MECH 420
Design of Machine Elements Major Project
Fall, 2013

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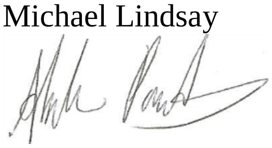
Dear Professor Le,

Enclosed is the report on the design of a single-stage gear box, including the Excel, and SolidWorks verification of its feasibility. This reports has five parts that were analyzed and they are gear design, shaft design, mechanical key design, bearing design, and housing design. This was then put into a final report with recommendations and conclusions.

After designing the gear box, it was concluded that the gear box design meets all the design specifications and it is safe for production. This can be verified with the Excel spreadsheets and SolidWorks. We look forward to your review and honest criticism.

Sincerely,

Michael Lindsay



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DESIGN OF MACHINE ELEMENTS

MECH420 Major Project

Single-stage Gear Box

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Executive Summary

Earlier this fall of 2013, engineers were able to hone their design skills by completing a design verification on a car jack. Now for the closure of this fall 2013, engineers were presented with another task. Along with the help of Dr. Le, engineers were required to use their mechanics of materials skills and statics knowledge in order to design a single-stage gear box that meets the design specifications. The design specifications were a delivered power of 20 hp, gear ratio of 7.2, 1% variation, input speed of 1800 rpm, pressure angle of the gear of 20 degrees, design life of 4,000 hours, factor of safety of 3.25 for the static design, factor of safety of 1.25 for the fatigue design, and reliability index of 0.90. By meeting the design specifications one would have the opportunity of reproducing one's gear box.

Using Excel, the engineers were able to do a gear analysis in order to come up with two gears that transfer power from one gear to another. Moreover, one did a shaft design by performing static and fatigue analysis through Excel in order to work with the bearing and key design. In addition, a mechanical key design was performed in order to transfer the power from the gears to the shaft or vice versa. Furthermore, a bearing analysis was performed in order to support the shaft and provide less friction when the shaft rotated. Finally, a housing design was created in order to hold all the parts.

After the analysis was made, the gears, shafts, bearings, mechanical key, and housing were created through SolidWorks in order to get a visual understanding how the gear box was put together. In addition, free body diagrams were developed in order to demonstrate the knowledge of structural analysis. The forces on each component were identified so that the maximum load was able to be determined. This helps identify if anything goes wrong, the location in which it failed.

Finally, a Finite Element Analysis was performed on the shaft in order to verify where in the parts the stress was most concentrated. While this seemed to be a simple design, the following sections show detailed descriptions on what was performed in order to ensure that the gear box could be put into the production line

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3. Introduction

A gear box is one of the most important designs that a Mechanical Engineer must learn upon graduation. The purpose of a gear box is to transfer energy from one device to another in order to increase torque while reducing speed. Mechanical Engineers must learn this because a gearbox contains several components that on a daily basis most Mechanical Engineers will have to deal with. For example, one of the components Mechanical Engineers deal with is determining the types of gears that transfer the power from one gear to another. This is important to know in order to come up with a design that could sustain the power delivered. After designing the gears Mechanical Engineers are then presented with the problem of modifying the gears based on the size of shaft. This means that given the dimensions of the gear and shaft one has to compromise by making either the shaft or the gears smaller than they were initially intended to be. Afterwards, a bearing analysis was created in order to prevent any type of friction from occurring when the shaft rotates. This is important to know in order to make the gear box long lasting and efficient. This now give rise to developing a mechanical key that would help transfer the power from the gears to the shaft or vice versa. This is important to know so that Mechanical Engineers are able to understand that if any goes wrong one would be able to identify where the problem is located. Not only are these concepts important to understand, but it is also important to understand what are design specifications. The following are the design specifications that were followed in order to create the design.

4. Design Specifications

- Delivered power of 20 hp with driven device that is treated as "light-shock" loadings
- Input speed of 1800 rpm
- Gear Ratio of 7.2 with 1% variation
- Pressure Angle of 20^0
- Design Life of 4,000 hours
- Factor of safety of 3.25 for the static design of the shaft
- Factor of safety of 1.25 for the fatigue design of the shaft
- Reliability index of 0.9

These design specifications were the guidelines that indicated if the design was being designed as specified. The design specifications affected how the gears, shafts, mechanical key, bearings, and housing designs were created. The following sections have detail descriptions of the different parts that create a gearbox.

5. Design and Analysis

5.1. Gear Design

The first component in creating a gear box are the gears. Gears are rotating objects that transfer energy from one place to another. In order to create a gear one must analyze the way it look so that it close to the gearbox layout. The figure (Appendix: Major Project Worksheet Instructions) shows that there are two gears present. One is the pinion which is a small size gear that is next to a large gear. The gears would transfer power from one gear to another. The following is a spreadsheet created through Excel in order to determine the shape of the gears:

	A	B	C	D	E	F	G	H	I	J	K	L	M	N
1		Light-Shock Loadings, Uniform								Table From Lecture 13 Slide 59				
2	Delivered Power	20	hp	Designed Power P_{des}	30	hp	Base Circle Diameter D_b	1.996847		Np	18			
3	Input Speed $n_{input/pinion}$	1800	rpm	Diametrial Pitch P_d	8	tooth/in	Maximum Output Speed n_{min}	252.5253	rpm	NG	48.29907			
4	Gear Ratio	7.2		Tooth Thickness t	0.19635	in	Minimum Output Speed n_{Max}	247.5248	rpm	Actual VR	2.683282			
5	Overload Factor K_o	1.5		Face Width F	1.5	in	Nominal Output Speed n_{Nom}	250	rpm	ng	32			
6	Variation	1	%	Minimum teeth # for Pinion N_P	17	gears	Required Reduction Ration e_{Min}	7.272						
7	Variation	0.01		Maximum teeth # for Gear N_G	122	gears	Required Reduction Ration e_{Max}	7.128						
8	Variation*Gear Ratio	0.072		Verifying Gear Ratio	7.176471		Required Reduction Ration e_{Nom}	7.2						
9	Gear Ratio Min	7.128		Addendum a	0.125	in	One Stage emax=Gear Ratio	7.2						
10	Gear Ratio Max	7.272		Dedendum b	0.15625	in	Two Stages emax=Gear Ratio	51.84						
11	Pressure Angle ϕ	20	°	Clearance c	0.03125	in	Equal Reduction Ration R1=R2	2.683282						
12	Pressure Angle ϕ	0.349066		Pitch Diameter D for pinion	2.125	in								
13	Recommended AGMA Quality #	5		Pitch Diameter D for gear	15.25	in								
14	Life of Gears and Bearings	4000	hours	Outside Diamter D_0	2.375	in								
15	Shaft (Static Design) Factor of Safety	3.25		Root Diameter D_R	1.8125	in								
16	Shaft (Fatigue Design) Factor of Safety	1.25		Working Depth h_k	0.25	in								
17	Index Reliablity (Gear,Shaft,Bearing)	0.9		Whole Depth h_t	0.28125	in								
18	Hub Length for Input Shaft	> 1.5	in	Center Distance C	8.6875	in								
19	Hub Length for Output Shaft	2	in											
20	Gaps between gears and station part	> 0.5	in											
21	Hub Diameter	3	in											
22	Hub Length	2	in											
23														
24														
25														
26														
27														

Gear Analysis | Gear Material | Input Shaft Analysis | Output Shaft Analysis | Input Bearing Analysis | Output Bearinç ...

Figure 1.0: Gear Analysis

The first parameter in determining the size of the gear is the designated power (P_{Des}). The following equation was used in order to solve for P_{des}:

$$P_{des} = K_0 P$$

Where K_0 is the designed factor and P is the power delivered. This would help determine the diametrical pitch using **Table From Lecture 16 Slide 26.** Based on the calculations obtained, it was concluded that the diametrical pitch P_d was 8 tooth/inch for each gear.

Another parameter that was used was the tooth thickness t and that was calculated by using the following formula:

$$t = \frac{\pi}{2 * P_d}$$

Furthermore, the face width, which is the width of the tooth, was calculated using the following equation:

$$F = \frac{12}{P_d}$$

This was then followed by verifying the gear ratio so that one could determine the number of teeth for each gear. Based on the minimum gear ratio of 7.128, it was concluded that the pinion gear had to have 17 teeth (N_P). Whereas, the larger gear had maximum gear ratio of 7.272 which means the larger gear had about 122 teeth (N_G). Based on the number of teeth the following equations were used in order to determine the pitch diameter of each gear:

$$D_{Pinion} = \frac{N_P}{P_d}$$

$$D_{Gear} = \frac{N_G}{P_d}$$

By determining the dimensions of the gears the next question that arises is what the material the gears are made out of? The next part of the gear analysis analyzes the AGMA Bending Stress, Allowable Bending Stress, AGMA Contact Stress, Allowable Contact Stress, and Hardness so that the material is determined.

Mike add your excel spread sheet here.

AGMA Bending Stress:

In order to choose the material the gears were made out of, the AGMA bending stress and the AGMA contact stress first had to be calculated. The required hardness was then calculated according to the required bending and contact stress in order to determine the proper material with the proper heat treatment to satisfy the required hardness. The AGMA bending stress was calculated using the equation:

$$\sigma = W_t * K_O * K_V * K_S * \frac{P_d}{F} * \frac{K_M * K_B}{J},$$

Where W_t is the tangential force, K_O is the overload factor, K_V is the dynamic factor, K_S is the size factor, P_d is the diametral pitch, F is the face width of the gear, K_M is the load distribution factor, K_B is the rim thickness factor and J is the geometry factor for bending stress for both the pinion and the gear.

The first part of the equation is the tangential force on the gear, W_t. This was calculated by dividing the power supplied to the gears, P by the pitch-line velocity, V_t with the pitch-line velocity being equal to pi multiplied by the pitch diameter and by the input pinion speed.

$$V_t = (\pi * D_p * n_p)/12$$

$$W_t = 33000 * \frac{P}{V_t}$$

With 12 and 33000 being conversion factors from inches to feet and horsepower to feet per minute respectively.

K_O is the overload factor which is the probability that application-specific conditions such as vibrations or shocks would cause peak loads to be greater than the tangential force, W_t being applied to the teeth on the gears. They are determined by the power source and for this gearbox, the overload factor was given as 1.5.

K_V is the dynamic factor which is based upon the fact that the actual load is high than the transmitted load alone and determined by the gear quality and pitch line velocity values. The dynamic factor was calculated by the equation:

$$K_V = \left[\frac{A+\sqrt{V_t}}{A}\right]^B$$

With A and B being values that are calculated through other equations with the gear quality number. A was calculated by the equation:

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

And B was calculated by the equation:

$$B = 0.25(12 - Q_v)^{\frac{2}{3}}$$

In this equation, Q_v is the gear quality number and for this gearbox the given gear quality number is 5.

K_S is the size factor and as the AGMA indicates, it can be taken as 1 for most gears but for gears with a large face width, a value greater than 1 is recommended. For this gearbox, the face width was relatively small so a size factor of 1 was used.

P_d is the diametral pitch of the gears and a value of 8 teeth per and inch was given as a design specification for the gearbox.

F is the face width which is the surface of a gear tooth from the pitch circle to the outside circle of the gear. For this gear box, the face width was calculated using the equation:

$$F = \frac{12}{P_d}$$

Which gave the face width a value of 1.5 inches.

K_M is the load distribution factor. The load distribution factor was calculated using the equation:

$$K_M = 1.0 + C_{pf} + C_{ma}$$

Where C_{pf} and C_{ma} are the pinion proportion factor and the mesh alignment factor respectively. C_{pf} is calculated using the equation:

$$C_{pf} = \frac{10}{P_d} - .025, \ when\ F < 1$$

And by:

$$C_{pf} = \frac{10}{Pd} - .0375 + .0125F\ , when\ 1 < F < 15$$

C_{ma} was calculated using the equation:

$$C_{ma} = .247 + .0167F - .765 * 10^{-4}F^2$$

K_B is the rim-thickness factor which Accounts for bending of rim on a gear that is not solid. The equation for calculating the rim-thickness factor is:

$$K_B = \begin{cases} 1.6 \ln(\frac{2.242}{m_B}) & m\_B < 1.2 \\ 1 & m\_B > 1.2 \end{cases}$$

J is the geometry factor for bending stress for both the pinion and the gear which is a modification factor that is based upon both the tooth geometry and the stress concentration. In order to calculate it for the pinion and the gear, **Figure 14-6 (lecture 14 slide 45)** was used in conjunction with the number of teeth for both the pinion and gear.

Allowable Bending Stress:

The allowable bending stress number equations for both the pinion and gear are used to calculate the allowable bending stress number, $S_{tp}$and S_{tg}, for the material that the gears will be made out of in order to make sure that the material is able to withstand the bending stress caused by the pinion and gear. The equation for the allowable bending stress for the pinion is:

$$S_{tp} > (S_F * K_R * K_T * \sigma_P)/Y_{NP}$$

And the equation for the allowable bending stress for the gear is:

$$S_{tg} > (S_F * K_R * K_T * \sigma_G)/Y_{NG}$$

Where S_{tp} and S_{tg} are the allowable bending stress numbers for the pinion and gear respectively, S_F is the factor of safety for bending, K_R is the reliability factor, K_T is the temperature factor, σ_P and σ_G are the bending stresses for the pinion and gear respectively, and Y_{NP} and Y_{NG} are the stress cycle factors for the pinion and the gear respectively.

S_F is the first variable in the equation for the allowable bending stress number and is the factor of safety for bending for the pinion and the gear. For this gearbox, the factor of safety was chosen to be 1.25.

K_R is the reliability factor and it was calculated to be .85 from table in the appendix using the reliability of .90 which was specified in the design specifications.

K_T is the temperature factor and was gives as a value of 1.

σ_P and σ_G are the bending stresses for the pinion and gear respectively and were calculated using the bending stress equation shown above.

Y_{NP} and Y_{NG} are the stress cycle factors for the pinion and the gear respectively. They were calculated using the equation:

$$Y_N = 1.3558(N)^{-.0178}$$

Where N is the number of cycles of loading and was be calculated using the equation:

$$N = 60 * L * n * q$$

Where L is the design life in hours, n is the rotational speed in rpm, q is the number of load applications per revolution which for this gear box is 1 and 60 is the conversion factor from hours to minutes. The equation for the stress cycle factors for the pinion and the gear was obtained from **Figure 14-14 (lecture 14 slide 27)** in the appendix.

AGMA Contact Stress:

The AGMA contact stress is calculated in a similar manner to the AGMA bending stress but not exactly. It was calculated using the equation:

$$\sigma = C_p\sqrt{W_t * K_O * K_V * K_S * \left(\frac{K_M}{d_p * F}\right) * \frac{C_f}{I}}$$

Where σ_P is the contact stress for the pinion, C_p is the elastic coefficient, W_t is the tangential force, K_O is the overload factor, K_V is the dynamic factor, K_S is the size factor, K_M is the load distribution factor, d_p is the pitch diameter of the pinion, F is the face width, C_f is the surface condition factor, and I is the geometry factor for pitting resistance.

C_p is the elastic coefficient and for this gearbox, steel was chosen based on the calculated values for the bending stress and from the data from table in the appendix .

C_f is the surface condition factor which accounts for detrimental surface finish. The value of 1 was used for it based on the fact that the value of 1 is used for normal commercial gears.

I is the geometry factor for pitting resistance which was determined to be .108 by using graph **Figure 9-23 (lecture 14 slide 46)** in the appendix in conjunction with the gear ratio of 7.2 and the number of teeth on the pinion which is 17.

Allowable Contact Stress:

The allowable contact stress number equation for both the pinion and gear are used to calculate the allowable contact stress number, $S_{CP}$and S_{CG}, for the material that the gears will be made out of in order to make sure that the material is able to withstand the bending stress caused by the pinion and gear. The equation for the allowable contact stress for the pinion is:

$$S_{CP} > \frac{S_H * K_T * K_R}{Z_{NP} * C_H}$$

And the allowable contact stress for the gear is:

$$S_{CG} > \frac{S_H * K_T * K_R}{Z_{NG} * C_H}$$

Where S_{CP} and S_{CG} are the allowable contact stress numbers for the pinion and gear respectively, S_H is the factor of safety for contact, K_T is the temperature factor, K_R is the reliability factor, Z_{NP} and Z_{NG} are the stress cycle factors for Contact Stress for the pinion and the gear respectively and C_H is the hardness-ratio factor.

S_H is the factor of safety for contact for the pinion and the gear and for this gearbox, the factor of safety was chosen to be 1.25.

Z_{NP} and Z_{NG} are the stress cycle factors for Contact Stress for the pinion and the gear respectively. They were calculated using the equation:

$$Z_N = 1.4488(N)^\wedge - .023$$

Where N is the number of cycles of loading and was be calculated using the equation:

$$N = 60 * L * n * q$$

Where L is the design life in hours, n is the rotational speed in rpm, q is the number of load applications per revolution which for this gear box is 1 and 60 is the conversion factor from hours to minutes. The equation for the stress cycle factors for the pinion and the gear was obtained from graph **Figure 14-15 (lecture 14 slide 28)** in the appendix.

C_H is the hardness-ratio factor which accounts for the increase in the gear capacity with regards to the pitting resistance. For the gearbox, the hardness-ratio factor was determined to be 1.

Hardness:

After all these values were calculated, the hardness was then determined for the pinion and the gear by the equation:

$$S_C = 349H_B + 34300$$

From graph **Figure 14-5 (lecture 14 slide 20)**.

Based on the hardness the material of both the pinion and the gear is AISI 1030 Q&T at 400 degrees Fahrenheit.

5.2 Shaft Analysis

For the design of the shaft, the first step was to choose a layout. One of the shaft layouts that was considered for the gearbox design was the shoulder layout. In this layout, there are two bearings and two shoulders on each side which block axial movement. Moreover the design layout has a rotating ring which is used in order to keep the gear in place.

Another shaft layout is the Shoulder Shoulder Spacers layout. In this layout there are two bearings on each side and one shoulder and spacer on one side. This varies from layout 1 as having a spacing enables the gear to move at a faster rate thus transfer power quicker.

Based on an extensive analysis it was concluded that shaft design layout #2 was the layout chosen for the final design as the group concluded that more power transfer would create a more efficient gearbox. This layout was used for both the input and output shafts. The chosen material for the shafts is AISI 1030 steel quenched and tempered at 400 °Fahrenheit. The final layout of the shaft is shown below:

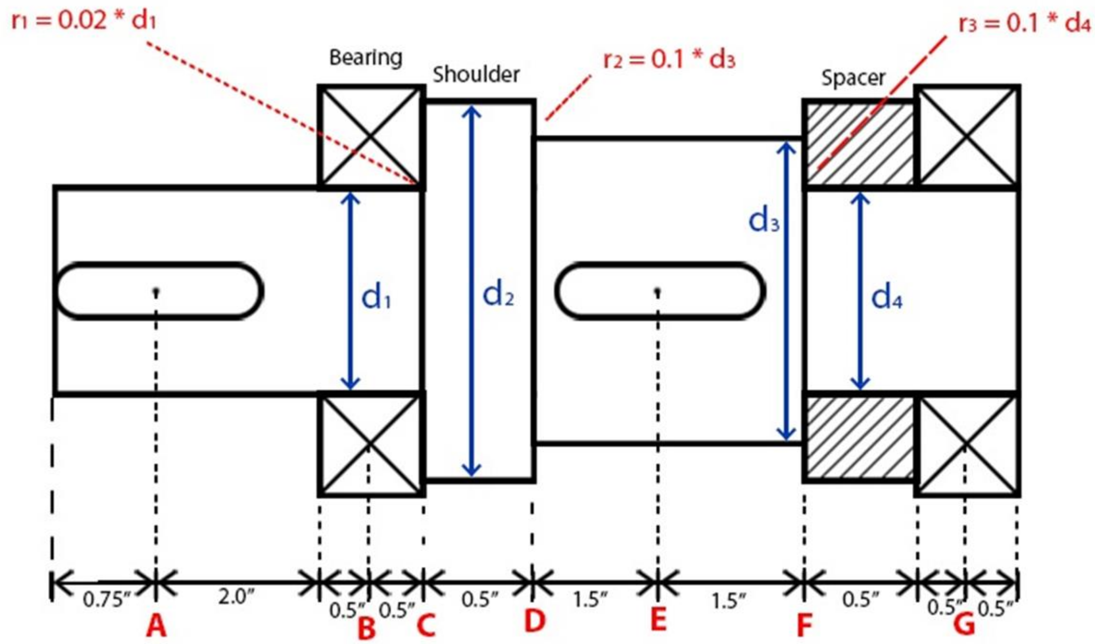


Figure: Shaft layout

The next step in designing the shaft was to create torsion, shear force, and moment diagrams of the shaft. To do this, the velocity was calculated based on the pitch diameter and input speed:

$$V_t = \frac{\pi d_p n}{12} = \frac{\pi \times 2.125 \times 1800}{12} = 1001.383 (\text{ft/min})$$

The velocity and output power were used to calculate the radial and tangential force on the shaft:

$$W_t = 33000\frac{P}{V_t} = 33000\frac{20}{1001.383} = 659.0887 (\text{lbs})$$
$$W_r = W_t \tan\phi = 659.0887 \tan 20^0 = 239.8887 (\text{lbs})$$

The radial and tangential force on the shaft were then used to calculate the total force and torque on the shaft at point E:

$$F_R = \sqrt{(W_t)^2 + (W_r)^2} = \sqrt{(659.0887)^2 + (239.8887)^2} = 701.3876 (lb)$$
$$T = W_t \frac{d_p}{2} = 659.0887 \times \frac{2.125}{2} = 700.2817 (lb - in)$$

The numbers used in the above calculations are for the input shaft, but the output shaft was analyzed using the same method and equations. The final torsion, shear force, and moment diagrams for both shafts are shown below:

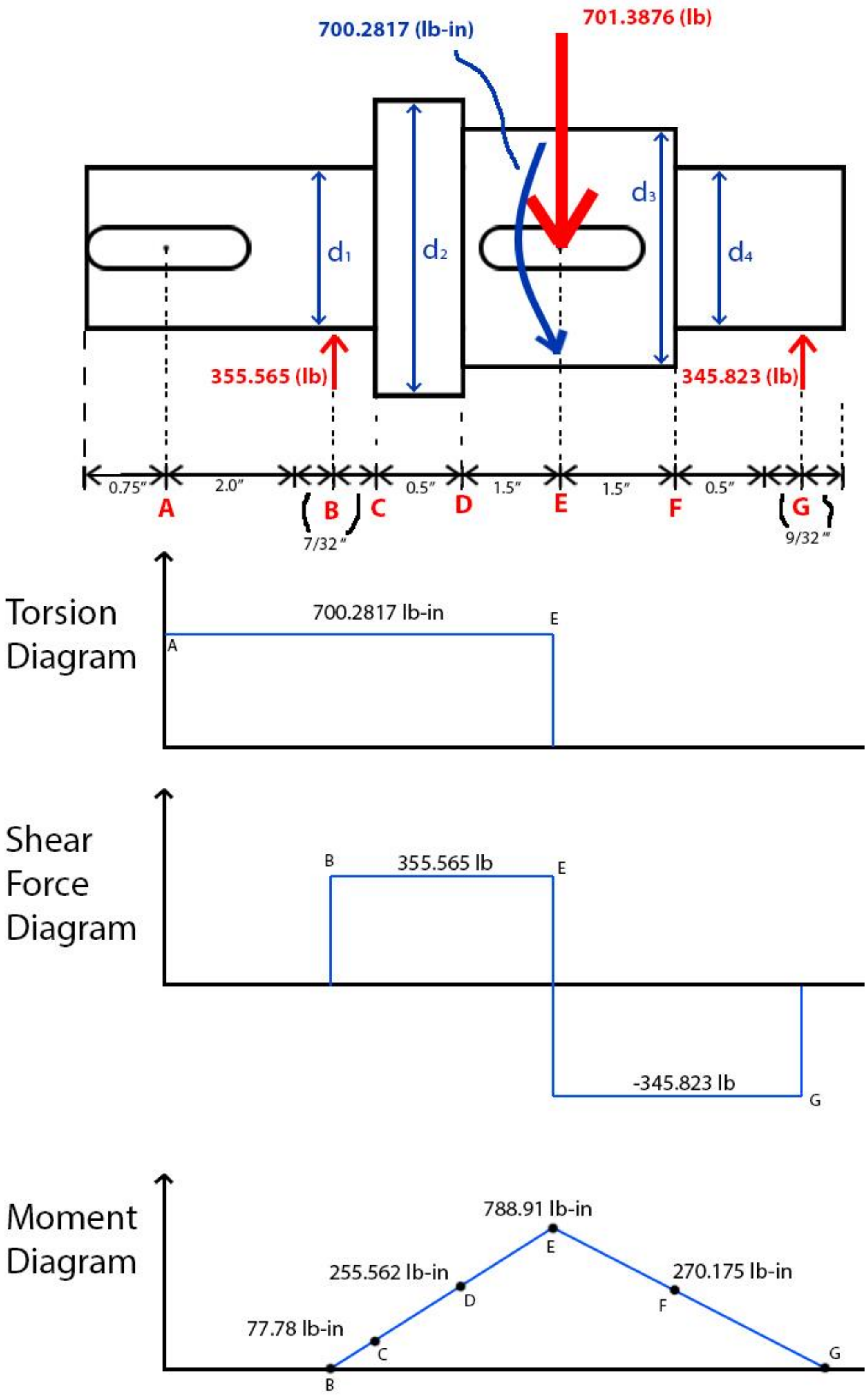


Figure: Input shaft torsion, shear force, and moment diagrams

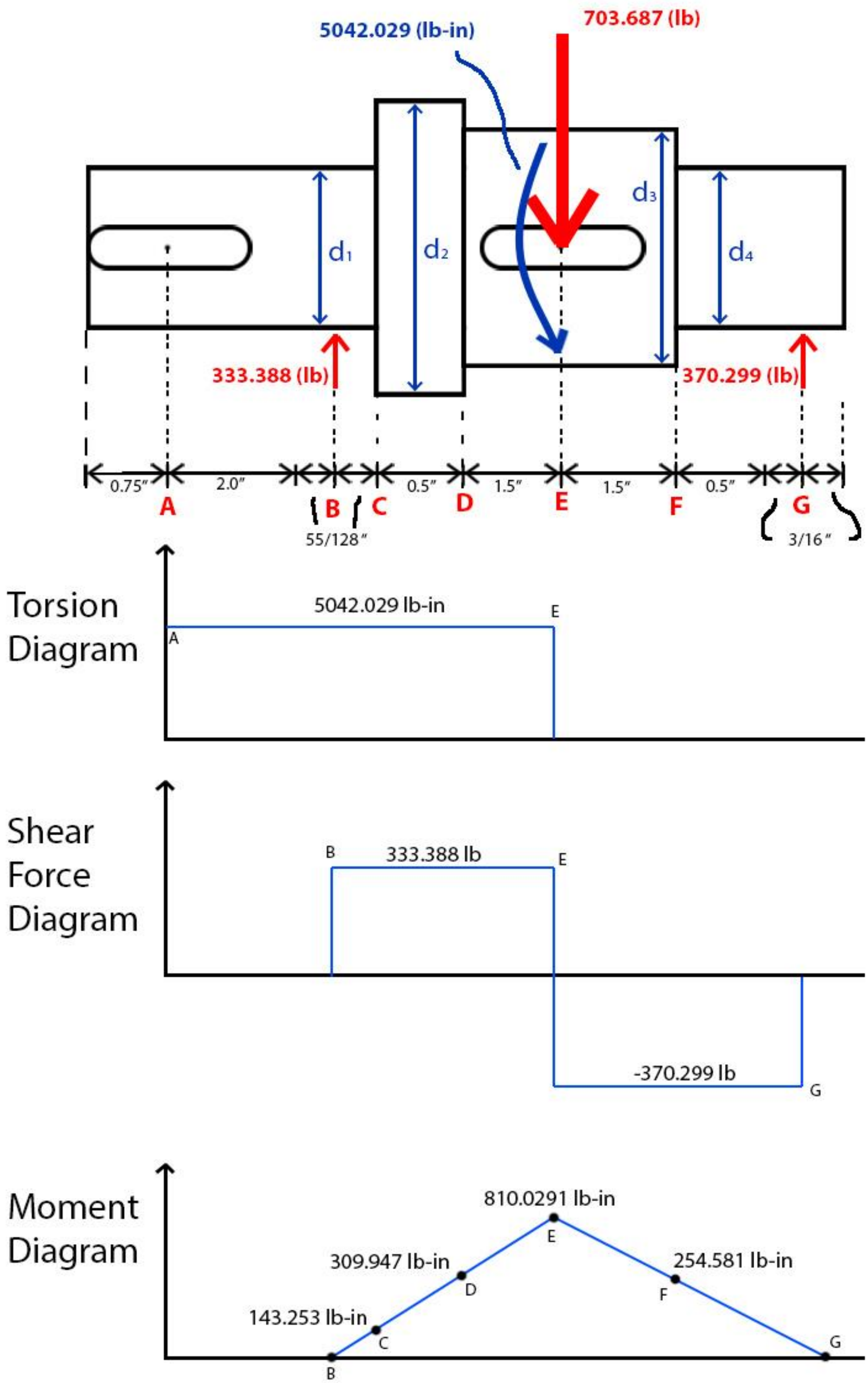


Figure: Output shaft torsion, shear force, and moment diagrams

Using these diagrams, the required minimum shaft diameters were determined based on static analysis. This was done using the following equation:

$$d=\left[\frac{16n_s\sqrt{4(K_tM_a)^2+3(K_{ts}T_m)^2}}{\pi S_y}\right]^{1/3}$$

In this equation, n_s is the factor of safety, K_t is the stress concentration for bending moment under static loading, M_a is the bending moment, K_{ts} is the stress concentration for torsion, T_m is torsion, and S_y is the yield strength of the material. For the static analysis, a factor of safety of 3.25 was used. The values of K_t and K_{ts} were determined using Table 7-1, which can be found in the appendix.

d_1 was analyzed at two critical areas: the keyseat at A and the fillet at C. d_2 had to be at least 1/8th of an inch larger than d_1 and d_3. d_3 was analyzed at critical areas D and E. d_4 only had to be analyzed at the fillet at F. The minimum diameters based on static analysis are as follows:

d_1 Critical area A	1.992168	in
d_1 Critical area C	1.496316	in
d_2	larger than d_1 and d_3	in
d_3 critical area D	1.318042	in
d_3 critical area E	1.663651	in
d_4	0.532286	in

Figure: Input shaft diameters

d1 Critical area A	1.031688	in
d1 Critical area C	0.777858	in
d2	larger than d1 and d3	in
d3 critical area D	0.705596	in
d3 critical area E	0.95277	in
d4	0.542939	in

Figure: Output Shaft Diameters

The next step in designing the shaft was to calculate the minimum diameter based on fatigue analysis. This was done using the following equation:

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

In this equation, n_f is the factor of safety, K_f is the fatigue stress concentration factor for bending, K_{fs} is the fatigue stress concentration factor for torsion, S_e is the endurance limit of the component for cyclic bending stress, and S_{ut} is the ultimate strength of the material. For the fatigue analysis, the factor of safety is 1.25.

In order to calculate S_e, the following set of equations was used:

$$S_e = k_a k_b k_c k_d k_e k_f S_e^{'};$$

$$S_e^{'} = 0.5 S_{ut};$$

$$k_a = 2.70 \times S_{ut}^{-0.265};$$

$$k_b = 0.879 d^{-0.107};$$

$$k_c = 1;$$

$$k_d = 1;$$

$$k_e = 0.814;$$

$$k_f = 1;$$

Here, k_a is the surface factor for machined surface, k_b is the size factor, k_c is the load factor, k_d is the temperature factor, k_e is the reliability factor, and k_f is the miscellaneous factor.

K_f and K_{fs} were calculated using the following equations:

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

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

For the analysis at the keyseats, K_t and K_{ts} are the same as those used for the static analysis, and q is equal to 0.8. For the analysis at the fillets, K_t and K_{ts} are found using a stress concentration resource called *eFatigue*, and q was determined using the following equations:

$$q = 1 \Big/ \left(1 + \frac{\sqrt{a}}{\sqrt{r}}\right); \text{ where r is fillet radius}$$

$$\sqrt{a}_{bending} = 0.246 - 3.08(10^{-3})S_{ut} + 1.51(10^{-5})S_{ut}^2 - 2.67(10^{-8})S_{ut}^3;$$

$$\sqrt{a}_{torsion} = 0.190 - 2.51(10^{-3})S_{ut} + 1.35(10^{-5})S_{ut}^2 - 2.67(10^{-8})S_{ut}^3;$$

The equation for $\sqrt{a}_{bending}$ is used when calculating K_f, and the equation for $\sqrt{a}_{torsion}$ is used when calculating K_{fs}.

Determining the final dimensions of the input and output shafts was an iterative process. First, the fatigue analysis was done using the diameters determined by static analysis. The new diameters found in this fatigue analysis were then used to determine the keyseat dimensions and which bearings to use. When this was done, the shaft was modified according to the keyseat and bearing analysis. The static and fatigue analysis were then repeated using these modifications. This entire process was repeated until a set of diameters was found for the shafts which satisfied static, fatigue, bearing, and keyseat analysis. This process was done in excel to keep the calculations organized and to save time.

Input Shaft Dimensions based on static loading			Fatigue Design for Shaft					
d_p	2.125	in	S_e'	61500	psi	d_1 based on static loading	1.04	inches
V_t	1001	ft/min	Reliability	90	%	d_2 based on static loading	1.17	inches
W_t	659.1	lbs	d_1, section A-C			d_3, section D		
W_r	239.9	lbs	k_a	0.75429		k_a	0.7543	
F_R	701.4	lbs	k_b	0.5696		k_b	0.692	
T	700.3	lb-in	k_c	1		k_c	1	
AISI 1030 Q & T 400 F, S_{ut}	123000	psi	k_d	1		k_d	1	
AISI 1030 Q & T 400 F, S_y	95000	psi	k_e	0.897		k_e	0.897	
K_t key	2.14	Table 7-1	k_f	1		k_f	1	
K_{ts} key	3	Table 7-1	S_e	23701.6	psi	S_e	28807	psi
K_t sharp fillet	2.7	Table 7-1	r_1 fillet at C	0.03	in	r_2 fillet at D	0.125	in
K_{ts} sharp fillet	2.2	from Table 7-1	K_t at C	2.23	Calculated using efatigue	K_t at D	1.7	Calculated using efatigue
K_t rounded fillet	1.7	from Table 7-1	K_{ts} at C	1.82	Calculated using efatigue	K_{ts} at D	1.36	Calculated using efatigue
K_{ts} rounded fillet	1.5	from Table 7-1	$\sqrt{a}$ for bending	0.04592		$\sqrt{a}$ for bending	0.0459	
d_1 Critical area A	#####	in	$\sqrt{a}$ for torsion	0.03583		$\sqrt{a}$ for torsion	0.0358	

Fatigue Design for Shaft						Final diameters based on bearing design			width of bearing	
d_3 based on static loading	0.96	inches	The fatigue analysis does not call for larger diameters than the static analysis, so the diameters from the static analysis can be used			d_1	1.5	in	7/16	in
d_4 based on static loading	0.543	inches				d_2	1.635	in		
d_3, section E			d_4, section F			d_3	1.25			
k_a	0.7543		k_a	0.7543		d_4	1.000	in	9/16	in
k_b	0.6923		k_b	0.879						
k_c	1		k_c	1						
k_d	1		k_d	1						
k_e	0.897		k_e	0.897						
k_f	1		k_f	1						
S_e	28807	psi	S_e	36576	psi					
q	0.8		r_3 fillet at F	0.1	in					
K_t at E	2.14	from Table 7-1	K_t at F	1.68	Calculated using efatigue					
K_{ts} at E	3	from Table 7-1	K_{ts} at F	1.35	Calculated using efatigue					
K_f at E	1.912		$\sqrt{a}$ for bending	0.0459						
K_{fs} at E	2.6		$\sqrt{a}$ for torsion	0.0358						

The final diameters of the shaft are as follows:

d1	1.5	in
d2	1.635	in
d3	1.25	in
d4	1.000	in

Figure: Input shaft diameters

d_1	2.25	in
d_2	2.375	in
d_3	1.687	in
d4	1.125	in

Figure: Output shaft diameters

These diameters were then used in order to determine the size of the mechanical key..

5.3 Mechanical Key Design

Add your part here Patricia.

5.4 Bearing Design

Before starting with the bearing analysis, one must define the application of bearings. The main design purpose of the bearings is to locate the shaft and provide less friction shaft rotation. Since there is an input and output shaft, and the shaft layout required two bearings for each shaft, meaning one had to pick four bearings from the website McMaster-Carr. To choose the proper bearings, it was recommended to use the bearings using the ABEC 1 tolerances. ABEC 1 are “dimensional tolerance standards in hard-to-find inch sizes, these quiet-running, electric-motor-quality bearings handle radial (perpendicular to the shaft) loads and small amounts of angular misalignment. Temperature range is –40° to +248° F” (Mc-Master-Carr). For both shafts one ended up choosing 2/4 bearings using the ABEC 1 tolerances. The following is a detailed analysis that describes how one came to the conclusion that different kinds of bearings are needed to be introduced in order for the design specifications to be met (Note: This procedure is the same for the input bearings and output bearings).

The first equation used was to determine the design life (L_D) of the bearings. The following equation was used to determine the design life:

$$L_D = 60 * h * n$$

Where *h* is the design life of the gears and bearings, and *n* is the input speed. Not only was L_D determined, but the department head informed one that L_D=L_{D10} making it simpler to solve for other necessary parameters. For example, now was able to calculate the design life ratio of L_D and L_{10} named x_D. The following equation was used in order to calculate x_D:

$$x_D = \frac{L_D}{L_{10}}$$

This was followed by selecting either a tapered or ball and straight roller bearing. Based on the choice of the bearing, the values from the table on **Lecture 18 Slide 26** was used in order to obtain the values L_{10},x_0, ɵ, and b. Another parameter that the department gave to criticism was a_f and based on the fact that a_f had to be in between 1.25-1.50, 1.35 seemed to be the best option for a_f as it was the average of the expected a_f. This was then followed by selecting a, which consisted of either 10/3 for roller bearings or 3 for ball bearings. Furthermore, now one was presented with the two most important concepts that would ultimately determine the bearing and they are the dynamic loading C_{10} and the minimum diameter $d_{\text{minimum shaft}}$. To find the dynamic loading C_{10} one had to use the following formula:

$$C_{10} = a_f * F_D * \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{\frac{1}{b}}}\right]^{1/a}$$

The only variable in the above formula that has not been defined is R_D with is the index reliability of 0.9. Not only is C_{10} is an important factor, but C_{D10} is important to consider in order to prove that if $C_{D10}<C_{10}$ then the design specifications are being met. The following is the equation for C_{D10}:

$$C_{D10} = a_f * V * F_D * \left[\frac{L_D}{L_{10}}\right]^{1/a}$$

After measuring the dynamic load, one had to compare the diameter of the static ($d_{\text{minimum shaft}}$) loading with the diameters from the bearings McMaster provided. In order to make a final decision, the dynamic loading C_{10} had to be more than the calculated value. Not only was C_{10} analyzed, but the diameter of the shaft also had to be greater than d_{bore}.

Based on the calculations for determining the size of the bearings, one used an open bearing of R24 and a high load open 2780T26 for the input shaft. Whereas, for the output shaft had a steel tapered-roller bearings, and an open bearing of size R18. These equations give rise to the concept of creating a housing for the assembly.

5.4 Housing Design

The housing is the part of the gear box that holds everything together. For our design, we picked a simple design in which the shaft bearings of the shaft rest on top of the housing. In order to determine the proper dimensions of the housing, some rough sketches were made, which can be found in the appendix.

The distance between both shafts must be the half the pitch diameter of the input shaft plus half the pitch diameter of the output shaft. This ensures that the gears come together in the right position so that they can work efficiently. One requirement for our design was that the walls of the housing must be at least 0.5 inches away from the moving parts of the gear. Therefore, the each shaft had to be half of the pitch diameter plus 0.5 inches away from the walls.

For the top view, the dimensions of the bearing holders were determined based on the dimensions of the chosen bearings. The width of the housing was determined by the lengths of the shafts. The SolidWorks model for the housing can be found in the next section.

6. Drawings for components and assemblies

6.1. Part Drawings

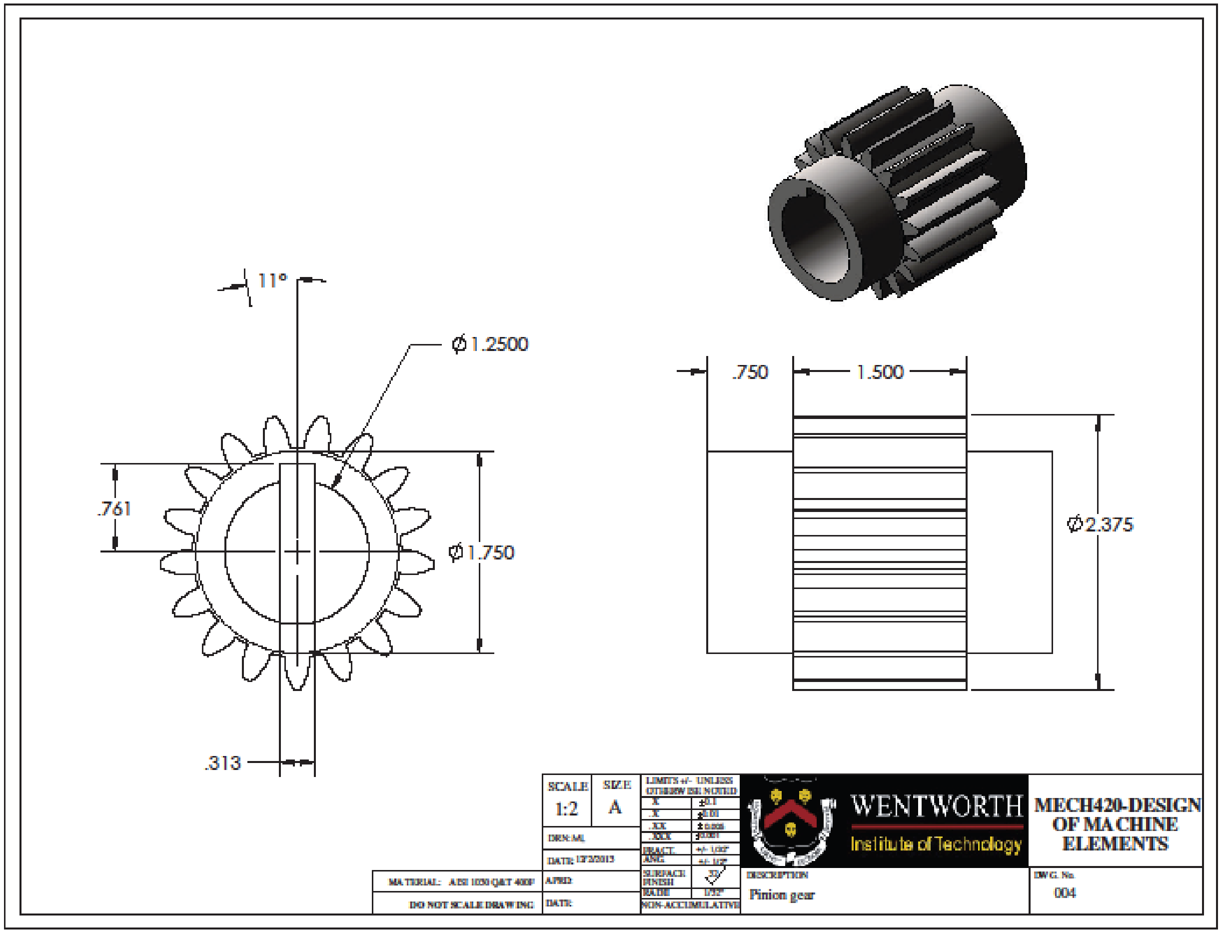


Figure: Pinion Drawing

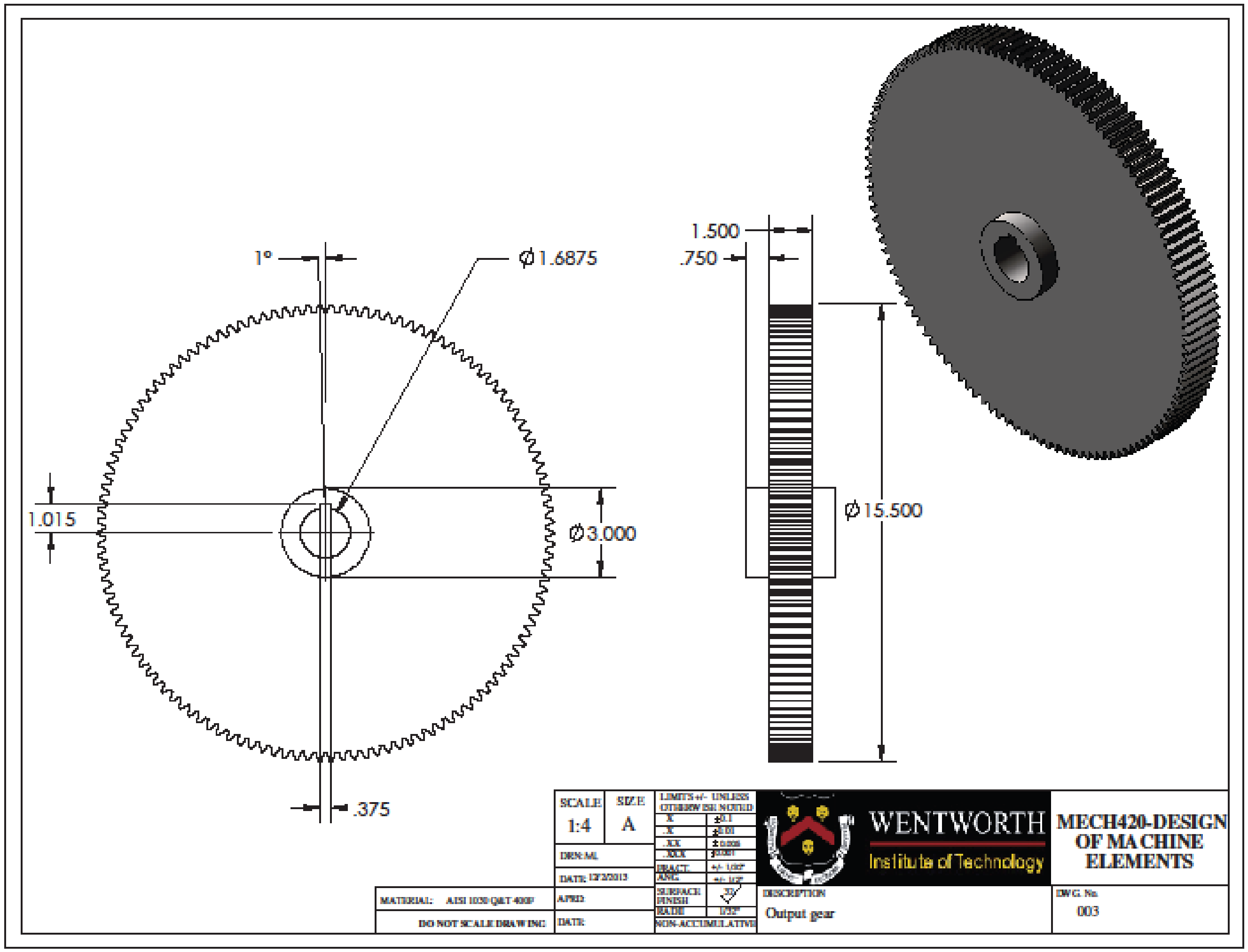


Figure: Output gear drawing

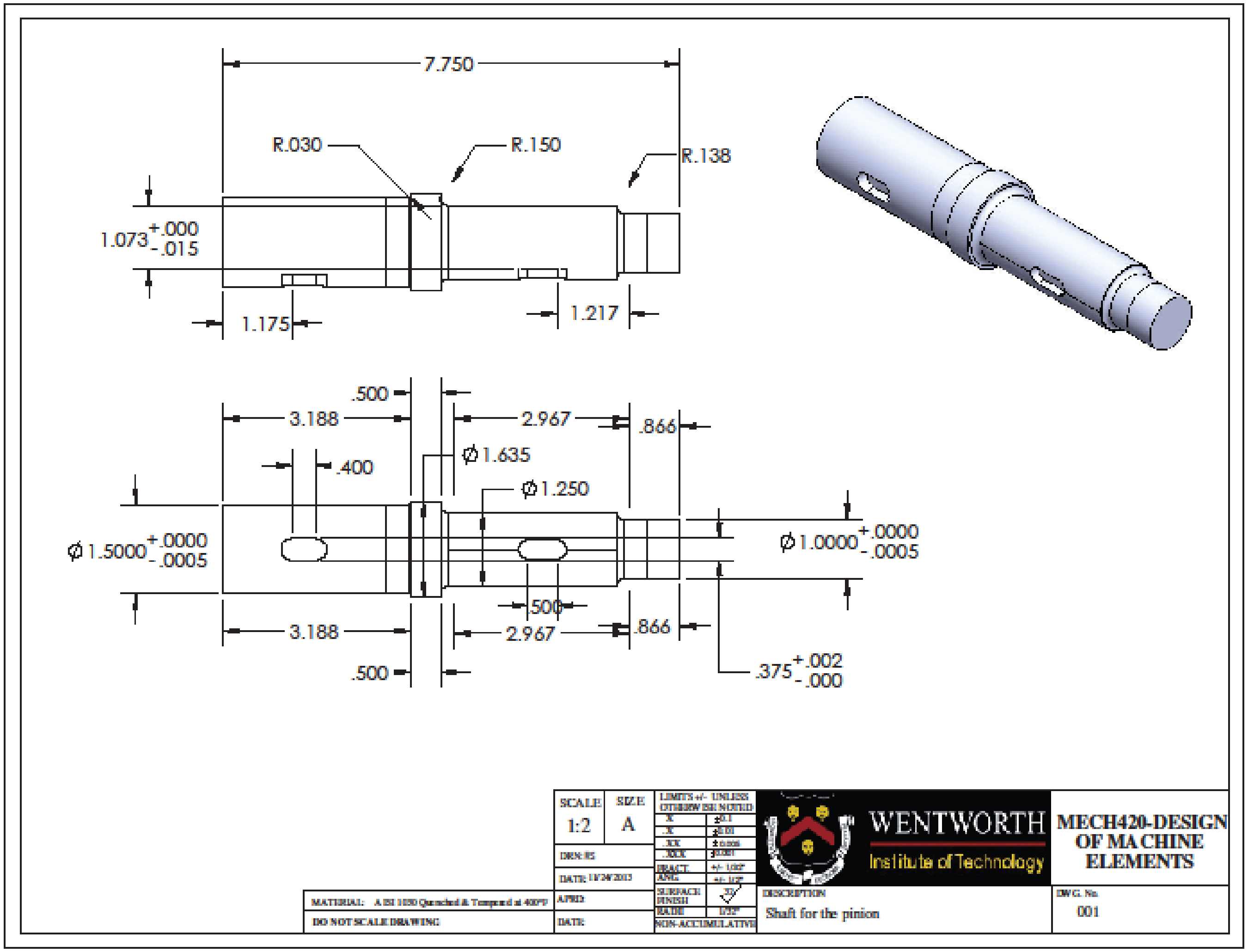


Figure: Input shaft drawing

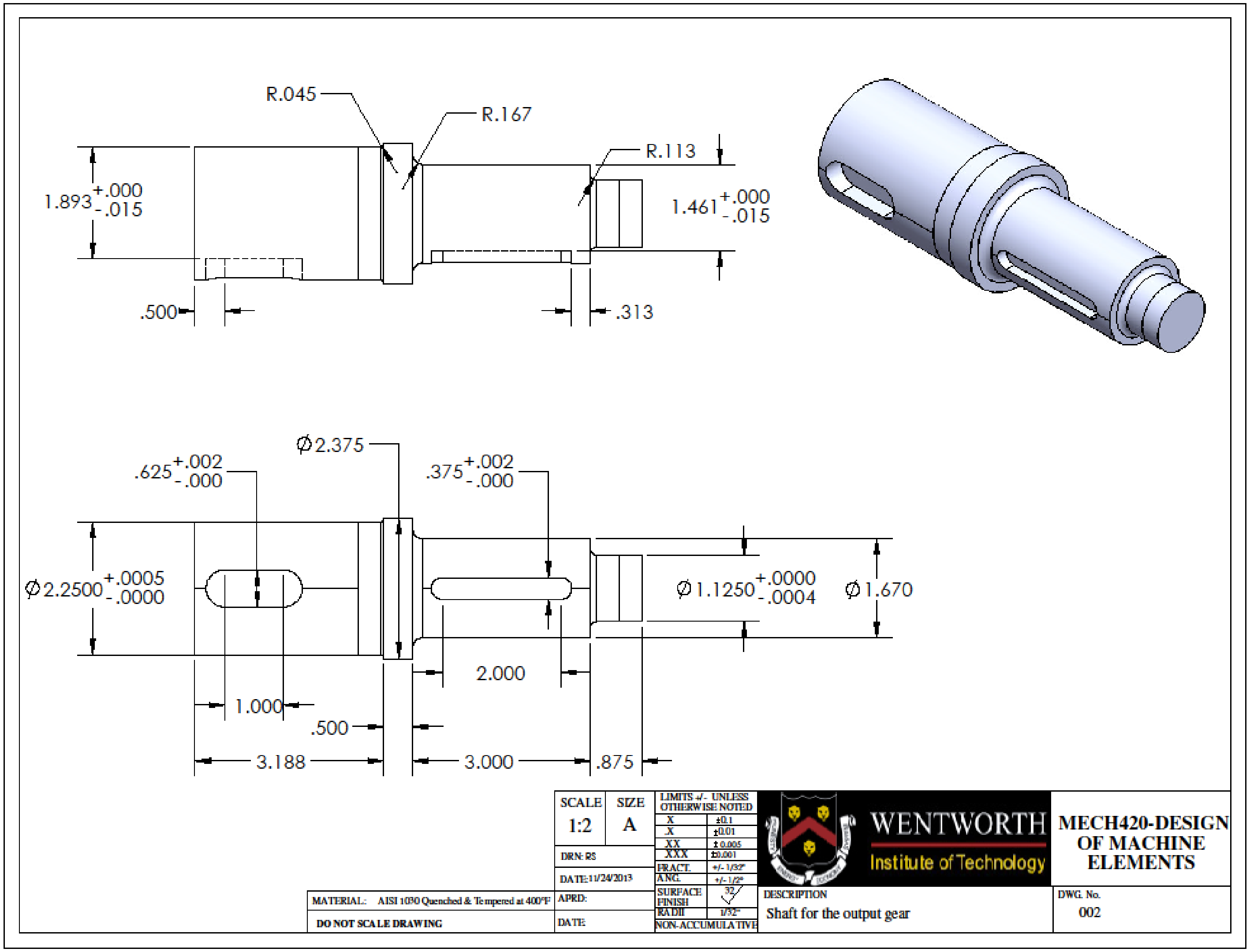


Figure: Output shaft drawing

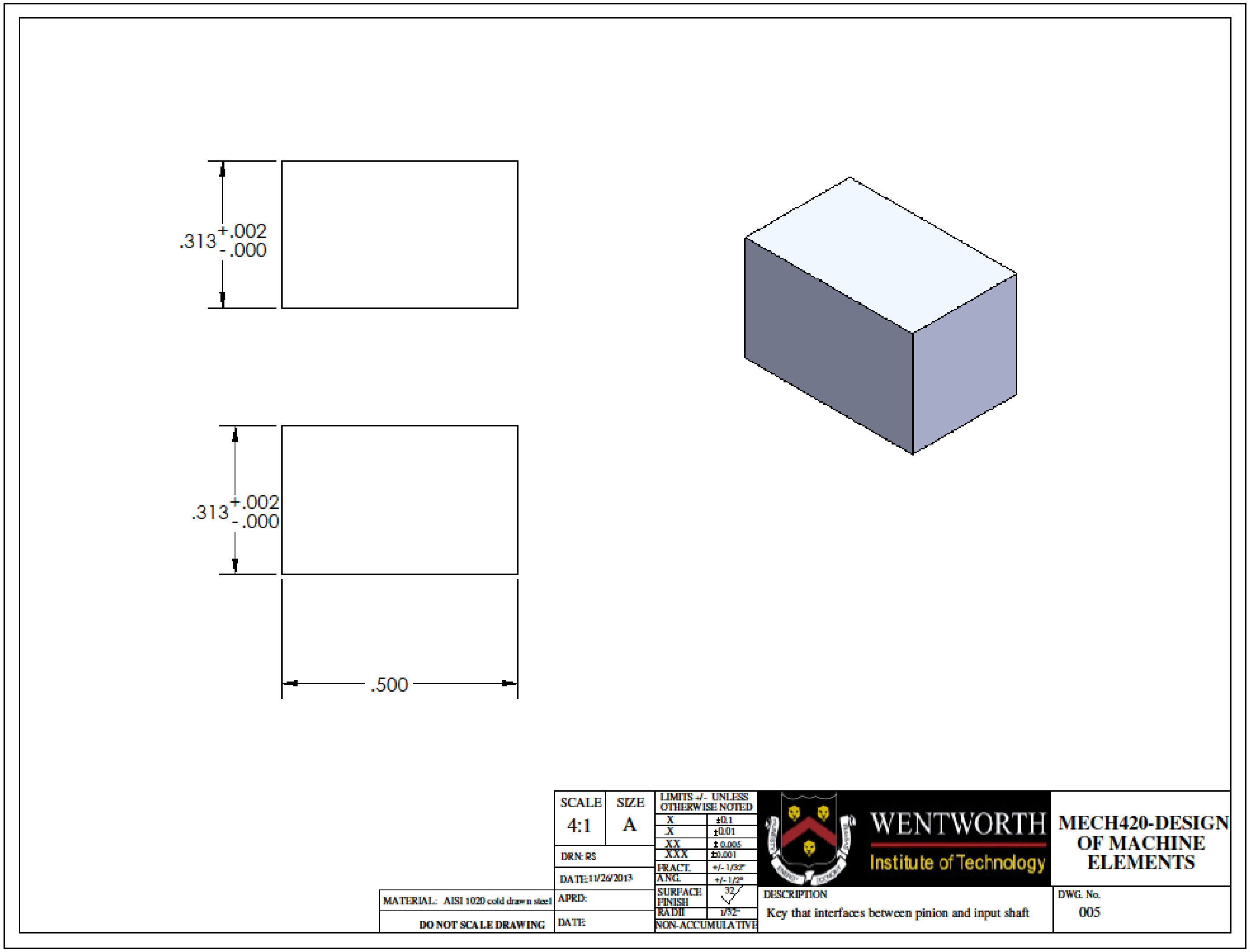


Figure: Input key

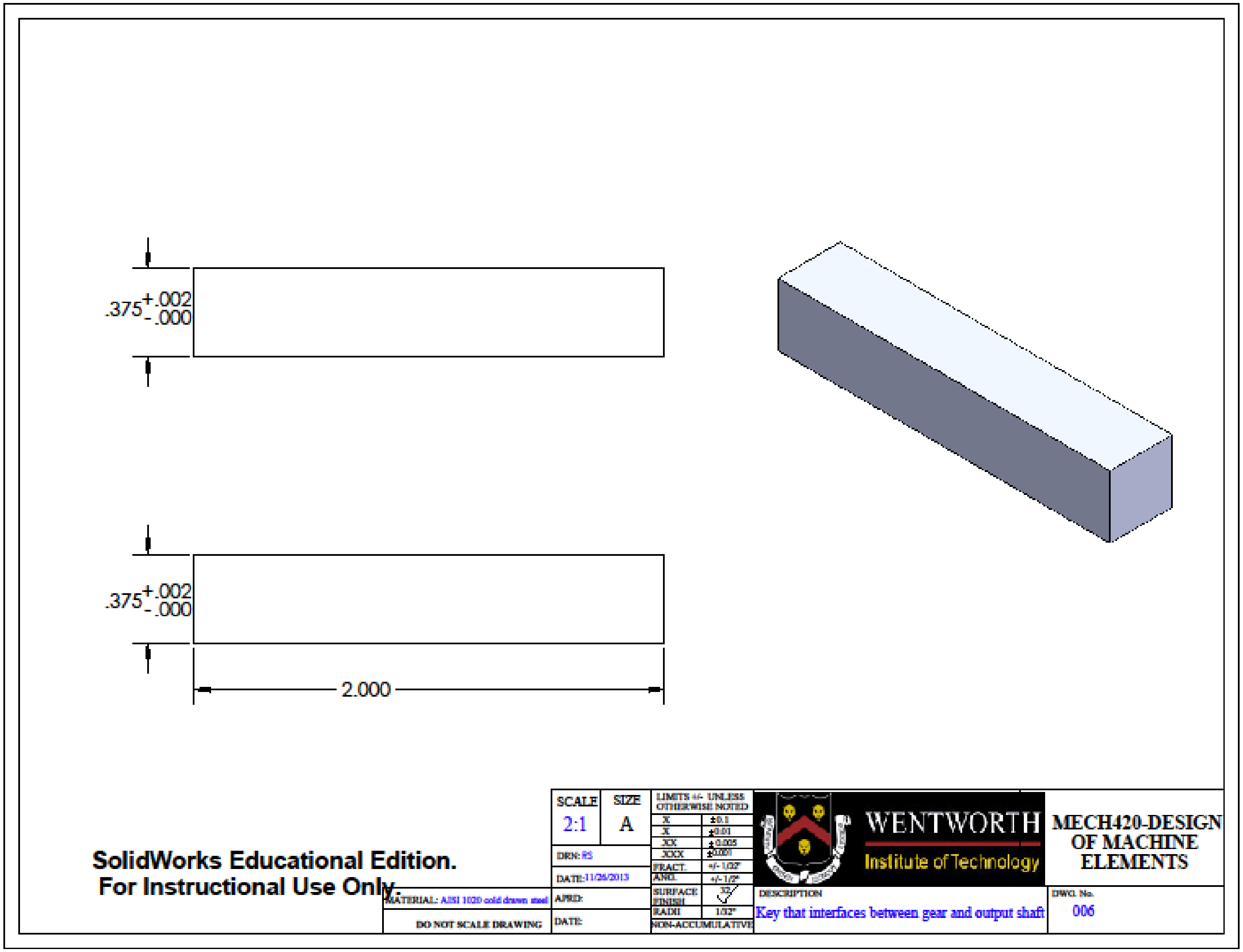


Figure: Output key

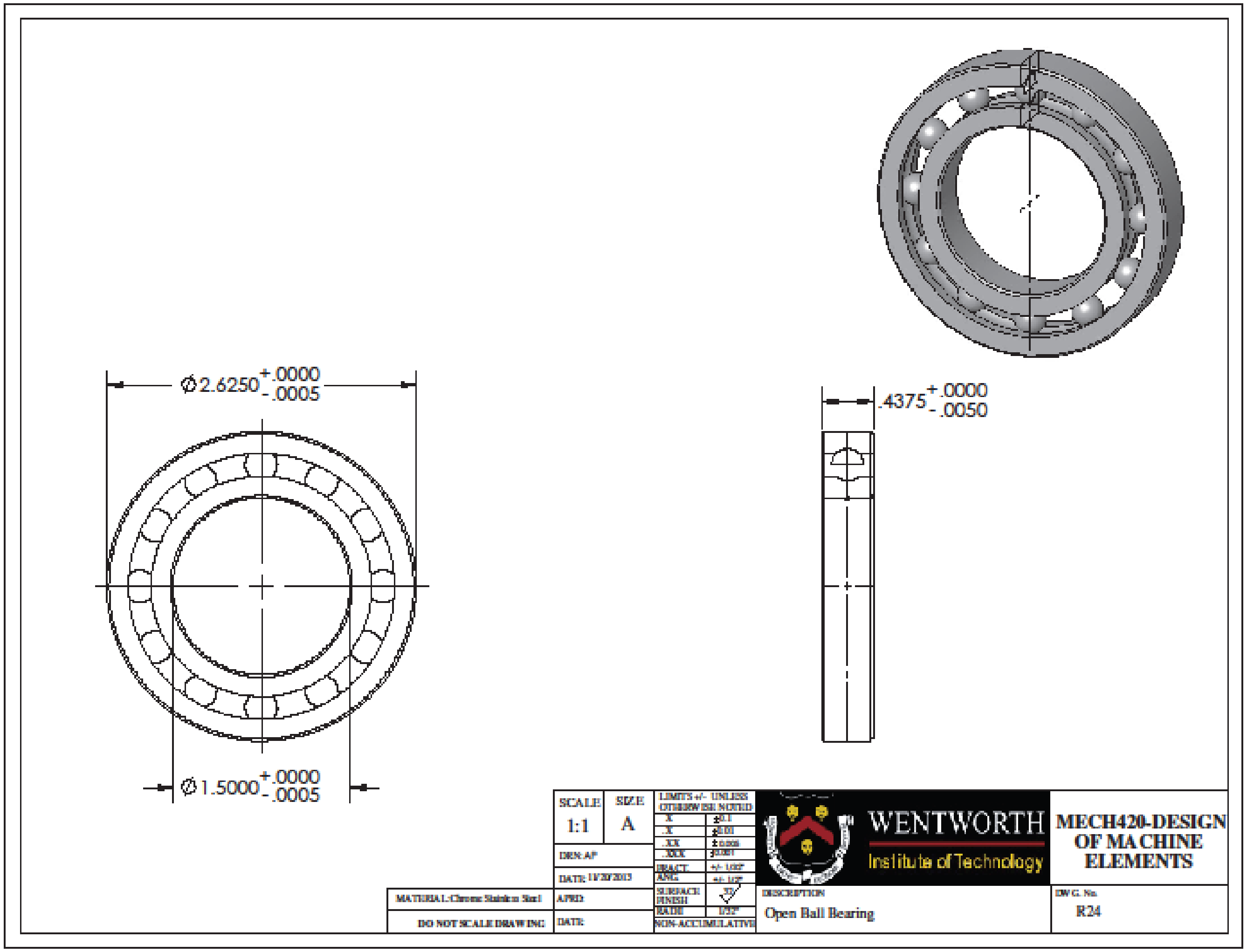


Figure: R24 Bearing

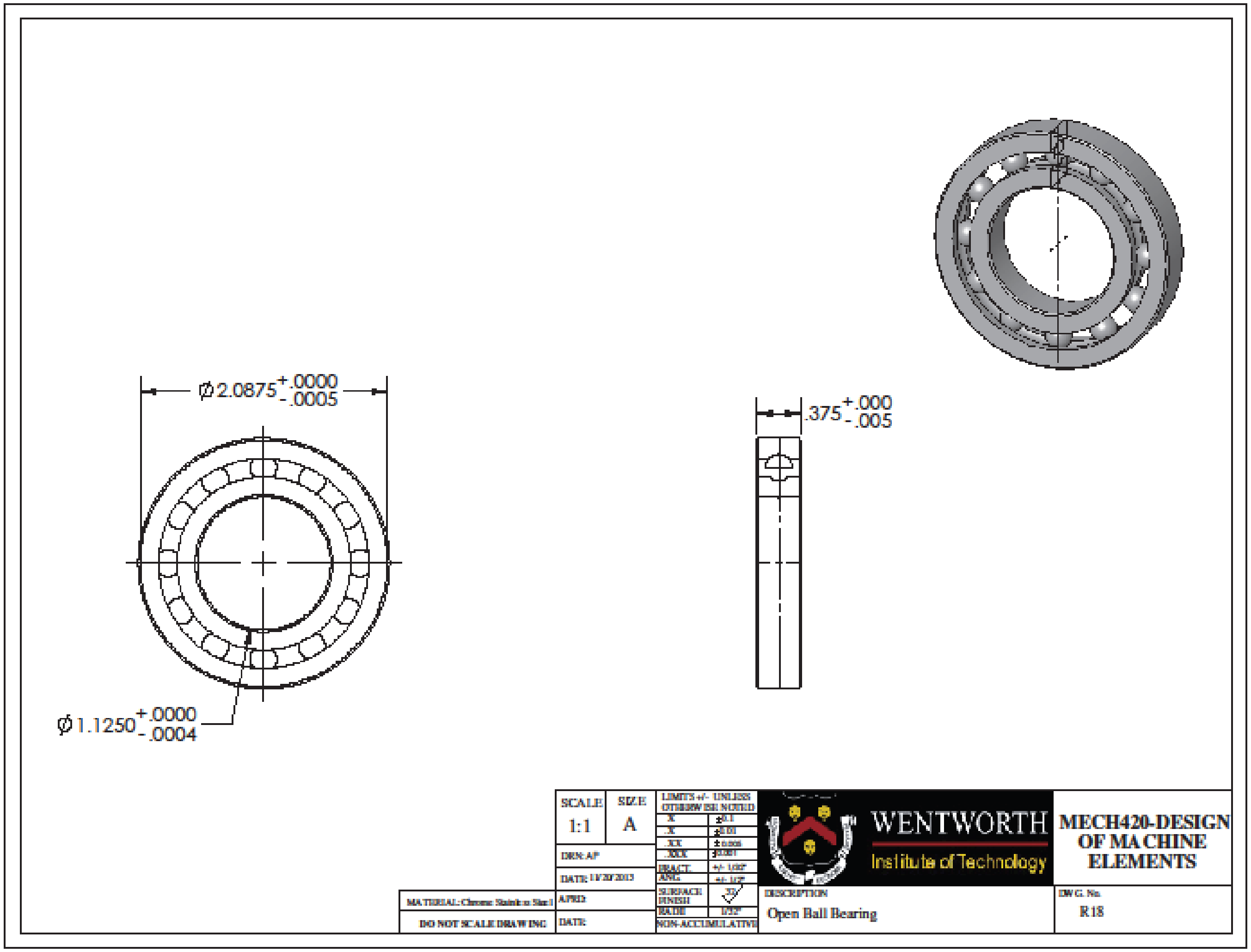


Figure: R18 Bearing

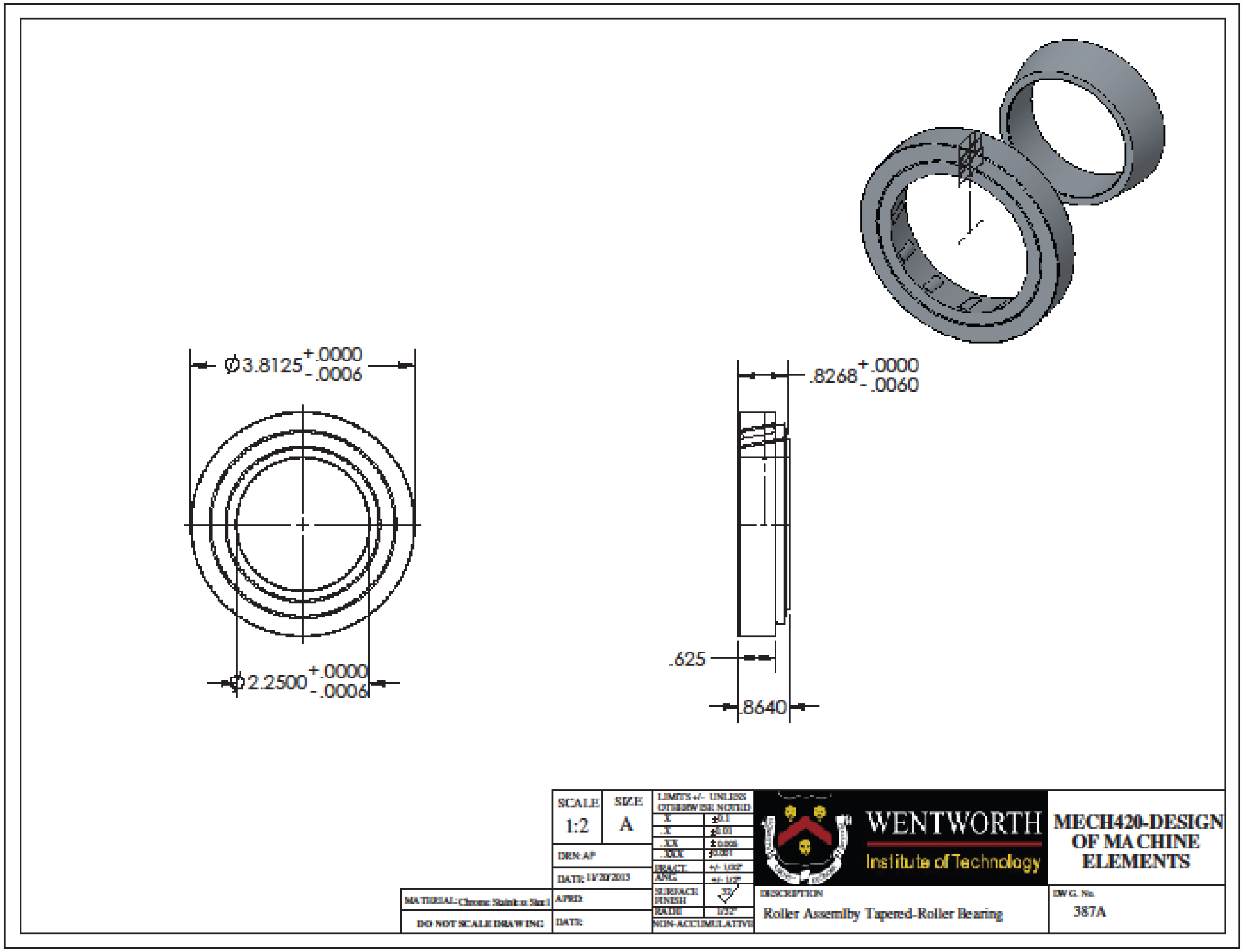


Figure: 387A Bearing

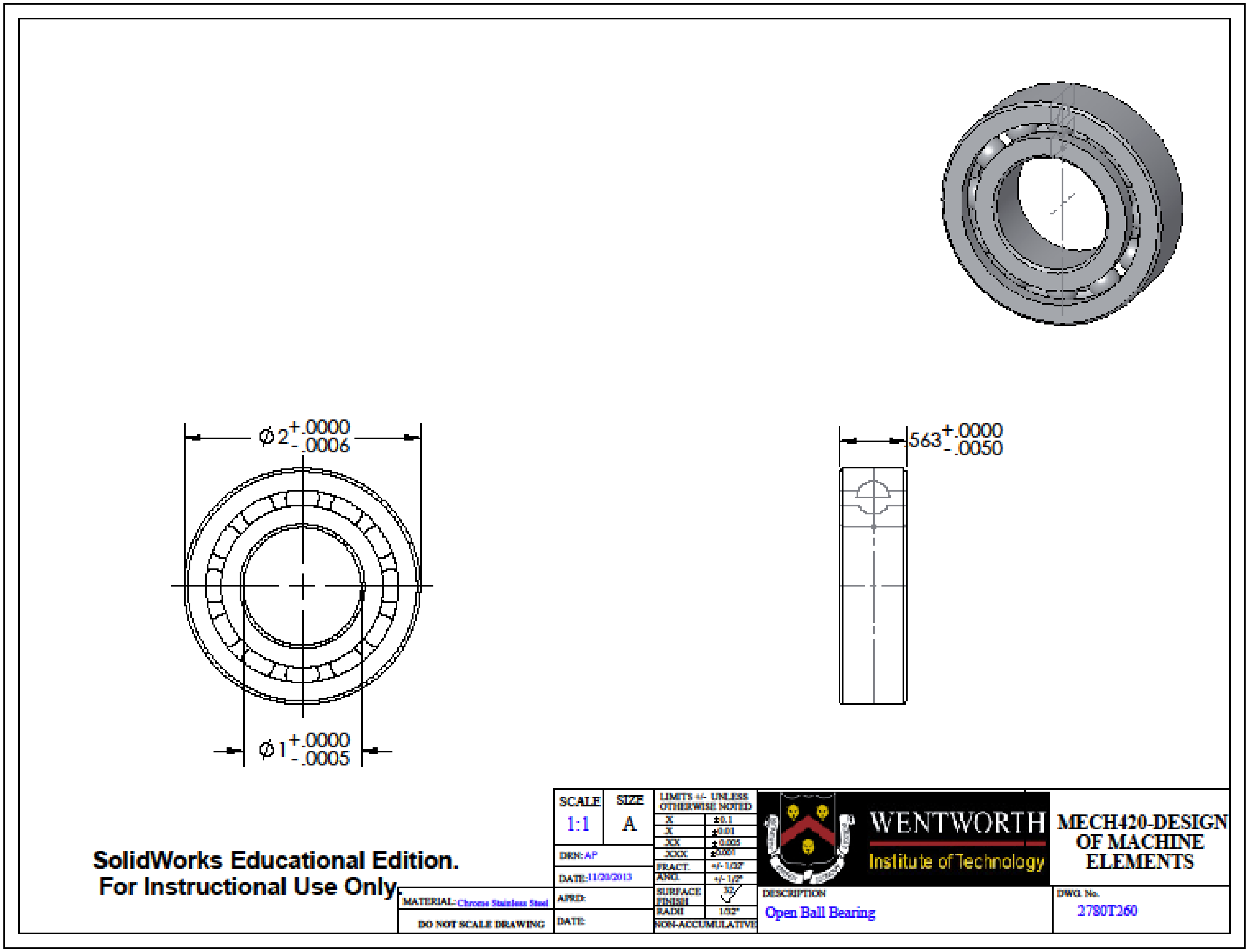


Figure: 2780T260 Bearing

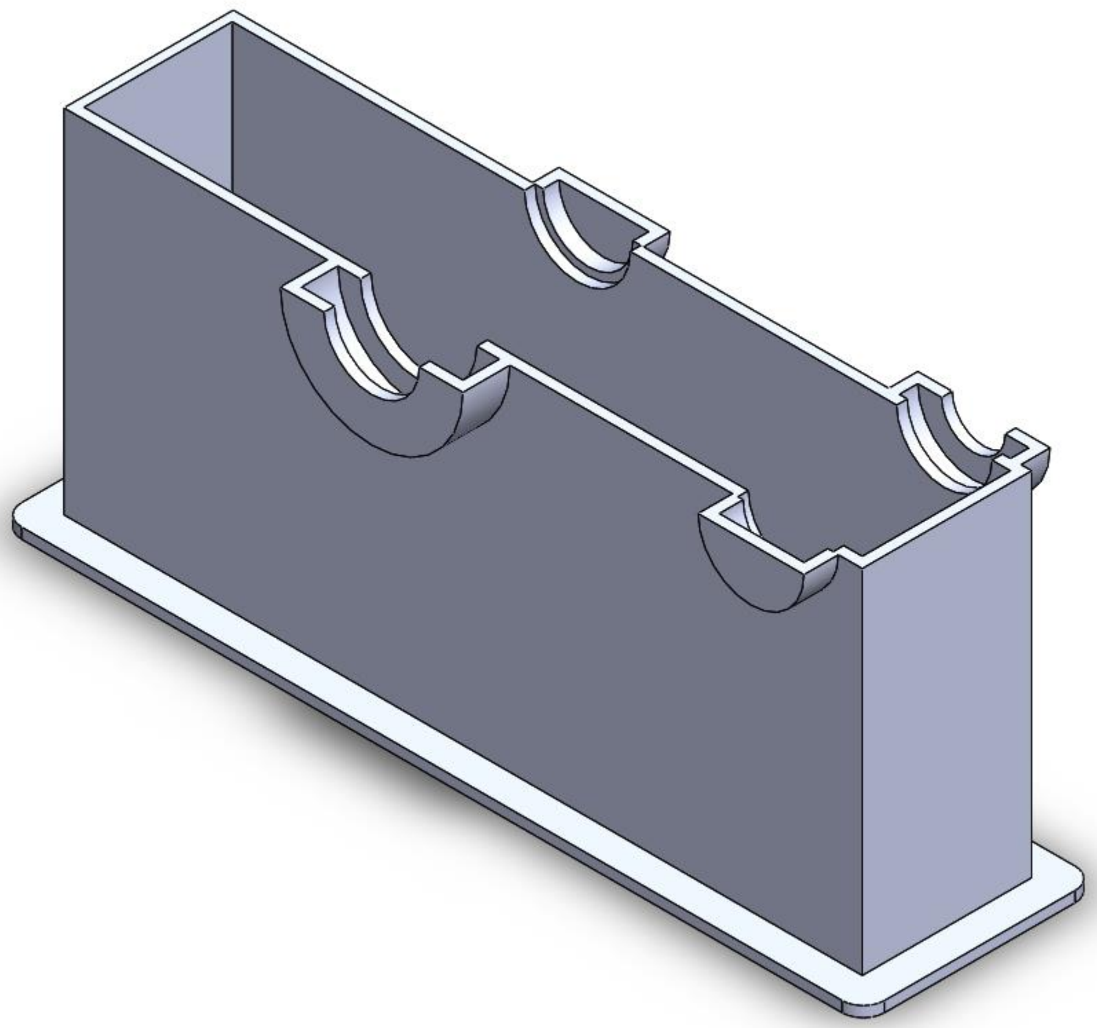


Figure: Housing SolidWorks Model

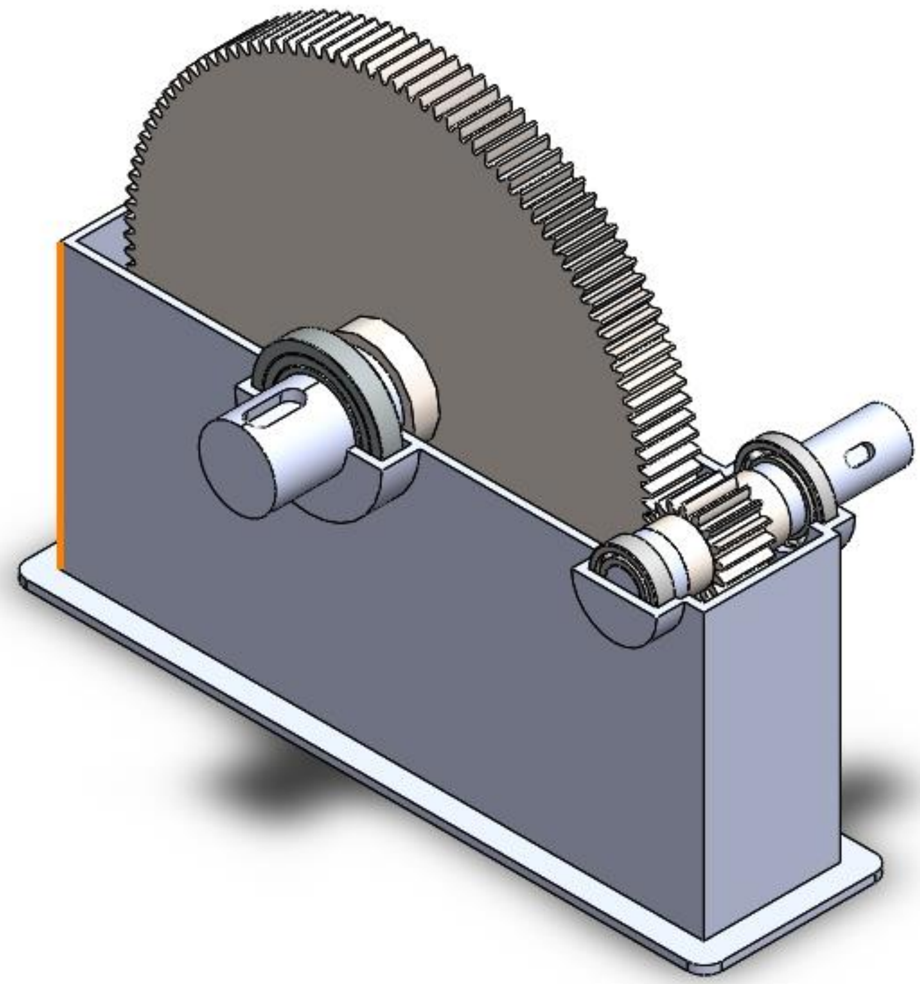


Figure: Gearbox Assembly

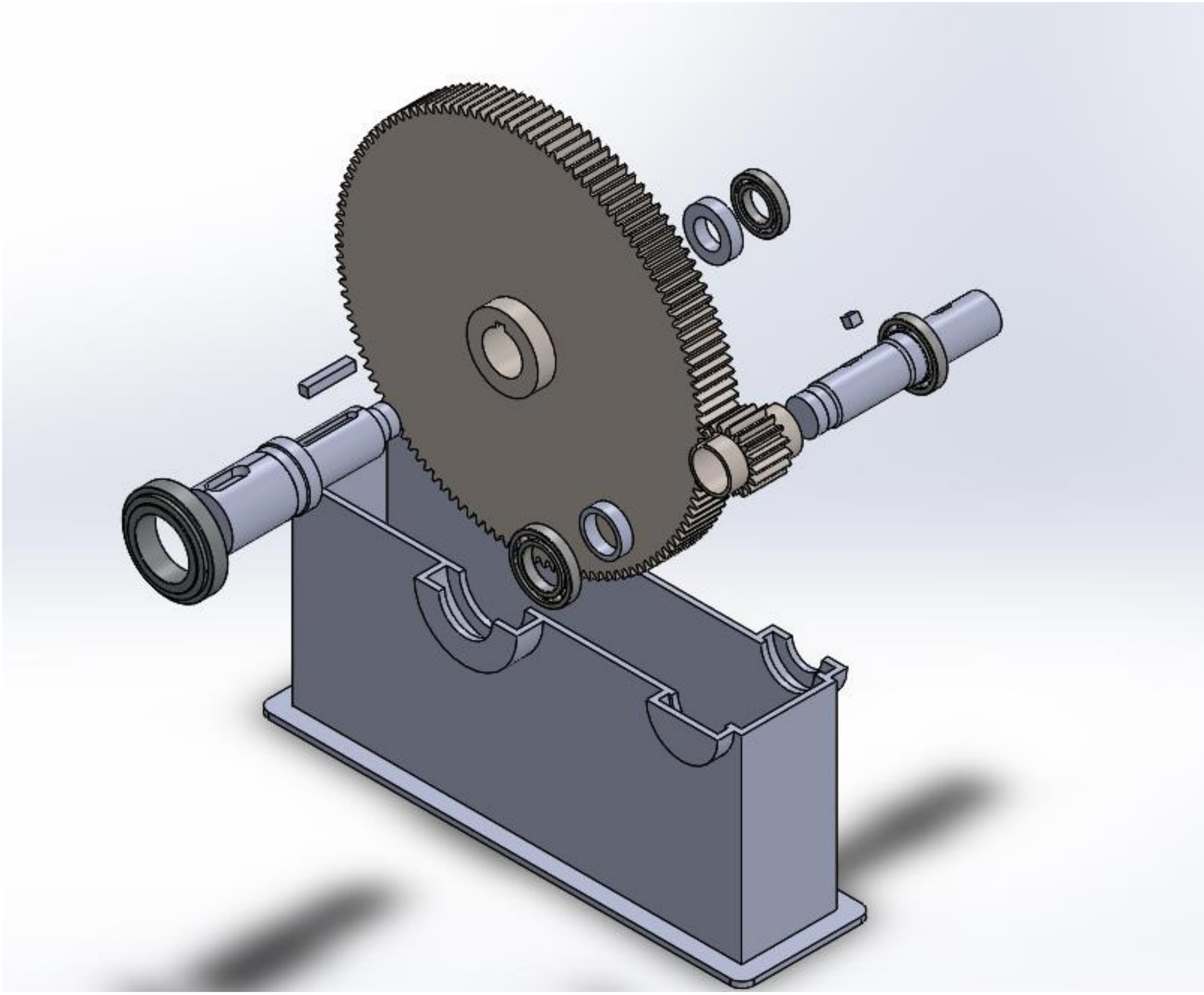
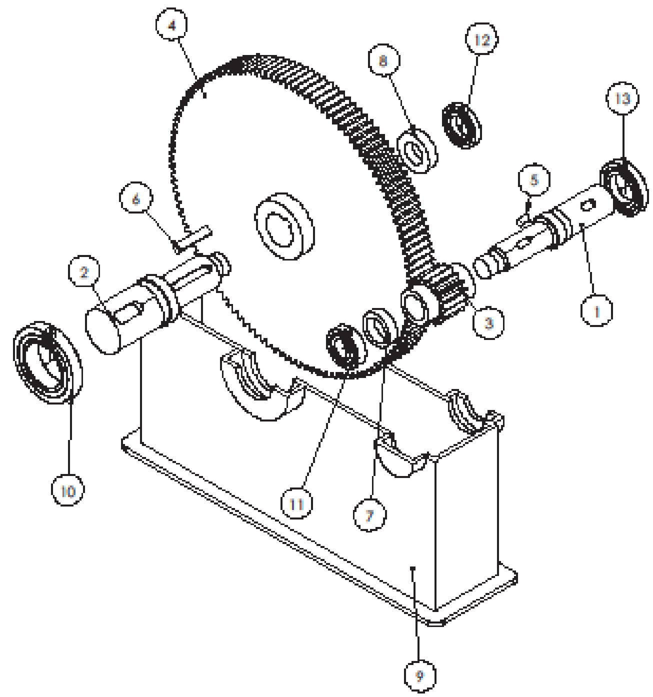


Figure: Gearbox Assembly exploded view

ITEM NO.	PartNo	DESCRIPTION	QTY.
1	001	Shaft for the pinion	1
2	002	Shaft for the output gear	1
3	003	Pinion gear	1
4	004	Output Gear	1
5	005	Key that interfaces between pinion and input shaft	1
6	006	Key that interfaces between gear and output shaft	1
7	008	Input spacer	1
8	009	Outuput spacer	1
9	010	Gearbox Housing	1
10	387A	Roller Assemby Tapered-Roller Bearing	1
11	2780T260	Open Ball Bearing	1
12	R18	Open Ball Bearing	1
13	R24	Open Ball Bearing	1



SCALE 1:12 SIZE B

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MECH420-DESIGN OF MACHINE ELEMENTS

Figure: Gearbox Bill of Materials

7. Findings, Discussion and Conclusions

Based on the presented calculations, graphs, and tables it is found that the current design does not meet the all the major design specifications. This means the design specifications were met for all parts of the carjack except the link bar. This could be due to the shape of the link or material.

While this project did not required a redesign, what could have been done was to use a material such as Alloy Steel and used the FEA Simulation through SolidWorks in order to determine if the material was the problem.

Moreover, this shows that the project was limited and there is a lot more to be done in order to make this car jack ready for reproduction. Thus it is concluded that the link bar was the main cause of failure and that one needs to work on finding a better link bar design. This was an adventurous venue and by working with a group, one would perhaps be able to expand its creativity in designing something creative such as a gearbox.

8. References

[1] Le, Xiaobin., Duva, Anthony., and Talpasanu, Ilie. 2013, *Design of Machine Elements MECH 420 Major Project Worksheet*, Boston, Print

[2] Carr, Mc-Master, 2013. “United States of America”

[3] Budynas, Richard G., and J. K. Nisbett. 2010, *Shigley's Mechanical Engineering Design*. 9th ed., McGraw-Hill Science/Engineering/Math, New York City, Print.

[4] Le, Xiaobin. 2013 Blackboard Lectures 10-21., 2013, Wentworth Institute of Technology, Boston, Print.