30{,}886\, psi$$

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

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

$$= \frac{16}{\pi 0.669^3}\left[4\left(\frac{1.702 \times 17.68}{30{,}886}\right)^2 + 3\left(\frac{1.492 \times 420.17}{90{,}000}\right)^2\right]^{\frac{1}{2}} = 0.208$$

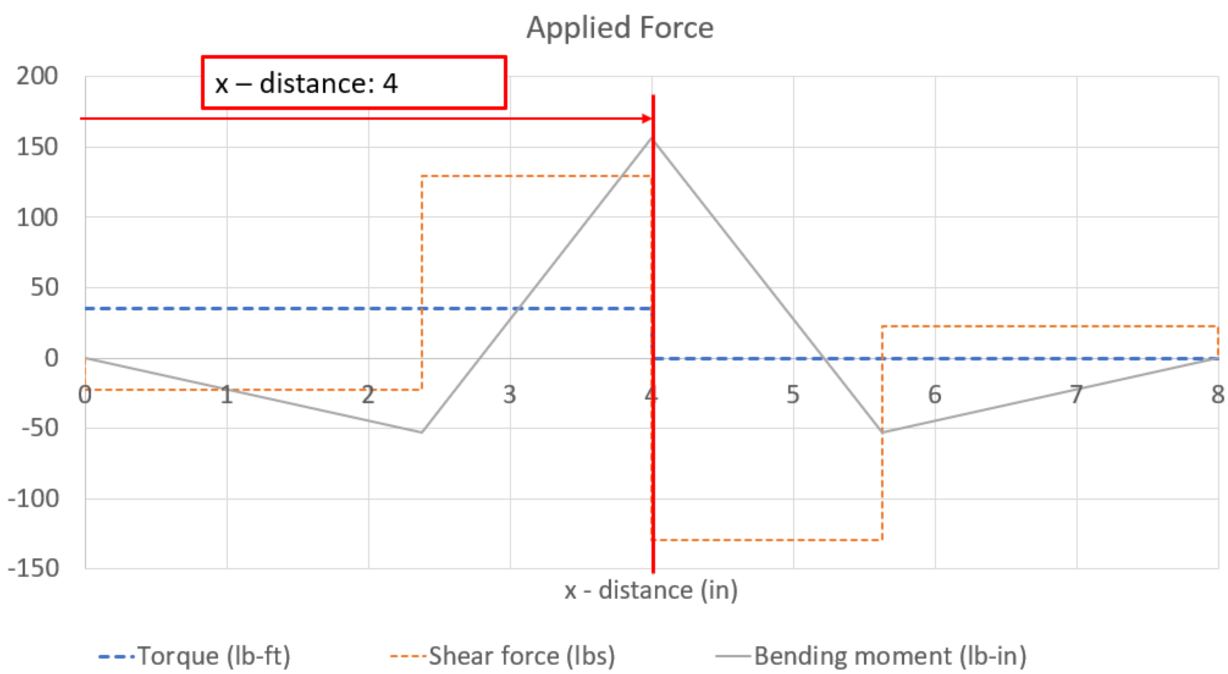
$$n = \frac{1}{0.208} = 4.81$$

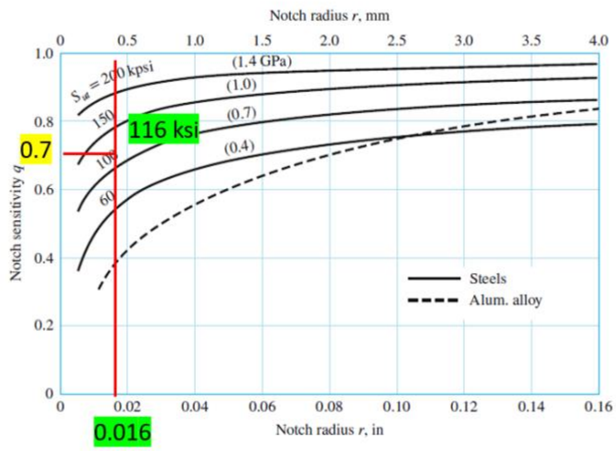
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	35.01	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	420.17	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert\frac{T_{max} - T_{min}}{2}\right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	17.68	lbf-in		$M_a = \left\lvert\frac{M_{max} - M_{min}}{2}\right\rvert$
S_{ut}	Ultimate Tensile Strength	116000	psi	Shaft material: AISI 1030 Q&T	
S_{yt}	Yield Tensile Strength	90000	psi	Shaft material: AISI 1030 Q&T	
q	Notch sensitivity	0.78			
qs	Notch sensitivity	0.82			
r	Notch radius	0.005	in		
K_t	Stress-Concentration factor	1.9			
K_{ts}	Stress-Concentration factor	1.6			
d_{input}	Shaft diameter (input)	0.669	in		
K_f	Fatigue Stess-Concentration Factor	1.702			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	1.492			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.71			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.918			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	58000	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	30886	psi		$S_e = k_a k_b k_c k_d k_e S_e'$
1/n	Utilization Factor	0.208			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	4.81			

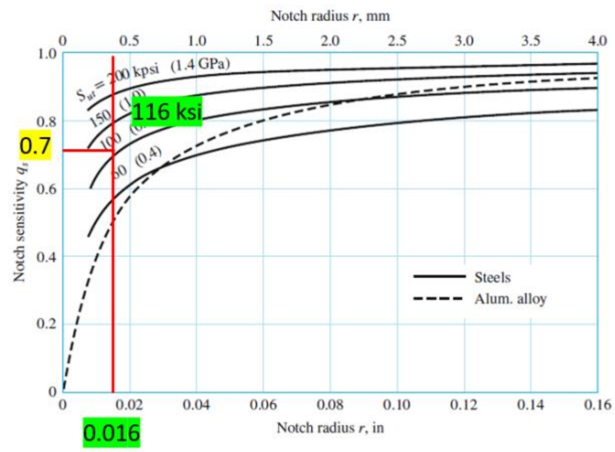
4.2.1.4 C – Key

The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

- Torque: 35.01 lb-ft
- Shear force: 129.28 lb
- Bending moment: 156.79 lb-in
- D: 0.875 in
- t: 0.0625 in
- q: 0.70
- q_s: 0.70
- r: 0.0156 in
- Kt: 2.158
- Kts: 2.235







Shaft diameter	Key size (Width X Height)	Depth of key seat	Fillet radius	K_{TB} (Bending)	K_s (Torsion)	K_{TA} (Axial)
3/8"	3/32" X 3/32"	3/64"	0.0078"	2.202	2.232	2.830
½"	1/8" X 1/8"	1/16"	0.0104"	2.150	2.234	2.830
11/16"	3/16" X 3/16"	3/32"	0.0156"	2.158	2.235	2.847
1 1/16"	¼" X ¼"	1/8"	0.0208"	2.280	2.244	2.778
1 5/16"	5/16" X 5/16"	5/32"	0.0260"	2.267	2.228	2.797
1 9/16"	3/8" X 3/8"	3/16"	0.0312"	2.312	2.246	2.824

Based on the shaft diameter of 0.875 inches in the course material 'ASEE-2013-6775', Kt and Kts values of 2.158 and 2.235 were used.

Also, the safety factor was calculated based on these inputs is 5.44.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.7(2.158 - 1) = 1.811$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.7(2.235 - 1) = 1.865$$

$$k_a = aS_{ut}^b = 2 \times 116^{-0.217} = 0.71$$

$$k_b = 0.879d^{-0.107} = 0.879(0.875)^{-0.107} = 0.89$$

$$Se' = 0.5S_{ut} = 0.5 \times 116{,}000 = 58{,}000psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S_e'} = 0.71(0.89)(1)(1)(0.814)(58{,}000) = 30{,}012\ psi$$

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

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

$$= \frac{16}{\pi 0.875^3}\left[4\left(\frac{1.811 \times 156.79}{30{,}012}\right)^2 + 3\left(\frac{1.865 \times 420.17}{90{,}000}\right)^2\right]^{\frac{1}{2}} = 0.184$$

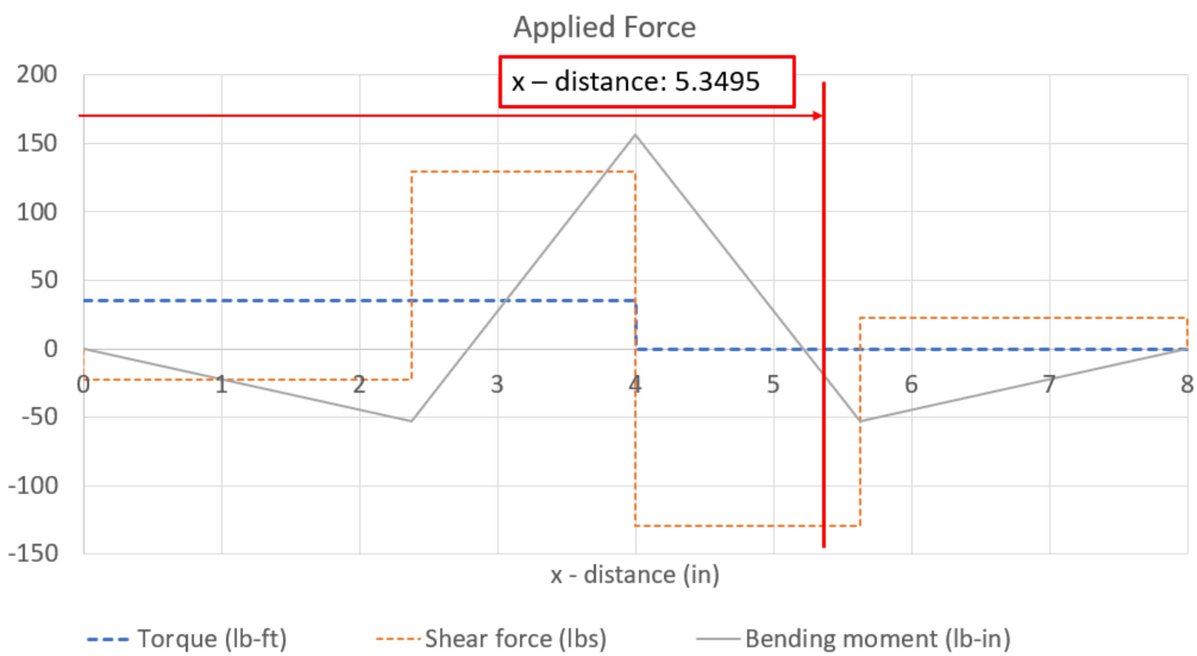
$$n = \frac{1}{0.184} = 5.44$$

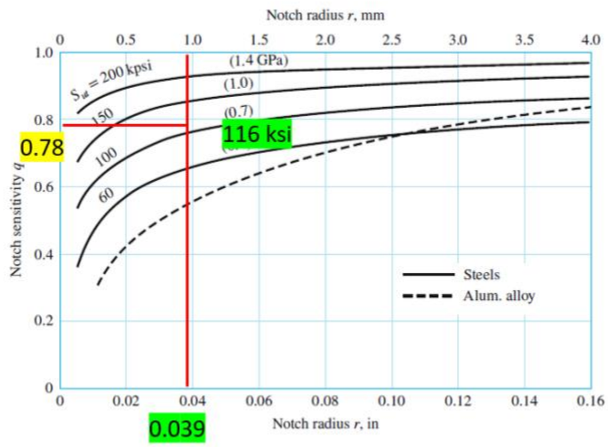
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	35.01	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	420.17	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert\frac{T_{max} - T_{min}}{2}\right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	156.79	lbf-in		$M_a = \left\lvert\frac{M_{max} - M_{min}}{2}\right\rvert$
S_{ut}	Ultimate Tensile Strength	116000	psi	Shaft material: AISI 1030 Q&T	
S_{yt}	Yield Tensile Strength	90000	psi	Shaft material: AISI 1030 Q&T	
q	Notch sensitivity	0.7			
qs	Notch sensitivity	0.7			
r	Notch radius	0.0156	in		
K_t	Stress-Concentration factor	2.158			
K_{ts}	Stress-Concentration factor	2.235			
d_{input}	Shaft diameter (input)	0.875	in		
K_f	Fatigue Stess-Concentration Factor	1.8106			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	1.8645			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.71			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.892			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	58000	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	30012	psi		$S_e = k_a k_b k_c k_d k_e S_e'$
1/n	Utilization Factor	0.184			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	5.44			

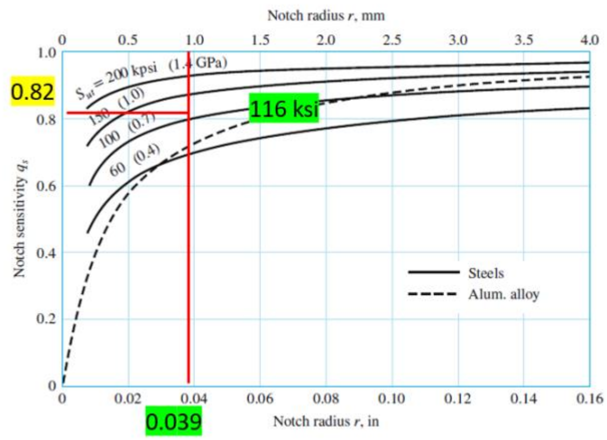
4.2.1.5 D – Step

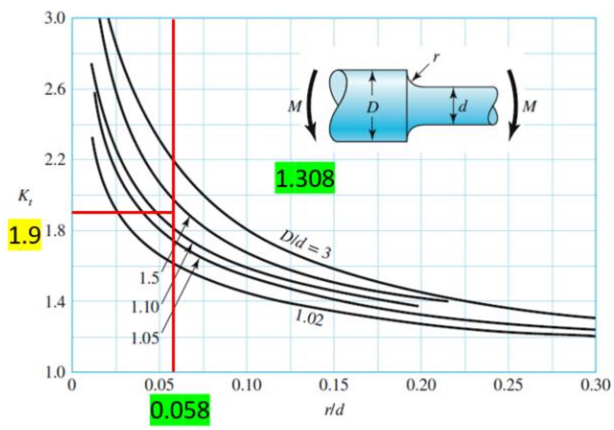
The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

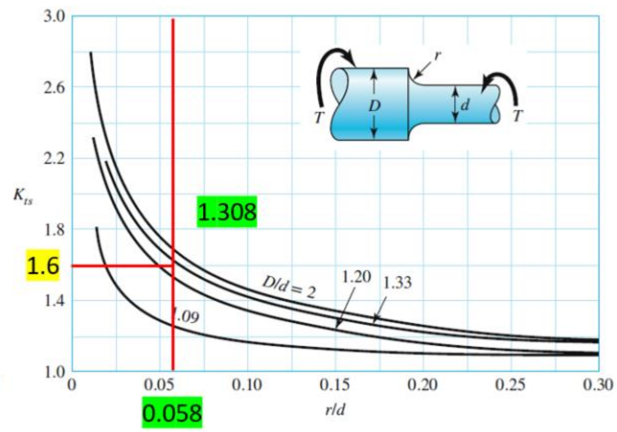
- Torque: 0 lb-ft
- Shear force: -129.28 lb
- Bending moment: -17.68 lb-in
- D: 0.669 in
- t: 0.103 in
- q: 0.78
- q_s: 0.82
- r: 0.039 in
- Kt: 1.9
- Kts: 1.6











Also, the safety factor was calculated based on these inputs is 30.17.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.78(1.9 - 1) = 1.702$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.82(1.6 - 1) = 1.492$$

$$k_a = aS_{ut}^b = 2 \times 116^{-0.217} = 0.71$$

$$k_b = 0.879d^{-0.107} = 0.879(0.669)^{-0.107} = 0.918$$

$$Se' = 0.5S_{ut} = 0.5 \times 116000 = 58{,}000psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e} S_e' = 0.71(0.918)(1)(1)(0.814)(58{,}000) = 30{,}886\,psi$$

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

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

$$= \frac{16}{\pi 0.669^3}\left[4\left(\frac{1.702 \times 17.68}{30{,}886}\right)^2\right]^{\frac{1}{2}} = 0.033$$

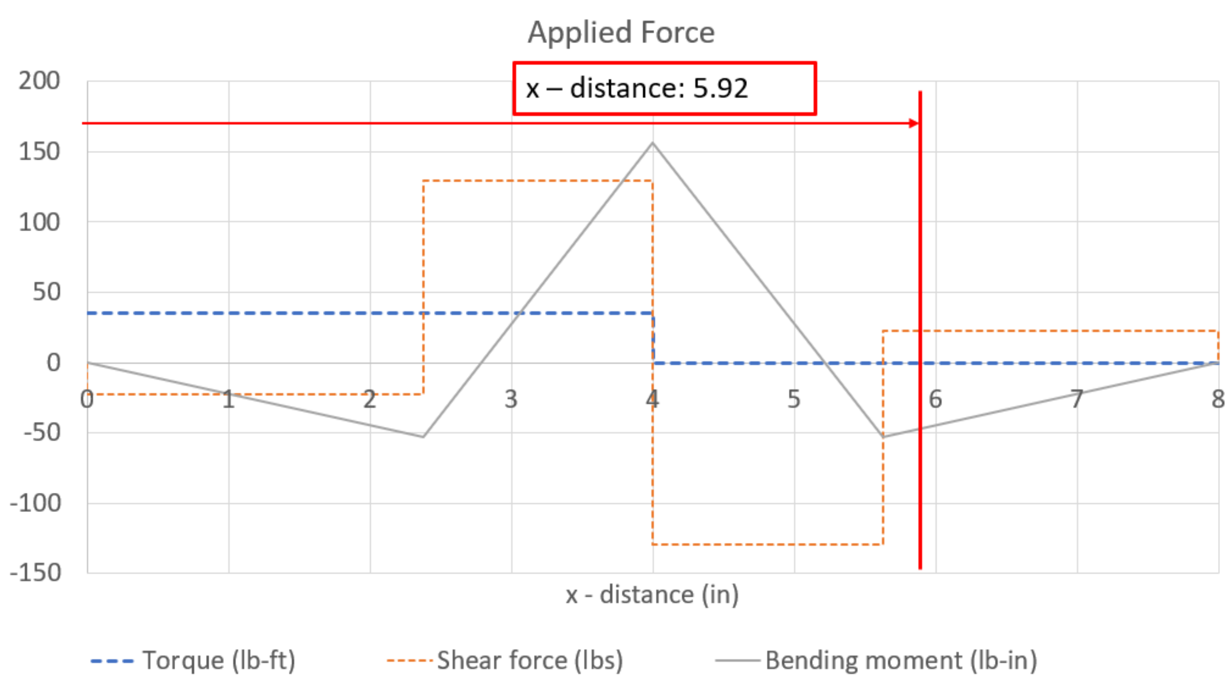
$$n = \frac{1}{0.033} = 30.17$$

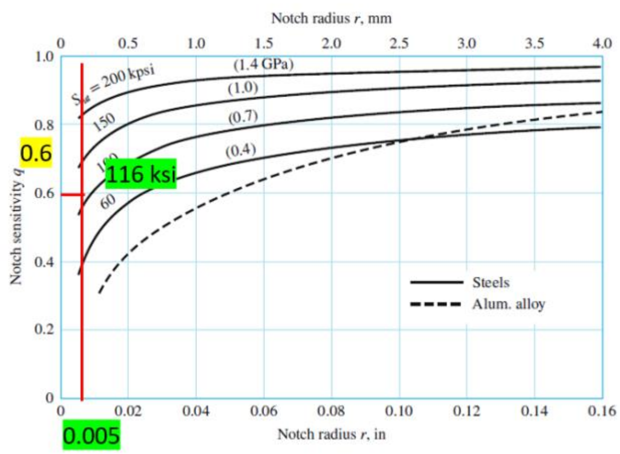
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	0.00	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	0.00	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert\frac{T_{max} - T_{min}}{2}\right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	17.68	lbf-in		$M_a = \left\lvert\frac{M_{max} - M_{min}}{2}\right\rvert$
S_{ut}	Ultimate Tensile Strength	116000	psi	Shaft material: AISI 1030 Q&T	
S_{yt}	Yield Tensile Strength	90000	psi	Shaft material: AISI 1030 Q&T	
q	Notch sensitivity	0.78			
qs	Notch sensitivity	0.82			
r	Notch radius	0.005	in		
K_t	Stress-Concentration factor	1.9			
K_{ts}	Stress-Concentration factor	1.6			
d_{input}	Shaft diameter (input)	0.669	in		
K_f	Fatigue Stess-Concentration Factor	1.702			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	1.492			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.71			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.918			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	58000	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	30886	psi		$S_e = k_a k_b k_c k_d k_e S'_e$
1/n	Utilization Factor	0.033			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	30.17			

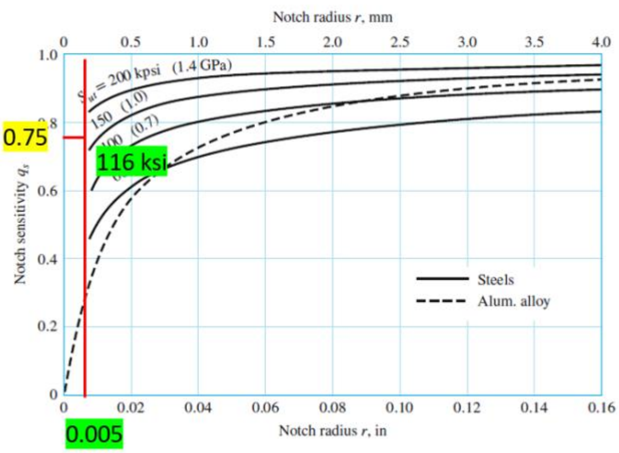
4.2.1.6 D – Groove

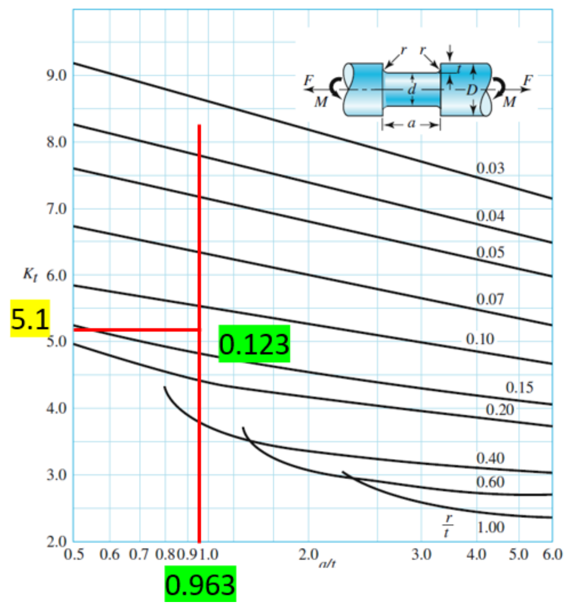
The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

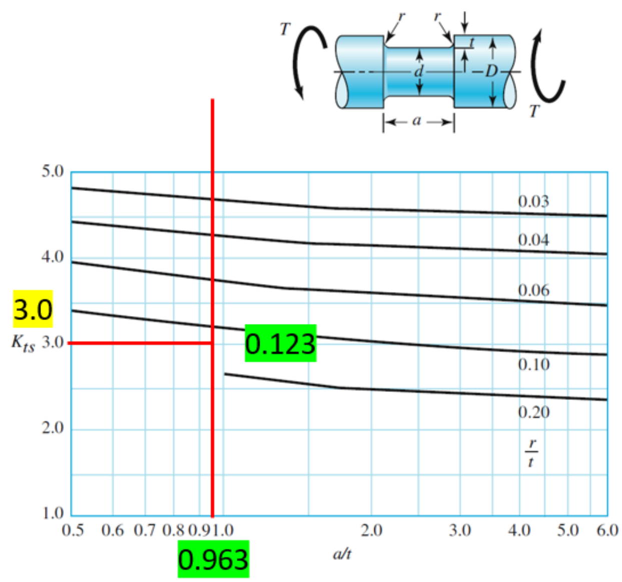
- Torque: 0 lb-ft
- Shear force: 22.44 lb
- Bending moment: -46.68 lb-in
- D: 0.669 in
- t: 0.0405 in
- a: 0.039 in
- q: 0.6
- q_s: 0.75
- r: 0.005 in
- Kt: 5.1
- Kts: 3











Also, the safety factor was calculated based on these inputs is 5.65.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.6(5.1 - 1) = 3.46$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.75(3 - 1) = 2.5$$

$$k_a = aS_{ut}^b = 2 \times 116^{-0.217} = 0.71$$

$$k_b = 0.879d^{-0.107} = 0.879(0.669)^{-0.107} = 0.918$$

$$Se' = 0.5S_{ut} = 0.5 \times 116000 = 58{,}000psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S'_e} = 0.71(0.918)(1)(1)(0.814)(58{,}000) = 30{,}886\,psi$$

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

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

$$= \frac{16}{\pi 0.669^3}\left[4\left(\frac{3.46 \times 46.68}{30{,}886}\right)^2\right]^{\frac{1}{2}} = 0.177$$

$$n = \frac{1}{0.177} = 5.65$$

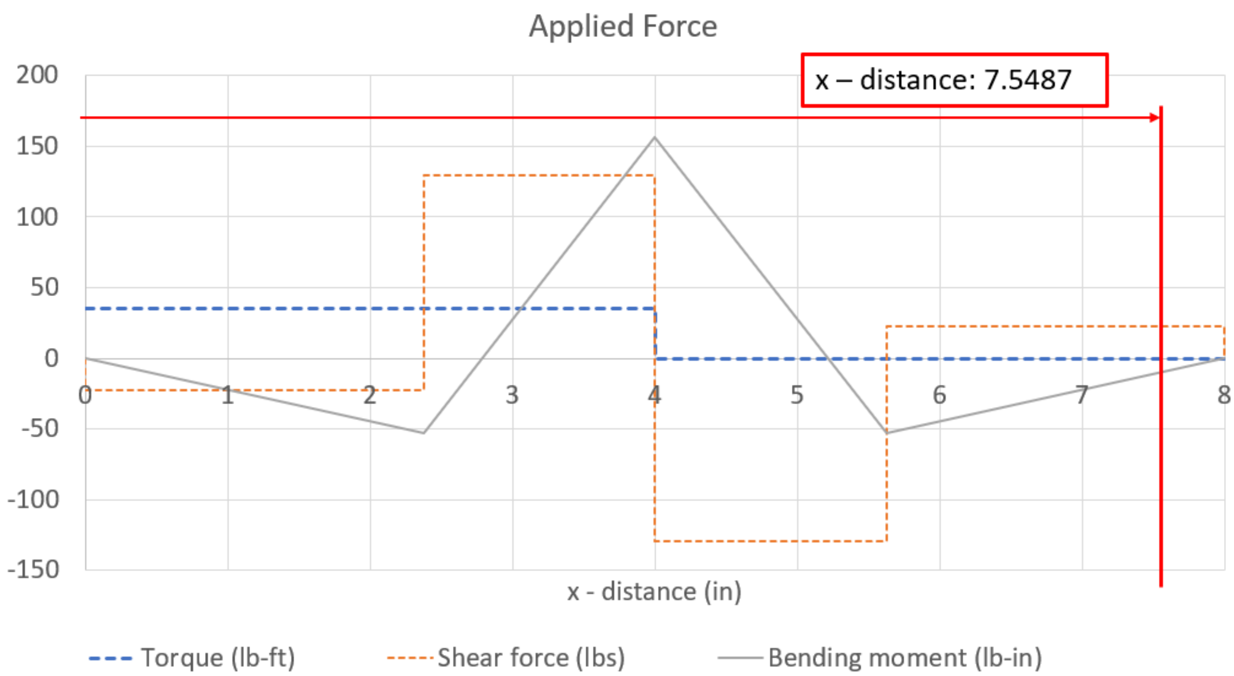
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	0.00	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	0.00	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert\frac{T_{max} - T_{min}}{2}\right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	46.48	lbf-in		$M_a = \left\lvert\frac{M_{max} - M_{min}}{2}\right\rvert$
S_{ut}	Ultimate Tensile Strength	116000	psi	Shaft material: AISI 1030 Q&T	
S_{yt}	Yield Tensile Strength	90000	psi	Shaft material: AISI 1030 Q&T	
q	Notch sensitivity	0.6			
qs	Notch sensitivity	0.75			
r	Notch radius	0.005	in		
K_t	Stress-Concentration factor	5.1			
K_{ts}	Stress-Concentration factor	3			
d_{input}	Shaft diameter (input)	0.669	in		
K_f	Fatigue Stess-Concentration Factor	3.46			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	2.5			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.71			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.918			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	58000	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	30886	psi		$S_e = \mathrm{k_a k_b k_c k_d k_e} S_e'$
1/n	Utilization Factor	0.177			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	5.65			

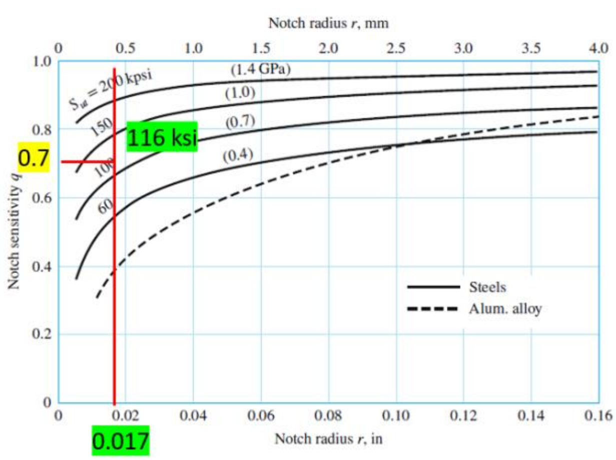
4.2.1.7 E – Step

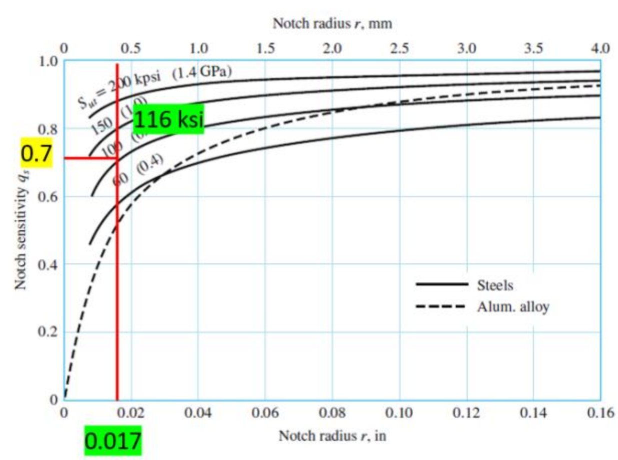
The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

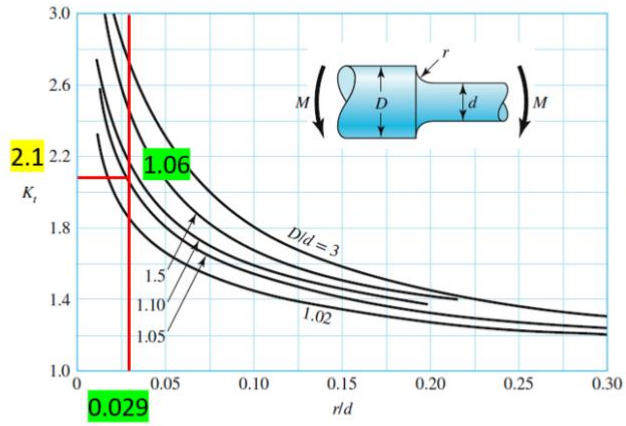
- Torque: 0 lb-ft
- Shear force: -22.44 lb
- Bending moment: -10.13 lb-in
- D: 0.591 in
- t: 0.017 in
- q: 0.7
- q_s: 0.7
- r: 0.017 in
- Kt: 2.1

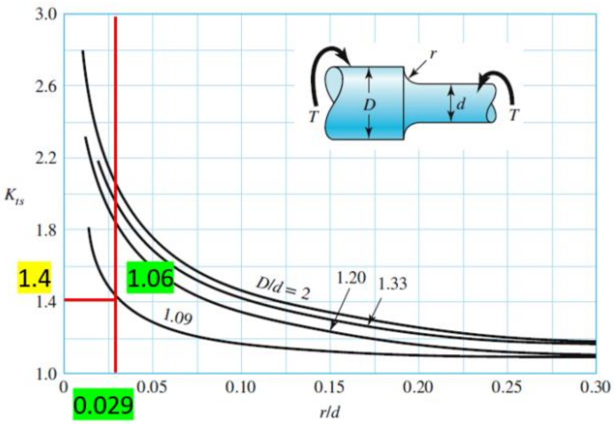
- Kts: 1.4


Applied Force
x – distance: 7.5487
200
150
100
50
0
-50
-100
-150
0
1
2
3
4
5
6
7
8
x - distance (in)
Torque (lb-ft)
Shear force (lbs)
Bending moment (lb-in)


Notch radius r, mm
S_{ut} = 200 kpsi
(1.4 GPa)
150
(1.0)
100
(0.7)
60
(0.4)
116 ksi
0.7
Notch sensitivity q
Steels
Alum. alloy
Notch radius r, in
0.017


Notch radius r, mm
S_{ut} = 200 kpsi
(1.4 GPa)
150
100
60
(0.4)
116 ksi
0.7
Notch sensitivity q_s
Steels
Alum. alloy
Notch radius r, in
0.017





Also, the safety factor was calculated based on these inputs is 35.38.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.7(2.1 - 1) = 1.77$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.7(1.4 - 1) = 1.28$$

$$k_a = aS_{ut}^b = 2 \times 116^{-0.217} = 0.71$$

$$k_b = 0.879d^{-0.107} = 0.879(0.591)^{-0.107} = 0.93$$

$$Se' = 0.5S_{ut} = 0.5 \times 116000 = 58{,}000psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S_e'} = 0.71(0.93)(1)(1)(0.814)(58{,}000) = 31{,}299\ psi$$

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

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

$$= \frac{16}{\pi 0.591^3}\left[4\left(\frac{1.77 \times 10.13}{31{,}299}\right)^2\right]^{\frac{1}{2}} = 0.028$$

$$n = \frac{1}{0.028} = 35.38$$

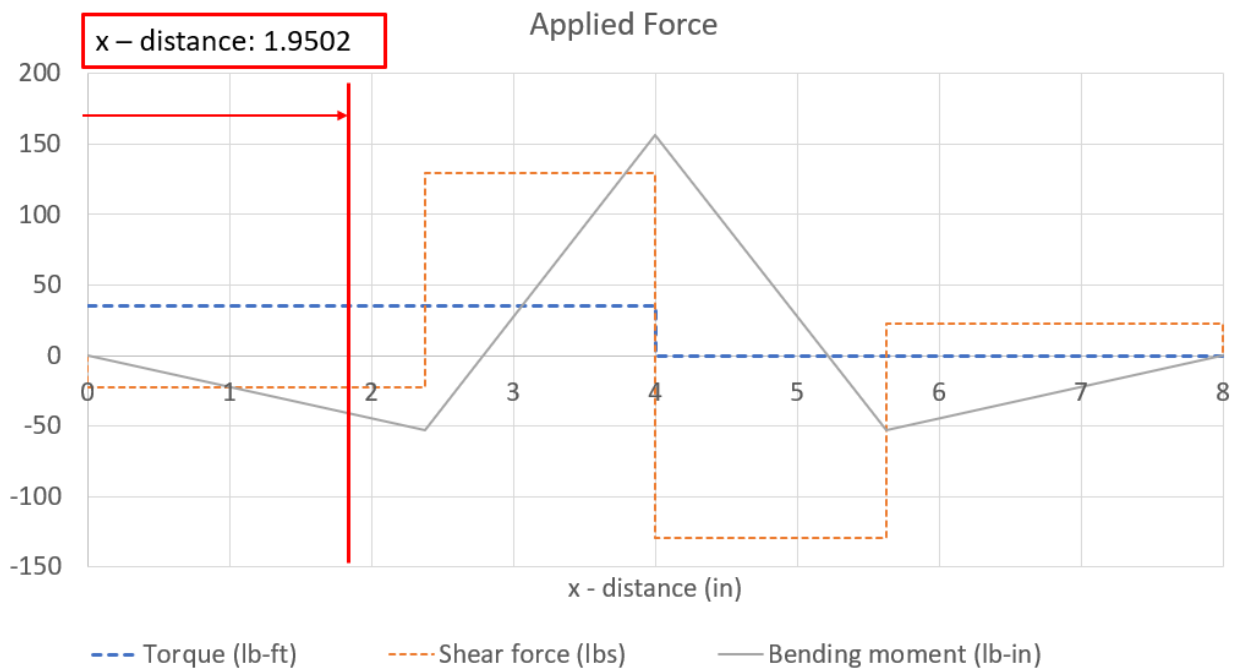
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	0.00	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	0.00	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert\frac{T_{max} - T_{min}}{2}\right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	10.13	lbf-in		$M_a = \left\lvert\frac{M_{max} - M_{min}}{2}\right\rvert$
S_{ut}	Ultimate Tensile Strength	116000	psi	Shaft material: AISI 1030 Q&T	
S_{yt}	Yield Tensile Strength	90000	psi	Shaft material: AISI 1030 Q&T	
q	Notch sensitivity	0.7			
qs	Notch sensitivity	0.7			
r	Notch radius	0.0156	in		
K_t	Stress-Concentration factor	2.1			
K_{ts}	Stress-Concentration factor	1.4			
d_{input}	Shaft diameter (input)	0.591	in		
K_f	Fatigue Stess-Concentration Factor	1.77			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	1.28			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.71			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.930			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	58000	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	31299	psi		$S_e = \mathrm{k_a k_b k_c k_d k_e} S_e'$
1/n	Utilization Factor	0.028			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	35.38			

4.2.1.8 F – Step

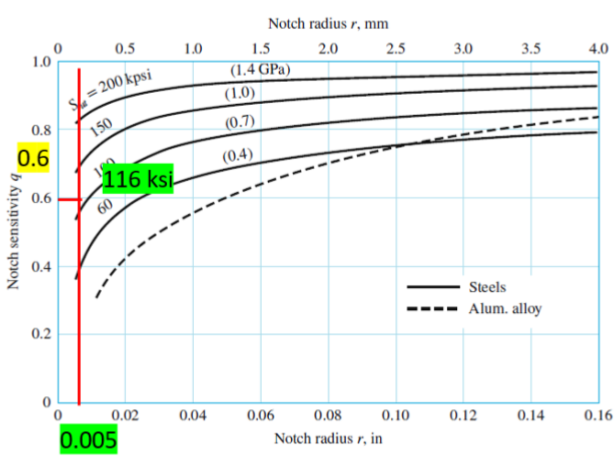
The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

- Torque: 35.01 lb-ft
- Shear force: -22.44 lb
- Bending moment: -43.76 lb-in
- D: 0.625 in
- t: 0.022 in
- q: 0.6
- q_s: 0.75
- r: 0.005 in

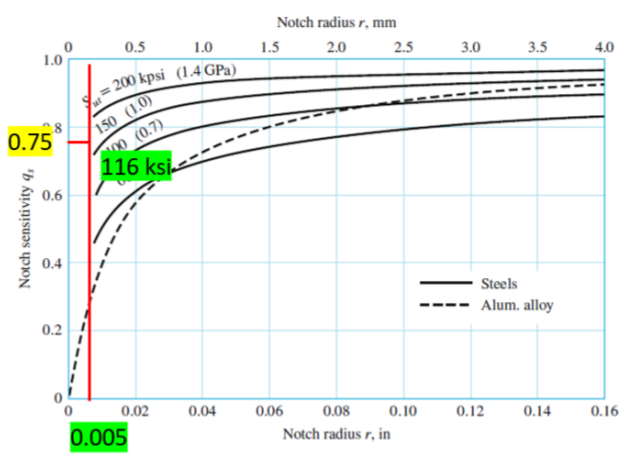
- Kt: 2.65
- Kts: 1.9



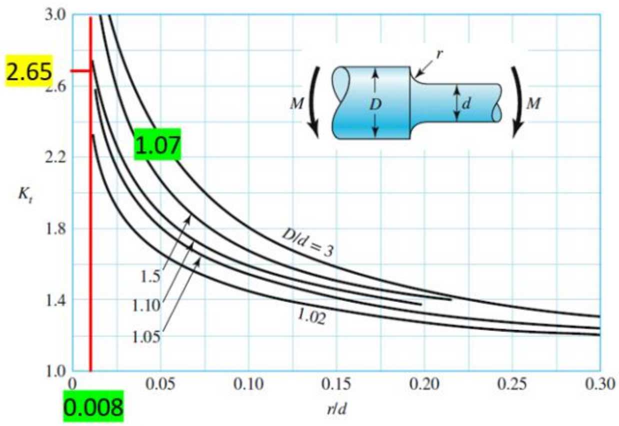
x – distance: 1.9502
Applied Force
200
150
100
50
0
-50
-100
-150
0
1
2
3
4
5
6
7
8
x - distance (in)
Torque (lb-ft)
Shear force (lbs)
Bending moment (lb-in)

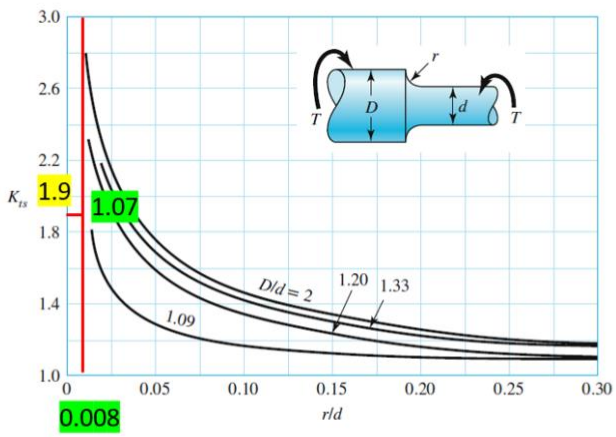


Notch radius r, mm
Notch sensitivity q
Notch radius r, in
0.6
116 ksi
0.005
S_{ut} = 200 kpsi
(1.4 GPa)
(1.0)
150
(0.7)
(0.4)
60
Steels
Alum. alloy



Notch radius r, mm
Notch sensitivity q_s
Notch radius r, in
0.75
116 ksi
0.005
S_{ut} = 200 kpsi
(1.4 GPa)
150
(1.0)
100
(0.7)
Steels
Alum. alloy





Also, the safety factor was calculated based on these inputs is 2.92.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.5(2.7 - 1) = 1.85$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.55(2.1 - 1) = 1.605$$

$$k_a = aS_{ut}^b = 2 \times 91^{-0.217} = 0.75$$

$$k_b = 0.879d^{-0.107} = 0.879(0.625)^{-0.107} = 0.92$$

$$Se' = 0.5S_{ut} = 0.5 \times 91000 = 45{,}500psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S'_e} = 0.75(0.92)(1)(1)(0.814)(45{,}500) = 25{,}727\ psi$$

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

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

$$= \frac{16}{\pi 0.625^3}\left[4\left(\frac{1.85 \times 43.76}{25{,}727}\right)^2 + 3\left(\frac{1.605 \times 420.17}{77{,}000}\right)^2\right]^{\frac{1}{2}} = 0.34$$

$$n = \frac{1}{0.32} = 2.92$$

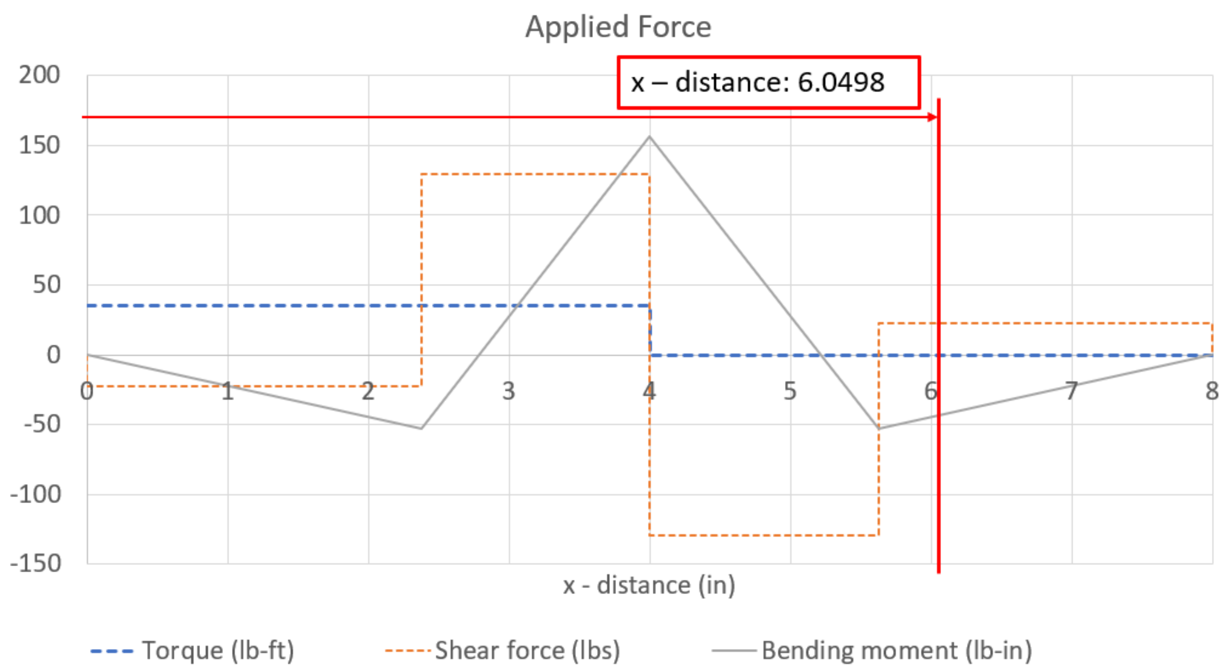
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	35.01	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	420.17	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert\frac{T_{max} - T_{min}}{2}\right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	43.76	lbf-in		$M_a = \left\lvert\frac{M_{max} - M_{min}}{2}\right\rvert$
S_{ut}	Ultimate Tensile Strength	91000	psi	Shaft material: AISI 1045 CD	
S_{yt}	Yield Tensile Strength	77000	psi	Shaft material: AISI 1045 CD	
q	Notch sensitivity	0.5			
qs	Notch sensitivity	0.55			
r	Notch radius	0.005	in		
K_t	Stress-Concentration factor	2.7			
K_{ts}	Stress-Concentration factor	2.1			
d_{input}	Shaft diameter (input)	0.625	in		
K_f	Fatigue Stess-Concentration Factor	1.85			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	1.605			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.75			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.92			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	45500	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	25727	psi		$S_e = k_a k_b k_c k_d k_e S'_e$
1/n	Utilization Factor	0.34			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	2.92			

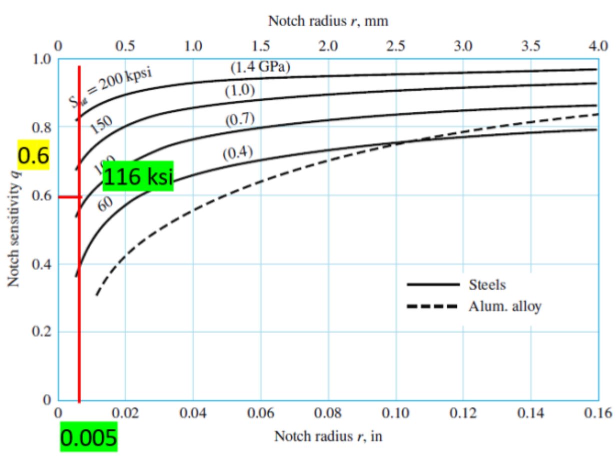
4.2.1.9 G – Step

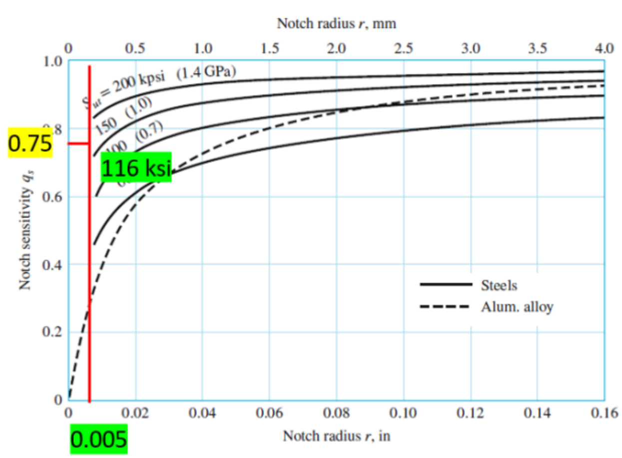
The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

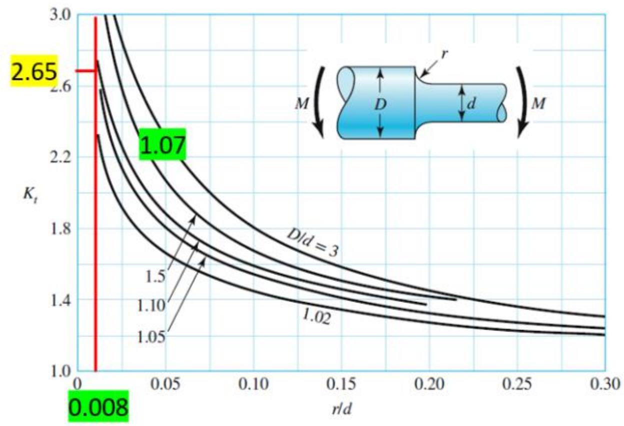
- Torque: 0 lb-ft
- Shear force: 22.44 lb
- Bending moment: -43.76 lb-in
- D: 0.625 in
- t: 0.022 in
- q: 0.5
- q_s: 0.55

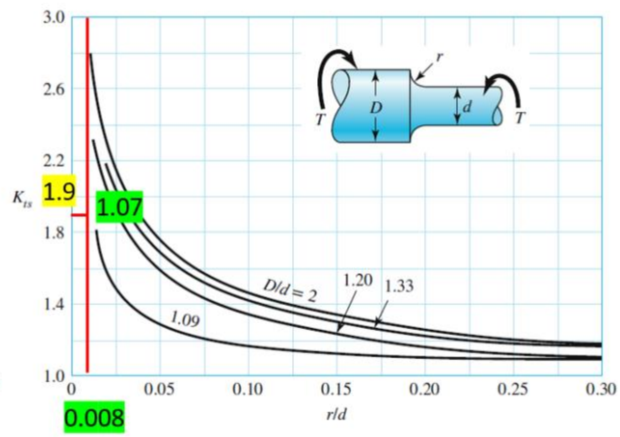
- r: 0.005 in
- Kt: 2.7
- Kts: 2.1











Also, the safety factor was calculated based on these inputs is 7.62.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.5(2.7 - 1) = 1.85$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.55(2.1 - 1) = 1.605$$

$$k_a = aS_{ut}^b = 2 \times 91^{-0.217} = 0.75$$

$$k_b = 0.879d^{-0.107} = 0.879(0.625)^{-0.107} = 0.92$$

$$Se' = 0.5S_{ut} = 0.5 \times 91000 = 45{,}500psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S_e'} = 0.75(0.92)(1)(1)(0.814)(45{,}500) = 25{,}727\ psi$$

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

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

$$= \frac{16}{\pi 0.625^3}\left[4\left(\frac{1.85 \times 43.76}{25{,}727}\right)^2\right]^{\frac{1}{2}} = 0.13$$

$$n = \frac{1}{0.13} = 7.62$$

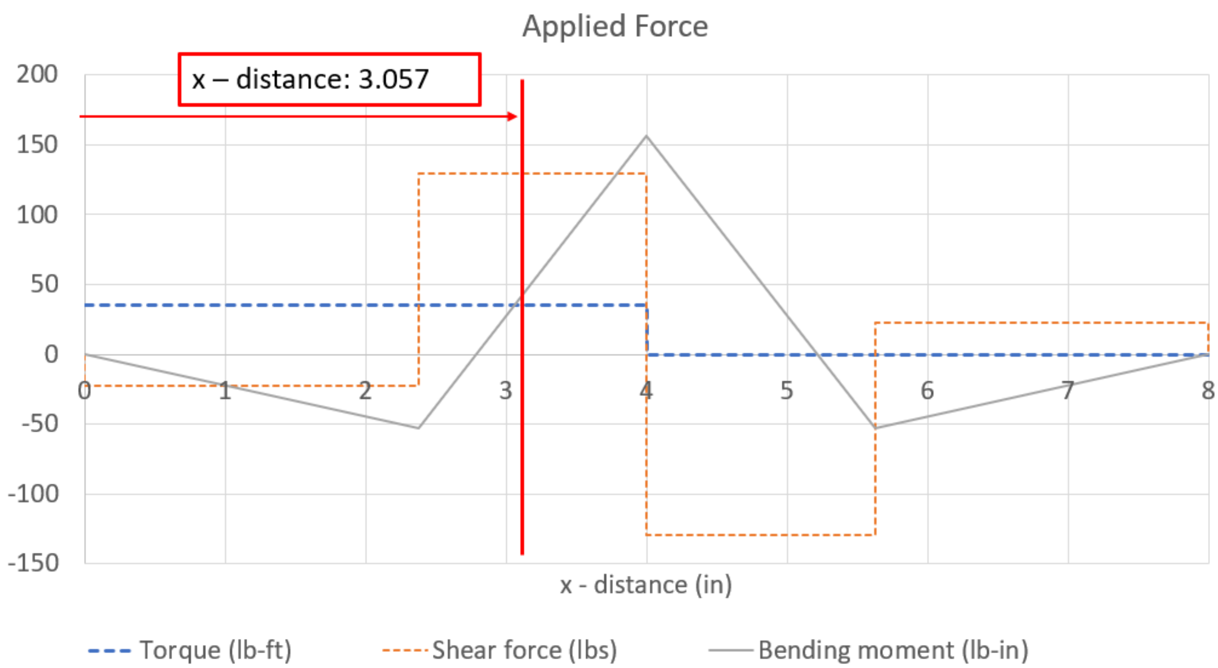
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	0.00	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	0.00	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert\frac{T_{max} - T_{min}}{2}\right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	43.76	lbf-in		$M_a = \left\lvert\frac{M_{max} - M_{min}}{2}\right\rvert$
S_{ut}	Ultimate Tensile Strength	91000	psi	Shaft material: AISI 1045 CD	
S_{yt}	Yield Tensile Strength	77000	psi	Shaft material: AISI 1045 CD	
q	Notch sensitivity	0.5			
qs	Notch sensitivity	0.55			
r	Notch radius	0.005	in		
K_t	Stress-Concentration factor	2.7			
K_{ts}	Stress-Concentration factor	2.1			
d_{input}	Shaft diameter (input)	0.625	in		
K_f	Fatigue Stess-Concentration Factor	1.85			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	1.605			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.75			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.92			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	45500	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	25727	psi		$S_e = k_a k_b k_c k_d k_e S_e'$
1/n	Utilization Factor	0.13			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	7.62			

4.2.1.11 H - Groove

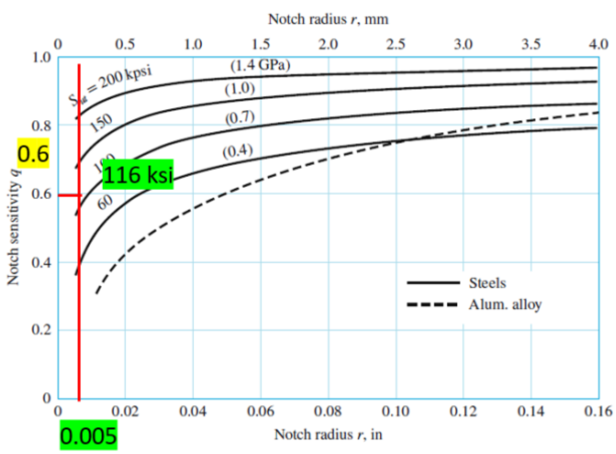
The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

- Torque: 35.01 lb-ft
- Shear force: 129.28 lb
- Bending moment: 34.87 lb-in
- D: 0.875 in
- t: 0.027 in
- a: 0.086 in
- q: 0.6
- q_s: 0.75
- r: 0.005 in

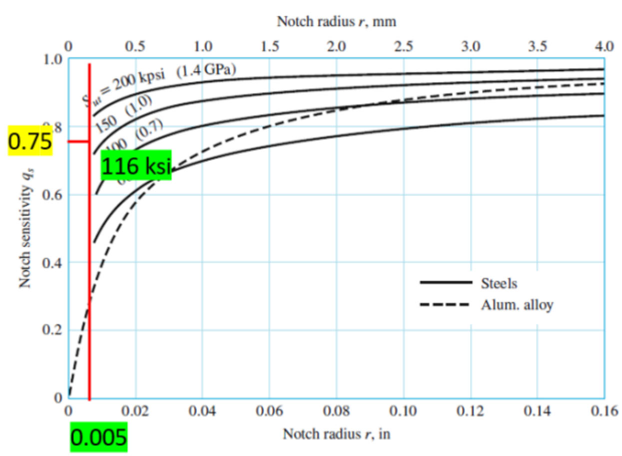
- Kt: 4.1
- Kts: 2.5



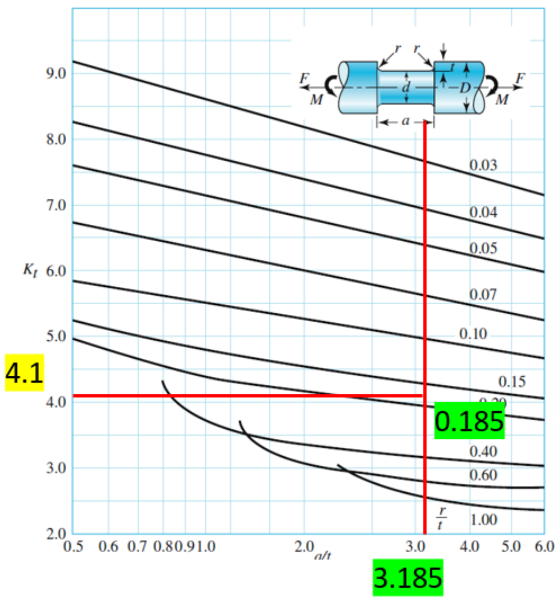
Applied Force
x – distance: 3.057
200
150
100
50
0
-50
-100
-150
0
1
2
3
4
5
6
7
8
x - distance (in)
Torque (lb-ft)
Shear force (lbs)
Bending moment (lb-in)

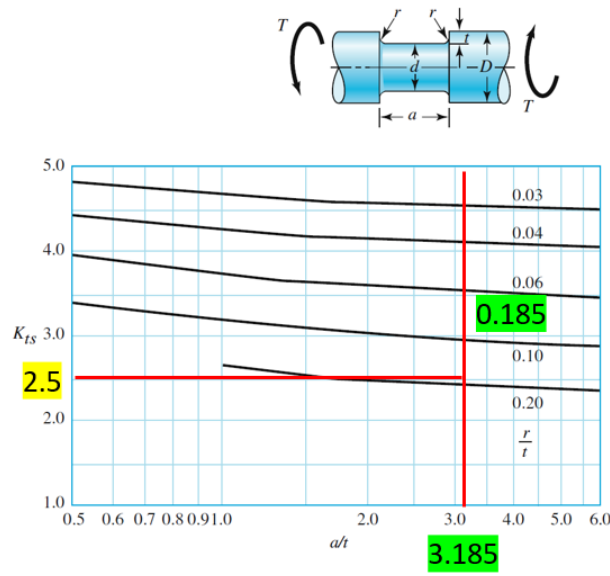


Notch radius r, mm
Notch radius r, in
Notch sensitivity q
S_{ut} = 200 kpsi
(1.4 GPa)
150
(1.0)
(0.7)
60
(0.4)
0.6
116 ksi
0.005
Steels
Alum. alloy



Notch radius r, mm
Notch radius r, in
Notch sensitivity q_s
S_{ut} = 200 kpsi
(1.4 GPa)
150
(1.0)
(0.7)
0.75
116 ksi
0.005
Steels
Alum. alloy





Also, the safety factor calculated based on these inputs is 7.14.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.6(4.1 - 1) = 2.86$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.75(2.5 - 1) = 2.125$$

$$k_a = aS_{ut}^b = 2 \times 116^{-0.217} = 0.713$$

$$k_b = 0.879d^{-0.107} = 0.879(0.875)^{-0.107} = 0.89$$

$$Se' = 0.5S_{ut} = 0.5 \times 116000 = 58{,}000psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S_e'} = 0.713(0.89)(1)(1)(0.814)(58{,}000) = 30{,}012\, psi$$

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

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

$$=\frac{16}{\pi 0.875^3}\left[4\left(\frac{2.86\times 34.87}{30{,}012}\right)^2+3\left(\frac{2.125\times 420.17}{90{,}000}\right)^2\right]^{\frac{1}{2}}=0.14$$

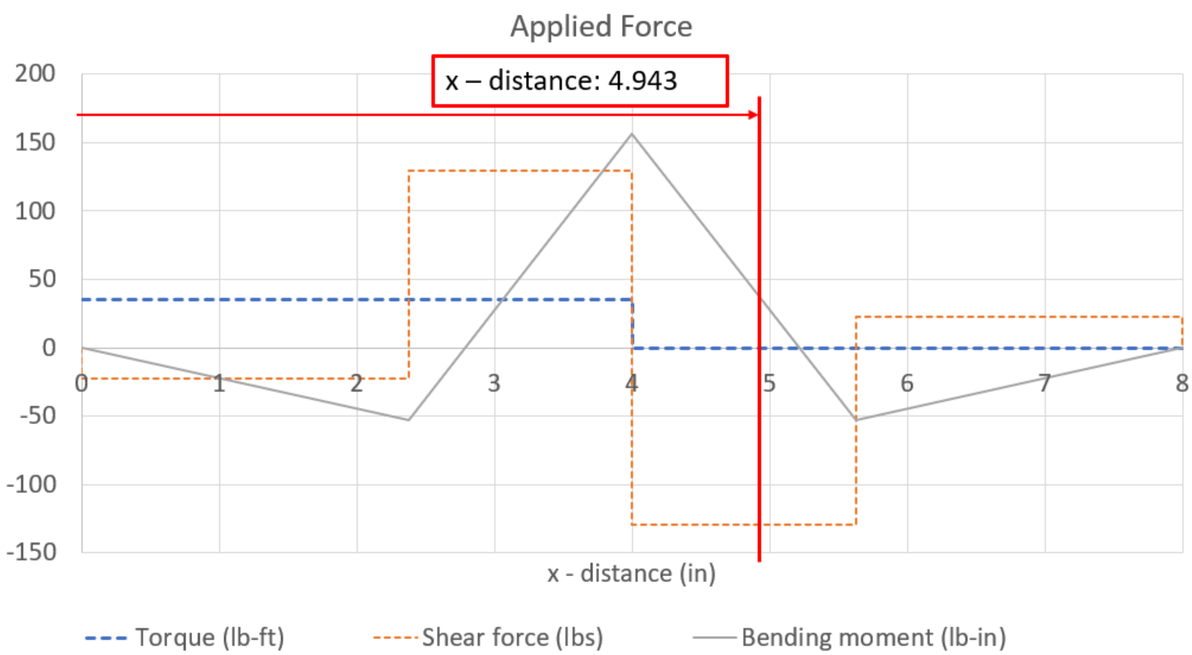
$$n=\frac{1}{0.14}=7.14$$

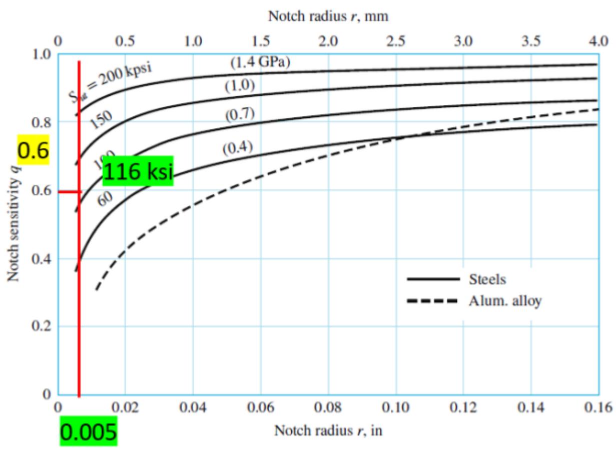
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	35.01	lbf-ft		$T_m=\frac{T_{max}+T_{min}}{2}$
T_m	Mean Torque (lbf-in)	420.17	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a=\left\lvert\frac{T_{max}-T_{min}}{2}\right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m=\frac{M_{max}+M_{min}}{2}$
M_a	Alternating Bending Moment	34.87	lbf-in		$M_a=\left\lvert\frac{M_{max}-M_{min}}{2}\right\rvert$
S_{ut}	Ultimate Tensile Strength	116000	psi	Shaft material: AISI 1030 Q&T	
S_{yt}	Yield Tensile Strength	90000	psi	Shaft material: AISI 1030 Q&T	
q	Notch sensitivity	0.6			
qs	Notch sensitivity	0.75			
r	Notch radius	0.005	in		
K_t	Stress-Concentration factor	4.1			
K_{ts}	Stress-Concentration factor	2.5			
d_{input}	Shaft diameter (input)	0.875	in		
K_f	Fatigue Stess-Concentration Factor	2.860			$K_f=1+q(K_t-1)$
K_{fs}	Fatigue Stess-Concentration Factor	2.125			$K_{fs}=1+q_s(K_{ts}-1)$
k_a	Surface factor	0.713			$k_a=aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.892			$k_b=0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	58000	psi	Sut < 200 ksi	$Se'=0.5S_{ut}$
S_e	Endurance limit at the critical location	30012	psi		$S_e=k_a k_b k_c k_d k_e S_e'$
1/n	Utilization Factor	0.140			$\frac{1}{n}=\frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2+3\left(\frac{K_{fs}T_a}{S_e}\right)^2+4\left(\frac{K_f M_m}{S_y}\right)^2+3\left(\frac{K_{fs}T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	7.14			

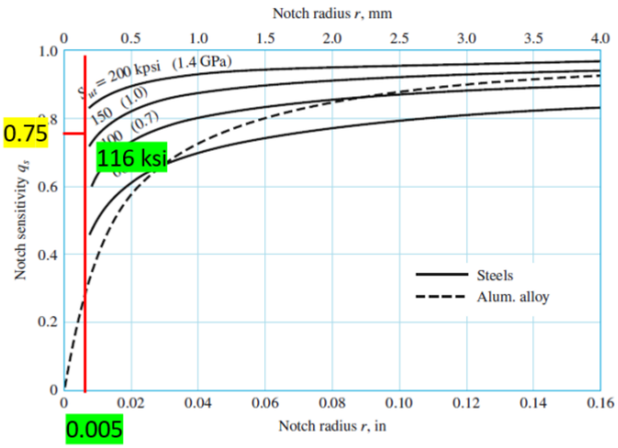
4.2.1.12 I - Groove

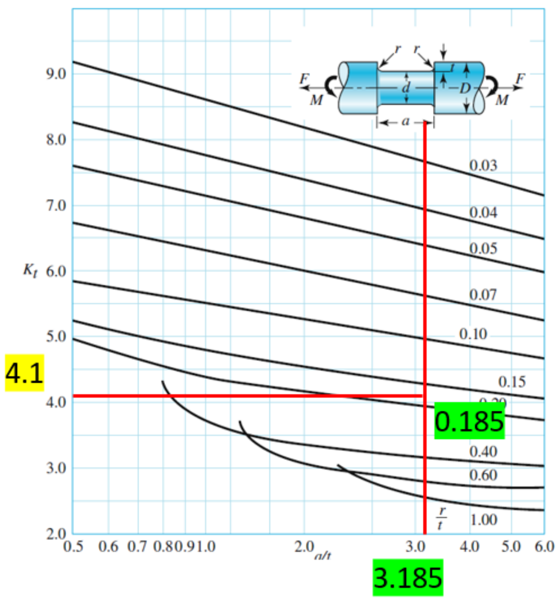
The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

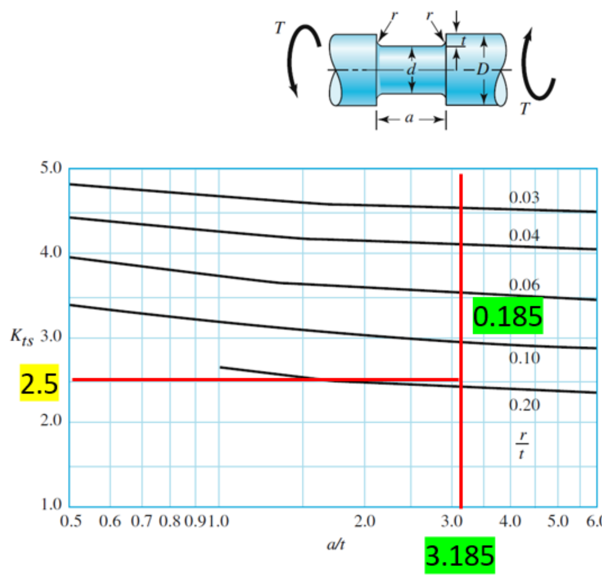
- Torque: 0 lb-ft
- Shear force: -129.28 lb
- Bending moment: 34.87 lb-in
- D: 0.875 in
- t: 0.027 in
- a: 0.086 in
- q: 0.6
- q_s: 0.75
- r: 0.005 in
- Kt: 4.1
- Kts: 2.5











Also, the safety factor calculated based on these inputs is 19.79.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.6(4.1 - 1) = 2.86$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.75(2.5 - 1) = 2.125$$

$$k_a = aS_{ut}^b = 2 \times 116^{-0.217} = 0.713$$

$$k_b = 0.879d^{-0.107} = 0.879(0.875)^{-0.107} = 0.89$$

$$Se' = 0.5S_{ut} = 0.5 \times 116000 = 58{,}000 psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e} S_e' = 0.713(0.89)(1)(1)(0.814)(58{,}000) = 30{,}012\, psi$$

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

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

$$= \frac{16}{\pi 0.875^3}\left[4\left(\frac{2.86 \times 34.87}{30{,}012}\right)^2\right]^{\frac{1}{2}} = 0.051$$

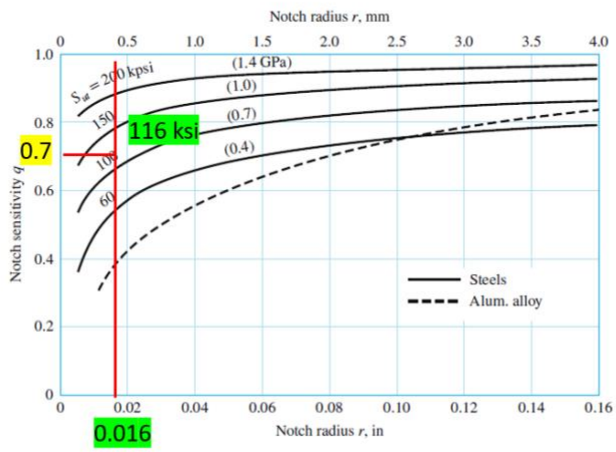
$$n = \frac{1}{0.051} = 19.79$$

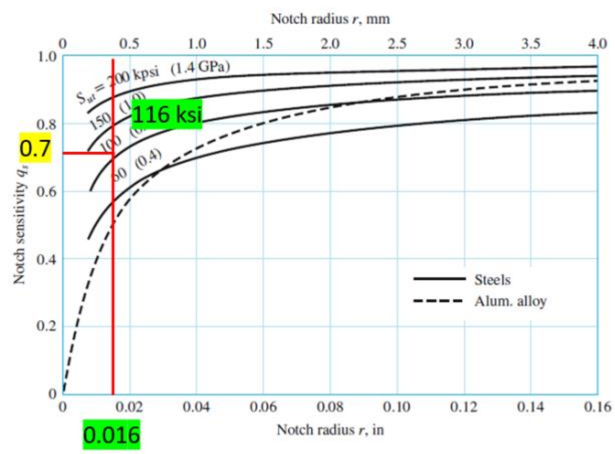
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	0.00	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	0.00	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert\frac{T_{max} - T_{min}}{2}\right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	34.87	lbf-in		$M_a = \left\lvert\frac{M_{max} - M_{min}}{2}\right\rvert$
S_{ut}	Ultimate Tensile Strength	116000	psi	Shaft material: AISI 1030 Q&T	
S_{yt}	Yield Tensile Strength	90000	psi	Shaft material: AISI 1030 Q&T	
q	Notch sensitivity	0.6			
qs	Notch sensitivity	0.75			
r	Notch radius	0.005	in		
K_t	Stress-Concentration factor	4.1			
K_{ts}	Stress-Concentration factor	2.5			
d_{input}	Shaft diameter (input)	0.875	in		
K_f	Fatigue Stess-Concentration Factor	2.860			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	2.125			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.713			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.892			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	58000	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	30012	psi		$S_e = k_a k_b k_c k_d k_e S_e'$
1/n	Utilization Factor	0.051			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	19.79			

4.2.1.13 J – Key

The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

- Torque: 35.01 lb-ft
- Shear force: 0 lb
- Bending moment: 0 lb-in
- D: 0.591 in
- t: 0.0625 in
- q: 0.7
- q_s: 0.7
- r: 0.0156 in
- Kt: 2.158
- Kts: 2.235





Shaft diameter	Key size (Width X Height)	Depth of key seat	Fillet radius	K_{TB} (Bending)	K_s (Torsion)	K_{TA} (Axial)
3/8”	3/32” X 3/32”	3/64”	0.0078”	2.202	2.232	2.830
½”	1/8” X 1/8”	1/16”	0.0104”	2.150	2.234	2.830
11/16”	3/16” X 3/16”	3/32”	0.0156”	2.158	2.235	2.847
1 1/16”	¼” X ¼”	1/8”	0.0208”	2.280	2.244	2.778
1 5/16”	5/16” X 5/16”	5/32”	0.0260”	2.267	2.228	2.797
1 9/16”	3/8” X 3/8”	3/16”	0.0312”	2.312	2.246	2.824

Based on the shaft diameter of 0.591 inches in the course material 'ASEE-2013-6775', Kt and Kts values of 2.158 and 2.235 were used.

Also, the safety factor was calculated based on these inputs is 2.69.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.7(2.158 - 1) = 1.811$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.7(2.235 - 1) = 1.865$$

$$k_a = aS_{ut}^b = 2 \times 116^{-0.217} = 0.71$$

$$k_b = 0.879d^{-0.107} = 0.879(0.591)^{-0.107} = 0.93$$

$$Se' = 0.5S_{ut} = 0.5 \times 116{,}000 = 58{,}000psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S_e'} = 0.71(0.93)(1)(1)(0.814)(58{,}000) = 31{,}299\ psi$$

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

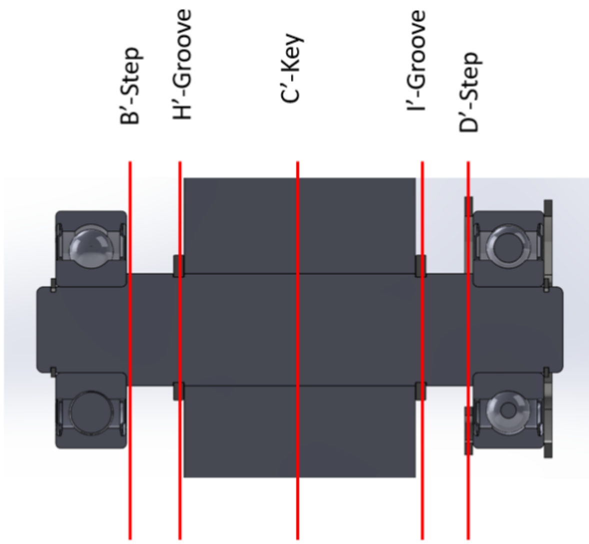
$$= \frac{16}{\pi d^3}\left[\cancel{4\left(\frac{K_f M_a}{S_e}\right)^2} + \cancel{3\left(\frac{K_{fs} T_a}{S_e}\right)^2} + \cancel{4\left(\frac{K_f M_m}{S_y}\right)^2} + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$$

$$= \frac{16}{\pi 0.591^3}\left[3\left(\frac{1.865 \times 420.17}{90{,}000}\right)^2\right]^{\frac{1}{2}} = 0.372$$

$$n = \frac{1}{0.372} = 2.69$$

Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	35.01	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	420.17	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert\frac{T_{max} - T_{min}}{2}\right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	0.00	lbf-in		$M_a = \left\lvert\frac{M_{max} - M_{min}}{2}\right\rvert$
S_{ut}	Ultimate Tensile Strength	116000	psi	Shaft material: AISI 1030 Q&T	
S_{yt}	Yield Tensile Strength	90000	psi	Shaft material: AISI 1030 Q&T	
q	Notch sensitivity	0.7			
qs	Notch sensitivity	0.7			
r	Notch radius	0.0156	in		
K_t	Stress-Concentration factor	2.158			
K_{ts}	Stress-Concentration factor	2.235			
d_{input}	Shaft diameter (input)	0.591	in		
K_f	Fatigue Stess-Concentration Factor	1.811			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	1.865			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.713			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.930			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	58000	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	31299	psi		$S_e = \mathrm{k_a k_b k_c k_d k_e S'_e}$
1/n	Utilization Factor	0.372			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	2.69			

4.2.2 Secondary Shaft Design and Analysis

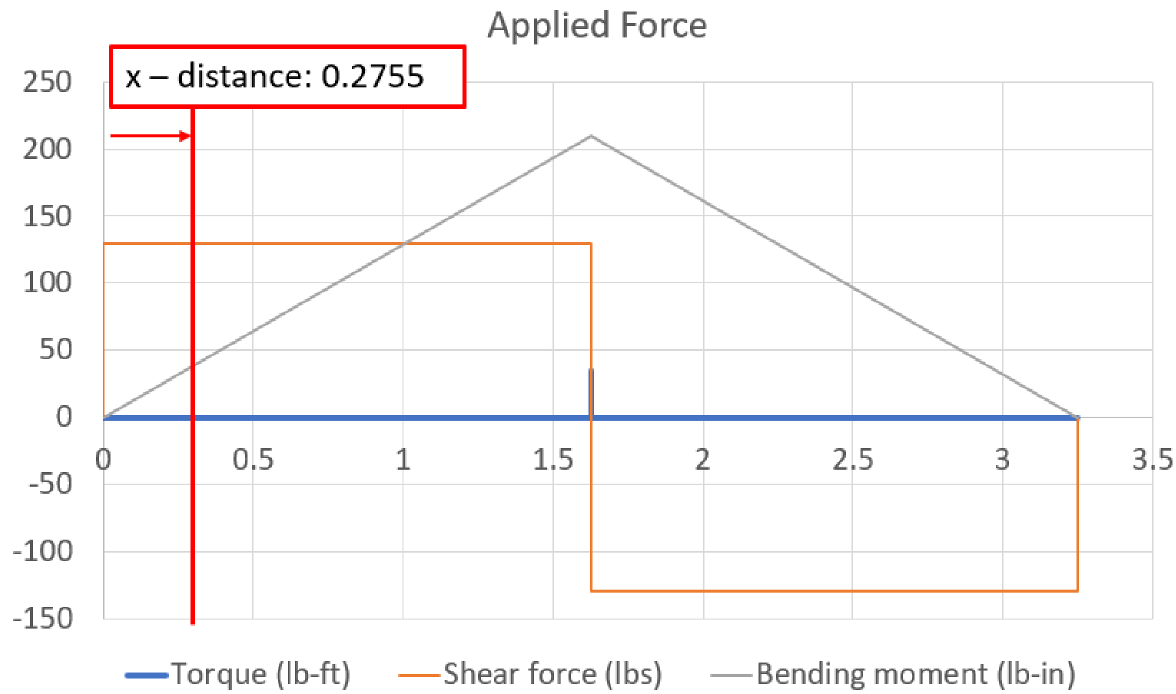


The secondary shaft was analyzed in multiple sections where stress concentrations occur. For evaluating the shaft's fatigue safety, the DE-ASME equation was used. The shaft is made of AISI 1045 CD steel, with an ultimate tensile strength of 91 ksi and a yield strength of 77 ksi.

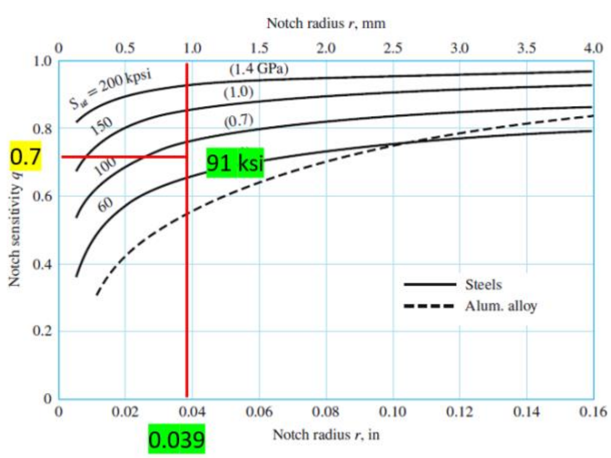
4.2.2.1 B' – Step

The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

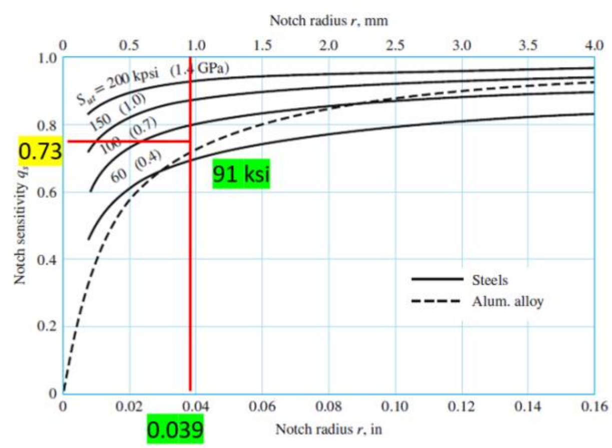
- Torque: 0 lb-ft
- Shear force: 129.28 lb
- Bending moment: 35.62 lb-in
- D: 0.669 in
- t: 0.103 in
- q: 0.7
- q_s: 0.73
- r: 0.039 in
- Kt: 1.92
- Kts: 1.62



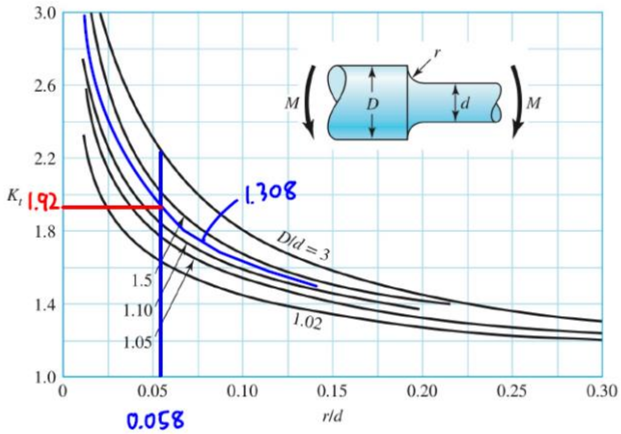
Applied Force
x – distance: 0.2755
Torque (lb-ft)
Shear force (lbs)
Bending moment (lb-in)

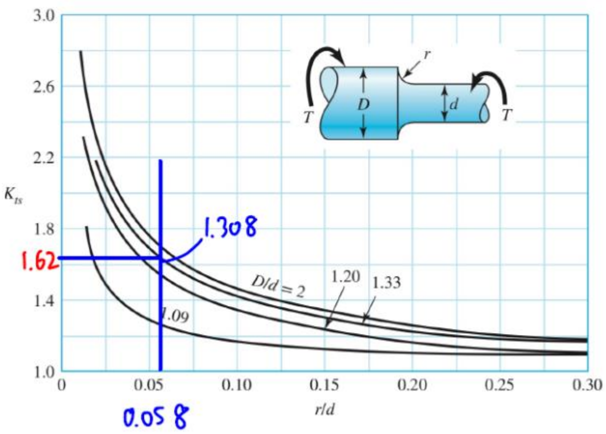


Notch radius r, mm
Notch radius r, in
Notch sensitivity q
Steels
Alum. alloy
0.7
91 ksi
0.039



Notch radius r, mm
Notch radius r, in
Steels
Alum. alloy
0.73
91 ksi
0.039





Also, the safety factor was calculated based on these inputs is 12.82.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.7(1.92 - 1) = 1.644$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.73(1.62 - 1) = 1.453$$

$$k_a = aS_{ut}^b = 2 \times 91^{-0.217} = 0.75$$

$$k_b = 0.879d^{-0.107} = 0.879(0.669)^{-0.107} = 0.918$$

$$Se' = 0.5S_{ut} = 0.5 \times 91000 = 45{,}500psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S_e'} = 0.75(0.918)(1)(1)(0.814)(45{,}500) = 25{,}540\ psi$$

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

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

$$= \frac{16}{\pi 0.669^3}\left[4\left(\frac{1.644 \times 17.68}{25{,}540}\right)^2\right]^{\frac{1}{2}} = 0.08$$

$$n = \frac{1}{0.08} = 12.82$$

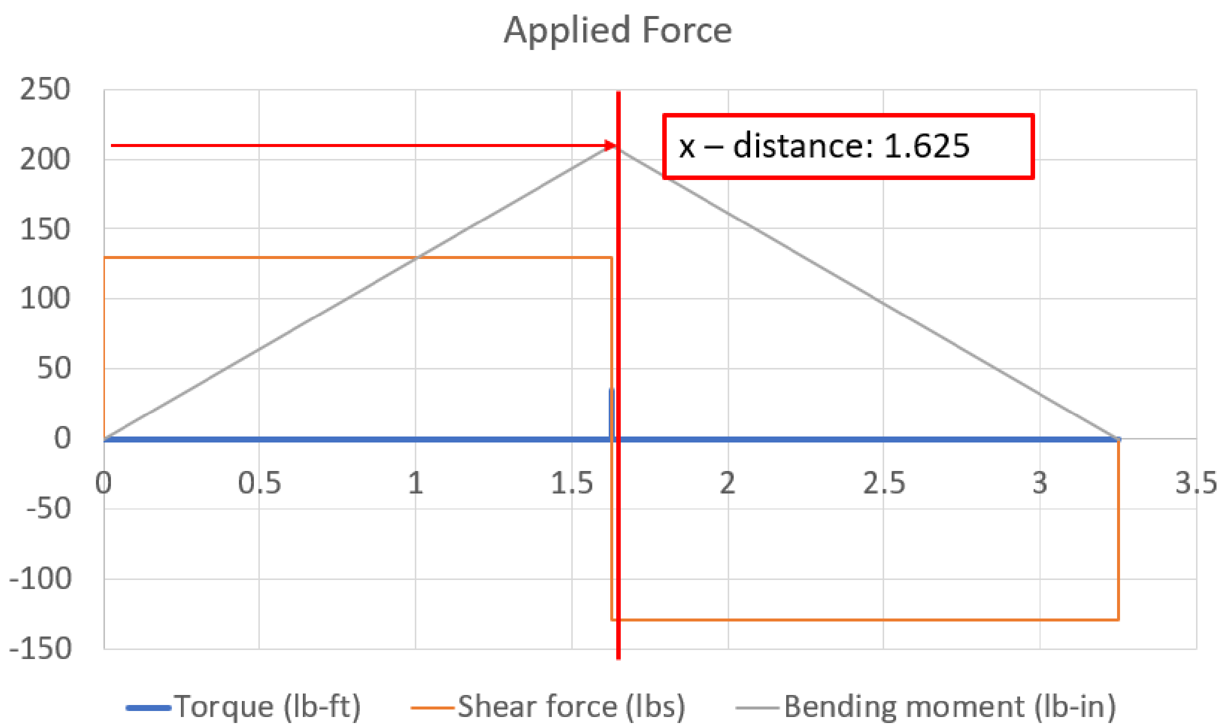
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	0.00	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	0.00	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert\frac{T_{max} - T_{min}}{2}\right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	35.62	lbf-in		$M_a = \left\lvert\frac{M_{max} - M_{min}}{2}\right\rvert$
S_{ut}	Ultimate Tensile Strength	91000	psi	Shaft material: AISI 1045 CD	
S_{yt}	Yield Tensile Strength	77000	psi	Shaft material: AISI 1045 CD	
q	Notch sensitivity	0.7			
qs	Notch sensitivity	0.73			
r	Notch radius	0.039	in		
K_t	Stress-Concentration factor	1.92			
K_{ts}	Stress-Concentration factor	1.62			
d_{input}	Shaft diameter (input)	0.669	in		
K_f	Fatigue Stess-Concentration Factor	1.644			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	1.4526			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.75			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.918			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	45500	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	25540	psi		$S_e = \mathrm{k_a k_b k_c k_d k_e S'_e}$
1/n	Utilization Factor	0.08			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	12.82			

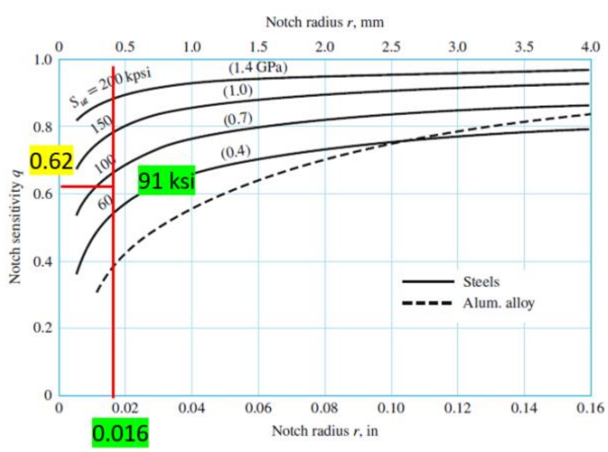
4.2.2.2 C' – Key

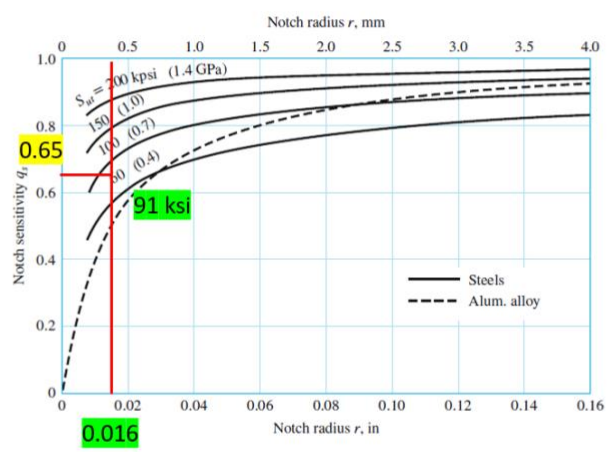
The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

- Torque: 35.01 lb-ft
- Shear force: 129.28 lb
- Bending moment: 210.08 lb-in
- D: 0.875 in
- t: 0.615 in
- q: 0.62
- q_s: 0.65
- r: 0.0156 in
- Kt: 2.158

- Kts: 2.235


Applied Force
x – distance: 1.625
Torque (lb-ft)
Shear force (lbs)
Bending moment (lb-in)


Notch radius r, mm
Notch sensitivity q
Notch radius r, in
S_{ut} = 200 kpsi
(1.4 GPa)
(1.0)
(0.7)
(0.4)
0.62
91 ksi
0.016
Steels
Alum. alloy


Notch radius r, mm
Notch sensitivity q_s
Notch radius r, in
S_{ut} = 200 kpsi
(1.4 GPa)
0.65
91 ksi
0.016
Steels
Alum. alloy

Shaft diameter	Key size (Width X Height)	Depth of key seat	Fillet radius	K_{TB} (Bending)	K_s (Torsion)	K_{TA} (Axial)
3/8”	3/32” X 3/32”	3/64”	0.0078”	2.202	2.232	2.830
½”	1/8” X 1/8”	1/16”	0.0104”	2.150	2.234	2.830
11/16”	3/16” X 3/16”	3/32”	0.0156”	2.158	2.235	2.847
1 1/16”	¼” X ¼”	1/8”	0.0208”	2.280	2.244	2.778
1 5/16”	5/16” X 5/16”	5/32”	0.0260”	2.267	2.228	2.797
1 9/16”	3/8” X 3/8”	3/16”	0.0312”	2.312	2.246	2.824

Based on the shaft diameter of 0.875 inches in the course material 'ASEE-2013-6775', Kt and Kts values of 2.158 and 2.235 were used.

Also, the safety factor was calculated based on these inputs is 3.9.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.62(2.158 - 1) = 1.718$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.65(2.235 - 1) = 1.803$$

$$k_a = aS_{ut}^b = 2 \times 91^{-0.217} = 0.75$$

$$k_b = 0.879d^{-0.107} = 0.879(0.875)^{-0.107} = 0.89$$

$$Se' = 0.5S_{ut} = 0.5 \times 91000 = 45{,}500psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S_e'} = 0.75(0.89)(1)(1)(0.814)(45{,}500) = 24{,}817\ psi$$

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

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

$$= \frac{16}{\pi 0.875^3}\left[4\left(\frac{1.718 \times 210.08}{24{,}817}\right)^2 + 3\left(\frac{1.803 \times 420.17}{77{,}000}\right)^2\right]^{\frac{1}{2}} = 0.256$$

$$n = \frac{1}{0.256} = 3.9$$

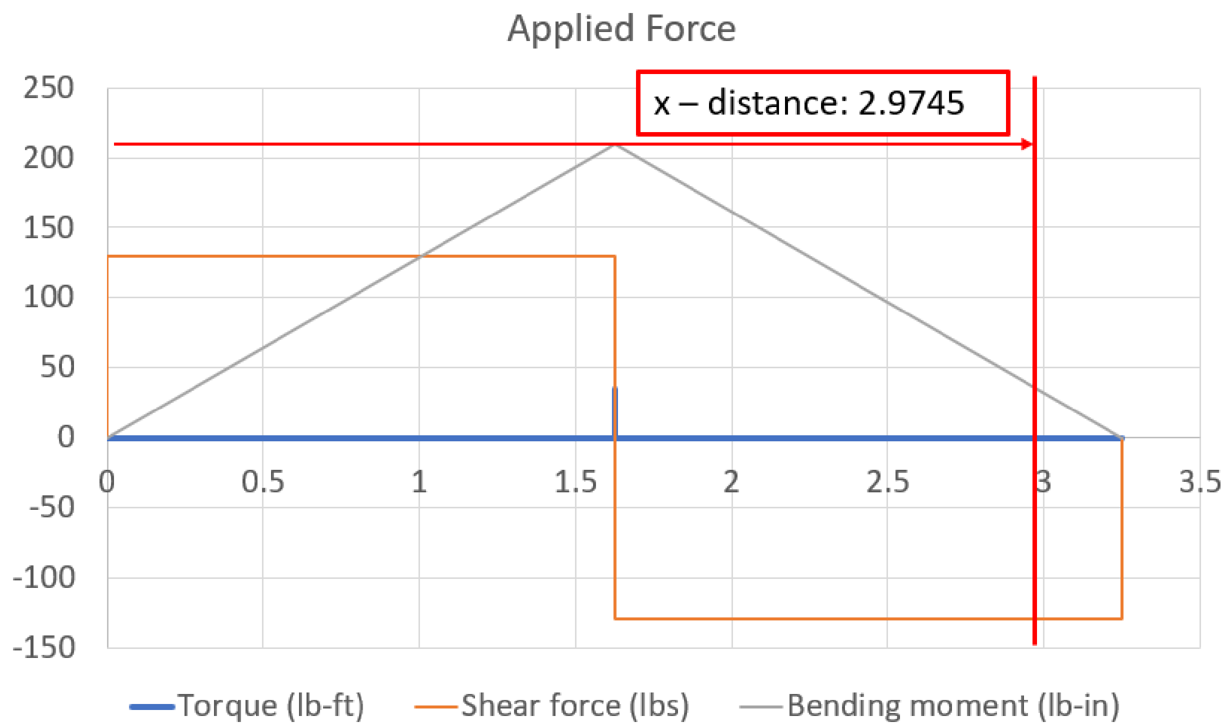
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	35.01	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	420.17	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert\frac{T_{max} - T_{min}}{2}\right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	210.08	lbf-in		$M_a = \left\lvert\frac{M_{max} - M_{min}}{2}\right\rvert$
S_{ut}	Ultimate Tensile Strength	91000	psi	Shaft material: AISI 1030 Q&T	
S_{yt}	Yield Tensile Strength	77000	psi	Shaft material: AISI 1030 Q&T	
q	Notch sensitivity	0.62			
qs	Notch sensitivity	0.65			
r	Notch radius	0.0156	in		
K_t	Stress-Concentration factor	2.158			
K_{ts}	Stress-Concentration factor	2.235			
d_{input}	Shaft diameter (input)	0.875	in		
K_f	Fatigue Stess-Concentration Factor	1.718			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	1.803			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.751			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.892			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	45500	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	24817	psi		$S_e = \mathrm{k_a k_b k_c k_d k_e} S_e'$
1/n	Utilization Factor	0.256			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	3.90			

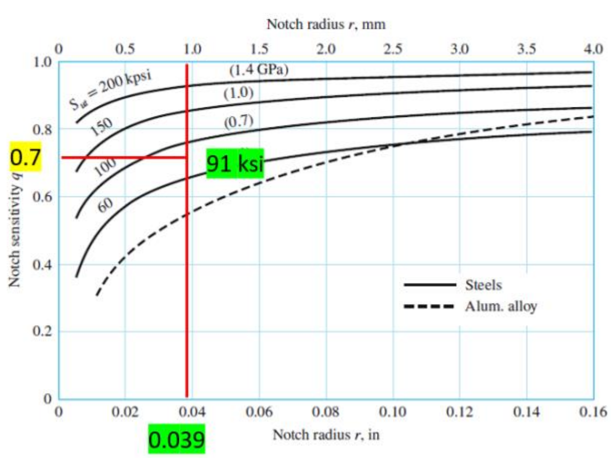
4.2.2.3 D’ – Step

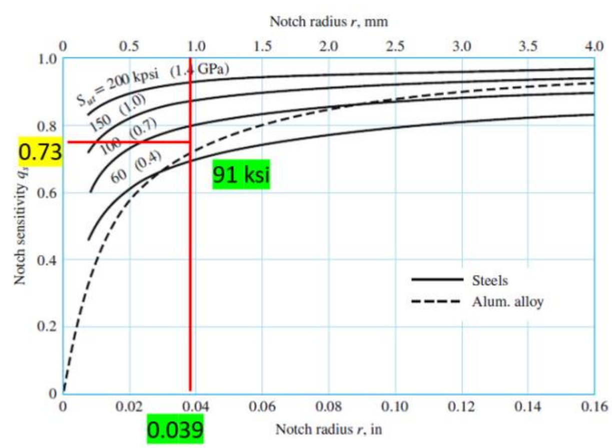
The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

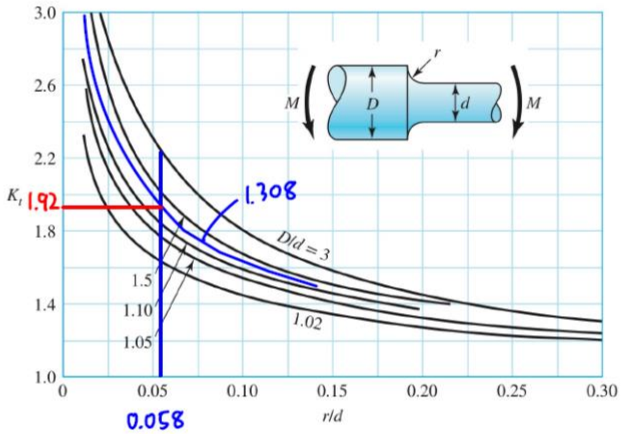
- Torque: 0 lb-ft
- Shear force: -129.28 lb
- Bending moment: 35.62 lb-in
- D: 0.669 in
- t: 0.103 in

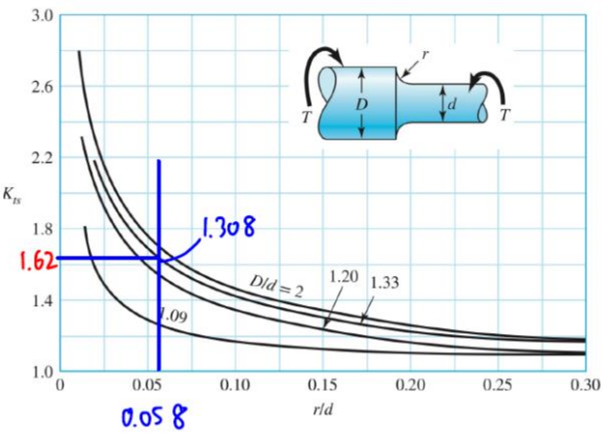
- q: 0.7
- q_s: 0.73
- r: 0.039 in
- Kt: 1.92
- Kts: 1.62











Also, the safety factor was calculated based on these inputs is 12.82.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.7(1.92 - 1) = 1.644$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.73(1.62 - 1) = 1.453$$

$$k_a = aS_{ut}^b = 2 \times 91^{-0.217} = 0.75$$

$$k_b = 0.879d^{-0.107} = 0.879(0.669)^{-0.107} = 0.918$$

$$Se' = 0.5S_{ut} = 0.5 \times 91000 = 45{,}500psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S'_e} = 0.75(0.918)(1)(1)(0.814)(45{,}500) = 25{,}540\,psi$$

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

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

$$= \frac{16}{\pi 0.669^3}\left[4\left(\frac{1.644 \times 35.62}{25{,}540}\right)^2\right]^{\frac{1}{2}} = 0.08$$

$$n = \frac{1}{0.08} = 12.82$$

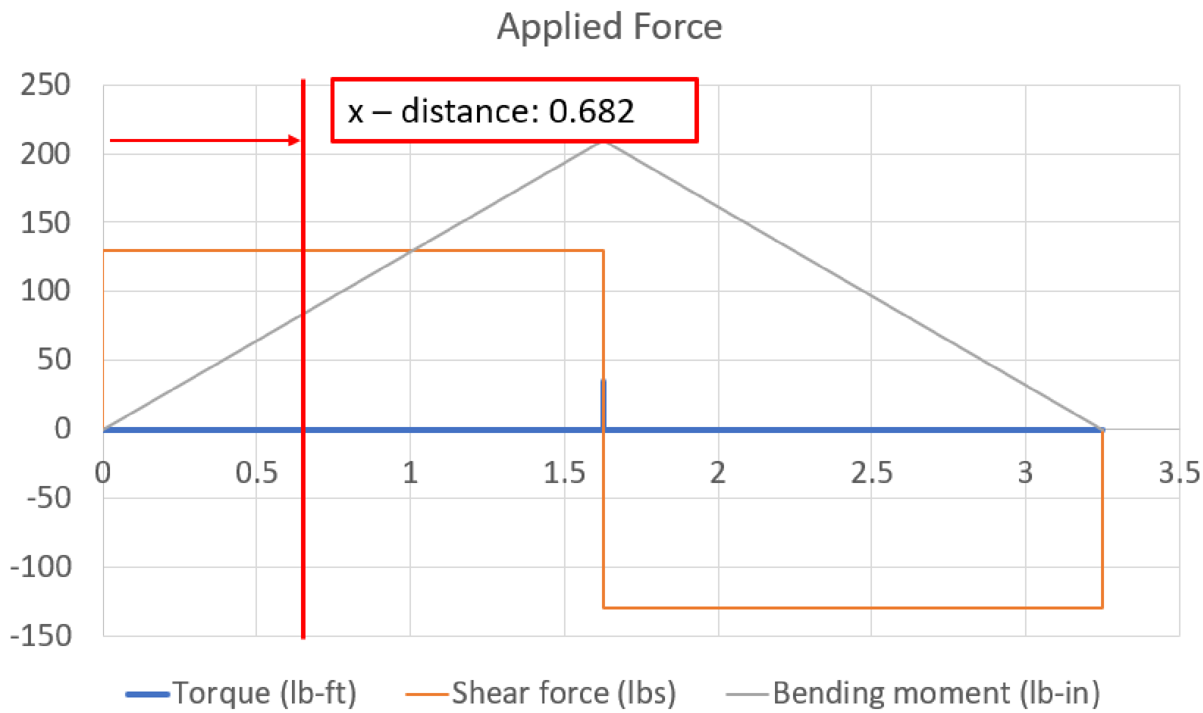
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	0.00	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	0.00	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert\frac{T_{max} - T_{min}}{2}\right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	35.62	lbf-in		$M_a = \left\lvert\frac{M_{max} - M_{min}}{2}\right\rvert$
S_{ut}	Ultimate Tensile Strength	91000	psi	Shaft material: AISI 1045 CD	
S_{yt}	Yield Tensile Strength	77000	psi	Shaft material: AISI 1045 CD	
q	Notch sensitivity	0.7			
qs	Notch sensitivity	0.73			
r	Notch radius	0.039	in		
K_t	Stress-Concentration factor	1.92			
K_{ts}	Stress-Concentration factor	1.62			
d_{input}	Shaft diameter (input)	0.669	in		
K_f	Fatigue Stess-Concentration Factor	1.644			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	1.4526			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.75			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.918			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	45500	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	25540	psi		$S_e = k_a k_b k_c k_d k_e S_e'$
1/n	Utilization Factor	0.08			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	12.82			

4.2.2.4 H' - Groove

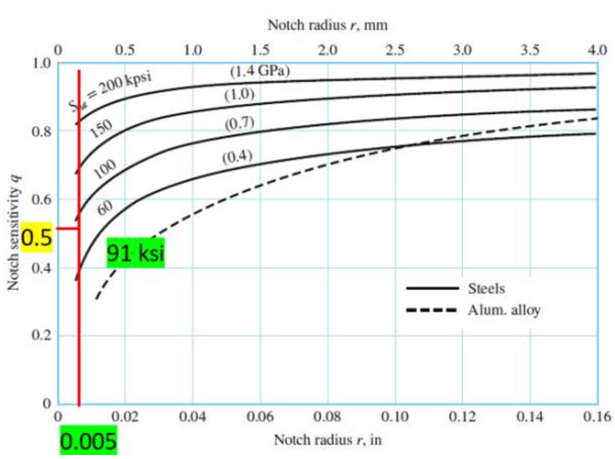
The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

- Torque: 0 lb-ft
- Shear force: 129.28 lb
- Bending moment: 88.17 lb-in
- D: 0.875 in
- t: 0.027 in
- a: 0.086 in
- q: 0.5
- q_s: 0.55
- r: 0.005 in

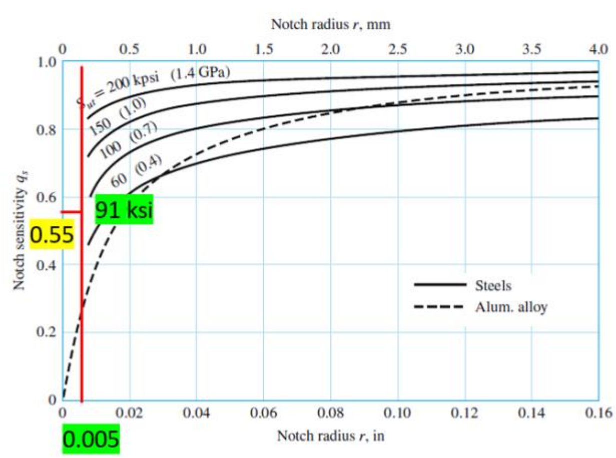
- Kt: 4.1
- Kts: 2.5



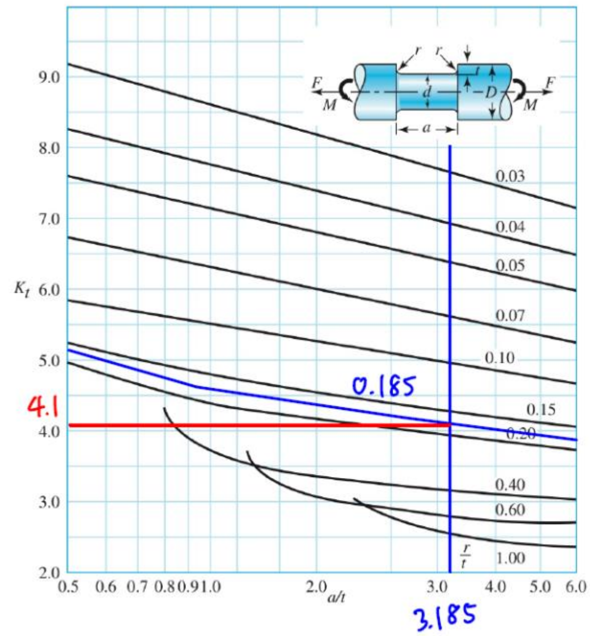
Applied Force
x – distance: 0.682
250
200
150
100
50
0
-50
-100
-150
0
0.5
1
1.5
2
2.5
3
3.5
Torque (lb-ft)
Shear force (lbs)
Bending moment (lb-in)

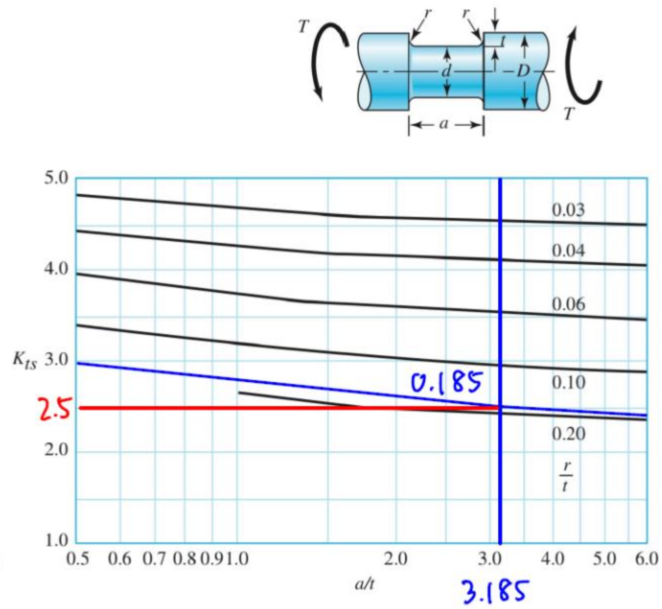


Notch radius r, mm
Notch radius r, in
Notch sensitivity q
S_{ut} = 200 kpsi
(1.4 GPa)
(1.0)
(0.7)
(0.4)
150
100
60
0.5
91 ksi
0.005
Steels
Alum. alloy



Notch radius r, mm
Notch radius r, in
Notch sensitivity q_s
S_{ut} = 200 kpsi
(1.4 GPa)
(1.0)
(0.7)
(0.4)
150
100
60
0.55
91 ksi
0.005
Steels
Alum. alloy





Also, the safety factor calculated based on these inputs is 7.26.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.5(4.1 - 1) = 2.55$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.55(2.5 - 1) = 1.825$$

$$k_a = aS_{ut}^b = 2 \times 91^{-0.217} = 0.75$$

$$k_b = 0.879d^{-0.107} = 0.879(0.875)^{-0.107} = 0.89$$

$$Se' = 0.5S_{ut} = 0.5 \times 91000 = 45{,}500 psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S_e'} = 0.75(0.89)(1)(1)(0.814)(45{,}500) = 24{,}817\ psi$$

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

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

$$= \frac{16}{\pi 0.875^3}\left[4\left(\frac{2.55 \times 88.17}{24{,}817}\right)^2\right]^{\frac{1}{2}} = 0.14$$

$$n = \frac{1}{0.14} = 7.26$$

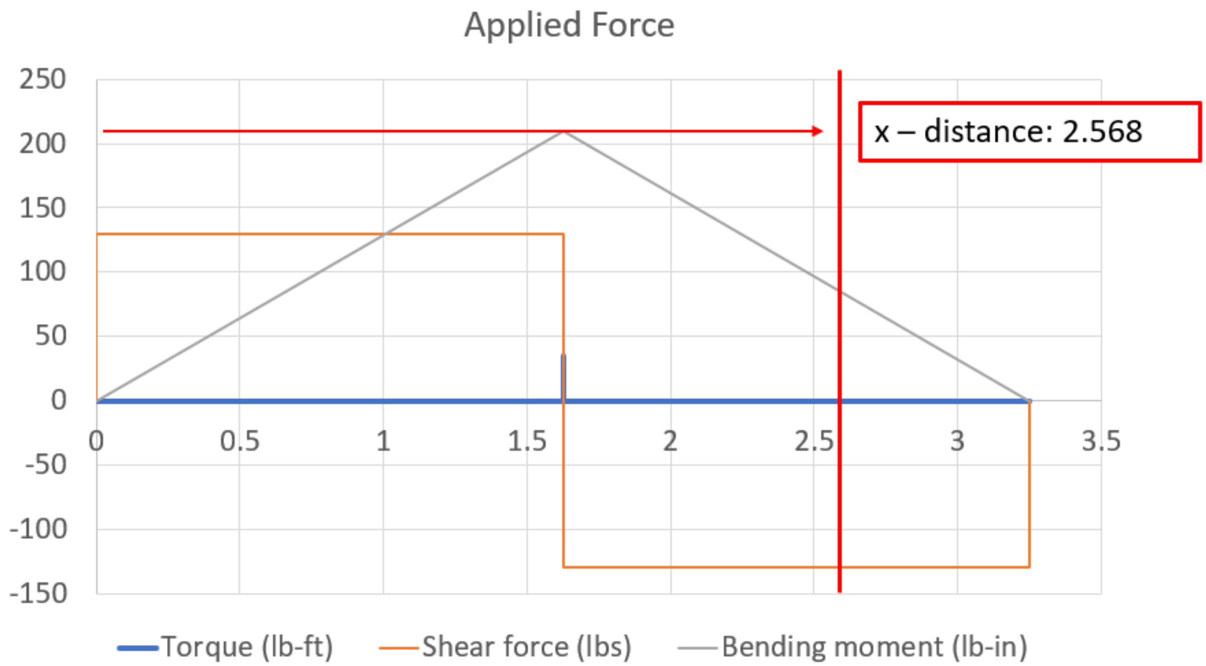
Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	0.00	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	0.00	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert \frac{T_{max} - T_{min}}{2} \right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	88.17	lbf-in		$M_a = \left\lvert \frac{M_{max} - M_{min}}{2} \right\rvert$
S_{ut}	Ultimate Tensile Strength	91000	psi	Shaft material: AISI 1045 CD	
S_{yt}	Yield Tensile Strength	77000	psi	Shaft material: AISI 1045 CD	
q	Notch sensitivity	0.5			
qs	Notch sensitivity	0.55			
r	Notch radius	0.005	in		
K_t	Stress-Concentration factor	4.1			
K_{ts}	Stress-Concentration factor	2.5			
d_{input}	Shaft diameter (input)	0.875	in		
K_f	Fatigue Stess-Concentration Factor	2.55			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	1.825			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.75			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.892			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	45500	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	24817	psi		$S_e = \mathrm{k_a k_b k_c k_d k_e S_e'}$
1/n	Utilization Factor	0.14			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	7.26			

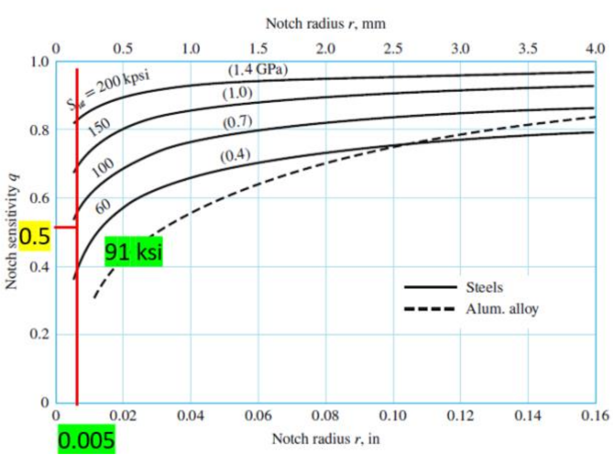
4.2.2.5 I' - Groove

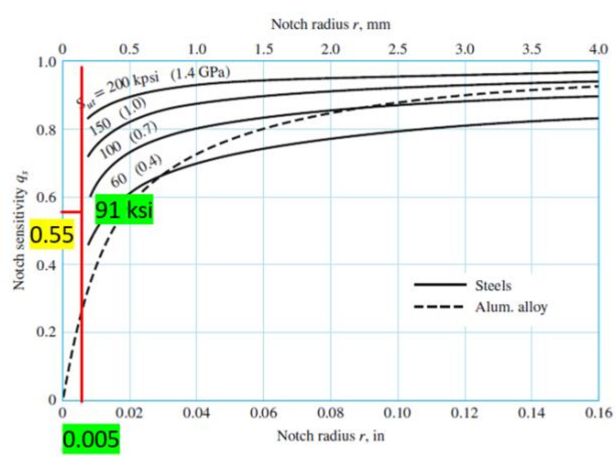
The forces, dimensions, notch sensitivity and stress concentration factors used for the calculation are as below.

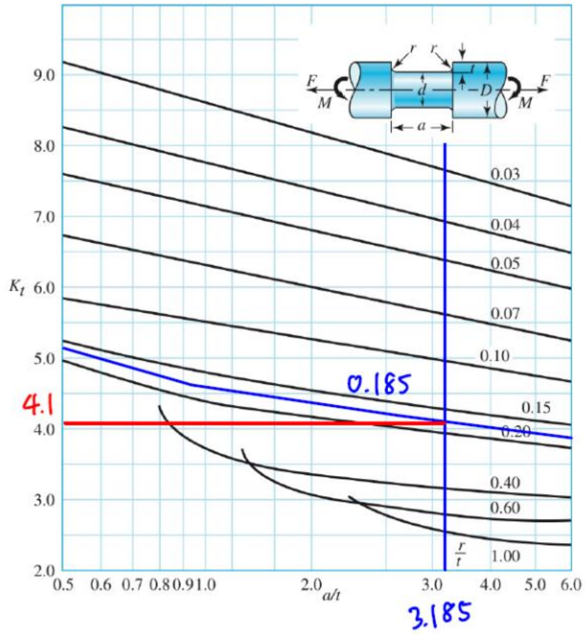
- Torque: 0 lb-ft
- Shear force: -129.28 lb
- Bending moment: 88.17 lb-in
- D: 0.875 in
- t: 0.027 in

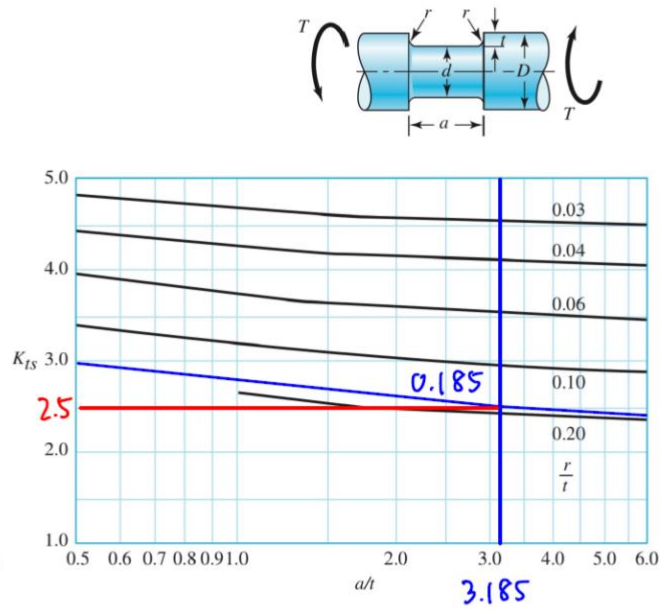
- a: 0.086 in
- q: 0.5
- q_s: 0.55
- r: 0.005 in
- Kt: 4.1
- Kts: 2.5











Also, the safety factor calculated based on these inputs is 7.26.

Calculation

$$K_f = 1 + q(K_t - 1) = 1 + 0.5(4.1 - 1) = 2.55$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.55(2.5 - 1) = 1.825$$

$$k_a = aS_{ut}^b = 2 \times 91^{-0.217} = 0.75$$

$$k_b = 0.879d^{-0.107} = 0.879(0.875)^{-0.107} = 0.89$$

$$Se' = 0.5S_{ut} = 0.5 \times 91000 = 45{,}500psi$$

$$S_e = \mathrm{k_a k_b k_c k_d k_e S_e'} = 0.75(0.89)(1)(1)(0.814)(45{,}500) = 24{,}817\ psi$$

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

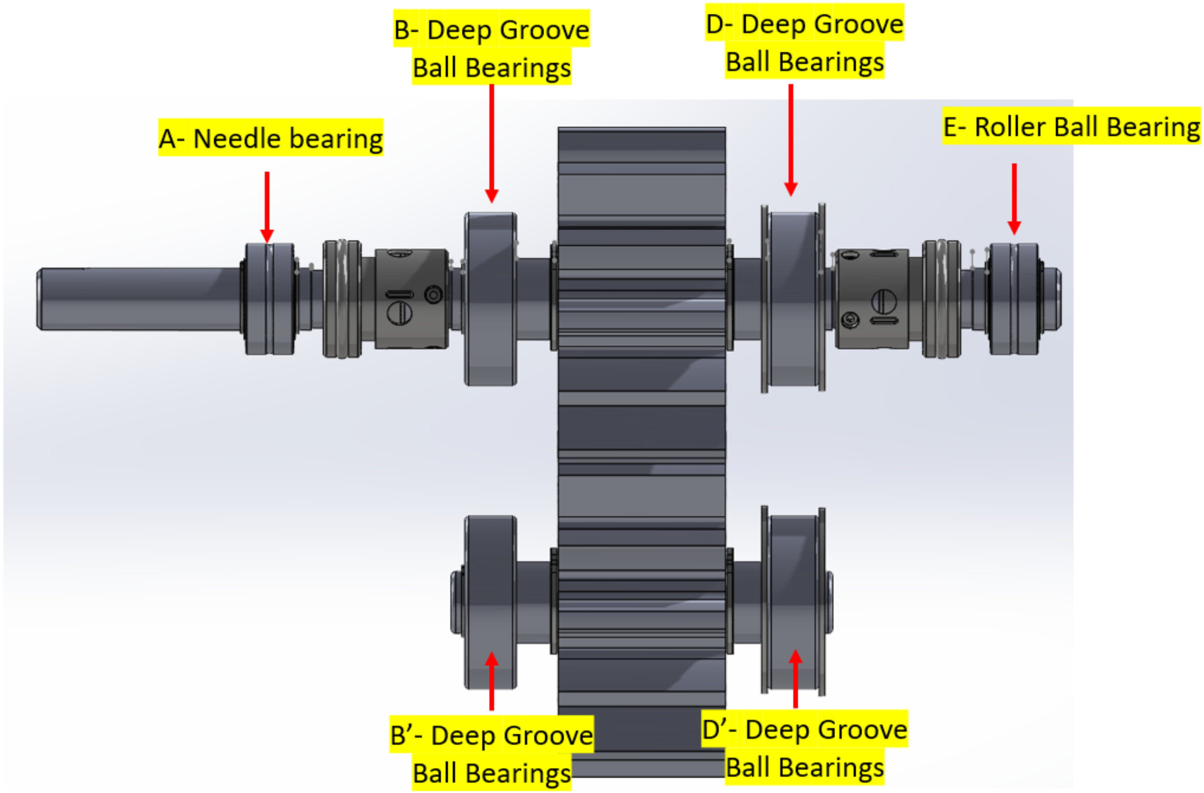
$$= \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\cancel{\frac{K_{fs}T_a}{S_e}}\right)^2 + 4\left(\cancel{\frac{K_f M_m}{S_y}}\right)^2 + 3\left(\cancel{\frac{K_{fs}T_m}{S_y}}\right)^2\right]^{\left(\frac{1}{2}\right)}$$

$$= \frac{16}{\pi 0.875^3}\left[4\left(\frac{2.55 \times 88.17}{24{,}817}\right)^2\right]^{\frac{1}{2}} = 0.14$$

$$n = \frac{1}{0.14} = 7.26$$

Symbol	Description	Value	Unit	Assumption	Equation
T_m	Mean Torque (lbf-ft)	0.00	lbf-ft		$T_m = \frac{T_{max} + T_{min}}{2}$
T_m	Mean Torque (lbf-in)	0.00	lbf-in		
T_a	Alternating Torque	0	lbf-ft		$T_a = \left\lvert\frac{T_{max} - T_{min}}{2}\right\rvert$
M_m	Mean Bending Moment	0.0	lbf-in		$M_m = \frac{M_{max} + M_{min}}{2}$
M_a	Alternating Bending Moment	88.17	lbf-in		$M_a = \left\lvert\frac{M_{max} - M_{min}}{2}\right\rvert$
S_{ut}	Ultimate Tensile Strength	91000	psi	Shaft material: AISI 1045 CD	
S_{yt}	Yield Tensile Strength	77000	psi	Shaft material: AISI 1045 CD	
q	Notch sensitivity	0.5			
qs	Notch sensitivity	0.55			
r	Notch radius	0.005	in		
K_t	Stress-Concentration factor	4.1			
K_{ts}	Stress-Concentration factor	2.5			
d_{input}	Shaft diameter (input)	0.875	in		
K_f	Fatigue Stess-Concentration Factor	2.55			$K_f = 1 + q(K_t - 1)$
K_{fs}	Fatigue Stess-Concentration Factor	1.825			$K_{fs} = 1 + q_s(K_{ts} - 1)$
k_a	Surface factor	0.75			$k_a = aS_{ut}^b$
a	Factor a	2		Cold-drwan	
b	Exponent b	-0.217		Cold-drwan	
k_b	Size factor	0.892			$k_b = 0.879d^{-0.107}$
k_c	Load factor	1		bending	
k_d	Temperature factor	1		The gear operates in a temperature range of 60°C to 80°C.	
k_e	Reliability factor	0.814		99% Reliability	
S'_e	Endurance limit	45500	psi	Sut < 200 ksi	$Se' = 0.5S_{ut}$
S_e	Endurance limit at the critical location	24817	psi		$S_e = \mathrm{k_a k_b k_c k_d k_e S'_e}$
1/n	Utilization Factor	0.14			$\frac{1}{n} = \frac{16}{\pi d^3}\left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2\right]^{\left(\frac{1}{2}\right)}$
n_{output}	Safety Factor (Output)	7.26			

4.3 Bearing



4.3.1 Bearing in Location B and D

The bearings at locations B and D need to withstand higher radial forces than those at locations A and E. Therefore, deep groove ball bearings were chosen. Given a required catalog load rating of 9.04 kN and a shaft diameter of 0.875 inches, 6303-2Z bearings were selected, which have a catalog load rating of 11.7 kN. Based on this load rating, the bearing's calculated rating life is 54,157 hours.

Calculation – Required Catalog load rating

$$L_D = \mathrm{L}\,(\mathrm{D}) \times n \times 60 = 25000 \times 1500 \times 60 = 2.25 \times 10^9$$

$$F_e = X_i V F_r + Y_i F_a = (1)(1)(0.58kN) + (0)(0) = 0.58\ kN$$

$$C_{10} = a_f F_e \left(\frac{L_D}{L_R}\right)^{\frac{1}{3}} = 1.2(0.58kN)\left[\frac{2.25 \times 10^9}{10^9}\right]^{\frac{1}{3}} = 9.04kN = 2033\ lb$$

Symbol	Description	Value	Unit	Assumption	Equation
a_f	Application factor	1.2		Applications with poor bearing seals	
F_r	Radial load (lb)	129.3	lb		
F_r	Radial load (kN)	0.58	kN		
F_a	Axial thrust	0	kN		
$\mathcal{L}$ (D)	rating life in hours	25000	hrs		
n	Gear Speed	1500	rpm		
L_D	rating life in rev	2.25E+09	rev		$L_D = \mathcal{L}\text{ (D)} \times n \times 60$
L_R	Deired life in rev	1000000	rev		
X_i	Ordinate intercept	1			
V	Rotation factor	1		inner ring torates	
Y_i	Slope	0			
Fe	Equivalent radial load	0.58	kN		$F_e = X_i V F_r + Y_i F_a$
C_{10}	Catalog load rating	9.04	kN		$C_{10} = a_f F_e \left(\frac{L_D}{L_R}\right)^{\frac{1}{3}}$
C_{10}	Catalog load rating	2032.80	lbs		

Calculation – Rating life

$$L_D = (C_{10}/a_f F_e)^3 \times L_R = \left(\frac{11.7}{1.2(0.58)}\right)^3 \times 10^9 = 4.87 \times 10^9\ rev$$

$$\text{L (D)} = \frac{L_D}{(n \times 60)} = \frac{4.87 \times 10^9}{1500 \times 60} = 54{,}157\ hr$$

Symbol	Description	Value	Unit	Assumption	Equation
C_{10}	Catalog load rating	11.7	kN		
L_D	rating life in rev	4.87E+09	rev		$L_D = (C_{10}/a_f F_e)^3 \times L_R$
$\mathcal{L}$ (D)	rating life in hours	54157.0	hrs		$\mathcal{L}\text{ (D)} = \frac{L_D}{(n \times 60)}$

4.3.2 Bearing in Location B' and D'

Since both the primary and secondary shafts have the same diameter and the radial load on B' and C' is lower than on B and C, the same bearings are used for all. This standardization reduces the variety of components needed and simplifies maintenance.

4.3.2 Bearing in Location A and E

The bearings at locations A and E only need to withstand lower radial loads, so single-row needle bearings were chosen. With a required catalog load rating of 353 lbs and a shaft diameter of 0.591 inches, NA 4902.2RS bearings were selected, offering a catalog load rating of 2053 lb. Based on this rating, the calculated bearing life is 5,579,069 hours.

Calculation – Required Catalog load rating

$$L_D = \mathrm{L}\ (\mathrm{D}) \times n \times 60 = 25000 \times 1500 \times 60 = 2.25 \times 10^9$$

$$F_e = X_i V F_r + Y_i F_a = (1)(1)(0.1kN) + (0)(0) = 0.1\ kN$$

$$C_{10} = a_f F_e \left(\frac{L_D}{L_R}\right)^{\frac{1}{3}} = 1.2(0.1kN)\left[\frac{2.25 \times 10^9}{10^9}\right]^{\frac{1}{3}} = 1.57kN = 353\ lb$$

Symbol	Description	Value	Unit	Assumption	Equation
a_f	Application factor	1.2		Applications with poor bearing seals	
F_r	Radial load (lb)	22.44	lb		
F_r	Radial load (kN)	0.1	kN		
F_a	Axial thrust	0	kN		
$\mathcal{L}$ (D)	rating life in hours	25000	hrs		
n	Gear Speed	1500	rpm		
L_D	rating life in rev	2.25E+09	rev		$L_D = \mathcal{L}\ (\mathrm{D}) \times n \times 60$
L_R	Deired life in rev	1000000	rev		
X_i	Ordinate intercept	1			
V	Rotation factor	1		inner ring torates	
Y_i	Slope	0			
Fe	Equivalent radial load	0.10	kN		$F_e = X_i V F_r + Y_i F_a$
C_{10}	Catalog load rating	1.57	kN		$C_{10} = a_f F_e \left(\frac{L_D}{L_R}\right)^{\frac{1}{3}}$
C_{10}	Catalog load rating	352.84	lbs		

Calculation – Rating life

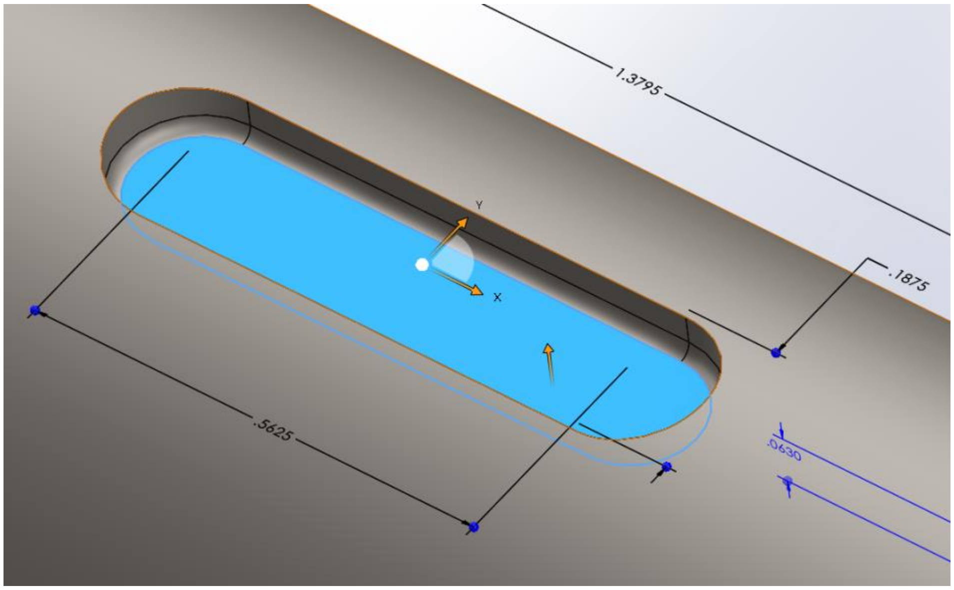
$$L_\mathrm{D} = (\mathrm{C}_{10}/\mathrm{a_f F_e})^3 \times L_R = \left(\frac{9.13}{1.2(0.1)}\right)^3 \times 10^9 = 4.43 \times 10^{11}\ rev$$

$$\mathrm{L}\ (\mathrm{D}) = \frac{L_D}{(n \times 60)} = \frac{4.43 \times 10^{11}}{1500 \times 60} = 4{,}921{,}111\ hr$$

Symbol	Description	Value	Unit	Assumption	Equation
C_{10}	Catalog load rating	9.13	kN		
L_D	rating life in rev	4.43E+11	rev		$L_\mathrm{D} = (\mathrm{C}_{10}/\mathrm{a_f F_e})^3 \times L_R$
$\mathcal{L}$ (D)	rating life in hours	4921111.3	hrs		$\mathcal{L}\ (\mathrm{D}) = \frac{L_D}{(n \times 60)}$

4.4 Key and Keyway

4.4.1 Key seat on the shaft



The shaft has a diameter of 0.875 inches, with a key seat measuring 1/8” x 3/16” x 3/4". Since the diameter of 0.875 inches, based on the course material 'ASEE-2013-6775', we used Kts value of 2.158 and 2.235. The shaft experiences a torsional shear stress of 8,944 psi, corresponding to an axial stress of 6,938 psi. Given that the primary shaft material is AISI 1030 Q & T, with a yield strength of 90,000 psi, the calculated safety factor is7.3, which is considered sufficiently high.

Shaft diameter	Key size (Width X Height)	Depth of key seat	Fillet radius	K_{TB} (Bending)	K_s (Torsion)	K_{TA} (Axial)
3/8”	3/32” X 3/32”	3/64”	0.0078”	2.202	2.232	2.830
½”	1/8” X 1/8”	1/16”	0.0104”	2.150	2.234	2.830
11/16”	3/16” X 3/16”	3/32”	0.0156”	2.158	2.235	2.847
1 1/16”	¼” X ¼”	1/8”	0.0208”	2.280	2.244	2.778
1 5/16”	5/16” X 5/16”	5/32”	0.0260”	2.267	2.228	2.797
1 9/16”	3/8” X 3/8”	3/16”	0.0312”	2.312	2.246	2.824

Calculation

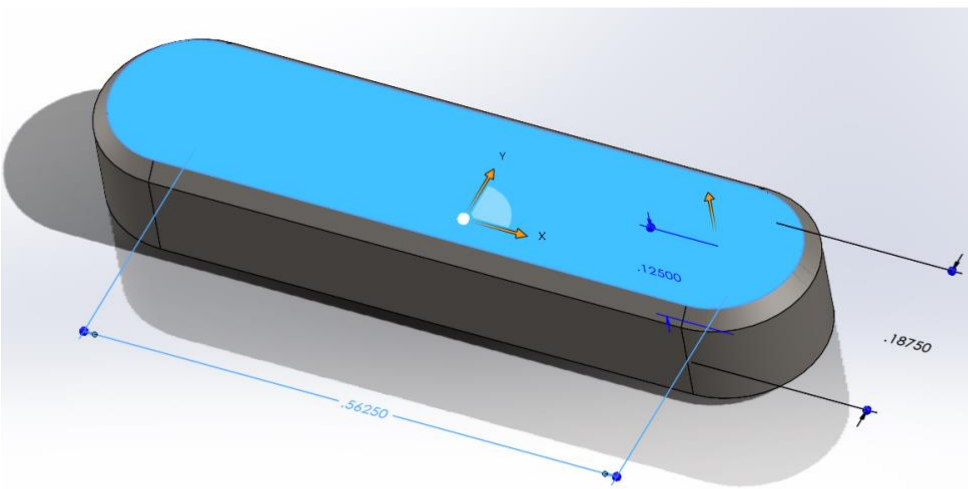
$$\tau = \frac{16TK_{ts}}{\pi d^3} = \frac{16(420.17)(2.235)}{\pi 0.875^3} = 7139.2\, psi$$

$$\sigma_{yt} = \frac{\tau}{0.577} = \frac{7139.2}{0.577} = 12372.9\, psi$$

$$n_{keyway} = \frac{S_{yt}}{\sigma_{yt}} = \frac{90000}{12372.9} = 7.3$$

Symbol	Description	Value	Unit	Assumption	Equation
d_{input}	Shaft diameter (input)	0.875	in		
H	Power	10	hp		
n	Gear Speed	1500	rpm		
T	Torque	420.17	lbf-in		
w	Key width	0.1875	in		
h	Key height	0.125	in		
K_{ts}	Static stress concentraion factor	2.235			
τ (shaft)	Shear Stress on shaft	7139.2	psi		$\tau = \frac{16TK_{ts}}{\pi d^3}$
σ_{yt}	Equivalent Yield stress	12372.9	psi		$\sigma_{yt} = \frac{\tau}{0.577}$
n_{keyway}	Safety factor in profile keyway	7.3			$n_{keyway} = \frac{S_{yt}}{\sigma_{yt}}$

4.4.2 Key



The key length should range between 0.625 inches and 1.3125 inches, equivalent to 70% to 150% of the shaft diameter. The selected key size is 1/8" x 3/16" x 3/4". The force acting on the key depends on the torque and the shaft radius. To evaluate its safety, both bearing and shear stresses are considered, with shear stress converted to axial stress for comparison with material properties. Since the key is designed to be the first component to fail, its safety factor is intentionally lower than that of other components. Accordingly, the relatively lower-strength material AISI 1010 CD, with a yield strength of 44,000 psi, was chosen. The safety factor for shear is 3.52, while for bearing, it is 2.15. The failure mode is governed by bearing stress, as its safety factor is lower than that of the shaft.

Calculation

$$A_s = 0.1875 \times (0.75 - 0.1875) + \pi\left(\frac{0.1875}{2}\right)^2 = 0.1331\ in^2$$

$$A_b = 0.5 \times h \times L_{key} = 0.5 \times 0.125 \times 0.75 = 0.0469 in^2$$

$$F_{key} = \frac{T}{\frac{d}{2}} = \frac{420.17}{\frac{0.875}{2}} = 960.4\ lb$$

$$\tau(key) = \frac{F_{key}}{A_s} = \frac{960.4}{0.1331} = 7216.6\ psi$$

$$\sigma_{yt} = \frac{\tau(key)}{0.577} = \frac{10624}{0.577} = 12507.1\ psi$$

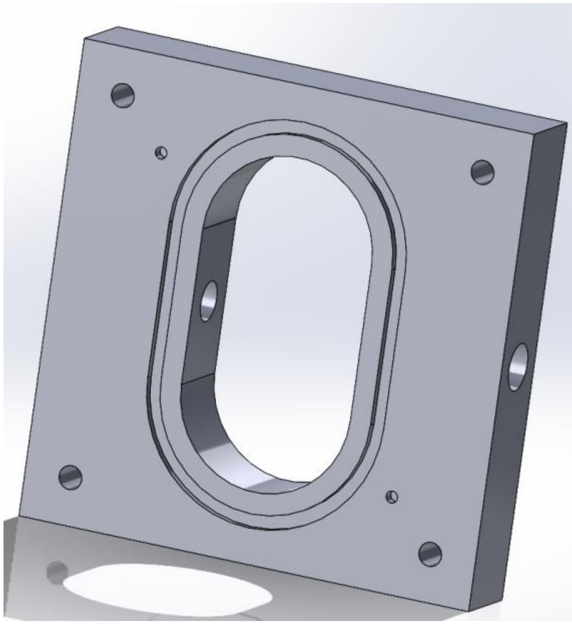
$$\sigma_b = \frac{F_{key}}{A_b} = \frac{960.4}{0.0469} = 20488\ psi$$

$$n_{shear} = \frac{S_{yt}(key)}{\sigma_{yt}} = \frac{44000}{12507.1} = 3.52$$

$$n_{bearning} = \frac{S_{yt}(key)}{\sigma_b} = \frac{44000}{20488} = 2.15$$

Symbol	Description	Value	Unit	Assumption	Equation
S_{yt} (key)	Key Yield Strength	44000	psi	AISI 1015 HR → AISI 1010 CD	
L_{key}	Key length	0.75	in	Length 70-150% of shaft dia	
A_s	Shear area	0.133080404	in^2		$A_s = w \times L_{key}$
A_b	bearing area	0.046875	in^2		$A_b = 0.5 \times h \times L_{key}$
F_{key}	Force acts on Key	960.4	lb		$F_{key} = \frac{T}{d/2}$
τ (key)	Shear stress on key	7216.6	psi		$\tau(key) = \frac{F_{key}}{A_s}$
σ_{yt}	Equivalent Yield stress	12507.1	psi		$\sigma_{yt} = \tau(key)/0.577$
σ_b	Bearing stress	20488.2	psi		$\sigma_b = F_{key}/A_b$
n_{shear}	Safety factor for shear	3.52			$n_{shear} = \frac{S_{yt}(key)}{\sigma_{yt}}$
$n_{bearing}$	Safety factor for bearning	2.15			$n_{bearning} = \frac{S_{yt}(key)}{\sigma_b}$

4.5 Gear Housing



The gear housing must be properly sealed to prevent external dirt from contaminating the gears. It features inlet and outlet ports to enable lubricant flow. When designing the inlet and outlet diameters, as well as the minimum wall thickness, it is essential to account for pressure changes caused by flow velocity. With a pressure drop of 4,398 psi, the flow velocity increases from 161 in/sec to 225 in/sec. To achieve this velocity increase, the outlet diameter is reduced to 0.847 inches, compared to the inlet diameter of 1 inch. The housing thickness, defined as the distance between the outer surface of the groove and any edge or hole, is 0.206 inches.

Calculation

$$D_P = \frac{\pi[bD^2][9N_P - 2.35]}{8N_P^2} = \frac{\pi[1.8(3.25^2)][9(13) - 2.35]}{8 \times 13^2} = 5.065\ in^3/rev$$

$$Tsmax = \frac{\pi d^3 \sigma_{ty}}{2 \times 16} \times \eta = \frac{\pi 0.875^3 \times 77000}{2 \times 16} \times 0.7 = 3545\ psi$$

$$P_1 - P_2 = \frac{Tsmax \times 2\pi}{D_P} = \frac{3545 \times 2\pi}{5.065} = 4398\ psi$$

$$t = \frac{(P_1 - P_2) \times r}{\sigma_y} \times SF = \frac{4398 \times 1.875}{40000} \times 1 = 0.206\ in$$

$$Q = D_P \times \frac{n}{60} = 5.065 \times \frac{1500}{60} = 126.6 in^3/sec$$

$$A_{inlet} = \frac{1}{4} \times \pi \times d_{inlet}^2 = \frac{1}{4} \times \pi \times 1^2 = 0.785\ in^2$$

$$V_{inlet} = \frac{Q}{A_{inlet}} = \frac{126.6}{0.785} = 161\ in/sec$$

$$V_{outlet}^2 - V_{inlet}^2 = (P_1 - P_2) \times \frac{2}{\rho} = 4398 \times \frac{2}{0.36} = 24431\ in^2/sec^2$$

$$V_{outlet} = \sqrt{V_{inlet}^2 + \frac{2 \times (P_1 - P_2)}{\rho}} = \sqrt{V_{inlet}^2 + V_{outlet}^2 - V_{inlet}^2} = \sqrt{161^2 + 24431}$$

$$= 225\ in/sec$$

$$A_2 = \frac{A_1 V_1}{V_2} = \frac{Q}{V_2} = \frac{126.6}{225} = 0.564\ in^2$$

$$D_2 = \sqrt{\frac{4A_{outlet}}{\pi}} = \sqrt{\frac{4 \times 0.564}{\pi}} = 0.847\ in$$

Symbol	Description	Value	Unit	Assumption	Equation
D_p	Displacement of the pump	5.065	in^3/rev		$D_P = \frac{\pi[bD^2][9N_P - 2.35]}{8N_P^2}$
σ_{ty}	Yield strength of shaft material	77000	psi	AISI 1045 CD	
η	Efficiency	0.7			
D_{shaft}	Shaft Diameter	0.875	in		
Tsmax	Torque required	3545	lbf-in		$Tsmax = \frac{\pi d^3 \sigma_{ty}}{2 \times 16} \times \eta$
P_1-P_2	Differential pressure	4398	psi		$P_1 - P_2 = \frac{Tsmax \times 2\pi}{D_P}$
r	Inside radius of the housing	1.875			
σ_y	Yiled strength of housing material	40000	psi	ASTM A48 Cast Iron	
SF	Service factor	1			
t	Thickness of the housing	0.206	in		$t = \frac{(P_1 - P_2) \times r}{\sigma_y} \times SF$
Q	Volume flow rate	126.62705	in^3/sec		$Q = D_P \times \frac{n}{60}$
d_{inlet}	Inlet hole diameter	1	in		
A_{inlet}	Inlet Area	0.785	in^2		$A_{inlet} = \frac{1}{4} \times \pi \times d_{inlet}^2$
V_{inlet}	Inlet velocity	161.227	in/sec		$V_{inlet} = \frac{Q}{A_{inlet}}$
p	Density	0.36	lb/in^3		
V_{outlet}^2-V_{inlet}^2		24431	In^2/sec^2		$V_{outlet}^2 - V_{inlet}^2 = (P_1 - P_2) \times 2/\rho$
V_{outlet}	Outlet velocity	225	in/sec		$V_{outlet} = \sqrt{V_{inlet}^2 + \frac{2 \times (P_1 - P_2)}{\rho}}$
A_{outlet}	Outlet Area	0.564	in^2		$A_2 = A_1 V_1 / V_2$
d_{outlet}	Outlet hole diameter	0.847	in		$D_2 = \sqrt{\frac{4A_{outlet}}{\pi}}$

4.6 Tie Rod

A ½”-13, 18-8 stainless steel tie rod was used to join the entire assembly. The mating nut used was a Grade 2H, ½”-13.

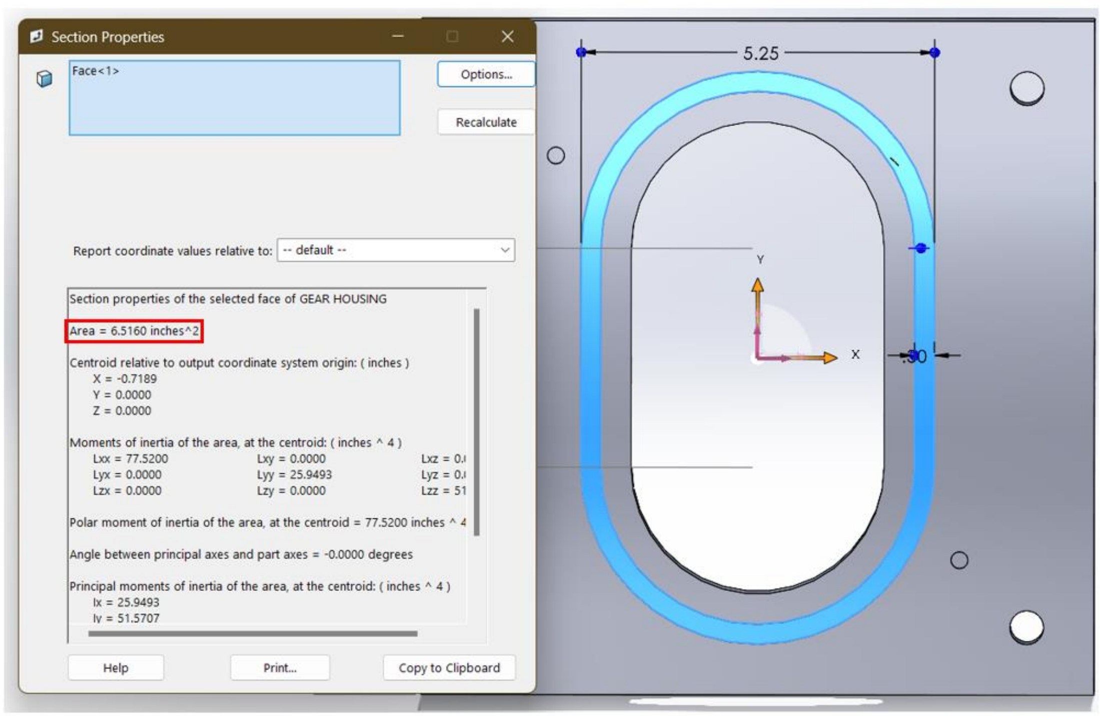


Table 8–5 Coefficients of Friction *f* for Threaded Pairs

	Nut Material			
Screw Material	**Steel**	**Bronze**	**Brass**	**Cast Iron**
Steel, dry	0.15–0.25	0.15–0.23	0.15–0.19	0.15–0.25
Steel, machine oil	0.11–0.17	0.10–0.16	0.10–0.15	0.11–0.17
Bronze	0.08–0.12	0.04–0.06	—	0.06–0.09

The bearing plate applies a pressure of 4398 psi, with a contact area of 6.516 in² against the O-ring. According to the course material, *‘TIE ROD TORQUE REQUIREMENTS’*, the required torque is 11.64 lb-ft. However, based on the *Shigley Textbook*, the required torque is 20.35 lb-ft. When comparing these values to the *‘Tie*

Rod Nut and Bolts' table, discrepancies arise in areas such as thread type and pressure. Generally, the calculated torque is lower than the table value of 52 lb-ft.

Calculation 1

$$F = PA = 4398 \times 6.516 = 28657.37\ lbs$$

$$W = \frac{F}{n_{tie\ rod}} \times SF \times 15\ \% = \frac{28657.37}{4} \times 2 \times 0.15 = 2149.3\ lbs$$

$$d_m = OD - \frac{0.6495}{Threads\ per\ inch} = 0.5 - \frac{0.6495}{13} = 0.45\ in^2$$

$$L = \frac{1}{13} = 0.076923$$

$$tan\lambda = \frac{L}{\pi d_m} = \frac{0.076923}{\pi(0.45)} = 0.05441$$

$$\lambda = tan^{-1}\, 0.05441 = 3.1145°$$

$$tan\alpha_n = tan\alpha \times cos\lambda = tan30 \times cos3.1145 = 0.5765$$

$$\alpha_n = tan^{-1}\, 0.5765 = 29.96$$

$$T = \frac{Wd_m}{2}\left[\frac{f\pi d_m + Lcos\alpha_n}{\pi d_m cos\alpha_n - fL}\right] = \frac{(2149.3)(0.45)}{2}\left[\frac{0.2\pi(0.45) + (0.076923)cos29.96}{\pi(0.45)cos29.96 - 0.2(0.076923)}\right]$$
$$= 139.7lb - in = 11.64\ lb - ft$$

Calculation 2

Table 8–15 Torque Factors *K* for Use with Equation (8–27)

Bolt Condition	*K*
Nonplated, black finish	0.30
Zinc-plated	0.20
Lubricated	0.18
Cadmium-plated	0.16
With Bowman Anti-Seize	0.12
With Bowman-Grip nuts	0.09

Table 8–2 **Diameters and Area of Unified Screw Threads UNC and UNF***

		Coarse Series—UNC			Fine Series—UNF		
Size Designation	Nominal Major Diameter in	Threads per Inch N	Tensile-Stress Area A_t in^2	Minor-Diameter Area A_r in^2	Threads per Inch N	Tensile-Stress Area A_t in^2	Minor-Diameter Area A_r in^2
0	0.0600				80	0.001 80	0.001 51
1	0.0730	64	0.002 63	0.002 18	72	0.002 78	0.002 37
2	0.0860	56	0.003 70	0.003 10	64	0.003 94	0.003 39
3	0.0990	48	0.004 87	0.004 06	56	0.005 23	0.004 51
4	0.1120	40	0.006 04	0.004 96	48	0.006 61	0.005 66
5	0.1250	40	0.007 96	0.006 72	44	0.008 80	0.007 16
6	0.1380	32	0.009 09	0.007 45	40	0.010 15	0.008 74
8	0.1640	32	0.014 0	0.011 96	36	0.014 74	0.012 85
10	0.1900	24	0.017 5	0.014 50	32	0.020 0	0.017 5
12	0.2160	24	0.024 2	0.020 6	28	0.025 8	0.022 6
$\frac{1}{4}$	0.2500	20	0.031 8	0.026 9	28	0.036 4	0.032 6
$\frac{5}{16}$	0.3125	18	0.052 4	0.045 4	24	0.058 0	0.052 4
$\frac{3}{8}$	0.3750	16	0.077 5	0.067 8	24	0.087 8	0.080 9
$\frac{7}{16}$	0.4375	14	0.106 3	0.093 3	20	0.118 7	0.109 0
$\frac{1}{2}$	0.5000	13	0.141 9	0.125 7	20	0.159 9	0.148 6
$\frac{9}{16}$	0.5625	12	0.182	0.162	18	0.203	0.189
$\frac{5}{8}$	0.6250	11	0.226	0.202	18	0.256	0.240
$\frac{3}{4}$	0.7500	10	0.334	0.302	16	0.373	0.351
$\frac{7}{8}$	0.8750	9	0.462	0.419	14	0.509	0.480
1	1.0000	8	0.606	0.551	12	0.663	0.625
$1\frac{1}{4}$	1.2500	7	0.969	0.890	12	1.073	1.024
$1\frac{1}{2}$	1.5000	6	1.405	1.294	12	1.581	1.521

$$F_P = A_t S_p = A_t 0.85\left(S_y\right) = 0.1419 \times 0.85(30000) = 3618.45\ lb$$

$$F_i = 0.75 \times F_P = 0.75 \times 3618.45 = 2713.84\ lb$$

$$T = KF_i d = 0.18(2713.84)(0.5) = 244.25\ lb - in = 20.35\ lb - ft$$

Appendices

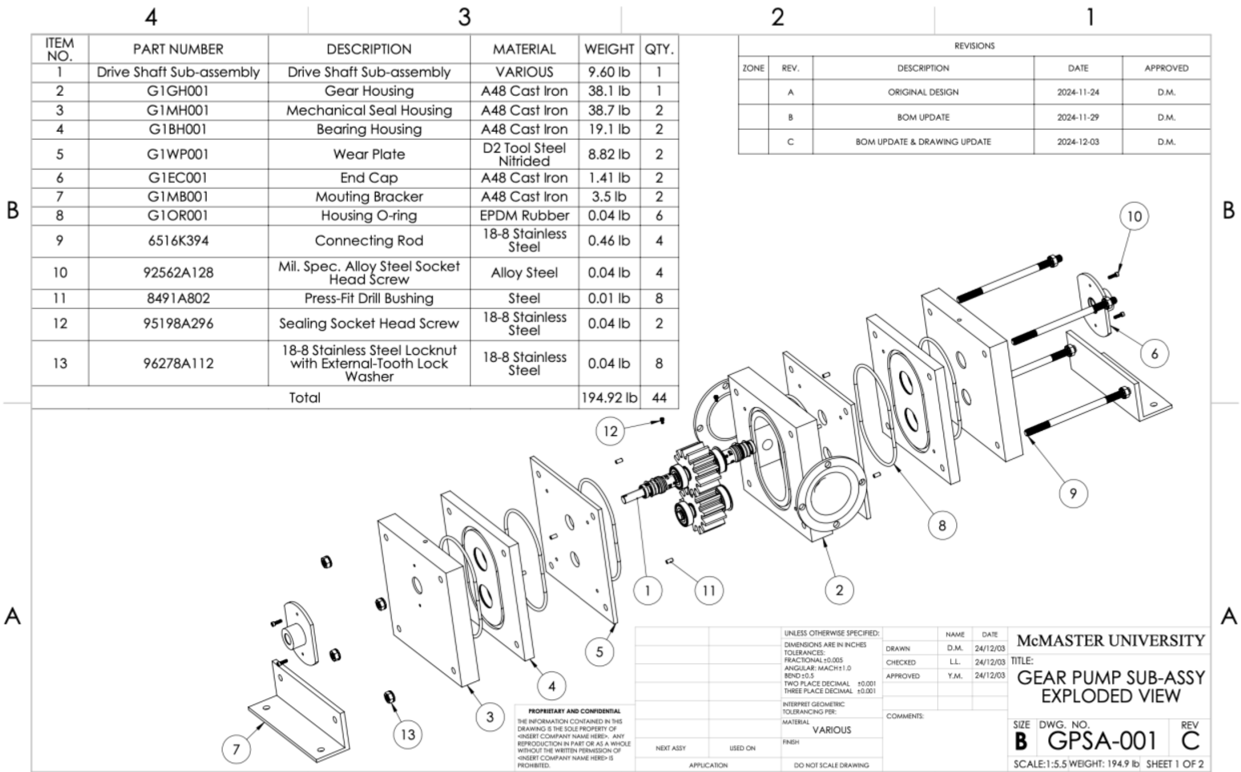
Glossary of Files

#	Items	3D	2D	Description
1	GEAR PUMP ASSEMBLY	GEAR PUMP ASSEMBLY.SLDASM	GEAR PUMP ASSEMBLY DRAWING.SLDDRW	Whole Gear Pump Assembly 3D Whole Exploded Assembly Drawing, BOM Whole Sectioned Assembly Drawing
2	Primary Shaft Sub Assembly	Primary Shaft Sub Assembly.SLDASM	Primary Shaft Sub Assembly.SLDDRW	Primay Shaft Assembly 3D Primary Shaft Exploded View, BOM Primary Shaft Broken-Out Section Details
3	Secondary Shaft Sub Assembly	Secondary Shaft Sub Assembly.SLDASM	Secondary Shaft Sub Assembly.SLDDRW	Secondary Shaft Assembly 3D Secondary Shaft Exploded View, BOM Secondary Shaft Broken-Out Section Details
4	Primary Shaft	DRIVE SHAFT.SLDPRT	DRIVE SHAFT.SLDDRW	Primay Shaft 3D & Drawing
5	Secondary Shaft	DRIVEN SHAFT.SLDPRT	DRIVEN SHAFT.SLDDRW	Secondary Shaft 3D & Drawing
6	Gear	GEAR.SLDPRT	GEAR.SLDPRW	Gear 3D & Drawing
7	Key	KEY.SLDPRT	KEY.SLDPRT	Key 3D & Drawing
8	Gear Housing	GEAR HOUSING.SLDPRT	GEAR HOUSING.SLDPRW	Gear Housing 3D & Drawing
9	Mechanical Seal Housing	MECHANICAL SEAL PLATE.SLDPRT	MECHANICAL SEAL PLATE.SLDPRW	Mechanical Seal Plate 3D
10	Wear Plate	WEAR PLATE.SLDPRT	WEAR PLATE.SLDPRW	Wear Plate 3D & Drawing
11	Bearing Housing	BEARING PLATE.SLDPRT	BEARING PLATE.SLDPRW	Bearing Housing 3D & Drawing
12	End Cap	END CAP.SLDPRT	END CAP.SLDPRW	End Cap 3D & Drawing
13	Mounting Bracket	MOUNTING BRACKET.SLDPRT	MOUNTING BRACKET.SLDPRW	Mounting Bracket 3D & Drawing
14	O-ring	O-RING.SLDPRT	O-RING.SLDPRW	O-ring 3D & Drawing
15	Mechanical Seal	1563N11_Mechanical Seal.SLDPRT	-	Mechanical Seal
16	Mechanical Seat	1563N11_Mechanical Seat.SLDPRT	-	Mechanical Seat 3D
17	Deep Groove Ball Bearing	skf_bearing_w_6303-2z_2.SLDASM	-	Deep Groove Ball Bearing 3D
18	Needle Bearing	skf_bearing_na_4902_2rs_2.SLDASM	-	Needle Bearing 3D
19	Retaining Ring	97633A230_External Retaining Ring.SLDPRT	-	Retaining Ring 3D
20	Retaining Ring	98585A118_Heavy Duty External Retaining Ring.SLDPRT	-	Retaining Ring 3D
21	Retaining Ring	99142A480_Internal Retaining Ring.SLDPRT	-	Retaining Ring 3D
22	Retaining Ring	99142A575_Internal Retaining Ring.SLDPRT	-	Retaining Ring 3D
23	Tie rod	6516K394_Connecting Rod.SLDPRT	-	Tie rod 3D
24	Nuts	96278A112_18-8 Stainless Steel Locknut with External-Tooth Lock Washer.SLDPRT	-	Nuts 3D
25	Screw	92562A434_Mil. Spec. Alloy Steel Socket Head Screw.SLDPRT	-	Screw 3D
26	Oil Seal	SKF_15X26X7 HMSA10 RG.sldprt		Oil Seal 3D
27	Dowel Pin	8491A802_Press-Fit Drill Bushing.SLDPRT	-	Dowel Pin 3D
28	Screw	95198A296_Sealing Socket Head Screw.SLDPRT		Screw for Mechanical Seal Adjustment 3D

Part Numbering System

Part Name	Part Number (Drawing No)	Group #1	Code	Number
Primary Shaft	G1PS001	G1	PS	001
Secondary Shaft	G1SS001	G1	SS	001
Gear	G1GE001	G1	GE	001
Key	G1KE001	G1	KE	001
Gear Housing	G1GH001	G1	GH	001
Wear Plate	G1WP001	G1	WP	001
Bearing Housing	G1BH001	G1	BH	001
End Cap	G1EC001	G1	EC	001
Mounting Bracket	G1MB001	G1	MB	001
O-ring	G1OR001	G1	OR	001

Drawing



ITEM NO.	PART NUMBER	DESCRIPTION	MATERIAL	WEIGHT	QTY.
1	Drive Shaft Sub-assembly	Drive Shaft Sub-assembly	VARIOUS	9.60 lb	1
2	G1GH001	Gear Housing	A48 Cast Iron	38.1 lb	1
3	G1MH001	Mechanical Seal Housing	A48 Cast Iron	38.7 lb	2
4	G1BH001	Bearing Housing	A48 Cast Iron	19.1 lb	2
5	G1WP001	Wear Plate	D2 Tool Steel Nitrided	8.82 lb	2
6	G1EC001	End Cap	A48 Cast Iron	1.41 lb	2
7	G1MB001	Mouting Bracker	A48 Cast Iron	3.5 lb	2
8	G1OR001	Housing O-ring	EPDM Rubber	0.04 lb	6
9	6516K394	Connecting Rod	18-8 Stainless Steel	0.46 lb	4
10	92562A128	Mil. Spec. Alloy Steel Socket Head Screw	Alloy Steel	0.04 lb	4
11	8491A802	Press-Fit Drill Bushing	Steel	0.01 lb	8
12	95198A296	Sealing Socket Head Screw	18-8 Stainless Steel	0.04 lb	2
13	96278A112	18-8 Stainless Steel Locknut with External-Tooth Lock Washer	18-8 Stainless Steel	0.04 lb	8
	Total			194.92 lb	44

REVISIONS

ZONE	REV.	DESCRIPTION	DATE	APPROVED
	A	ORIGINAL DESIGN	2024-11-24	D.M.
	B	BOM UPDATE	2024-11-29	D.M.
	C	BOM UPDATE & DRAWING UPDATE	2024-12-03	D.M.

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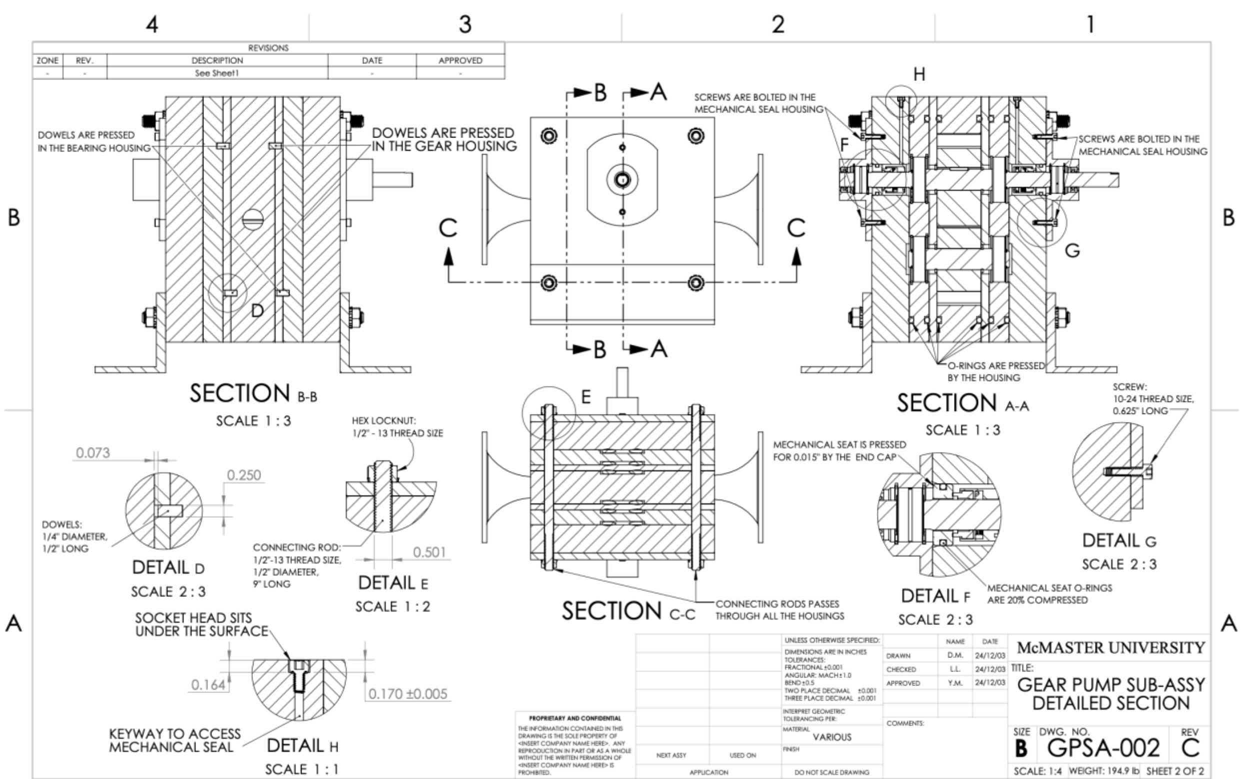
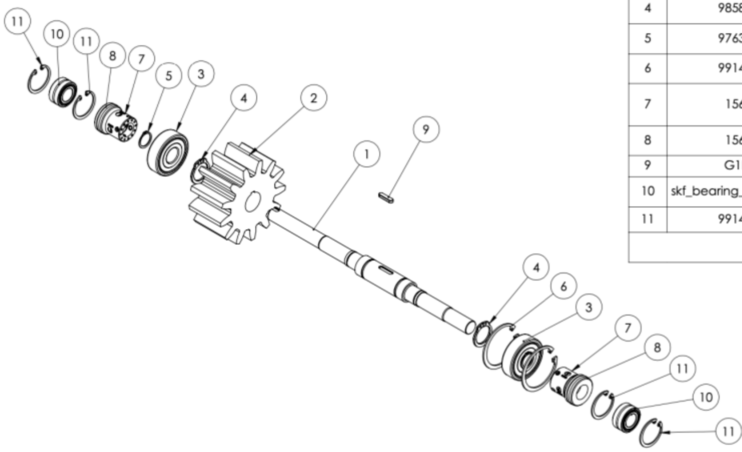


REVISIONS

ZONE	REV.	DESCRIPTION	DATE	APPROVED
-	-	See Sheet1	-	-

SOLIDWORKS Educational Product. For Instructional Use Only.

ITEM NO.	PART NUMBER	DESCRIPTION	Material	Weight	QTY.
1	G1PS001	Primay Shaft	AISI 1030 Q&T	1.1852	1
2	G1GE001	Gear	AISI1045, Carburzied Steel Grade 1	3.5333	1
3	skf_bearing_w_6303-2z_2	Deep Groove Ball Bearing	High-carbon chromium	0.247	2
4	98585A118	Heavy Duty External Retaining Ring	1060-1090 Spring Steel	0.003	2
5	97633A230	External Retaining Ring	1060-1090 Spring Steel	0.001	2
6	99142A575	Internal Retaining Ring	1060-1090 Spring Steel	0.00	2
7	1563N11	Heavy Duty Mechanical Pump Shaft Seal	AISI 316 Stainless Steel Sheet (SS)	0.11	2
8	1563N11	Heavy Duty Mechanical Pump Shaft Seal Seat	Alumina	0.05	2
9	G1KE001	Key	AISI1010	0.0046	1
10	skf_bearing_na_4902_2rs_2	Needle Bearing	High-carbon chromium	0.0816	2
11	99142A480	Internal Retaining Ring	1060-1090 Spring Steel	0.00	4
Total				5.2157	21



REVISIONS

ZONE	REV.	DESCRIPTION	DATE	APPROVED
	A	ORIGINAL DESIGN	2024-11-22	K.L.

PROPRIETARY AND CONFIDENTIAL

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NEXT ASSY | USED ON | APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±0.005
ANGULAR: MACH ±1.0
BEND ±0.5
TWO PLACE DECIMAL ±0.001
THREE PLACE DECIMAL ±0.001
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL VARIOUS
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	K.L	24.11.22
DRAWN	D.M.	24.11.22
DRAWN	Y.M.	24.11.22

COMMENTS:

McMASTER UNIVERSITY

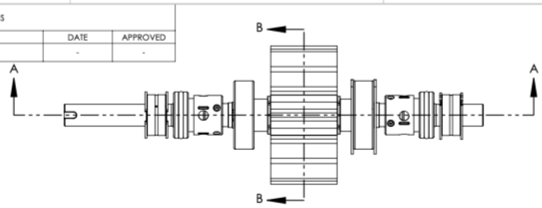
TITLE: PRIMARY SHAFT SUB-ASSY EXPLODED VIEW

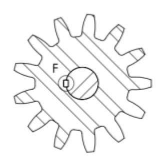
SIZE B DWG. NO. PSSA-001 REV A

SCALE: 1:8 WEIGHT:5.22 LB SHEET 1 OF 2

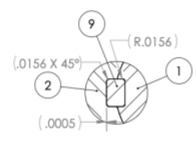
REVISIONS

ZONE	REV.	DESCRIPTION	DATE	APPROVED
-	-	See Exploded View	-	-

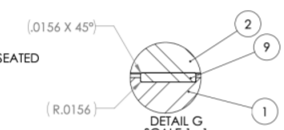




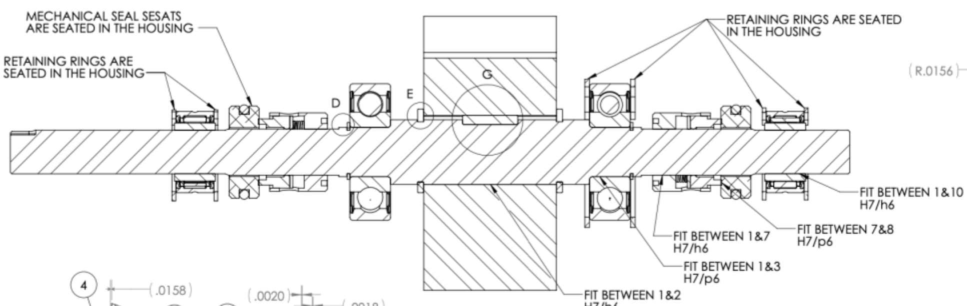
SECTION B-B



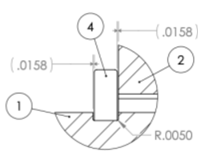
DETAIL F
SCALE 2 : 1



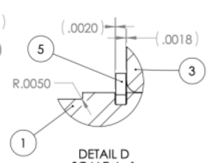
DETAIL G
SCALE 1 : 1



SECTION A-A
SCALE 1 : 1



DETAIL E
SCALE 4 : 1



DETAIL D
SCALE 4 : 1

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NEXT ASSY | USED ON | APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±0.005
ANGULAR: MACH ±1.0
BEND ±0.5
TWO PLACE DECIMAL ±0.001
THREE PLACE DECIMAL ±0.001
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL VARIOUS
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	K.L	24.11.22
DRAWN	D.M.	24.11.22
DRAWN	Y.M.	24.11.22

COMMENTS:

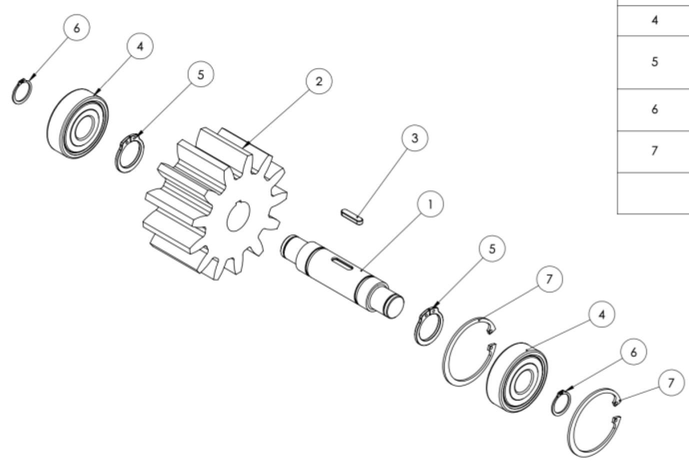
McMASTER UNIVERSITY

TITLE: PRIMARY SHAFT SUB-ASSY DETAILED SECTION

SIZE B DWG. NO. PSSA-001 REV A

SCALE: 1:2 WEIGHT:5.22 LB SHEET 2 OF 2

ITEM NO.	PART NUMBER	DESCRIPTION	Material	Weight	QTY.
1	G1SS001	Secondary Shaft	AISI 1045 Steel, cold drawn	0.5925	1
2	G1GE001	Gear	AISI1045, Carburzied Steel Grade 1	3.5333	1
3	G1KE001	Key	AISI1010	0.0046	1
4	skf_bearing_w_6303-2z_2	Deep Groove Ball Bearing	High-carbon chromium	0.247	2
5	98585A118	Heavy Duty External Retaining Ring	1060-1090 Spring Steel	0.003	2
6	97633A230	External Retaining Ring	1060-1090 Spring Steel	0.001	2
7	99142A575	Internal Retaining Ring	1060-1090 Spring Steel	0.0016	2
TOTAL				4.383	11



REVISIONS				
ZONE	REV.	DESCRIPTION	DATE	APPROVED
	A	ORIGINAL DESIGN	2024-11-22	K.L.

PROPRIETARY AND CONFIDENTIAL
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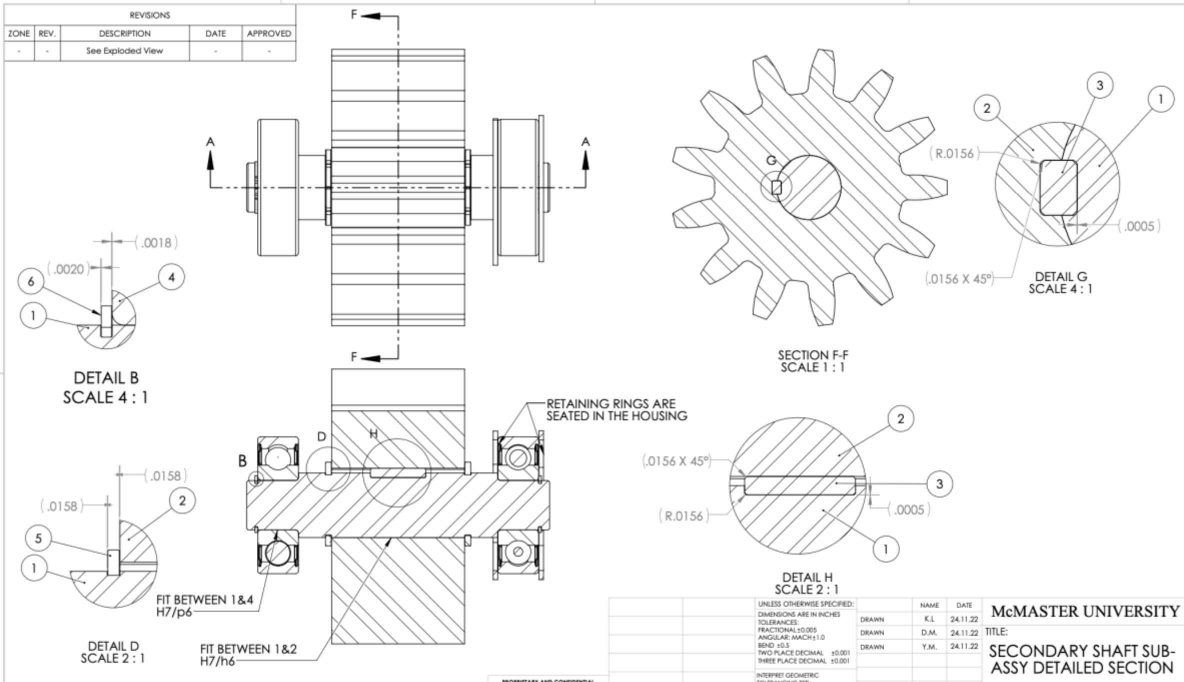
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±0.005
ANGULAR: MACH ±1.0
BEND ±0.5
TWO PLACE DECIMAL ±0.001
THREE PLACE DECIMAL ±0.001
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL VARIOUS
FINISH
NEXT ASSY | USED ON
APPLICATION | DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	K. L	24.11.22
DRAWN	D.M.	24.11.22
DRAWN	Y.M.	24.11.22

COMMENTS:

McMASTER UNIVERSITY
TITLE: SECONDARY SHAFT SUB ASSY EXPLODED VIEW
SIZE B | DWG. NO. SSSA-001 | REV A
SCALE: 1:2 | WEIGHT:4.38 lb | SHEET 1 OF 2

REVISIONS				
ZONE	REV.	DESCRIPTION	DATE	APPROVED
-	-	See Exploded View	-	-



DETAIL B
SCALE 4 : 1

DETAIL G
SCALE 4 : 1

SECTION F-F
SCALE 1 : 1

DETAIL D
SCALE 2 : 1

DETAIL H
SCALE 2 : 1

SECTION A-A
SCALE 1 : 1

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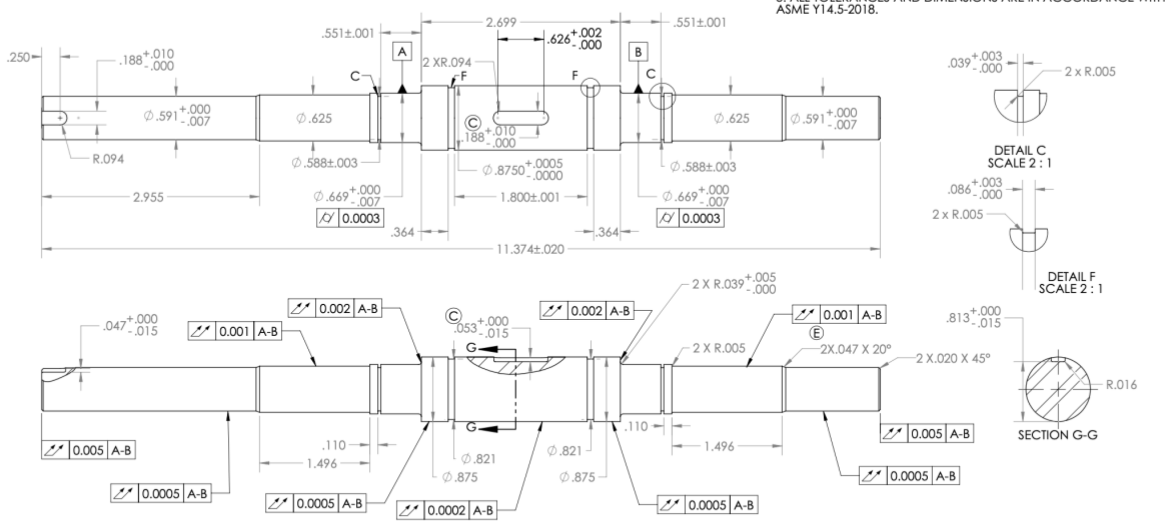
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±0.005
ANGULAR: MACH ±1.0
BEND ±0.5
TWO PLACE DECIMAL ±0.001
THREE PLACE DECIMAL ±0.001
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL VARIOUS
FINISH
NEXT ASSY | USED ON
APPLICATION | DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	K.L	24.11.22
DRAWN	D.M.	24.11.22
DRAWN	Y.M.	24.11.22

COMMENTS:

McMASTER UNIVERSITY
TITLE: SECONDARY SHAFT SUB-ASSY DETAILED SECTION
SIZE B | DWG. NO. SSSA-001 | REV A
SCALE: 1:2 | WEIGHT:4.38 LB | SHEET 2 OF 2

GENERAL NOTES:
1. ALL DIMENSIONS ARE IN INCHES UNLESS OTHERWISE SPECIFIED.
2. REMOVE ALL BURRS AND SHARP EDGES TO 0.010 INCH MAXIMUM
3. ALL TOLERANCES AND DIMENSIONS ARE IN ACCORDANCE WITH ASME Y14.5-2018.



REVISIONS				
ZONE	REV.	DESCRIPTION	DATE	APPROVED
	A	ORIGINAL DRAWING	2024.11.15	Y.M
	B	MATERIAL CHANGE: 1040 CD TO 1045 CD	2024.11.17	K.L
B3	C	KEY SEAT CHANGE: 1/8 X 1/8 TO 1/8 X 3/16	2024.11.20	K.L
	D	MATERIAL CHANGE: 1045 CD TO 1030 Q &T	2024.11.22	K.L
A2	E	FILLET TO CHAMFER	2024.11.27	K.L

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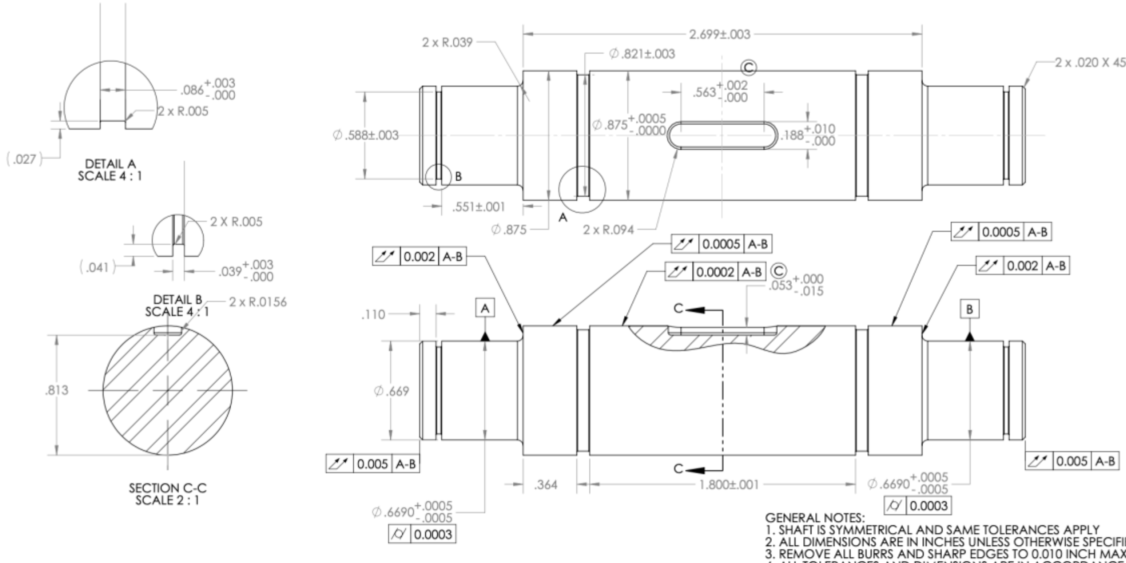
NEXT ASSY | USED ON | APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± 0.25
BEND ±0.001
TWO PLACE DECIMAL ±0.005
THREE PLACE DECIMAL ±0.003
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
AISI 1030 Q&T

	NAME	DATE
DRAWN	YM	24.11.15
DRAWN	DM	24.11.22
DRAWN	LL	24.11.22
Q.A.		
COMMENTS:		

McMASTER UNIVERSITY
TITLE:
DRIVE SHAFT
SIZE B | DWG. NO. G1PS001 | REV E
SCALE: 1:1 | WEIGHT: 1.81 lb

SOLIDWORKS Educational Product. For Instructional Use Only.



GENERAL NOTES:
1. SHAFT IS SYMMETRICAL AND SAME TOLERANCES APPLY
2. ALL DIMENSIONS ARE IN INCHES UNLESS OTHERWISE SPECIFIED.
3. REMOVE ALL BURRS AND SHARP EDGES TO 0.010 INCH MAXIMUM
4. ALL TOLERANCES AND DIMENSIONS ARE IN ACCORDANCE WITH ASME Y14.5-2018.

ZONE	REV.	DESCRIPTION	DATE	APPROVED
B2	C	KEY SEAT CHANGE: 1/8 X 1/8 TO 1/8 X 3/16	2024.11.20	K. L
	B	MATERIAL CHANGE: 1040 CD TO 1045 CD	2024.11.17	K. L
	A	ORIGINAL DESIGN	2024.11.14	Y.M
REVISIONS				

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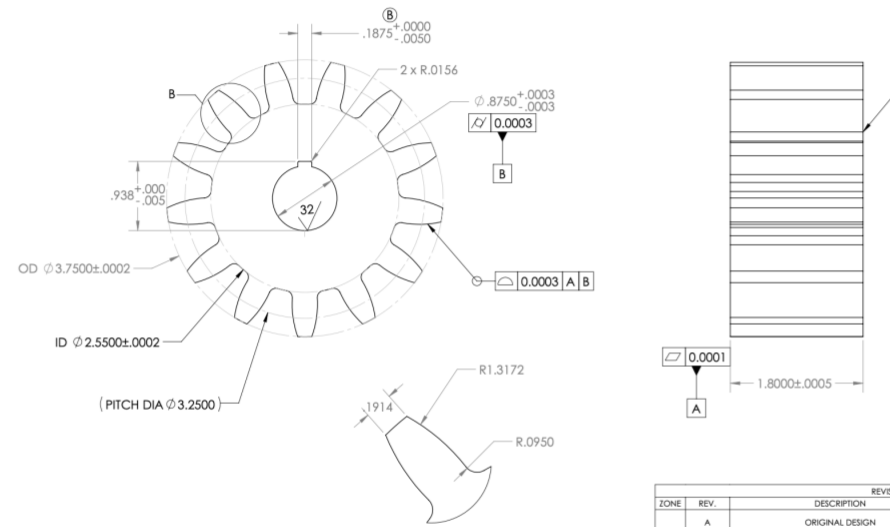
NEXT ASSY | APPLICATION | DO NOT SCALE DRAWING

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH±0.25
BEND ±0.001
TWO PLACE DECIMAL ±0.005
THREE PLACE DECIMAL ±0.003
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
AISI 1045 CD

	NAME	DATE
DRAWN	YM	24.11.15
	DM	24.11.20
	LL	24.11.20

McMASTER UNIVERSITY
TITLE:
DRIVEN SHAFT
SIZE B | DWG. NO. G1SS001 | REV C
WEIGHT:0.59 LB

SOLIDWORKS Educational Product. For Instructional Use Only.



DETAIL B
SCALE 2 : 1

# OF TEETH	13
MODULE	.25
BORE DIA	.875
PRESSURE ANGLE	20
OUTER DIA	3.75
PITCH DIA	3.25
BACKLASH	0.075
ARC LENGTH	0.467
FILLET RADIUS	0.095
TOOTH DEPTH	0.6

GENERAL NOTES:
1. ALL DIMENSIONS ARE IN INCHES UNLESS OTHERWISE SPECIFIED.
2. REMOVE ALL BURRS AND SHARP EDGES TO 0.010 INCH MAXIMUM
3. ALL TOLERANCES AND DIMENSIONS ARE IN ACCORDANCE WITH ASME Y14.5-2018.

REVISIONS

ZONE	REV.	DESCRIPTION	DATE	APPROVED
	A	ORIGINAL DESIGN	2024.11.18	Y.M
B3	B	KEY WAY WIDTH 1/8" TO 3/16"	2024.11.22	K. L

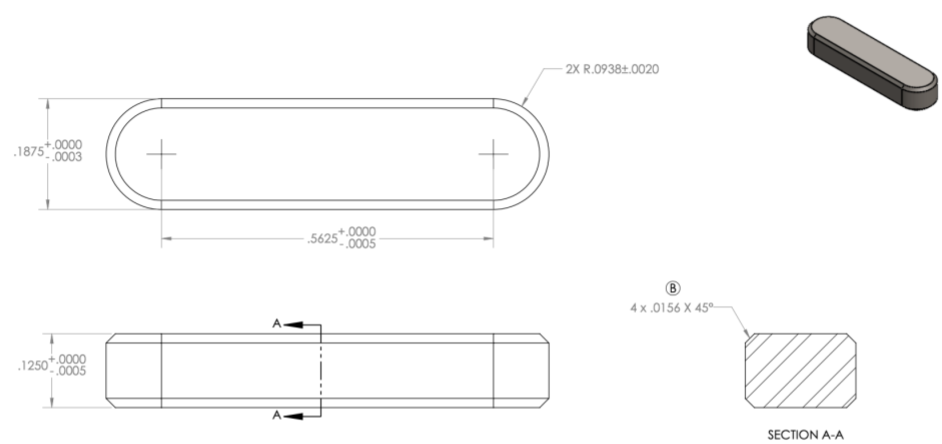
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH± 0.25
BEND ±0.001
TWO PLACE DECIMAL ±0.005
THREE PLACE DECIMAL ±0.002

INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
1045 CARBURIZED STEEL, GRADE 1

	NAME	DATE
DRAWN	YM	24.11.18
	DM	24.11.22
	LL	24.11.22

NEXT ASSY
APPLICATION
DO NOT SCALE DRAWING

McMASTER UNIVERSITY
TITLE:
GEAR
SIZE B DWG. NO. G1GE001 REV B
WEIGHT: 3.1LB



SECTION A-A

REVISIONS

ZONE	REV.	DESCRIPTION	DATE	APPROVED
B2	B	FILLET CHANGE TO CHAMFER	2024.11.22	K.L
	A	ORIGINAL DESIGN	2024.11.21	K.L

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ±0.25
BEND ±0.001
TWO PLACE DECIMAL ±0.005
THREE PLACE DECIMAL ±0.003

INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
AISI 1010
FINISH

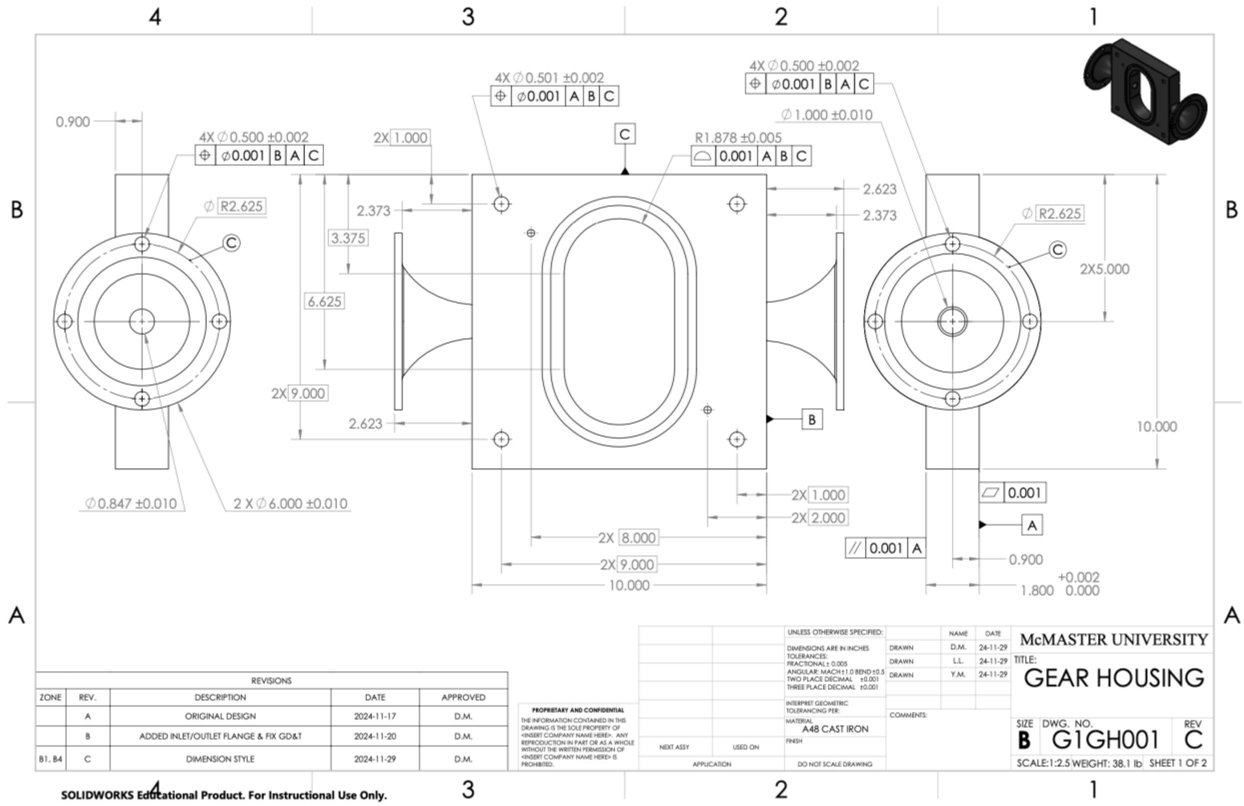
	NAME	DATE
DRAWN	K.L	24.11.22
DRAWN	D.M.	24.11.22
DRAWN	Y.F.	24.11.22

COMMENTS:

NOTE:
1. CHAMFERS ARE ALL AROUND

NEXT ASSY
USED ON
APPLICATION
DO NOT SCALE DRAWING

McMASTER UNIVERSITY
TITLE:
KEY
SIZE B DWG. NO. G1KE001 REV B
SCALE: 8:1 WEIGHT: 0.0046 SHEET 1 OF 1


4X ∅0.501 ±0.002
⌖ ∅0.001 A B C
4X ∅0.500 ±0.002
⌖ ∅0.001 B A C
∅1.000 ±0.010
0.900
2X 1.000
C
R1.878 ±0.005
0.001 A B C
∅R2.625
2.373
2.623
3.375
6.625
2X 9.000
2X5.000
10.000
∅0.847 ±0.010
2 X ∅6.000 ±0.010
B
0.001
A
2X 1.000
2X 2.000
2X 8.000
2X 9.000
10.000
// 0.001 A
0.900
1.800 +0.002 0.000
REVISIONS
ZONE | REV. | DESCRIPTION | DATE | APPROVED
A | ORIGINAL DESIGN | 2024-11-17 | D.M.
B | ADDED INLET/OUTLET FLANGE & FIX GD&T | 2024-11-20 | D.M.
B1, B4 | C | DIMENSION STYLE | 2024-11-29 | D.M.
PROPRIETARY AND CONFIDENTIAL
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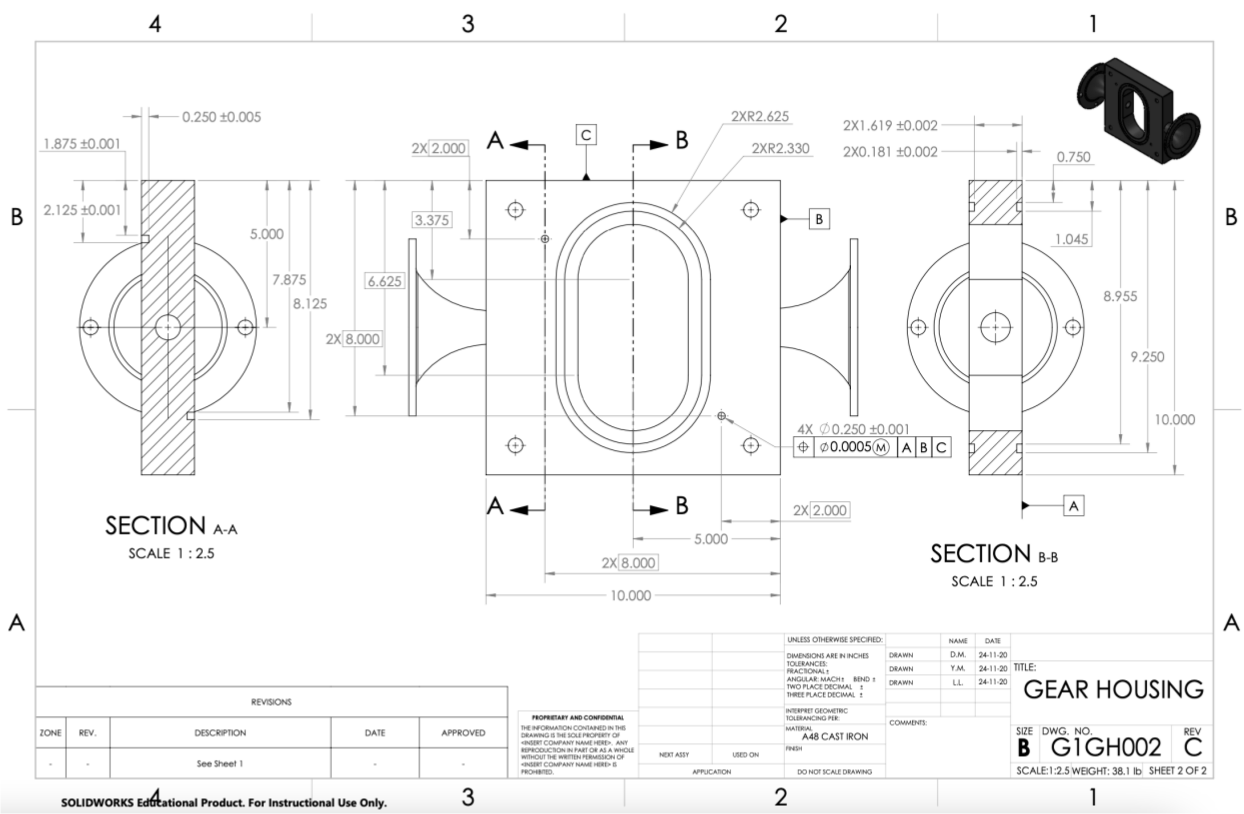
NEXT ASSY | USED ON
APPLICATION
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 0.005
ANGULAR: MACH ±1.0 BEND ±0.5
TWO PLACE DECIMAL ±0.001
THREE PLACE DECIMAL ±0.001
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
A48 CAST IRON
FINISH
DO NOT SCALE DRAWING
NAME | DATE
DRAWN | D.M. | 24-11-29
DRAWN | L.L. | 24-11-29
DRAWN | Y.M. | 24-11-29
COMMENTS:
McMASTER UNIVERSITY
TITLE:
GEAR HOUSING
SIZE B DWG. NO. G1GH001 REV C
SCALE:1:2.5 WEIGHT: 38.1 lb SHEET 1 OF 2
SOLIDWORKS Educational Product. For Instructional Use Only.


0.250 ±0.005
1.875 ±0.001
2.125 ±0.001
5.000
7.875
8.125
2X 2.000
3.375
6.625
2X 8.000
C
A
B
2XR2.625
2XR2.330
2X1.619 ±0.002
2X0.181 ±0.002
0.750
1.045
8.955
9.250
10.000
4X ∅0.250 ±0.001
⌖ ∅0.0005Ⓜ A B C
2X 2.000
5.000
2X 8.000
10.000
SECTION A-A
SCALE 1 : 2.5
SECTION B-B
SCALE 1 : 2.5
REVISIONS
ZONE | REV. | DESCRIPTION | DATE | APPROVED
- | - | See Sheet 1 | - | -
PROPRIETARY AND CONFIDENTIAL
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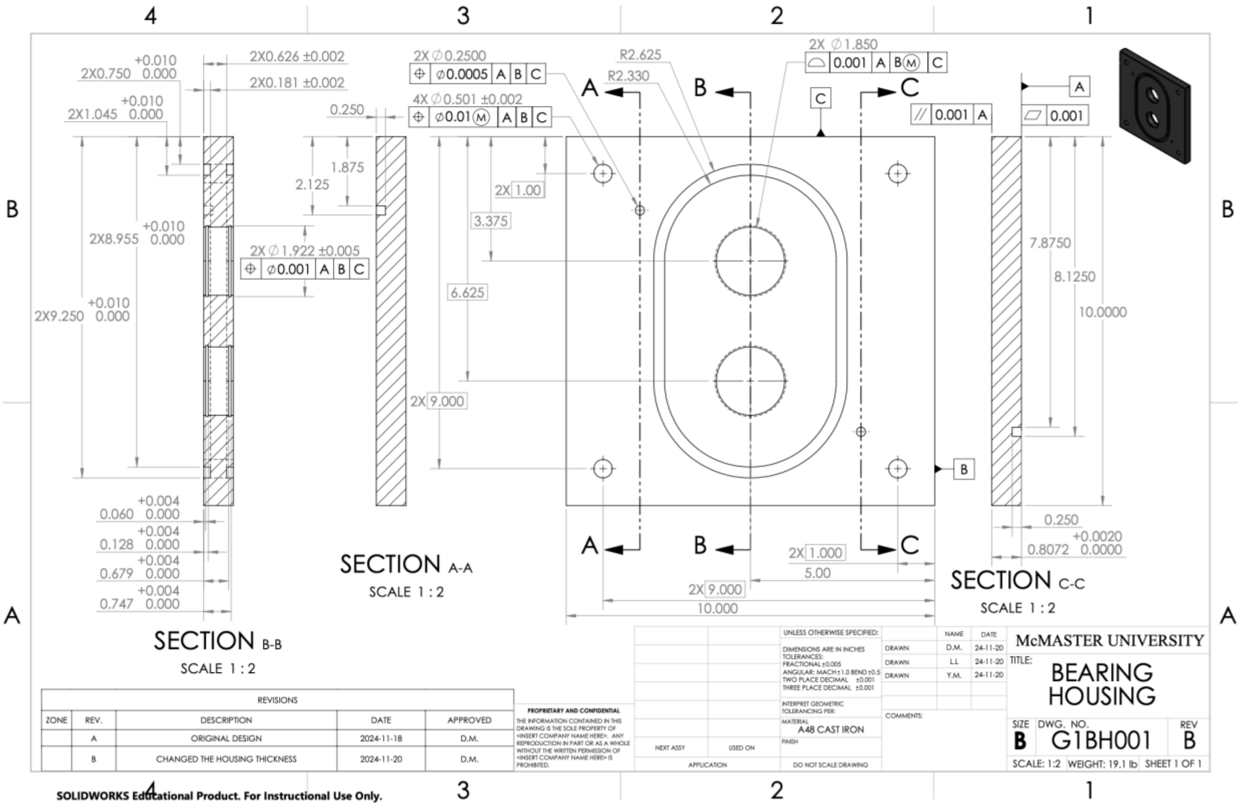
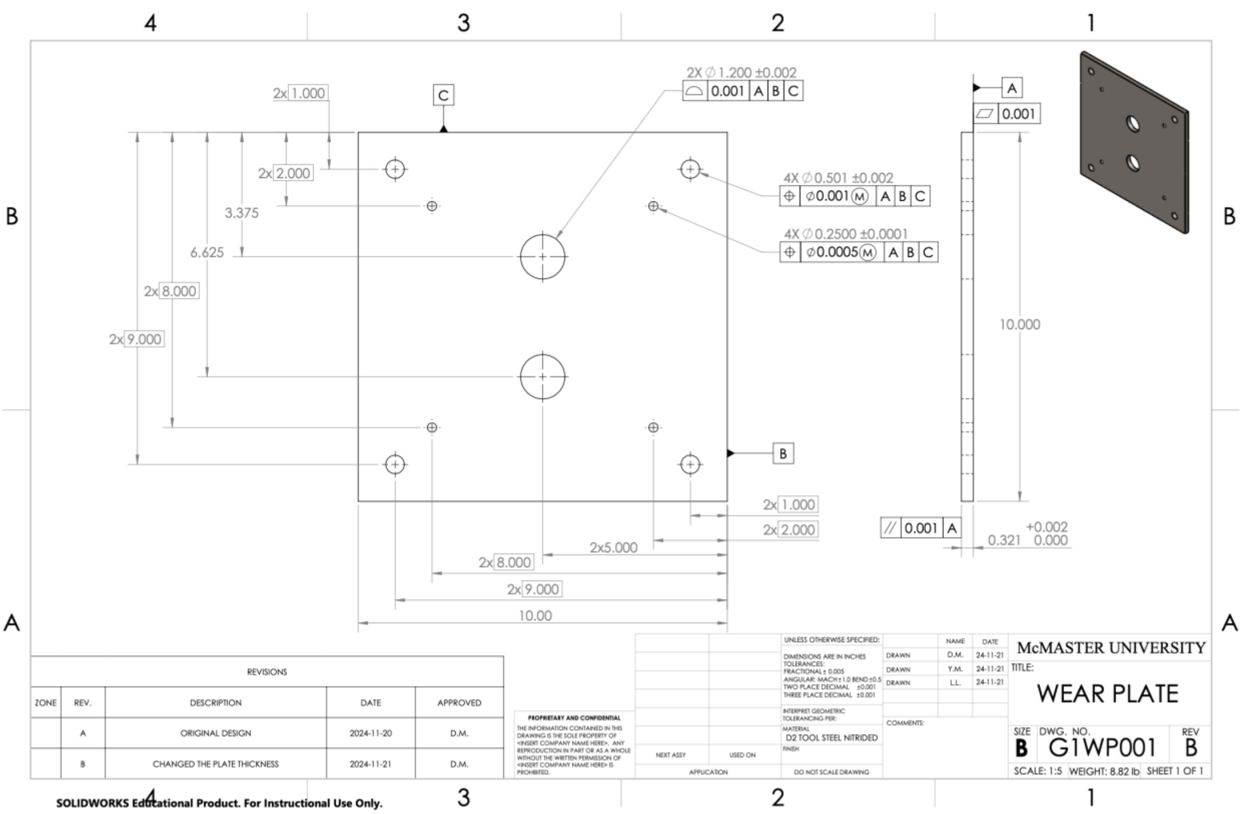
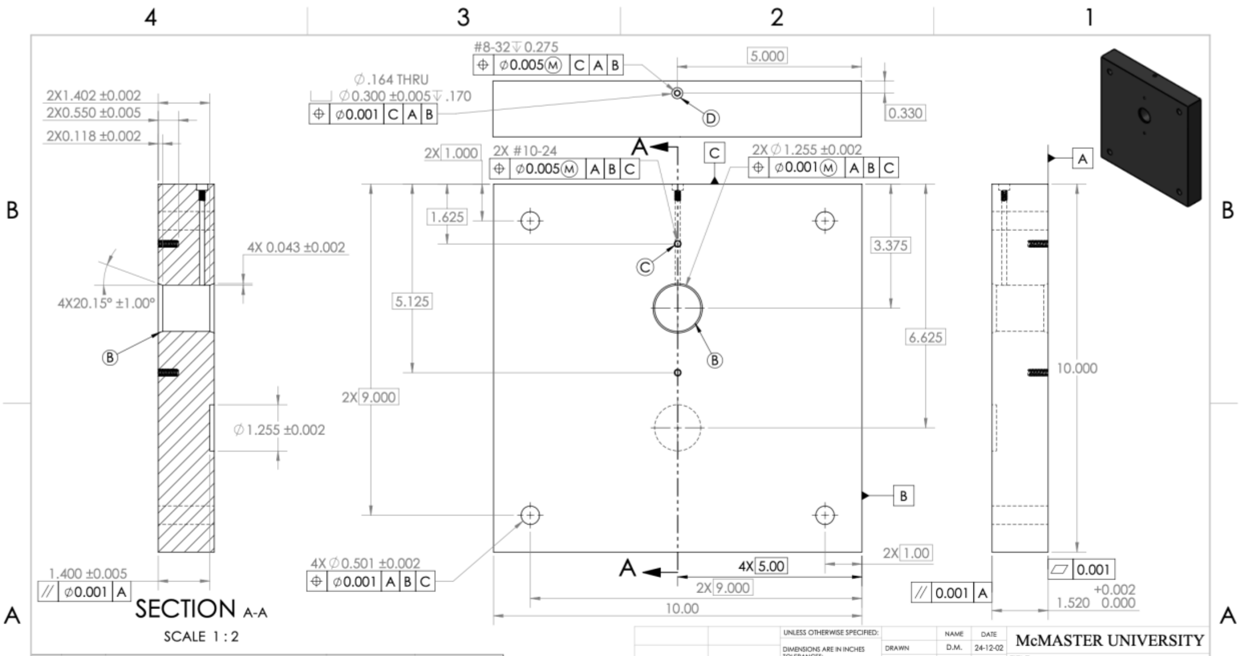
NEXT ASSY | USED ON
APPLICATION
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
A48 CAST IRON
FINISH
DO NOT SCALE DRAWING
NAME | DATE
DRAWN | D.M. | 24-11-20
DRAWN | Y.M. | 24-11-20
DRAWN | L.L. | 24-11-20
COMMENTS:
TITLE:
GEAR HOUSING
SIZE B DWG. NO. G1GH002 REV C
SCALE:1:2.5 WEIGHT: 38.1 lb SHEET 2 OF 2
SOLIDWORKS Educational Product. For Instructional Use Only.


SECTION A-A
SCALE 1 : 2
SECTION B-B
SCALE 1 : 2
SECTION C-C
SCALE 1 : 2
2X ∅ 1.850
R2.625
R2.330
2X ∅ 0.2500
4X ∅ 0.501 ±0.002
2X ∅ 1.922 ±0.005
10.000
REVISIONS
ZONE | REV. | DESCRIPTION | DATE | APPROVED
A | ORIGINAL DESIGN | 2024-11-18 | D.M.
B | CHANGED THE HOUSING THICKNESS | 2024-11-20 | D.M.
PROPRIETARY AND CONFIDENTIAL
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
MATERIAL A48 CAST IRON
DRAWN D.M. 24-11-20
DRAWN L.L. 24-11-20
DRAWN Y.M. 24-11-20
McMASTER UNIVERSITY
TITLE: BEARING HOUSING
SIZE B DWG. NO. G1BH001 REV B
SCALE: 1:2 WEIGHT: 19.1 lb SHEET 1 OF 1
SOLIDWORKS Educational Product. For Instructional Use Only.


2X ∅ 1.200 ±0.002
4X ∅ 0.501 ±0.002
4X ∅ 0.2500 ±0.0001
10.000
10.00
REVISIONS
ZONE | REV. | DESCRIPTION | DATE | APPROVED
A | ORIGINAL DESIGN | 2024-11-20 | D.M.
B | CHANGED THE PLATE THICKNESS | 2024-11-21 | D.M.
PROPRIETARY AND CONFIDENTIAL
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
MATERIAL D2 TOOL STEEL NITRIDED
DRAWN D.M. 24-11-21
DRAWN Y.M. 24-11-21
DRAWN L.L. 24-11-21
McMASTER UNIVERSITY
TITLE: WEAR PLATE
SIZE B DWG. NO. G1WP001 REV B
SCALE: 1:5 WEIGHT: 8.82 lb SHEET 1 OF 1
SOLIDWORKS Educational Product. For Instructional Use Only.



REVISIONS				
ZONE	REV.	DESCRIPTION	DATE	APPROVED
	A	ORIGINAL DESIGN	2024-11-19	D.M.
B2, B4	B	CHANGED THICKNESS, INNER DIAMETER & ADDED CHAMFER	2024-11-21	D.M.
B2	C	REPLACED SCREW FROM #10-32 TO #10-24	2024-11-29	D.M.
B2	D	MECHANICAL SEAL KEY WAY	2024-12-02	D.M.

PROPRIETARY AND CONFIDENTIAL

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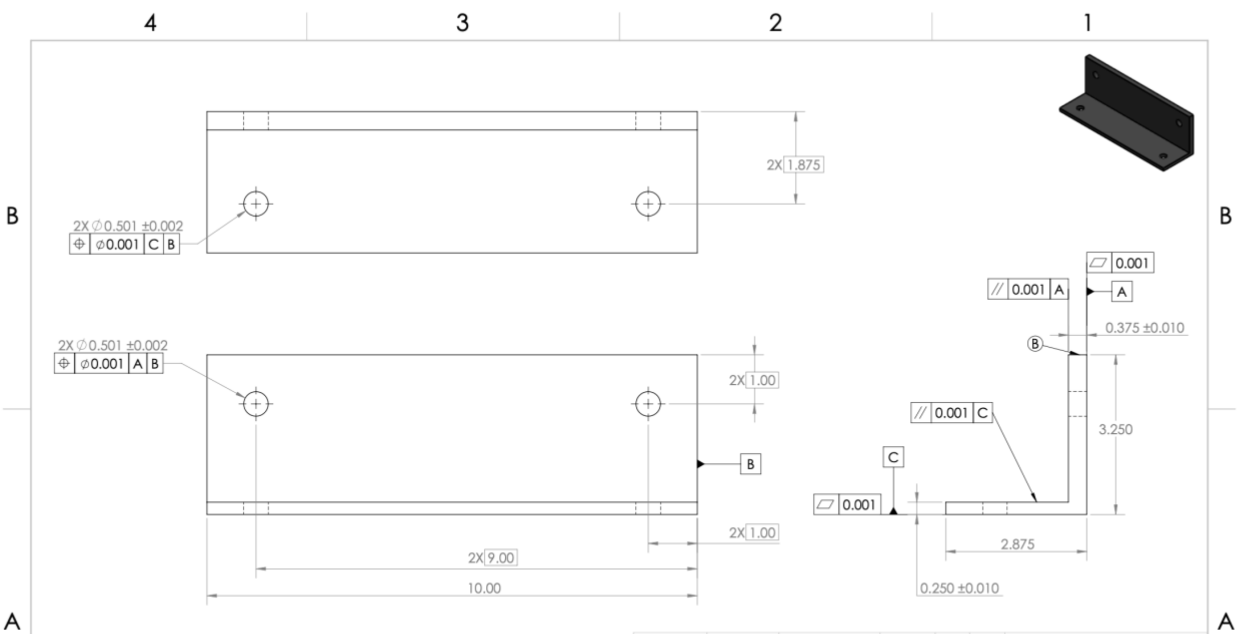
NEXT ASSY | USED ON | APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 0.005
ANGULAR: MACH ±1.0 BEND ±0.5
TWO PLACE DECIMAL ±0.001
THREE PLACE DECIMAL ±0.001
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL A48 CAST IRON
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	D.M.	24-12-02
DRAWN	L.L.	24-12-02
DRAWN	Y.M.	24-12-02

COMMENTS:

McMASTER UNIVERSITY
TITLE: MECHANICAL SEAL HOUSING
SIZE B | DWG. NO. G1MH001 | REV D
SCALE: 1:2 | WEIGHT: 38.7 lb | SHEET 1 OF 1



REVISIONS				
ZONE	REV.	DESCRIPTION	DATE	APPROVED
	A	ORIGINAL DESIGN	2024-11-23	D.M.
B1	B	MADE THE VERTICAL WALL THICKER	2024-11-29	D.M.

PROPRIETARY AND CONFIDENTIAL

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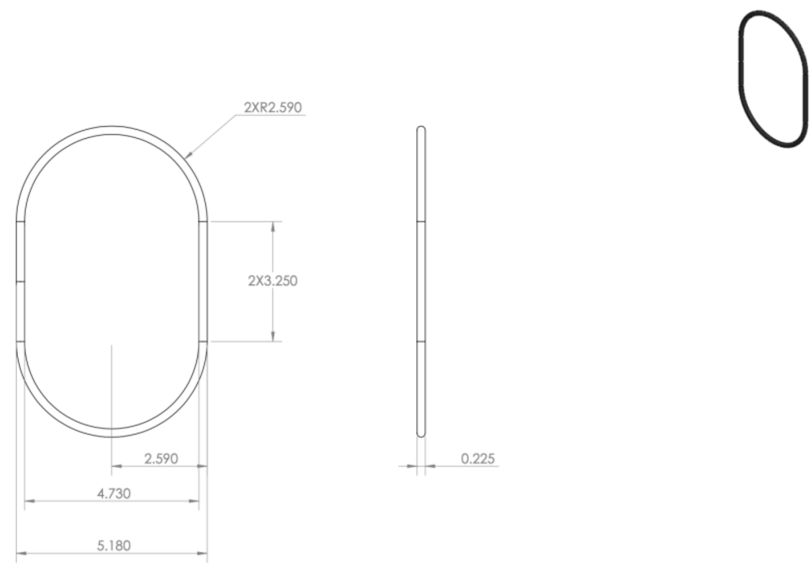
NEXT ASSY | USED ON | APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 0.005
ANGULAR: MACH ±1.0BEND ±0.5
TWO PLACE DECIMAL ±0.001
THREE PLACE DECIMAL ±0.001
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL A48 CAST IRON
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	D.M.	24-11-29
DRAWN	L.L.	24-11-29
DRAWN	Y.M.	24-11-29

COMMENTS:

McMASTER UNIVERSITY
TITLE: MOUTING BRACKET
SIZE B | DWG. NO. G1MB001 | REV B
SCALE:1:1.5 | WEIGHT: 3.69 lb | SHEET 1 OF 1



ZONE	REV.	DESCRIPTION	DATE	APPROVED
	A	ORIGINAL DESIGN	2024-11-19	D.M.
	B	ADDED MISSING DIMENSIONS	2024-11-20	D.M.

REVISIONS

PROPRIETARY AND CONFIDENTIAL

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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 0.005
ANGULAR: MACH ± 1.0 BEND ± 0.5
TWO PLACE DECIMAL ±0.001
THREE PLACE DECIMAL ±0.001

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
EPDM RUBBER

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	D.M.	24-11-20
DRAWN	L.L.	24-11-20
DRAWN	Y.M.	24-11-20

COMMENTS:

THE TOTAL LENGTH OF THE O-RING IS 11.680 ± 0.005.

McMASTER UNIVERSITY

TITLE:

O-RING

SIZE	DWG. NO.	REV
B	G1OR001	B

SCALE: 1:2 WEIGHT:0.04 lb SHEET 1 OF 1

Purchased Components

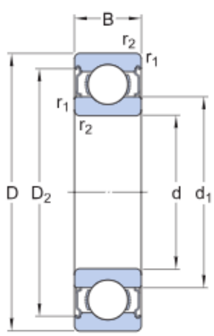
Groove Bearing – SKF 6303-2Z

W 6303-2Z

Stainless steel deep groove ball bearing with integral sealing

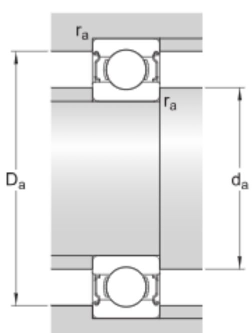
Stainless steel single row deep groove ball bearing with seals or shields on both sides, provide greater chemical and corrosion resistance. As with deep groove ball bearings generally, they are particularly versatile, have low frictio and are optimized for low noise and low vibration, which enables high rotational speeds. They accommodate radial and axial loads in both directions, are easy to mount, and require less maintenance than other bearing types. The integral sealing can significantly prolong bearing service life because it keeps lubricant in the bearings and contaminants out.

- Greater chemical and corrosion resistance
- Integral sealing prolongs bearing service life
- Typical benefits of single row deep groove ball bearings



Dimensions

d	0.669 in	Bore diameter
D	1.85 in	Outside diameter
B	0.551 in	Width
d_1	≈ 1.083 in	Shoulder diameter
d_2	≈ 1.083 in	Recess diameter
D_2	≈ 1.618 in	Recess diameter
$r_{1,2}$	min. 0.039 in	Chamfer dimension



Abutment dimensions

d_a	min. 0.866 in	Diameter of shaft abutment
d_a	max. 1.063 in	Diameter of shaft abutment
D_a	max. 1.654 in	Diameter of housing abutment
r_a	max. 0.039 in	Radius of shaft or housing fillet

Dimensions

Bore diameter	0.669 in
Outside diameter	1.85 in
Width	0.551 in

Performance

Basic dynamic load rating	2 630 lbf
Basic static load rating	1 472 lbf
Reference speed	36 000 r/min
Limiting speed	18 000 r/min

Properties

Filling slots	Without
Number of rows	1
Locating feature, bearing outer ring	None
Bore type	Cylindrical
Cage	Sheet metal
Matched arrangement	No
Radial internal clearance	CN
Material, bearing	Stainless steel
Coating	Without
Sealing	Shield on both sides
Sealing type	Non-contact
Lubricant	Grease
Relubrication feature	Without

Logistics

Product net weight	0.2447 lb
eClass code	23-05-08-01
UNSPSC code	31171504

Calculation data

Basic dynamic load rating	C	2 630 lbf
Basic static load rating	C_0	1 472 lbf
Fatigue load limit	P_u	63 lbf
Reference speed		36 000 r/min
Limiting speed		18 000 r/min
Minimum load factor	k_r	0.035
Calculation factor	f_0	12.4

Tolerance class

Dimensional tolerances	Normal
Radial run-out	Normal

Needle Bearing – SKF NA 4903.2RS

NA 4902.2RS

Single row needle roller bearing with machined rings, with flanges and integral sealing

Single row needle roller bearings incorporate cylindrical rollers that are small in diameter relative to their length. Owing to their large number of rollers, the bearings have a high load carrying capacity. The outer ring includes two integral flanges to guide the bearing axially and an annular groove with one or more lubrication holes to facilitate relubrication. The inner ring contains an additional lubrication hole.

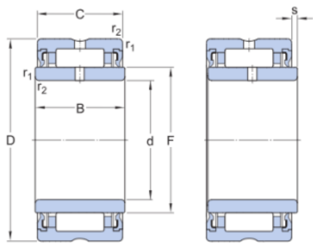
- High radial load carrying capacity
- High stiffness and low cross-sectional height
- Accommodate axial displacement in both directions
- Separable design
- Integral sealing prolongs bearing service life

Dimensions	
Bore diameter	0.591 in
Outside diameter	1.102 in
Width	0.551 in

Properties	
Bearing part	Complete bearing
Number of rows	1
Outer ring type	Machined (solid)
Aligning feature	Without
Cage	Sheet metal
Number of flanges, outer ring	2
Radial internal clearance	CN
Tolerance class	Normal
Material, bearing	Bearing steel
Coating	Without
Sealing	Seal on both sides
Sealing type	Contact
Lubricant	Grease
Relubrication feature	With

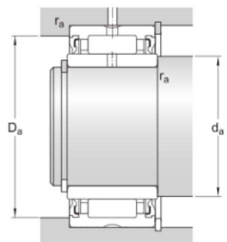
Performance	
Basic dynamic load rating	2 053 lbf
Basic static load rating	2 698 lbf
Limiting speed	9 500 r/min

Logistics	
Product net weight	0.07826 lb
eClass code	23-05-09-03
UNSPSC code	31171512



Dimensions

d	0.591 in	Bore diameter
D	1.102 in	Outside diameter
B	0.551 in	Width
C	0.512 in	Width outer ring
F	0.787 in	Raceway diameter inner ring
$r_{1,2}$	min. 0.012 in	Chamfer dimension outer ring
s	max. 0.02 in	Permissible axial displacement from the normal position of one bearing ring relative to the other



Abutment dimensions

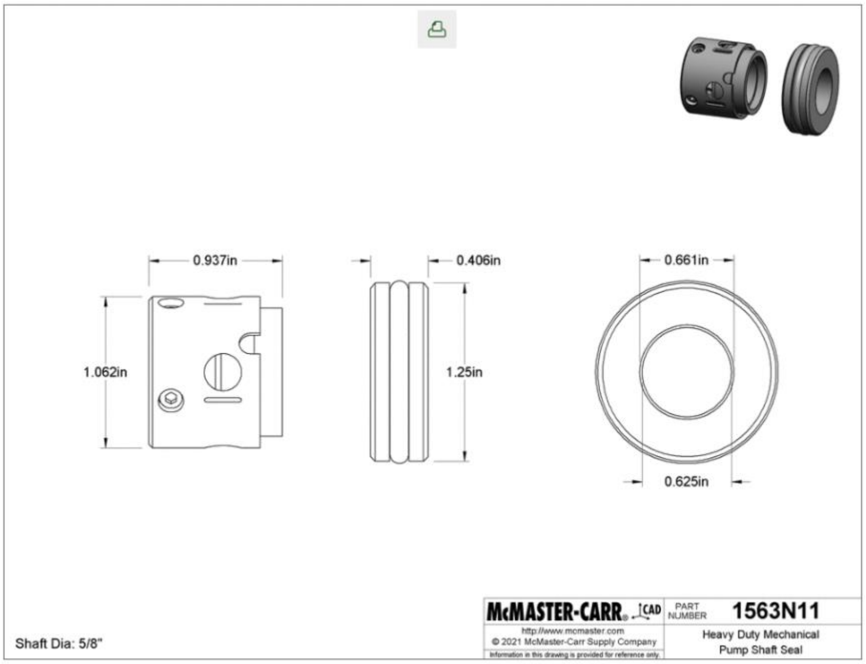
d_a	min. 0.669 in	Smallest permissible abutment diameter shaft, bearings with flanges
D_a	max. 1.024 in	Diameter of housing abutment
r_a	max. 0.012 in	Fillet radius

Calculation data

Basic dynamic load rating	C	2 053 lbf
Basic static load rating	C_0	2 698 lbf
Fatigue load limit	P_u	321 lbf
Limiting speed		9 500 r/min

Mechanical Seals – 1563N11

Heavy Duty Mechanical Pump Shaft Seal for 5/8" Shaft Diameter

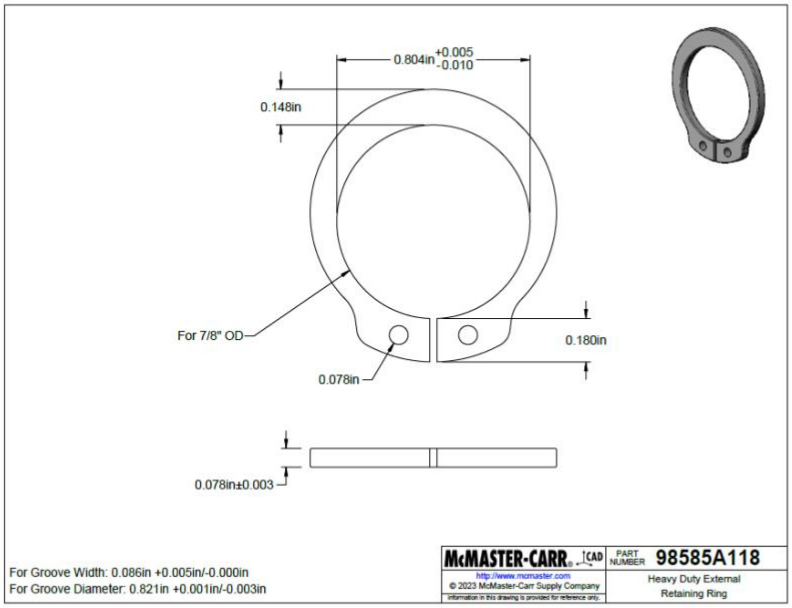


For Motion Type	Rotary, Oscillating
For Sealing	Shafts
For Shaft Diameter	5/8"
Manufacturer/ Brand Equivalent	John Crane Type 8T Pac-Seal Type 8T
Seal	
ID	0.625"
OD	1.062"
Width	0.937"
Seat	
ID	0.661"
OD	1.250"
Width	0.406"
Material	
Case	316 Stainless Steel
Spring	316 Stainless Steel
Diaphragm	Viton® Fluoroelastomer Rubber
Washer	Carbon
Seat	Ceramic
Gasket	Viton® Fluoroelastomer Rubber
Maximum Speed	
For Rotary Motion	3,600 rpm
For Oscillating Motion	83.3 feet/sec.
Maximum Pressure	300 psi
Temperature Range	0° to 400° F
Diaphragm Hardness	Durometer 70A (Hard)

Environment	Food Industry
System of Measurement	Inch
Cannot Be Sold To	Outside North America
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/14/2023, 235 SVHC) Compliant
DFARS	Specialty Metals COTS-Exempt
Country of Origin	Mexico
USMCA Qualifying	No
Schedule B	848420.0000
ECCN	EAR99

Retaining Ring - 98585A118

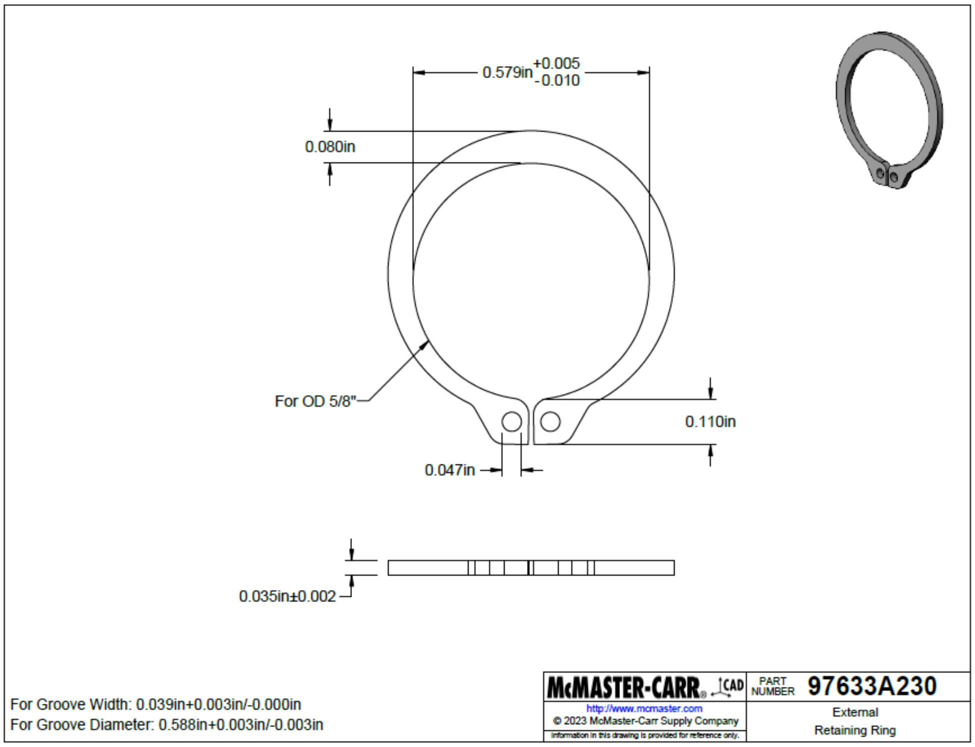
Heavy Duty External Retaining Ring for 7/8" OD, Black-Phosphate 1060-1090 Spring Steel



Retaining Ring Type	External
Retaining Ring Style	Standard
System of Measurement	Inch
Material	1060-1090 Spring Steel
Finish	Black Phosphate
For OD	7/8"
For Groove	
Diameter	0.821"
Diameter Tolerance	-0.003" to 0.001"
Width	0.086"
Width Tolerance	0" to 0.005"
Ring	
ID	0.804"
ID Tolerance	-0.01" to 0.005"
Thickness	0.078"
Thickness Tolerance	-0.003" to 0.003"
Min. Hardness	Rockwell C47
Thrust Load Capacity	10,500 lbs.
Magnetic Properties	Magnetic
Specifications Met	ASME B18.27.2
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/14/2023, 235 SVHC) Compliant
DFARS	Specialty Metals COTS-Exempt
Country of Origin	United States
USMCA Qualifying	No
Schedule B	731824.0000
ECCN	EAR99

Retaining Ring - 97633A230

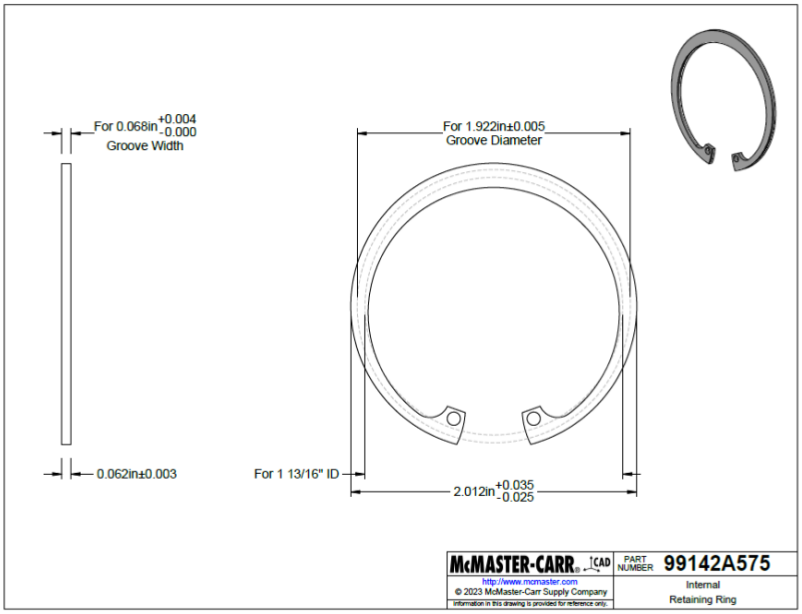
External Retaining Ring for 5/8" OD, Black-Phosphate 1060-1090 Spring Steel



Retaining Ring Type	External
Retaining Ring Style	Standard
System of Measurement	Inch
Material	1060-1090 Spring Steel
Finish	Black Phosphate
For OD	5/8"
For Groove	
Diameter	0.588"
Diameter Tolerance	-0.003" to 0.003"
Width	0.039"
Width Tolerance	0" to 0.003"
Ring	
ID	0.579"
ID Tolerance	-0.01" to 0.005"
Thickness	0.035"
Thickness Tolerance	-0.002" to 0.002"
Min. Hardness	Rockwell C40
Thrust Load Capacity	2,090 lbs.
Magnetic Properties	Magnetic
Specifications Met	ASME B18.27.1
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/14/2023, 235 SVHC) Compliant
DFARS	Specialty Metals COTS-Exempt
Country of Origin	United States
USMCA Qualifying	No
Schedule B	731824.0000
ECCN	EAR99

Retaining Ring - 99142A575

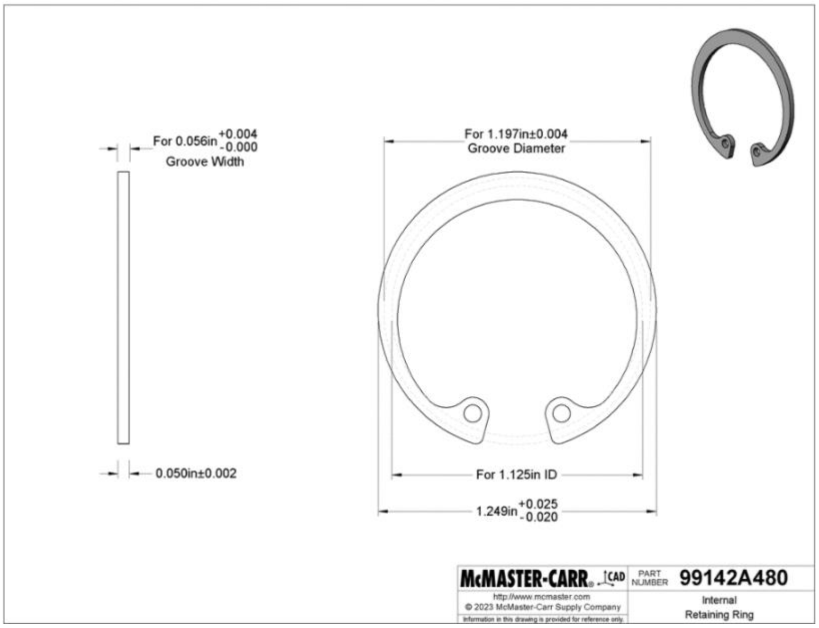
Internal Retaining Ring for 1-13/16" ID, Black-Phosphate 1060-1090 Spring Steel



Retaining Ring Type	Internal
Retaining Ring Style	Standard
System of Measurement	Inch
Material	1060-1090 Spring Steel
Finish	Black Phosphate
For ID	1 13/16"
For Groove	
Diameter	1.922"
Diameter Tolerance	-0.005" to 0.005"
Width	0.068"
Width Tolerance	0" to 0.004"
Ring	
OD	2.012"
OD Tolerance	-0.025" to 0.035"
Thickness	0.062"
Thickness Tolerance	-0.003" to 0.003"
Min. Hardness	Rockwell C40
Thrust Load Capacity	16,100 lbs.
Magnetic Properties	Magnetic
Specifications Met	ASME B18.27.1
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/14/2023, 235 SVHC) Compliant
DFARS	Specialty Metals COTS-Exempt
Country of Origin	United States
USMCA Qualifying	No
Schedule B	731824.0000
ECCN	EAR99

Retaining Ring - 99142A575

Internal Retaining Ringfor 1-1/8" ID, Black-Phosphate 1060-1090 Spring Steel



Retaining Ring Type	Internal
Retaining Ring Style	Standard
System of Measurement	Inch
Material	1060-1090 Spring Steel
Finish	Black Phosphate
For ID	1 1/8"
For Groove	
Diameter	1.197"
Diameter Tolerance	-0.004" to 0.004"
Width	0.056"
Width Tolerance	0" to 0.004"
Ring	
OD	1.249"
OD Tolerance	-0.02" to 0.025"
Thickness	0.05"
Thickness Tolerance	-0.002" to 0.002"
Min. Hardness	Rockwell C40
Thrust Load Capacity	8,010 lbs.
Magnetic Properties	Magnetic
Specifications Met	ASME B18.27.1
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/14/2023, 235 SVHC) Compliant
DFARS	Specialty Metals COTS-Exempt
Country of Origin	United States
USMCA Qualifying	No
Schedule B	731824.0000
ECCN	EAR99

Oil Seal – SKF 17 x 28 x 7 CRW1R

17X28X7 CRW1 R

Radial shaft seal with metal case and SKF Wave lip, for oil or grease

Radial shaft seals are used between rotating and stationary machine components, or between components in relative motion. CRW1 seals are designed with a metal case, an SKF WAVE lip made of elastomer with higher pumping rate to reduce heat generation, and a garter spring to optimize sealing against the shaft. Most of the seals feature the SKF Bore Tite coating on the outside diameter that helps fill small imperfections in the housing bore.

- For oil or grease
- With garter spring
- WAVE lip creates higher pumping rate
- WAVE lip has less friction between shaft and lip
- WAVE lip has a lower temperature at the contact point

Dimensions

Shaft diameter	0.669 in
Housing bore diameter	1.102 in
Seal width	0.276 in

Performance

Limiting speed	3 900 r/min
Maximum operating temperature	212 °F
Minimum operating temperature	-40 °F
Permissible circumferential speed	590.551 ft/min
Rotational speed	3 900 r/min

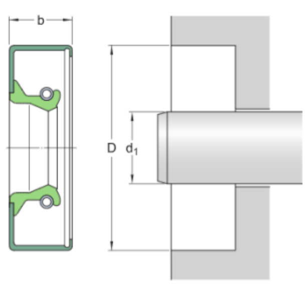
Properties

Design	CRW1
Auxiliary lip	No
Sealing lip material	Nitrile rubber (NBR)
Type of outside diameter	Metal-cased with sealant coating on the outside diameter
Unit system	Metric

Logistics

Product net weight	0.009921 lb
eClass code	23-07-08-01
UNSPSC code	31181602

Type of outside diameter	Metal-cased with sealant coating on the outside diameter
Lip material	Nitrile rubber (NBR)
Seal design	CRW1



Dimensions

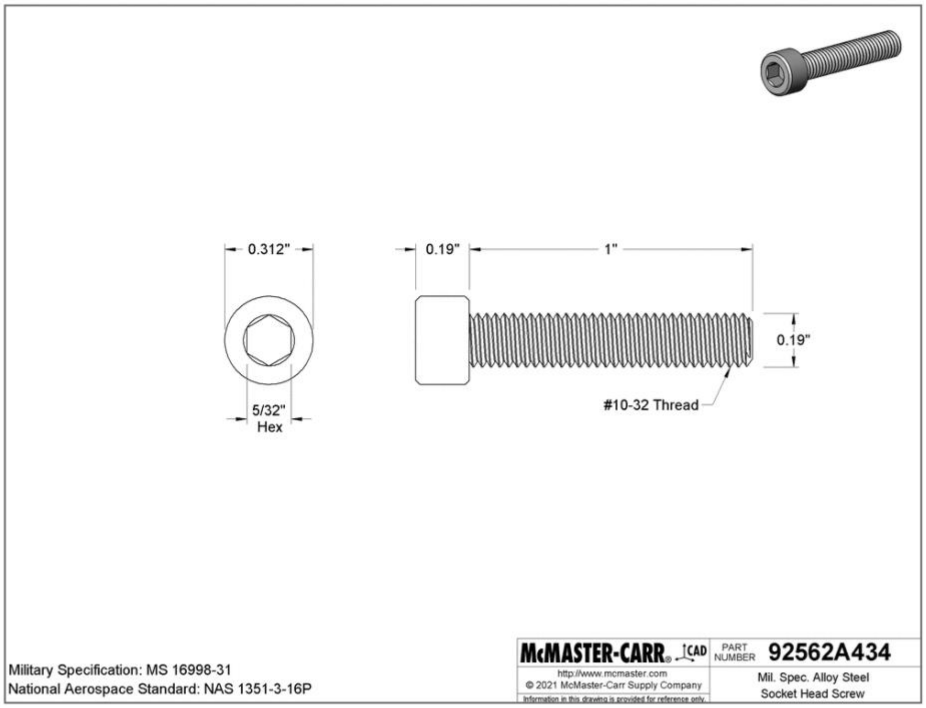
d_1	0.669 in	Shaft diameter
D	1.102 in	Housing bore diameter
b	0.276 in	Seal width

Application and operating conditions

Operating temperature	min. -40 °F
Operating temperature	max. 212 °F
Circumferential speed	max. 590.551 ft/min
Rotational speed	max. 3 900 r/min
Pressure differential	10.153 psi

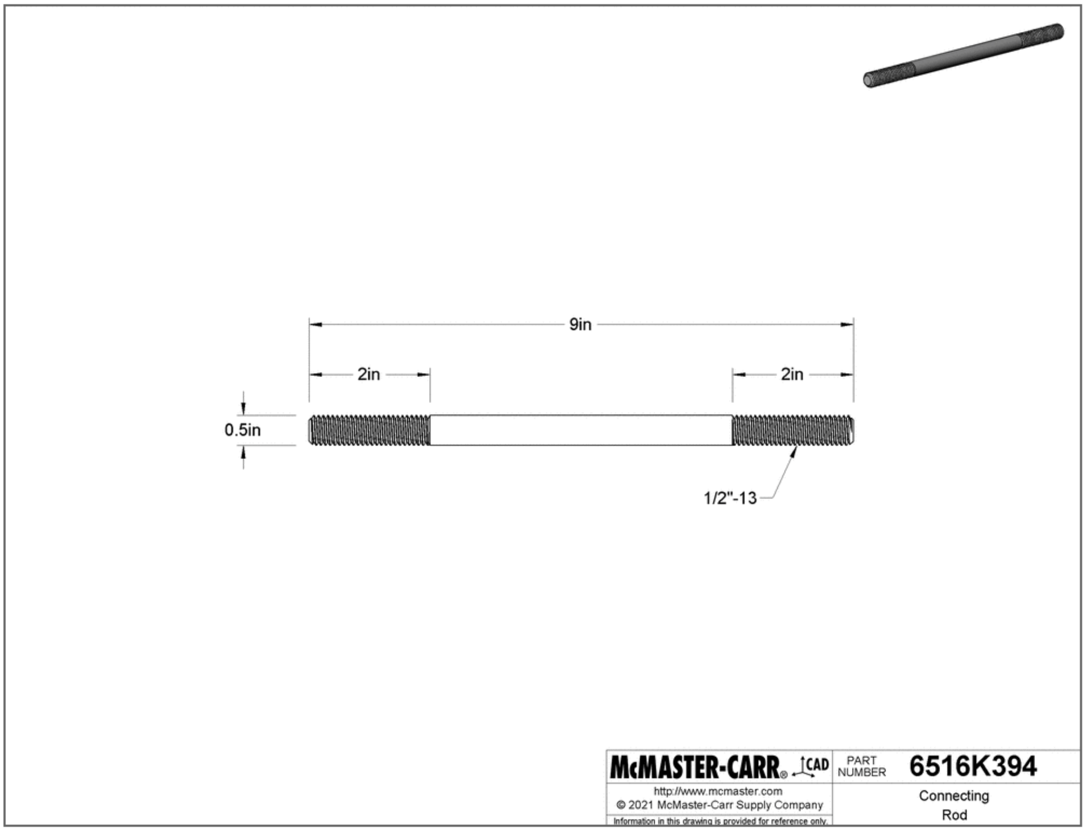
Screw - 92562A434

Mil. Spec. Alloy Steel Socket Head Screw, 10-32 Thread Size, 1" Long



Head Type	Socket
Socket Head Profile	Standard
Drive Style	Hex
System of Measurement	Inch
Thread Direction	Right Hand
Thread Size	10-32
Screw Size Decimal Equivalent	0.19"
Thread Type	UNRF
Thread Fit	Class 3A
Length	1"
Threading	Fully Threaded
Thread Spacing	Fine
Head	
Diameter	5/16"
Height	0.19"
Drive Size	5/32"
Material	Cadmium-Plated Alloy Steel
Tensile Strength	160,000 psi
Hardness	Rockwell C39
Specifications Met	MS16998-31, NAS1351-3-16P
RoHS	Not Compliant
REACH	Not Compliant
DFARS	Specialty Metals Compliant (252.225-7009)
Country of Origin	United States
USMCA Qualifying	No
Schedule B	731815.9000
ECCN	EAR99

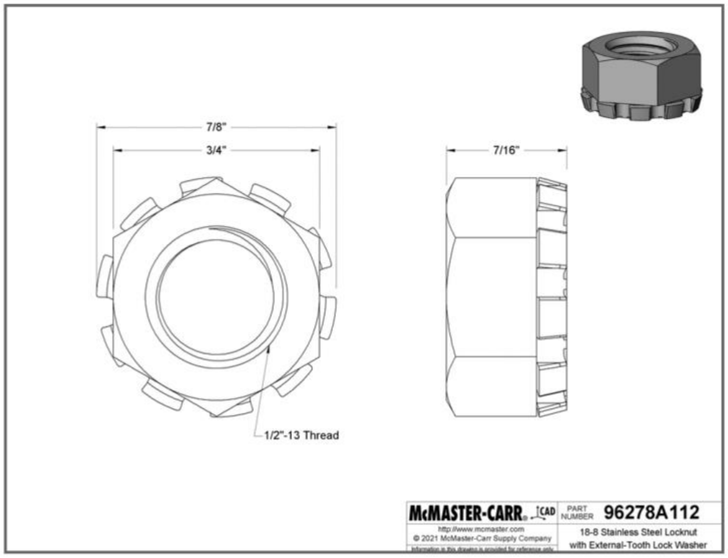
Connecting Rod - 18-8 Stainless Steel, 9" Overall Length, 1/2"-13 Thread



Overall Length	9"
Thread Size	1/2"-13
Thread Length	2"
OD	1/2"
Material	18-8 Stainless Steel
Rod Shape	Round
Threading	Both Ends Threaded
Gender	Male
Thread Direction	Right Hand
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/27/2024, 241 SVHC) Compliant
DFARS	Specialty Metals Compliant (252.225-7009)
Country of Origin	United States
USMCA Qualifying	No
Schedule B	731815.5000
ECCN	EAR99

18-8 Stainless Steel Locknut with External-Tooth Lock Washer

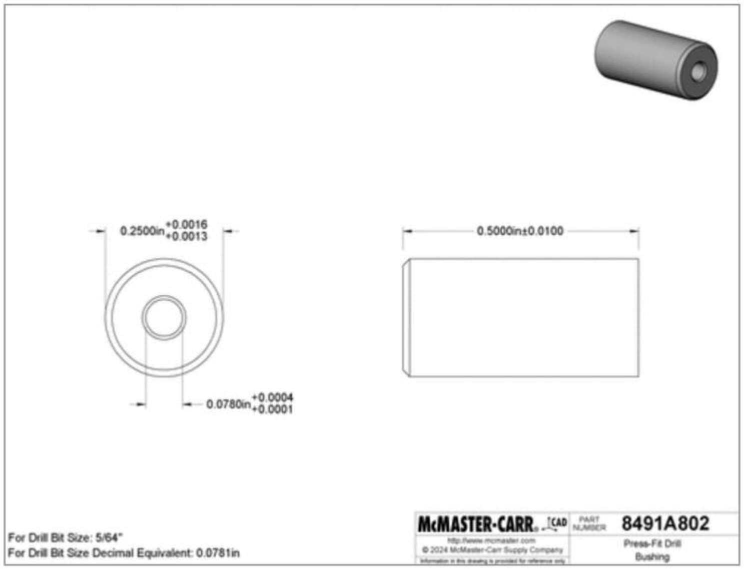
1/2"-13 Thread Size



Hex Nut Profile	Standard
Locking Type	Lock Washer
Thread Direction	Right Hand
Tooth Location	External
Washer Type	Tooth
Finished Material	18-8 Stainless Steel
Thread Size	1/2"-13
Width	3/4"
Height	7/16"
Washer	
OD	0.871"
Material	400 Series Stainless Steel
Drive Style	External Hex
Nut Type	Hex, Locknut
Performance	Vibration Resistant, Corrosion Resistant
System of Measurement	Inch
Thread Fit	Unified Standard Class 2B
Thread Spacing	Coarse
Thread Type	UNC
Country of Origin	China, Taiwan
DFARS Compliance	Not Specialty Metals Compliant
ECCN	EAR99
REACH Compliance	REACH (EC 1907/2006) (06/27/2024, 241 SVHC) Compliant
RoHS Compliance	RoHS 3 (2015/863/EU) Compliant
Schedule B Number	731816.0000
USMCA Qualifying	No

Dowel Pin - Press-Fit Drill Bushing

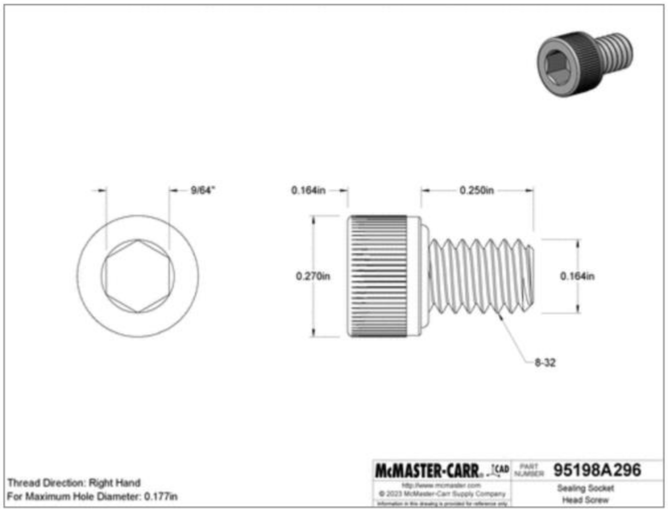
0.0781" ID, 1/4" OD, 1/2" Long



Installation Type	Permanent
Drill Bushing Style	Smooth
ID	0.0781"
OD	1/4"
Length	1/2"
For Drill Bit Size	5/64"
For Drill Bit Size Decimal Equivalent	0.0781"
Tolerance	
ID	0.0001" to 0.0004"
OD	0.0013" to 0.0016"
Length	-0.01" to 0.01"
Drill Bushing Type	P
Material	Steel
Hardness	Rockwell C61
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/14/2023, 235 SVHC) Compliant
DFARS	Specialty Metals COTS-Exempt
Country of Origin	United Kingdom
Schedule B	846620.8035
ECCN	EAR99

Sealing Socket Head Screw

18-8 Stainless Steel with Buna-N O-Ring, 8-32 Thread, 1/4" Long



Thread Size	8-32
Length	1/4"
Threading	Fully Threaded
Thread Spacing	Coarse
Head	
Diameter	0.27"
Height	0.164"
Drive Size	9/64"
For Maximum Hole Diameter	0.177"
Maximum Pressure	1,500 psi
O-Ring Material	Buna-N Rubber
O-Ring Temperature Range	-40° to 225° F
Material	18-8 Stainless Steel
Hardness	Rockwell B85
Tensile Strength	100,000 psi
Screw Size Decimal Equivalent	0.164"
Thread Type	UNC
Thread Fit	Class 3A
Thread Direction	Right Hand
Head Type	Socket
Socket Head Profile	Standard
Screw Features	Sealing
Drive Style	Hex
Specifications Met	ASME B18.3, DIN 912, ISO 4762
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (01/17/2022, 223 SVHC) Compliant
DFARS	Not Specialty Metals Compliant
Country of Origin	United States
USMCA Qualifying	No
Schedule B	731815.9000
ECCN	EAR99

References

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