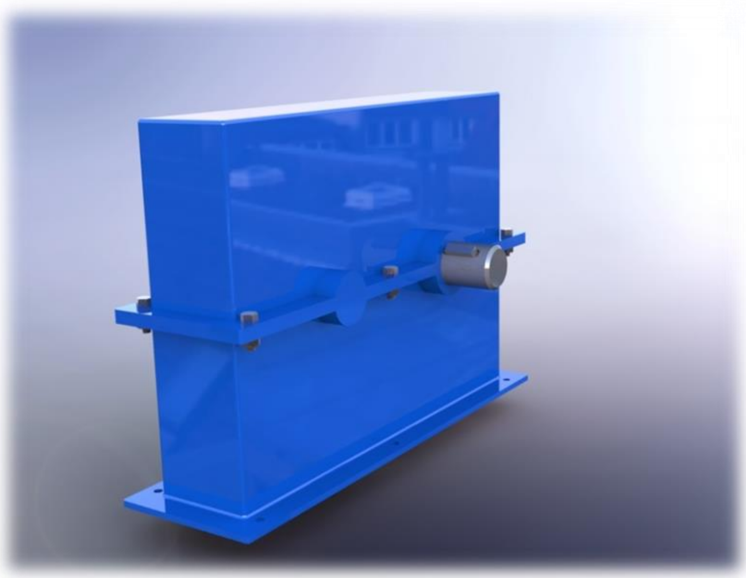
12/9/2014

Gear Box Design

MECH 420 MAJOR PROJECT



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Executive Summary

The company has planned to develop a series of single-reduction spur gearbox reducers. The task for each team working on this project was to design a single-reduction spur gearbox which includes a housing, ball bearings, spur gears, mechanical keys and shafts to meet specifications given for each team. The task for this team was to design a spur gearbox reducer which could handle a 10 horsepower input power with a gear ratio of 3.5 with 1% variation.

This design was conducted by the creation and implementation of a Microsoft Excel spreadsheet to create an equation driven design interface program for the design of each component of the gearbox reducer. The design interface program was used as a mathematical basis to perform the component design as well as provide a tool to base future designs off of. This design interface allowed for the modification of input values should they be needed as the design process progressed. In addition to conducting the design with equations, a FEA simulation was performed on the shafts to determine the deflection as well as the natural frequency of the components. A model of each component as well as an assembly of the device was created in SolidWorks; the associated drawings are presented in this report.

The final design uses all in house manufactured components with the exception of the ball bearings which will be purchased from McMaster-Carr. The associated engineering documents including the sketch layout, detailed calculations, component drawings, and assemblies are all included with this report. By using the design interface program as well as the FEA simulation, a final design was ultimately synthesized which satisfies the design intent.

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Introduction

A gearbox can be simple or complex and is a machine that is used to transfer rotational energy from a motor to another device. They are generally used to increase the torque while decreasing rotational speed, they do not have any effect on the power developed by the motor because as torque increases, rotational speed decreases and vice versa. The following figure 1 is an example of the gear box, which is for a manual Toyota.

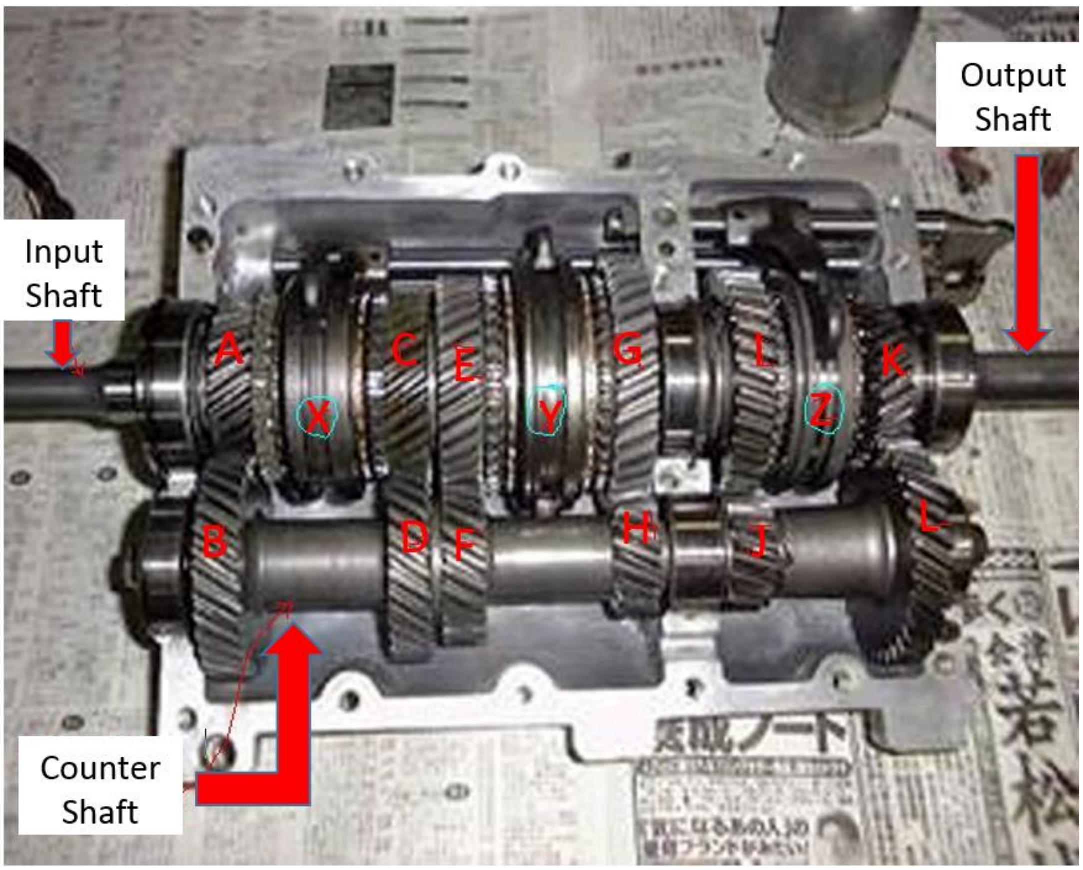


Figure 1: Gearbox of a Manual Toyota T50 [4]

Many machines that are used today are made up of a power source and a gearbox. Gearboxes are essential in vehicles because without a gearbox cars would have very limited top speed. A gearbox can be used in many different applications such as, generation power and construction. For this project a simple single-reduction spur gearbox was designed. The design involved shafts, housing, ball bearings, mechanical key, and the design of gears based on a desired horse power of 10 and a gear ratio of 3.5 with 1% variation.

Design Specifications

The purpose of this design project was to design a single-reduction spur gearbox, which includes a housing, ball bearings, spur gears, mechanical keys, and shafts.

The design specifications include:

- ✓ WIT-MEGB-01, 10 horsepower input, 3.5 gear ratio with 1% variation.
- ✓ Delivered power of 15~30 horsepower with an input speed of 1800 rpm.
- ✓ The design life of the gears and bearings will be 4000 hours.
- ✓ A factor of safety of 3.25 for shaft static design and 1.25 for shaft fatigue design.
- ✓ The reliability index for the gear, bearing, and shaft will be 0.90.
- ✓ Mechanical keys will be used for the connection between the power source and the driven device.

Design and Analysis

Gear Design

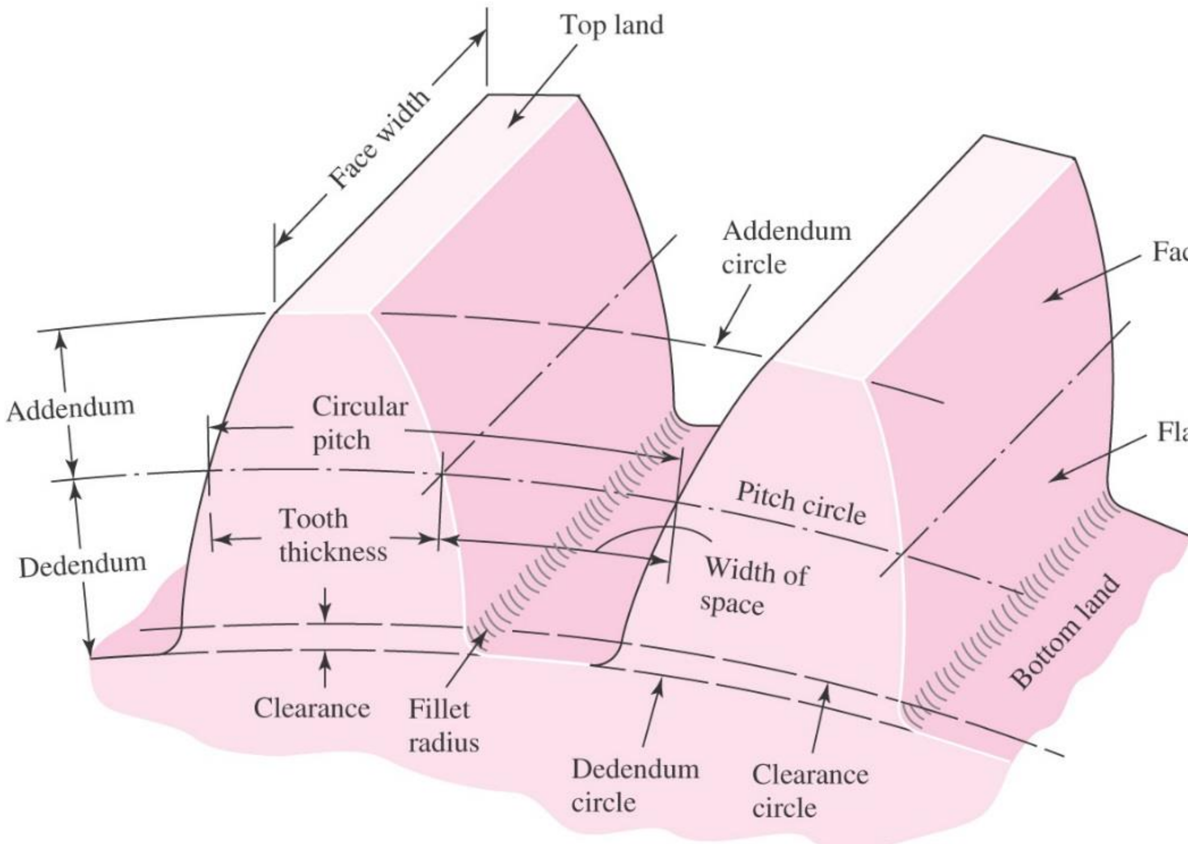


Figure 2: Nomenclature of Gear Sections [1]

There were two aspects to the Gear Design which needed to be considered:

1: The geometry of the gears to allow for the needed power transmission.

2: The strength design of the gear to allow for the use a Grade 1 steel.

By using the given rotational speed of 1800 rpm, input power of 10 horsepower, gear ratio of 3.5, and an overload factor of 1.5, the rough dimensions of the gear and pinion were determined. By using the standard information for the relation between diametral pitch and output power, it was determined that the diametral pitch had to be at least 10 inches. Using the pressure angle of 20 degrees and table 1, the minimum number of teeth for the pinion was found to be 24. Using these constraints, a

Microsoft Excel model of the gears was established which allowed these values to be changed within the allowable limits if needed as a result of the strength design.

N_P	Computed $N_G = (5.308)(N_P)$	Nearest integer N_G	Actual VR: $VR = N_G/N_P$	Actual output speed (rpm $n_G = n_P/VR = n_P(N_P/N_G$
17	90.23	90	90/17 = 5.29	651.7
18	95.54	96	96/18 = 5.33	646.9
19	100.85	101	101/19 = 5.32	649.0
20	106.15	106	106/20 = 5.30	650.9
21	111.46	111	111/21 = 5.29	652.7
22	116.77	117	117/22 = 5.32	648.7
23	122.08	122	122/23 = 5.30	650.4
24	127.38	127	127/24 = 5.29	652.0
25	132.69	133	133/25 = 5.32	648.5
26	138.00	138	138/26 = 5.308	650.0 Exact
27	143.31	143	143/27 = 5.30	651.4
28	148.61	149	149/28 = 5.32	648.3
29	153.92	154	**Too large**	

Table 1: Allowable Number of Pinion Teeth for a Pressure Angle of 20 Degree [2]

The major aspects of the strength design were to calculate the hardness required for the bending and contact stresses of the gear and pinion. The contact stress was calculated to be much larger than the bending stress of the pinion and gear and therefore the hardness was solely based off of the allowable contact stress. It was clear that once the strength calculations were finished for the first time that the geometry must be modified as the strength calculations were suggested a required hardness of around 600 HB which is well beyond the range of grade 1 steel. The final values for hardness can be seen in table 2 below.

Gear Bending Stress Number (psi) σ_BG	15241.38414	**Pinion Contact Stress Number (psi) σ_c**	116643.873
Pinion Bending Stress Number (psi) σ_BP	19912.13089	**Gear Contact Stress Number (psi) σ_cG**	114851.0364
Gear Allowable Bending Stress (psi) S_tG	16641.21064	**Gear Allowable Contact Stress Number (psi) S_Cg**	129287.9297
Pinion Allowable Bending Stress (psi) S_tP	22231.18733	**Pinion Allowable Contact Stress Number (psi) S_Cp**	135144.557
Hardness for Gear Bending Stress	**49.69224625**	**Hardness for Gear Contact Stress**	**311.1426388**
Hardness for Pinion Bending Stress	**122.0075981**	**Hardness for Pinion Contact Stress**	**329.3309222**

Table 2: Gear Strength Values

The number of teeth and the diametral pitch needed to be changed in order to obtain a hardness that was more acceptable. Ultimately a diametral pitch of 10 inches was decided to be optimal along with 17 teeth for the pinion. These values allowed for a required pinion hardness of 329 HB and a Gear Hardness of 311 HB. By using these hardness values and table 3, AISI 4340 Q&T 1000 deg F was selected for the pinion as it had a hardness of 360 HB and AISI 4130 Q&T 1000 deg F was selected for the gear as it has a hardness of 315 HB. The difference in materials is important because the pinion would become damaged by the gear if the two were the same material.

Table A–21 Useful Tables | **1041**

Mean Mechanical Properties of Some Heat-Treated Steels

1 AISI No.	2 Treatment	3 Temperature °C (°F)	4 Tensile Strength MPa (kpsi)	5 Yield Strength, MPa (kpsi)	6 Elongation, %	7 Reduction in Area, %	8 Brinell Hardness
1095	Q&T	315 (600)	1260 (183)	813 (118)	10	30	375
	Q&T	425 (800)	1210 (176)	772 (112)	12	32	363
4130	Q&T*	425 (800)	1280 (186)	1190 (173)	13	49	380
4140	Q&T	425 (800)	1250 (181)	1140 (165)	13	49	370
4340	Q&T	540 (1000)	1170 (170)	1080 (156)	13	51	360
4130	Q&T*	540 (1000)	1030 (150)	910 (132)	17	57	315
1060	Q&T	425 (800)	1080 (156)	765 (111)	14	41	311

Table 3: Mean Mechanical Properties of Some Heat-Treated Steels [1]

Detailed Calculations

Pitch Diameter

$$Pitch\ Diameter = \frac{Number\ of\ Teeth}{Diametral\ Pitch}$$

$$Pitch\ Diameter_{pinion} = \frac{24}{10}$$

$$Pitch\ Diameter_{pinion} = 2.4$$

$$Pitch\ Diameter_{gear} = \frac{84}{10}$$

$$Pitch\ Diameter_{gear} = 8.4$$

The Diametral Pitch is found based on the speed of the pinion and the power transmission that is desired to go through the pinion. The number of teeth on the pinion/gear is chosen based on the gear ratio with the minimum number of teeth possible for the pinion being 17 and the maximum for the gear being 150. The pitch diameter measures the diameter from the center of the pinion/gear to the middle of the teeth. The pitch line is also the point where the teeth from the gear and the pinion meet while the two of them are in motion.

Addendum

$$Addendum = \frac{1}{Diametral\ Pitch}$$

$$Addendum = \frac{1}{10}$$

Addendum = .1

The Addendum is the distance measured from the middle of the tooth to the top of the tooth.

Dedendum

$$Dedendum = \frac{1.25}{Diametral\ Pitch}$$

$$Dedendum = \frac{1.25}{10}$$

Dedendum= .125

The Dedendum is the distance measured from the middle of the tooth to the bottom of the tooth.

Clearance

$$Clearance = \frac{.25}{Diametral\ Pitch}$$

$$Clearance = \frac{.25}{10}$$

Clearance = .025

The Clearance is the distance between the top of the tooth and the bottom of the tooth space on the mating gear.

Outside Diameter

$$Outside\ Diameter = Pitch\ Diameter + (2 * Addendum)$$

$$Outside\ Diameter_{pinion} = 2.4 + (2 * .1)$$

$$Outside\ Diameter_{pinion} = 2.6$$

$$Outside\ Diameter_{gear} = 8.4 + (2 * .1)$$

$$Outside\ Diameter_{gear} = 8.6$$

The outside diameter is measured from the inside of the gear/pinion to the very outer edge of the teeth.

Root Diameter

$$Root\ Diameter = Pitch\ Diameter - (2 * Dedendum)$$

$$Root\ Diameter_{pinion} = 2.4 - (2 * .125)$$

$$Root\ Diameter_{pinion} = 2.15$$

$$Root\ Diameter_{gear} = 8.4 - (2 * .125)$$

$$Root\ Diameter_{gear} = 8.15$$

The root diameter is measured from the inside of the gear/pinion to the very base of the teeth.

<u>Hole Depth</u>

$$Hole\ Depth = Addendum + Dedendum$$

$$Hole\ Depth = .1 + .125$$

$$Hole\ Depth = .225$$

The hole depth is the direct distance from the top of the tooth to the base of the tooth.

<u>Working Depth</u>

$$Working\ Depth = 2 * Addendum$$

$$Working\ Depth = 2 * .1$$

$$Working\ Depth = .2$$

The working depth is how far into the mating gear the teeth actually go

<u>Tooth Thickness</u>

$$Tooth\ Thickness = \frac{\pi}{2 * Diametral\ Pitch}$$

$$Tooth\ Thickness = \frac{\pi}{2 * 10}$$

$$Tooth\ Thickness = .157$$

The tooth thickness is based off of the diametral pitch using it as a way to figure out how thick and how spaced out the teeth will need to be in order to fit the desired amount of teeth per inch.

<u>Base Circle Diameter</u>

$$Base\ Circle\ Diameter = Pitch\ Diameter * \cos(Pressure\ Angle)$$

$$Base\ Circle\ Diameter_{pinion} = 2.4 * \cos(20)$$

$$Base\ Circle\ Diameter_{pinion} = 2.255$$

$$Base\ Circle\ Diameter_{gear} = 8.4 * \cos(20)$$

$$Base\ Circle\ Diameter_{gear} = 7.89$$

The diameter of the pinion/gear using the pressure angle in order to find a tangential line to measure to which is always smaller than the pitch diameter.

Face Width

$$Face\ Width = \frac{12}{Diametral\ Pitch}$$

$$Face\ Width = \frac{12}{10}$$

$$Face\ Width = 1.2$$

The Face Width is how wide the teeth are measured parallel to the axis of the gear/pinion.

Pitch Line Velocity

$$Pitch\ Line\ Velocity = \frac{\pi * Pitch\ Diameter * Rotational\ Speed}{12}$$

$$Pitch\ Line\ Velocity_{pinion} = \frac{\pi * 2.4 * 1800}{12}$$

$$Pitch\ Line\ Velocity_{pinion} = 1130.973$$

$$Pitch\ Line\ Velocity_{gear} = \frac{\pi * 8.4 * 514.28}{12}$$

$$Pitch\ Line\ Velocity_{gear} = 1130.973$$

The pitch line velocity is how many feet per minute the gear/pinion is rotating at during operation.

Dynamic Factor

$$Dynamic\ Factor = \left(\frac{A + \sqrt{Pitch\ Line\ Velocity}}{A}\right)^{B}$$

$$Dynamic\ Factor = \left(\frac{59.77 + \sqrt{1130.973}}{59.77}\right)^{.825}$$

$$Dynamic\ Factor = 1.4455$$

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

$$A = 50 + 56(1 - .125)$$

$$A = 59.77$$

$$B = .25(12 - Q_v)^{\frac{2}{3}}$$

$$B = .25(12 - 6)^{\frac{2}{3}}$$

$$B = .825$$

Q_v is the AGMA transmission accurancy level number

<u>Pinion Proportion Factor</u>

$$Pinion\ Proportion\ Factor = \frac{Face\ Width}{10 * Pitch\ Diameter} - .025$$

$$Pinion\ Proportion\ Factor = \frac{1.2}{10 * 2.4} - .025$$

$$Pinion\ Proportion\ Factor = .025$$

The pinion proportion factor is used to figure out the size of the pinion overall comparing the width of the teeth with the pitch diameter of the pinion.

<u>Mesh Alignment Factor</u>

$$Mesh\ Alignment\ Factor = .127 + (.0158 * Face\ Width) - \left((1.093 * 10^{-4}) * (Face\ Width)^2\right)$$

$$Mesh\ Alignment\ Factor = .127 + (.0158 * 1.2) - ((1.093 * 10^{-4}) * (1.2^2))$$

$$Mesh\ Alignment\ Factor = .1458$$

<u>Center Distance</u>

$$Center\ Distance = \frac{Number\ of\ Pinion\ Teeth + Number\ of\ Gear\ Teeth}{2 * Diametral\ Pitch}$$

$$Center\ Distance = \frac{24 + 84}{2 * 10}$$

$$Center\ Distance = 5.4$$

The center distance is the distance measured from the center of the pinion to the center of the gear.

Gear Proportion Factor

$$Gear\ Proportion\ Factor = \frac{Face\ Width}{10 * Pitch\ Diameter} - .025$$

$$Gear\ Proportion\ Factor = \frac{1.2}{10 * 10} - .025$$

$$Gear\ Proportion\ Factor = -.01$$

The gear proportion factor just like the pinion proportion factor compares the size of the gear teeth to the overall size of the gear.

Pinion Load Distribution Factor

$$Pinion\ Load\ Distribution\ Factor = 1 + Mesh\ Alignment\ Factor + Pinion\ Proportion\ Factor$$

$$Pinion\ Load\ Distribution\ Factor = 1 + .1458 + .025$$

$$Pinion\ Load\ Distribution\ Factor = 1.17$$

The load distribution factor for the pinion gets taken into account when looking at the bending stress being applied to the pinion.

Gear Load Distribution Factor

$$Gear\ Load\ Distribution\ Factor = 1 + Mesh\ Alignment\ Factor + Gear\ Proportion\ Factor$$

$$Gear\ Load\ Distribution\ Factor = 1 + .1458 - .01$$

$$Gear\ Load\ Distribution\ Factor = 1.135$$

The gear load distribution factor is taken into account when looking at the bending stress being applied to the gear.

Backup Ratio

$$Backup\ Ratio = \frac{Dedendum}{Addendum}$$

$$Backup\ Ratio = \frac{.125}{.1}$$

$$Backup\ Ratio = 1.25$$

Loading Cycles

$$Number\ Of\ Cycles\ of\ Loading = 60 * Design\ Life * Revolutions\ Per\ Minute * Load\ Application\ per\ Revolution$$

$$Number\ of\ Cycles_{pinion} = 60 * 4000 * 1800 * 1$$

$$Number\ of\ Cycles_{pinion} = 432000000$$

$$Number\ of\ Cycles_{gear} = 60 * 4000 * 514.28 * 1$$

$$Number\ of\ Cycles_{gear} = 123428571$$

<u>Stress Cycle Factor</u>

$$Stress\ Cycle\ Factor = 1.3558 * Number\ of\ Cycles\ of\ Loading^{-0.0178}$$

$$Stress\ Cycle\ Factor_{pinion} = 1.3558 * 432000000^{-0.0178}$$

$$Stress\ Cycle\ Factor_{pinion} = .951$$

$$Stress\ Cycle\ Factor_{gear} = 1.3558 * 123428571^{-0.0178}$$

$$Stress\ Cycle\ Factor_{gear} = .973$$

<u>Contact Stress Cycle Factor</u>

$$Contact\ Stress\ Cycle\ Factor = 1.4488 * Number\ of\ Cycles\ of\ Loading^{-0.023}$$

$$Contact\ Stress\ Cycle\ Factor_{pinion} = 1.4488 * 432000000^{-0.023}$$

$$Contact\ Stress\ Cycle\ Factor_{pinion} = .917$$

$$Contact\ Stress\ Cycle\ Factor_{gear} = 1.4488 * 123428571^{-0.023}$$

$$Contact\ Stress\ Cycle\ Factor_{gear} = .943$$

<u>Tangential Force</u>

$$Tangential\ Force = \frac{33000 * Input\ Power}{Pitch\ Line\ Velocity}$$

$$Tangential\ Force = \frac{33000 * 10}{1130.973}$$

$$Tangential\ Force = 291.78$$

<u>Radial Force</u>

$$Radial\ Force = Tangential\ Force * \tan(Pressure\ Angle)$$

$$Radial\ Force = 291.78 + \tan(20)$$

$$Radial\ Force = 106.2$$

Design Power

$$Design\ Power = Overload\ Factor * Input\ Power$$

$$Design\ Power = 1.5 * 10$$

$$Design\ Power = 15$$

Bending Stress Number

$$Bending\ Stress\ Number = T * \mathrm{K_u} * \mathrm{K_v} * S * \frac{D}{F} * \frac{L * R}{B}$$

T=Tangential Force, K_u=Overload Factor, K_v=Dynamic Factor, S=Size Factor, F=Face Width, D=Diametral Pitch, L=Distribution Factor, R=Rim Thickness, B=Bending Geometry Factor

$$Bending\ Stress\ Number_{pinion} = 291.78 * 1.5 * 1.4455 * 1 * \frac{10}{1.2} * \frac{1.17 * 1}{.31}$$

$$Bending\ Stress\ Number_{pinion} = 19912.13$$

$$Bending\ Stress\ Number_{gear} = 291.78 * 1.5 * 1.4455 * 1 * \frac{10}{1.2} * \frac{1.135 * 1}{.405}$$

$$Bending\ Stress\ Number_{gear} = 14776.46$$

Allowable Bending Stress

$$Allowable\ Bending\ Stress = \frac{Factor\ of\ Safety\ for\ Bending * Reliability\ Factor * Temperature\ Factor}{Stress\ Cycle\ Factor} * \sigma$$

$$\sigma = Bending\ Stress\ Number$$

$$Allowable\ Bending\ Stress_{pinion} = \frac{1.25 * .85 * 1}{.951} * 19912.13$$

$$Allowable\ Bending\ Stress_{pinion} = 22231.19$$

$$Allowable\ Bending\ Stress_{gear} = \frac{1.25 * .85 * 1}{.973} * 14776.46$$

$$Allowable\ Bending\ Stress_{gear} = 16133.59$$

Hardness for Bending Stress

$$Hardness\ for\ Bending\ Stress = \frac{Allowable\ Bending\ Stress - 12800}{77.3}$$

$$Hardness\ for\ Bending\ Stress_{pinion} = \frac{22231.19 - 12800}{77.3}$$

$$Hardness\ for\ Bending\ Stress_{pinion} = 122$$

$$Hardness\ for\ Bending\ Stress_{gear} = \frac{16133.59 - 12800}{77.3}$$

$$Hardness\ for\ Bending\ Stress_{gear} = 43.12$$

<u>Contact Stress Number</u>

$$Contact\ Stress\ Number = C_p * \sqrt{T * K_u * K_v * S * \left(\frac{L}{D_p * F}\right) * \frac{C_f}{I}}$$

$C_p = Elastic\ Coefficient$, $T = Tangential\ Force, K_u = Overload\ Factor, K_v = Dynamic\ Factor, S = Size\ Factor, L = Distribution\ Factor, D_p = Pitch\ Diameter$, $F = Face\ Width, C_f = Surface\ Condition\ Factor$,I=Pitting Geometry Factor

$$Contact\ Stress\ Number_{pinion} = 2300 * \sqrt{291.78 * 1.5 * 1.4455 * 1 * \left(\frac{1.17}{2.4 * 1.2}\right) * \frac{1}{.1}}$$

$$Contact\ Stress\ Number_{pinion} = 116643.873$$

$$Contact\ Stress\ Number_{gear} = 2300 * \sqrt{291.78 * 1.5 * 1.4455 * 1 * \left(\frac{1.135}{8.4 * 1.2}\right) * 1/.1}$$

$$Contact\ Stress\ Number_{gear} = 114851.036$$

<u>Allowable Contact Stress Number</u>

$$Allowable\ Contact\ Stress\ Number = (S_h * K_t * K_r / Z * C_h) * \sigma$$

$S_h = Contact\ Stress\ Factor\ of\ Safety, K_t = Temperature\ Factor, K_r = Reliability\ Factor,$
$Z = Contact\ Stress\ Cycle\ Number, C_h = Hardness\ Ratio,$
$\sigma = Contact\ Stress\ Number$

$$Allowable\ Contact\ Stress\ Number_{pinion} = \frac{1.25 * 1 * .85}{.917 * 1} * 116643.873$$

$$Allowable\ Contact\ Stress\ Number_{pinion} = 135144.557$$

$$Allowable\ Contact\ Stress\ Number_{gear} = \frac{1.25 * 1 * .85}{.943 * 1} * 114851.036$$

$$Allowable\ Contact\ Stress\ Number_{gear} = 129287.929$$

Hardness for Contact Stress

$$Hardness\ for\ Contact\ Stress = \frac{Allowable\ Contact\ Stress\ Number - 29100}{322}$$

$$Hardness\ for\ Contact\ Stress_{pinion} = \frac{135144.557 - 29100}{322}$$

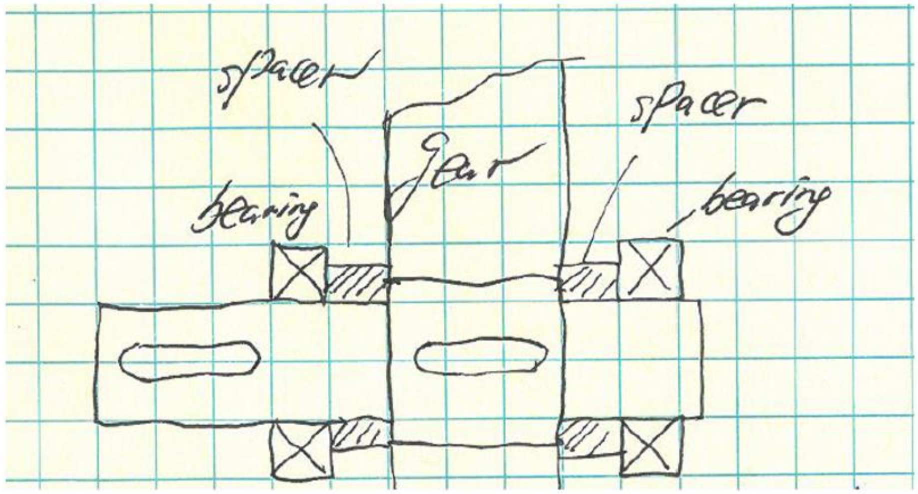
$$Hardness\ for\ Contact\ Stress_{pinion} = 329.33$$

$$Hardness\ for\ Contact\ Stress_{gear} = \frac{129287.929 - 29100}{322}$$

$$Hardness\ for\ Contact\ Stress_{gear} = 311.14$$

Shaft Design

All effort thus far has been put into the gear design. However, gears are useless without a shaft to rotate on. The shaft design was done in distinct, subsequent steps in order to create a prototype shaft based on assumptions which was then modified and refined to better suit the specific gearbox being created. The first major step was to choose a shaft layout which defined the basic shape of the piece as well as what other components would be used and their relative location on the shaft. Five possible layouts were suggested and it was decided to use layout number five for the design of the shaft because of several of its desirable traits. One, it was symmetric about the fixture points of the bearings which would make for easy force calculations and machining. Two, it did not use any retaining rings to hold components in place which would have to be cut into the shaft thereby weakening it. And three, it contained at least one shoulder which would prevent the shaft from shifting or sliding out of the assembly along its axis, and the thicker shoulder would strengthen the weakness caused by the key seat groove in the shaft. These things can be observed in figure 3.



Layout 5:

Spacer + shoulder + shoulder +spacer layout

Figure 3: Layout 5

Once the basic layout of the shaft was determined, the lengths of the components and therefore the total shaft length could be estimated to perform the force analysis on the part. Much of the lengths were provided as guidelines in the original specifications. For instance the length of the input connection for the engine should be 1.5 inches. Spacer lengths would be 0.5 inches, while 1 inch bearing widths were assumed. Also an end length of 0.25 was used on the opposite side of the input. One dimension was driven by the gear calculations which was the shoulder length for the gear to rest on, defined as 1.2 in the Excel sheet for the face width of the gear but rounded up to 1.5 to allow room for the hub. These estimates gave a total shaft length of 6.25 inches and a length of 2.5 inches of critical length on the inside of the bearings.

Now that the length of the shaft was defined the force analysis could commence. For the analysis the shaft was simplified to a supported beam with an applied external load and a torque force. These resultant forces on the shaft were calculated from the tangential and radial forces experienced by the meshing gears as well as the gears' pitch diameter. Then the reaction forces and torques at each point along the shaft were calculated using the static equations of sum of the forces and sum of the moments. The resultant force was the same for the pinion shaft and the gear shaft at 310.5 lbs_f but the torques were different with 350 ft-lbs on the pinion shaft and 1225.5 ft-lbs on the gear shaft. These were subsequently used to create shear force and bending moment diagrams for the shaft as can be seen in Figure 4.

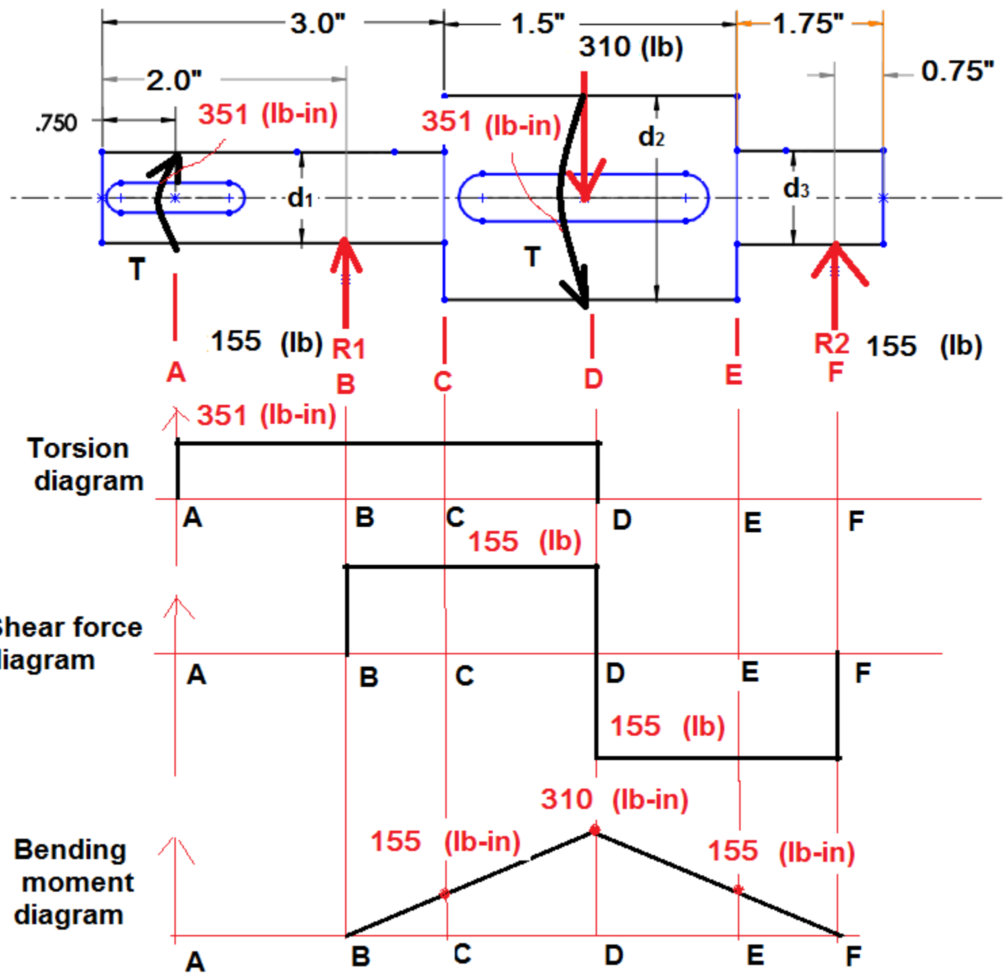


Figure 4: Shear Force, Bending Moment, Torsion Diagrams

When the internal forces were known at every point along the shaft, the diameters of the shaft section could be found. However, before any calculations a material needed to be specified for the shaft. The material selected for both shafts was AISI 1020 steel because it is a high yield strength, ductile material that can bend without breaking and is common and relatively cheap. With material specified step four of the shaft design process (defining the diameters) could continue.

The diameters of the shaft sections were determined by two different methods in which the larger, safer shaft was ultimately selected. The first method was using a static analysis of the shaft with a factor of safety of 3.25, and aimed to prevent bending and shearing. This gave dimensions for each of the critical sections A, B, C, D, E, and F. Since there were only three shaft sections of different diameters, the max diameter of the critical sections was chosen for a shaft section. For example for d1 on the pinion shaft the calculations showed A=0.84 in, B=0.58 in, and C=0.70 in, therefore a diameter of 0.84 in was picked for d1 under static analysis. Likewise, d2 was found to be 0.9 inches and d3 equaled 0.56 inches.

A very similar procedure was used for the second method used to define the shaft diameters, which was a fatigue analysis with a factor of safety of 1.25, aimed at preventing failure from repeated loadings and unloading or negative loadings. The main difference in the equation being that the endurance limit of the component was incorporated and included the endurance limit for the material and many stress concentration factors. These calculations produce the following diameters d1= 0.61 in, d2=0.80 in, d3=0.51 in.

Table 7–1

First Iteration Estimates for Stress-Concentration Factors K_t and K_{ts}.

Warning: These factors are only estimates for use when actual dimensions are not yet determined. Do *not* use these once actual dimensions are available.

	Bending	Torsional	Axial
Shoulder fillet—sharp (r/d = 0.02)	2.7	2.2	3.0
Shoulder fillet—well rounded (r/d = 0.1)	1.7	1.5	1.9
End-mill keyseat (r/d = 0.02)	2.14	3.0	—
Sled runner keyseat	1.7	—	—
Retaining ring groove	5.0	3.0	5.0

Missing values in the table are not readily available.

Figure 5: First Iteration Estimates for Stress Concentration Factors K_t,K_ts [1]

The last step of the diameter analysis was to pick the safest largest diameters for the shaft. The maximum values for each section was chosen. Since all of the dimension under static analysis were bigger, those diameters were used to define the actual minimum diameters of the pinion shaft. These unique minimum numbers were then rounded up to fit standard bearing ID sizes defined by bearing manufacturer McMaster-Carr Supply Company. Therefore d1 was bumped up from 0.84 to 0.875, and d2 from 0.9 to 1 inch. Besides matching standard part sizes, the shaft diameters were also modified to consider ease of manufacturing and assembly. In order to minimize the number of different parts it was decided to make d3 equal to d1 which would then allow the same bearing to be used for both shaft ends. A similar thought process was used when defining the gear shaft diameters. One end of the gear shaft, d3 was calculated to only be 0.55 in which rounded up to a standard size was still only 0.625 in. It was decided to make this match the smaller diameter of the pinion to make manufacturing easier and use the same bearing used on the pinion.

The shaft design had now been complete and simply needed to be verified by the fifth and final step in the shaft design process. The deflection under loading needed to be determined and compared to the maximum allowable deflection which is defined as 0.003 in for this particular design. Also the natural frequency of the shaft would be found and compared to the rpm required to avoid resonance. Both of these things were determined using SolidWorks Finite Element Analysis simulation and passed as can be seen from Figures 7-10. The maximum deflection experienced by the shaft was 0.0001361 inches for the pinion shaft and 0.0001022 inches for the gear shaft both of which were much less than the max deflection. The natural frequency for the pinion was 515340 rpm and 443316 rpm for the gear, which were both very far away from the desired rpms of 1800 rpm and 514 rpm.

Table 7–2

Typical Maximum Ranges for Slopes and Transverse Deflections

Slopes	
Tapered roller	0.0005–0.0012 rad
Cylindrical roller	0.0008–0.0012 rad
Deep-groove ball	0.001–0.003 rad
Spherical ball	0.026–0.052 rad
Self-align ball	0.026–0.052 rad
Uncrowned spur gear	< 0.0005 rad
Transverse Deflections	
Spur gears with $P < 10$ teeth/in	0.010 in
Spur gears with $11 < P < 19$	0.005 in
Spur gears with $20 < P < 50$	0.003 in

Figure 6: Maximum Deflections [1]

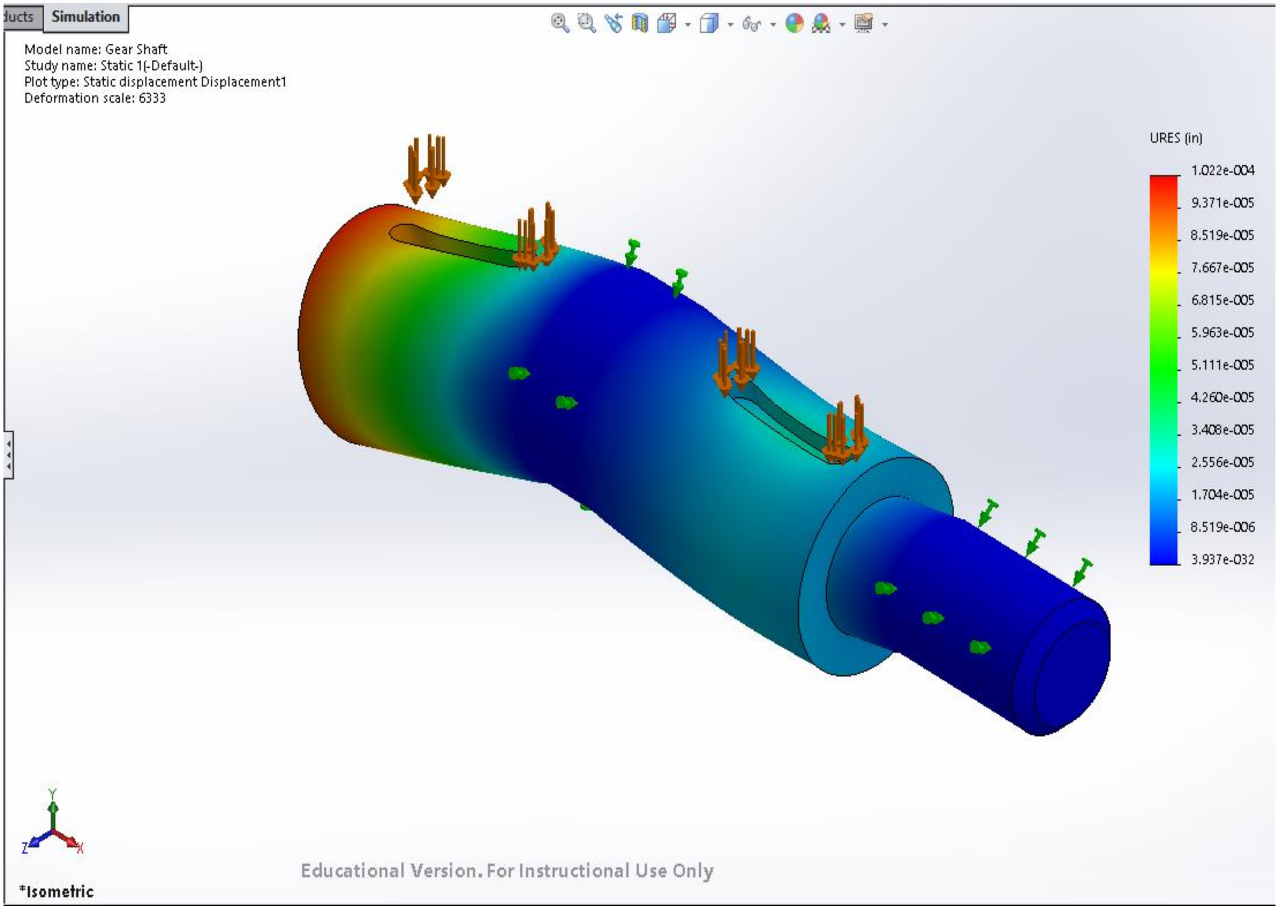


Figure 7: Deflection of Gear Shaft

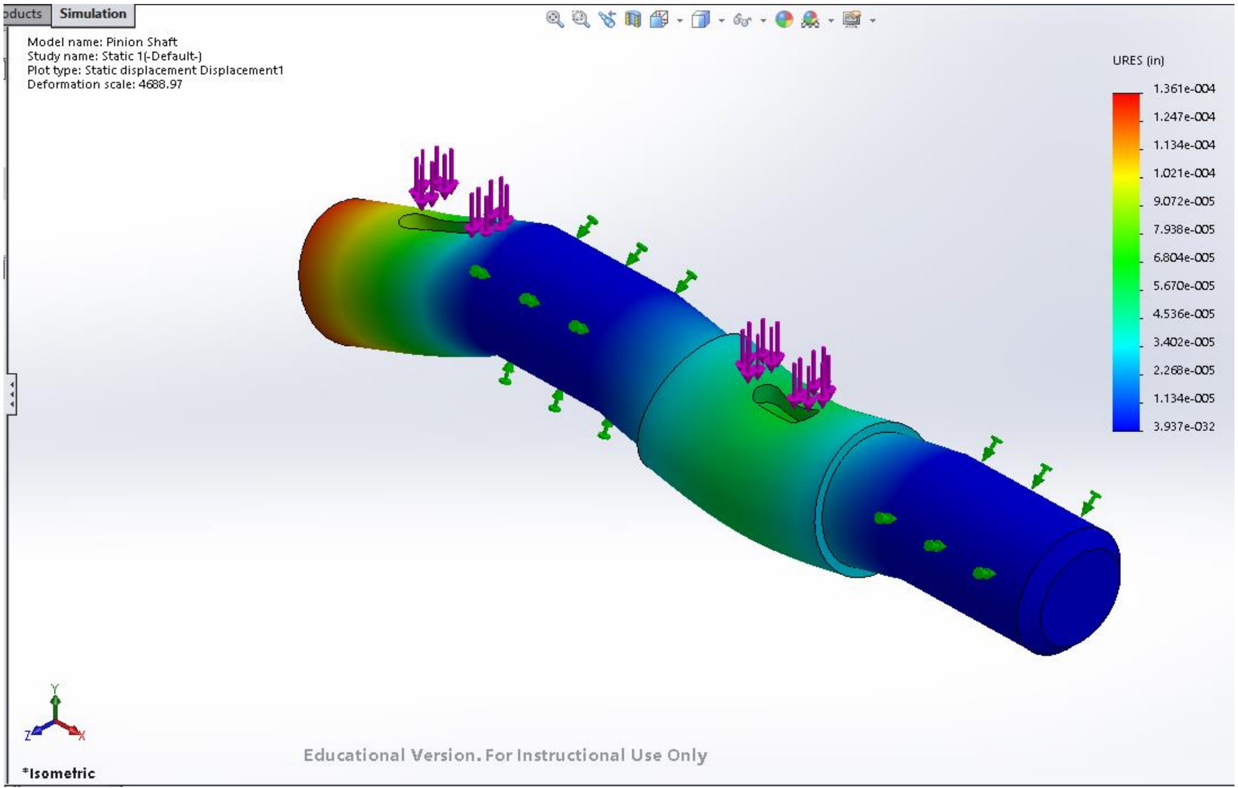


Figure 8: Deflection of Pinion Shaft

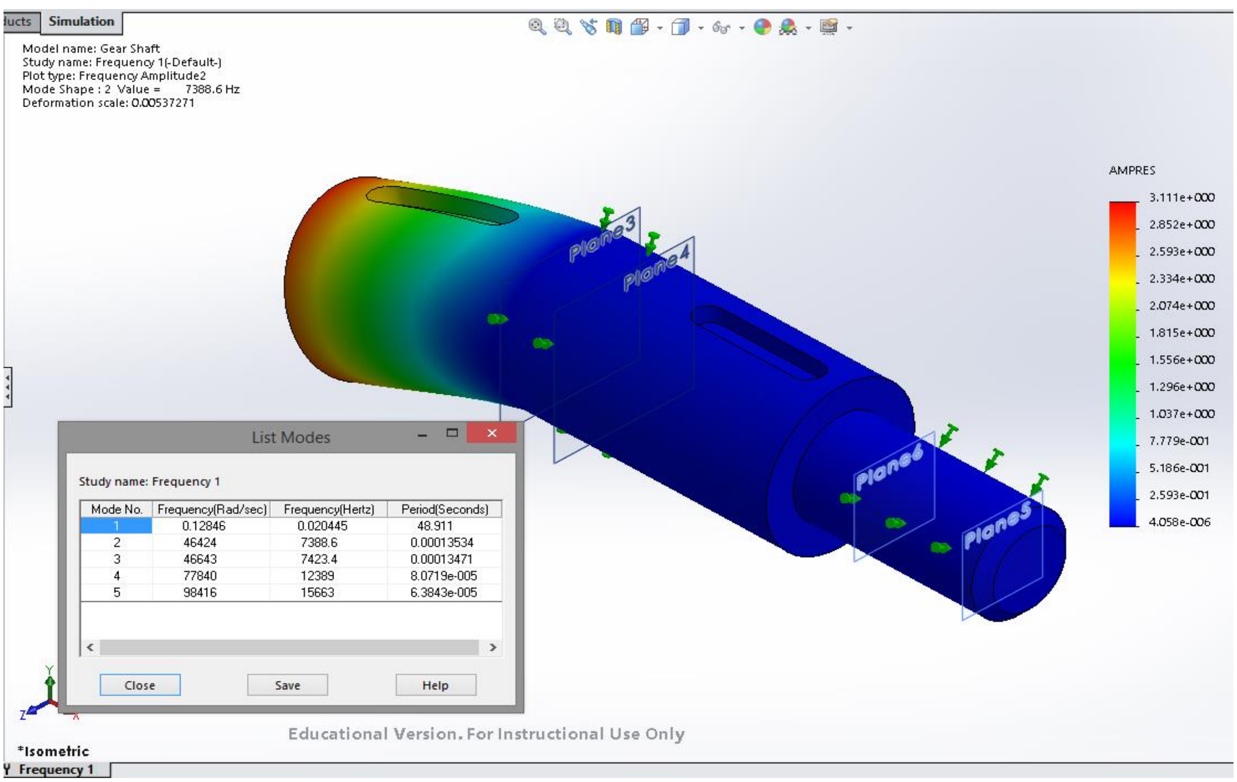


Figure 9: Natural Frequency of Gear Shaft

Gear Shaft	
Frequency(Hertz)	RPM
7388.6	443316
7423.4	445404
12389	743340
15663	939780

Table 4: Gear Shaft Natural Frequencies

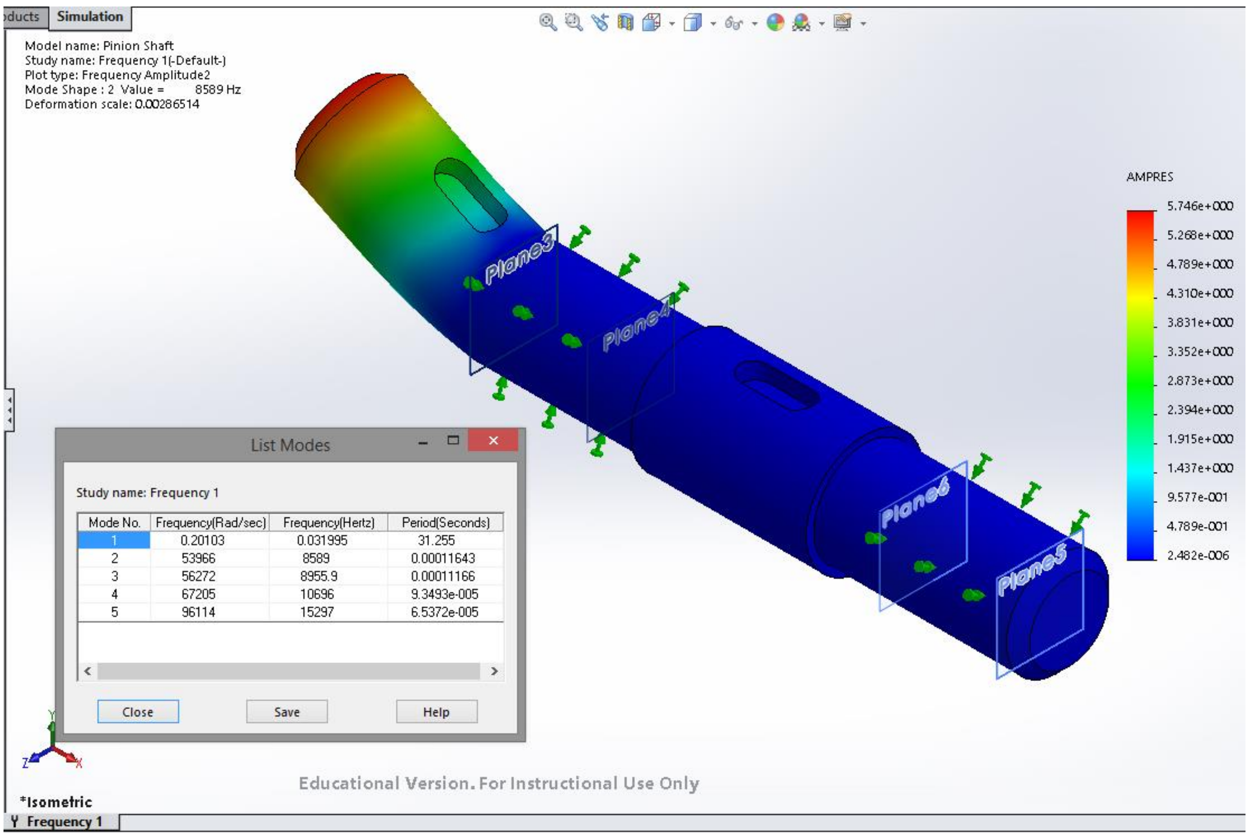


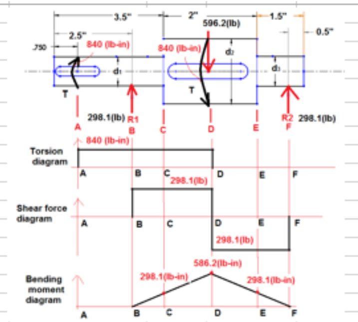
Figure 10: Natural Frequencies of Pinion Shaft

Pinion Shaft	
Frequency(Hertz)	RPM
8589	515340
8955.9	537354
10696	641760
15297	917820

Table 5: Natural Frequencies of Pinion Shaft

Detailed Calculations

Shaft calculations were done with the aid of a Microsoft Excel spreadsheet, and followed equations given in lecture slides.



Pitch Diameter Pinion D_p	2.4
Tangential Force W_t	291.7840623
Radial Force W_r	106.2007135
Resultant Force Fr	310.5101135
Torque T	350.1408748
Reaction Force 1 (R1)	155.2550567
Reaction Force 2 (R2)	155.2550567
Reaction Torque 1	350.1408748
Material for Shaft	AISI 1020 steel
Ultimate Strength (Sut)	60989.38 psi
Yield Strength (Sy)	50991.06 psi

3. Shaft design based on static loading

We need to consider static failure, even in fatigue situation. Use the strain energy theory to run the design based on static loading

$$n_s = \frac{S_y \pi d^3}{16\sqrt{4(K_t M_a)^2 + 3(K_{ts} T_m)^2}} \qquad (7-15)$$

$$d = \left[\frac{16 n_s \sqrt{4(K_t M_a)^2 + 3(K_{ts} T_m)^2}}{\pi S_y}\right]^{1/3} \qquad (7-16)$$

n_s is the factor of safety under static loading;
S_y is the yield strength, d is the diameter of the shaft
K_t is the stress concentration under static bending
K_{ts} is the stress concentration factor under static torsion
M_a and T_m are the bending moment and the torsion

2 Shaft Design Equation based on fatigue

The design equation for the infinite life by using Modified-Goodman approach is:

$$\frac{1}{n_f} = \frac{16}{\pi d^3}\left(\frac{2K_f M_a}{S_e} + \frac{\sqrt{3}K_{fs}T_m}{S_{ut}}\right) \qquad (7-7)$$

The design equation for diameter :

$$d = \left[\frac{16 n_f}{\pi}\left(\frac{2K_f M_a}{S_e} + \frac{\sqrt{3}K_{fs}T_m}{S_{ut}}\right)\right]^{1/3} \qquad (7-8)$$

d is the daimeter of the shaft under the consideration
n_f is the fatigue factor of safety for infinite fatigue life
S_e is the endurance limit of the component for cyclic bending stress
S_{ut} is the untimate strength of the materials
K_f and K_{fs} are the fatigue stress bending and shearing concentration factors
M_a and T_m are the bending moment and torsion

Calculations for Static Loading

Factor of safety (ns) 3.25

Section	Bending Stress Concentration Factor (Kt)	Torsion Stress Concentration Factor (Kts)	Bending Moment (Ma)	Torsion™	Diameter (in)
A	2.14	3	0	350.1408748	0.838998868
B	1	1	0	350.1408748	0.581729325
C	1.7	1.5	155.2550567	350.1408748	0.698915075
D	2.14	3	310.5101135	350.1408748	0.900973203
E	1.7	1.5	155.2550567	0	0.555428338
F	1	1	0	0	0

d1 Section A-C	0.838998868
d2 Section C-E	0.900973203
d3 Section E-F	0.555428338

Final Minimum Diameters	
d1	0.838998868
d2	0.900973203
d3	0.555428338

from mcmaster 7/8 1

ndard Diameters based on Bear	
d1	0.875
d2	1
d3	0.875

from solidworks

11.2

Calculations for Fatigue Loading

Assuming for all cases q = 0.8 Factor of safety (nf) 1.25

Section	Bending Stress Concentration Factor (Kf)	Torsion Stress Concentration Factor (Kfs)	Bending Moment (Ma)	Torsion™	Endurance Limit (Se)	Diameter (in)
A	1.912	2.6	0	350.14	22255.3521	0.5480257
B	1	1	0	350.14	22255.3521	0.3985442
C	1.56	1.4	155.2550567	350.14	22255.3521	0.6101852
D	1.912	2.6	310.5101135	350.14	22086.2898	0.7973064
E	1.56	1.4	155.2550567	0	23259.573	0.5099093
F	1	1	0	0	23259.573	0

d1 Section A-C	0.61018522
d2 Section C-E	0.797306384
d3 Section E-F	0.509909275

Endurance Limit Calcs

Endurance Limit of Material (S'e)	30495	
Surface factor (ka)	0.9084	machine
Size factor (kb) d1	0.8957	
Size factor (kb) d2	0.8889	
Size factor (kb) d3	0.9361	
Load factor (kc)	1	
Temp factor (kd)	1	
Reliability factor (ke)	0.897	

Reliability, %	Transformation Variate z_a	Reliability Factor k_e
50	0	1.000
90	1.288	0.897
95	1.645	0.868
99	2.326	0.814
99.9	3.091	0.753
99.99	3.719	0.702
99.999	4.265	0.659
99.9999	4.753	0.620

First Iteration Estimates for Stress-Concentration Factors K_t and K_{ts}.
Warning: These factors are only estimates for use when actual dimensions are not yet determined. Do *not* use these once actual dimensions are available.

	Bending	Torsional	Axial
Shoulder fillet—sharp (r/d = 0.02)	2.7	2.2	3.0
Shoulder fillet—well rounded (r/d = 0.1)	1.7	1.5	1.9
End-mill keyseat (r/d = 0.02)	2.14	3.0	—
Sled runner keyseat	1.7	—	—
Retaining ring groove	5.0	3.0	5.0

Missing values in the table are not readily available.

Gears | Pinion Shaft | Sheet1 | Gear Shaft | Pinion Key | Gear Key | Bearing

Figure 11: Excel Screenshot of Shaft Design Calculations

Resultant Force

$$Resultant\ Force\ F_R = \sqrt{(Tangential\ Force)^2 + (Radial\ Force)^2}$$

$$Resultant\ Force\ F_R = \sqrt{(291.8)^2 + (106.2)^2}$$

$$Resultant\ Force\ F_R = 310.5$$

Torque

$$Torque\ T = Tangential\ Force * \frac{Pitch\ Diameter}{2}$$

$$Torque_{Pinion} = 291.8 * \frac{2.4}{2}$$

$$Torque_{Pinion} = 350$$

$$Torque_{Gear} = 291.8 * \frac{8.4}{2}$$

$$Torque_{Gear} = 1225.5$$

Shaft Diameter

3. Shaft design based on static loading

We need to consider static failure, even in fatigue situation. Use the strain energy theory to run the design based on static loading

$$n_s = \frac{S_y \pi d^3}{16\sqrt{4(K_t M_a)^2 + 3(K_{ts} T_m)^2}} \qquad (7-15)$$

$$d = \left[\frac{16 n_s \sqrt{4(K_t M_a)^2 + 3(K_{ts} T_m)^2}}{\pi S_y}\right]^{1/3} \qquad (7-16)$$

n_s is the factor of safety under static loading;

S_y is the yield strength, d is the diameter of the shaft

K_t is the stress concentration under static bending

K_{ts} is the stress concentration factor under static torsion

M_a and T_m are the bending moment and the torsion

$$d_{A_{Gear}} = \left[\frac{16 * 3.25 * \sqrt{4 * (2.14 * 0)^2 + 3 * (3 * 1225.5)^2}}{\pi * 50991.06}\right]^{\frac{1}{3}}$$

$$d_{A_{Gear}} = 1.27$$

2 Shaft Design Equation based on fatigue

The design equation for the infinite life by using Modified-Goodman approach is:

$$\frac{1}{n_f} = \frac{16}{\pi d^3}\left(\frac{2K_f M_a}{S_e} + \frac{\sqrt{3}K_{fs}T_m}{S_{ut}}\right) \tag{7-7}$$

The design eqauation for diameter :

$$d = \left[\frac{16 n_f}{\pi}\left(\frac{2K_f M_a}{S_e} + \frac{\sqrt{3}K_{fs}T_m}{S_{ut}}\right)\right]^{1/3} \tag{7-8}$$

d is the daimeter of the shaft under the consideration

n_f is the fatigue factor of safety for infinite fatigue life

S_e is the endurance limit of the component for cyclic bending stress

S_{ut} is the untimate strength of the materials

K_f and K_{fs} are the fatigue stress bending and shearing concentration factors

M_a and T_m are the bending moment and torsion

$$d_{A_{Gear}} = \left[\frac{16 * 1.25}{\pi}\left(\frac{2 * 1.912 * 0}{21282.83} + \frac{\sqrt{3} * 2.6 * 1225.5}{60989.38}\right)\right]^{\frac{1}{3}}$$

$$d_{A_{Gear}} = 0.832$$

<u>Bending Stress Concentration Factor</u>

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

$$K_f A = 1 + 0.8 * (2.14 - 1) = 1.912$$

<u>Endurance Limit</u>

$$S_e = S'_e * K_a * K_b * K_c * K_d * K_e$$

$$S_e = 30494.7 * 0.908 * 0.857 * 1 * 1 * 0.897$$

Mechanical Key Design

Once the diameters of the shaft were finalized, it was possible to begin the key design process. While following ANSI B17.1-1967, standards for the keys were established. These standards such as width with tolerance, height with tolerances, key seat depth, fillet radius, and chamfer size were all based off of the nominal shaft diameter and extracted from tables 10, 11, 12 in the appendix. The key standards for the gear and pinion key design can be seen in tables 6 and 7 below.

Key Standards						
Nominal Shaft Diameter			**Nominal Key Size**			
Over	1.250	in		Width, W	0.31	in
To (incl.)	1.375	in		Height, H (Square)	0.31	in
Tolerances			**Keyseat**			
Width	+0.002	-0.000		Depth	0.15625	in
Height	+0.002	-0.000		Fillet Radius	0.03125	in
				45 deg Chamfer	0.046875	in

Table 6: Gear Key Standards

Key Standards						
Nominal Shaft Diameter			**Nominal Key Size**			
Over	0.875	in		Width, W	0.25	in
To (incl.)	1.25	in		Height, H (Square)	0.25	in
Tolerances			**Keyseat**			
Width	+0.001	-0.000		Depth	0.125	in
Height	+0.001	-0.000		Fillet Radius	0.0313	in
				45 deg Chamfer	0.04688	in

Table 7: Pinion Key Standards

In addition to the key standards being used for the ultimate key design calculation, were the torque, key factor of safety, shaft diameter, and the yield strength of AISI 1015 CD steel. AISI 1015 CD steel was chosen for the material of the key because it is weaker than the material of the shaft and gear. This was a preference because the key acts as a fuse, which breaks in the case of extreme stress.

The torque being applied to the gear and the pinion shafts were calculated to be 1225 in-lb and 350 in-lb respectively. A factor of safety value of 3 was used for both shafts as well as the yield strength of 47137 psi for AISI 1015 CD for both shafts. The gear shaft diameter was 1.375 inches and the diameter of the pinion shaft was 1 inch. Both sections of each individual shaft with key slots were made equal to allow for only 1 key size to be used for each shaft as well as to require only 2 different sized bearings for the entire design.

Using the key standards and the variables outlined above, the key length, key seat width, shaft key seat, and the hub key seat were calculated. The results for these pinion and gear key calculations are represented in tables 5 and 6 below and they outline the dimensions of the keys. Figures 12 and 13 below show the dimension with which the shaft key seat and hub key seat represent.

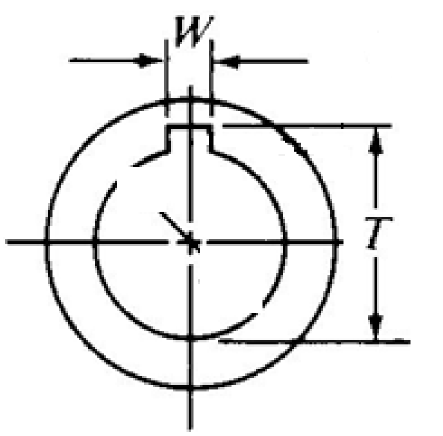


Figure 13: Depth of Hub Key Seat [2]

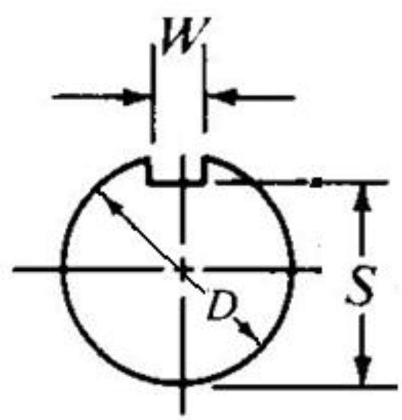


Figure 12: Depth of Shaft Key Seat [2]

Outputs		
Key Length from Shear L_s	0.726068	
Key Length from Compression L_c	0.726068	
Keyseat Width W	0.3125	+0.002/-0.000
Shaft Keyseat S	1.200759	+0.000/-0.015
Hub Keyseat T	1.513259	+0.010/-0.000

Table 8: Gear Key Dimensions

	Outputs		
	Key Length from Shear L_s	0.356551	
	Key Length from Compression L_c	0.356551	
	Keyseat Width W	0.25	+0.002/-0.000
	Shaft Keyseat S	0.859123	+0.000/-0.000
	Hub Keyseat T	1.109123	+0.010/-0.000

Table 9: Pinion Key 1 Dimensions

	Outputs		
	Key Length from Shear L_s	0.407487	
	Key Length from Compression L_c	0.407487	
	Keyseat Width W	0.25	+0.002/-0.000
	Shaft Keyseat S	0.731763	+0.000/-0.000
	Hub Keyseat T	0.981763	+0.010/-0.000

Table 10: Pinion Key 2 Dimensions

These calculations established the dimensions for the keys as well as the key seats. The key seats were implemented into the shafts as well as the gears. It was important to ensure that the width of the gear hub was wide enough to accommodate the length of the keys.

Detailed Calculations

Gear Key Design

Key Length from Shear

$$L_s = \frac{(4Tn_{key})}{S_y dW} = \frac{(4*1225.49*3)}{(47137*1.375*0.3125)} = 0.72607 \text{ in}$$

$$L_s = Key\ Length\ from\ Shear, T = Torque, n_{key} = Key\ Factor\ of\ Safety, S_y = AISI\ 1015\ Yield\ Strength\ (psi), d = Shaft\ Diameter, W = Key\ Width$$

Key Length from Compression

$$L_C = \frac{(4Tn_{key})}{S_y dH} = \frac{(4*1225.49*3)}{(47137*1.375*0.3125)} = 0.72607 \text{ in}$$

$$L_C = Key\ Length\ from\ Compression, T = Torque, n_{key} = Key\ Factor\ of\ Safety, S_y = AISI\ 1015\ Yield\ Strength\ (psi), d = Shaft\ Diameter, H = Key\ Height$$

Shaft Key Seat

$$S = \frac{\left(d - H + \sqrt{d^2 - W^2}\right)}{2} = \frac{1 - 0.25 + \sqrt{1^2 - 0.3125^2}}{2} = 1.201 \frac{+0.000}{-0.015}$$

$$S = Shaft\ Key\ Seat, d = Shaft\ Diameter, H = Key\ Height, W = Key\ Width$$

Hub Key Seat

$$T = \frac{\left(d + H + \sqrt{d^2 - W^2}\right)}{2} = \frac{1 + 0.25 + \sqrt{1^2 - 0.3125^2}}{2} = 1.513 \frac{+0.010}{-0.000}$$

$$T = Hub\ Key\ Seat, d = Shaft\ Diameter, H = Key\ Height, W = Key\ Width$$

Pinion Key Design

(Same process repeated for second key for use with a diameter of 0.875 in)

Key Length from Shear

$$L_s = \frac{(4Tn_{key})}{S_y dW} = \frac{(4*350*3)}{(47137*1*0.25)} = 0.357 \text{ in}$$

$$L_s = Key\ Length\ from\ Shear, T = Torque, n_{key} = Key\ Factor\ of\ Safety, S_y = AISI\ 1015\ Yield\ Strength\ (psi), d = Shaft\ Diameter, W = Key\ Width$$

Key Length from Compression

$$L_C = \frac{(4Tn_{key})}{S_y dW} = \frac{(4*350*3)}{(47137*1*0.25)} = 0.357 \text{ in}$$

$$L_C = Key\ Length\ from\ Compression, T = Torque, n_{key} = Key\ Factor\ of\ Safety, S_y = AISI\ 1015\ Yield\ Strength\ (psi), d = Shaft\ Diameter, H = Key\ Height$$

Shaft Key Seat

$$S = \frac{\left(d - H + \sqrt{d^2 - W^2}\right)}{2} = \frac{1 - 0.25 + \sqrt{1^2 - 0.3125^2}}{2} = 0.859 \frac{+0.000}{-0.0000}$$

$$S = Shaft\ Key\ Seat, d = Shaft\ Diameter, H = Key\ Height, W = Key\ Width$$

Hub Key Seat

$$T = \frac{\left(d + H + \sqrt{d^2 - W^2}\right)}{2} = \frac{1 + 0.25 + \sqrt{1^2 - 0.3125^2}}{2} = 1.109 \frac{+0.010}{-0.000}$$

$$T = Hub\ Key\ Seat, d = Shaft\ Diameter, H = Key\ Height, W = Key\ Width$$

Bearing Design

The function of a bearing is to allow constrained rotation movement between two or more parts. It supports the shaft and is located in between the shaft and the housing. Bearing come in standard sizes to reduce the cost of manufacturing new ones for each design. The bearing used in this gear box design came for McMaster-Carr (see Appendix table #). In order to choose the bearings the required dynamic load at rated life must be calculated. The variables used in the calculation are shown in Figure 14.

Pinion Shaft								
	Reaction Force 1 (R1)	155.255			**Shaft Diameters**			
	Reaction Force 2 (R2)	155.255			Final Minimum Diameters			
	Reaction Torque 1	350.141			d1	0.84451		
					d2	0.90689		
					d3	0.55907		
	Resultant Force Fr	310.51			Standard Diameters based on Bearings			
	Lr	10^6			d1	0.875		
	a	3			d2	1.0625		
	af	1			d3	0.875		
	v	1						
	Input Rotational Speed		1800	rpm	h	25		
				Bearing	R14	Thickness	0.3750	
	Cr10	432.378						

Figure 14: Dynamic Load Variables

$L_D = 60hn$

L_D – the design life(revoulatio), F_D – the design load

h – design life in hours from table 11 - 4, n – The bearing rotation speed (rpm)

When the bearing is selected, the application factor a_f will be used.

The application factor a_f is the factor of safety, from table 11 - 5.

$$C_{R10} = a_f V F_D \left(\frac{L_D}{L_R}\right)^{1/a} = a_f V F_D (x_D)^{1/a} = a_f V F_D \left(\frac{60hn}{L_{10}}\right)^{1/a} \qquad (11\text{-}3)$$

$x_D = L_D / L_R$ – the dimensionless multiple of the rating life.

a_f – The load application factor, that is, the factor of safety, from table 11 - 5.

C_{R10} – the required dynamic load at the rated life L_{10}. This is the value for choosing the bearing. For the selected bearing $C_{R10} \le C_{10}$

V – the rotation factor, V = 1 for inner ring rotation, V = 1.2 for outer ring rotation

C_{10} – the dynamic load rating of the bearing.

Design life in hours was chosen from table 15 and the load application factor was picked from table 16.

Detailed Calculations

$$C_{R10} = 1 * 1 \left(\frac{60 * 25 * 1800rpm}{10^6}\right)^{\frac{1}{3}} = 432.378lb$$

Once the required dynamic load is calculated a standard bearing has to be pick with a dynamic load rating equal or greater, with the right inner diameter for the shaft to fit. The McMaster bearings (figure 15) used in this design are R14 for the pinion shaft and R22 for the gear shaft.

Housing Design

The housing for this design is a welded plate housing. The housing has two halves a top and bottom that are bolted together in the final assembly. It is made out of four parts and the supports that are welded to the side of the case under each shaft. The bearing holding plates are CNC machined and support the bearings and in turn the shafts. The top and bottom plats are held to their respective halves of the case.

Drawings for Components and Assemblies

Manufacturing Routine of Component Assembly

Shafts:

1. Start with steel rod of two diameters corresponding with the largest portions of the pinion and gear shafts which are 1” for the pinion and 1.375” for the gear.
2. Use the lathe to machine to smaller diameters specified in drawings.
3. Use milling machine to cut in key seats as specified in drawings.
4. Chamfer the ends of the shafts to specified dimensions.

Housing:

1. Take the plate of sheet metal and bend it into the required shape for both the top and bottom case sections of the gearbox.
2. Use CNC milling machine to create the bearing holders from blocks of steel.
3. Weld the bearing holders to the top and bottom portions of the case respectively.
4. Cut out gussets from sheet metal and weld them to the bearing holders and housing for support.

Spacers:

1. Start with brass cylinder and hollow it out using the lathe machine to the specified dimensions.
2. Chamfer the inside edges of the spacers.

Assembly Procedures

Sub Assembly 1: Pinion Assembly

1. Start with the pinion shaft and place the keys into the key seats.
2. Slide the pinion onto the pinion shaft lining up the key with the keyway on the pinion.
3. Slide the spacers onto both ends of the pinion shaft flush with the pinion.
4. Press fit the bearings onto the ends of the shaft.

Sub Assembly 2: Gear Assembly

1. Start with the gear shaft and place the keys into the key seats.
2. Slide the gear onto the gear shaft lining up the key with the keyway on the gear.
3. Slide the spacers onto both ends of the gear shaft flush with the gear.
4. Press fit the bearings onto the ends of the shaft.

Complete Assembly

1. Using the lower portion of the housing as a base, place the pinion assembly into the allotted slot in the case.
2. Place the gear assembly into the allotted slot in the case making sure the gear and pinion teeth are intertwined.
3. Place the top portion of the housing on and bolt it down to the lower portion of the housing

Drawings for Component and Assemblies

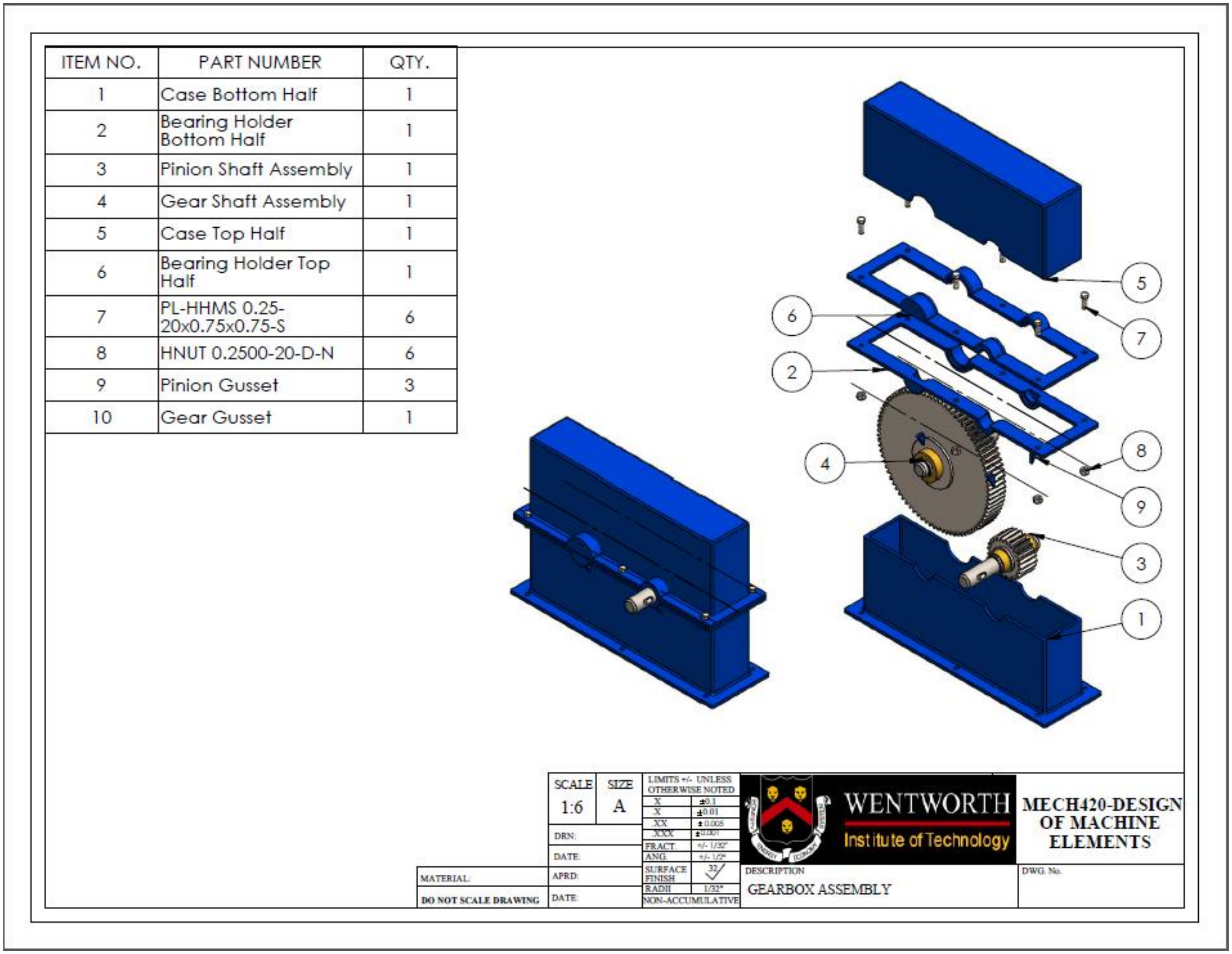
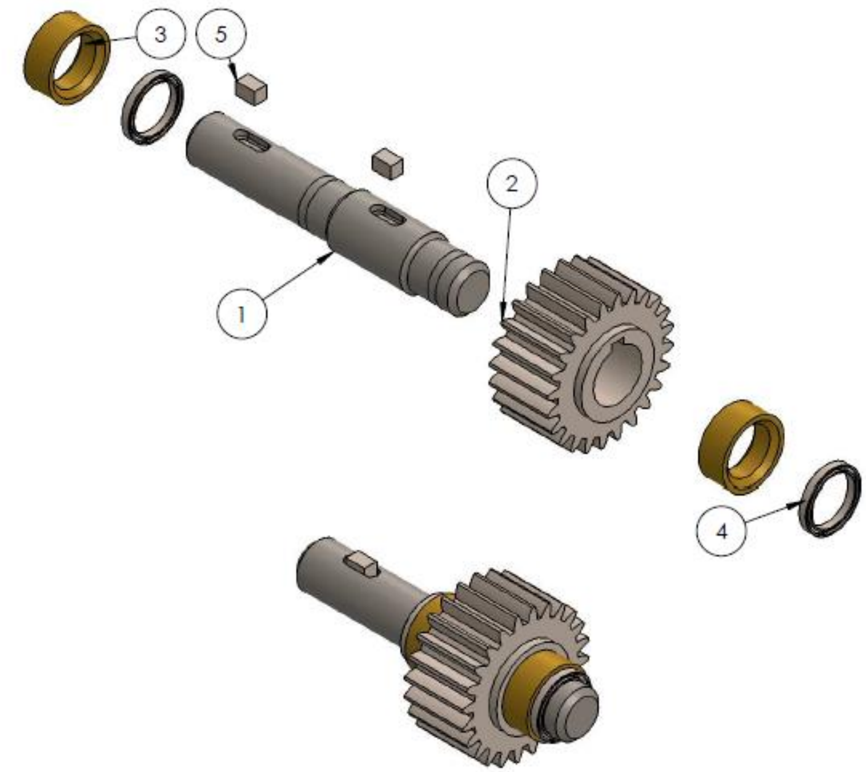


Figure 15: Gearbox Assembly Drawing

ITEM NO.	PART NUMBER	QTY.
1	Pinion Shaft	1
2	Inch - Spur gear 10DP 24T 20PA 1.2FW --- S24B1.5H1.5L1.0S1	1
3	Pinion Shaft Spacer	2
4	MCMASTER-CARR BEARING NO. R14	2
5	Key B17.1 0.25x0.25x.38	2



SCALE	SIZE	LIMITS +/- UNLESS OTHERWISE NOTED		WENTWORTH Institute of Technology	MECH420-DESIGN OF MACHINE ELEMENTS
1:2	A	.X	±0.1		
		.X	±0.01		
		.XX	±0.005		
DRN:GROUP 1		.XXX	±0.001		
DATE:12/7/14		FRACT.	+/- 1/32"		
		ANG.	+/- 1/2°		
MATERIAL: APRD:		SURFACE FINISH	32	DESCRIPTION	DWG. No.
DO NOT SCALE DRAWING DATE:		RADII	1/32"	PINION SHAFT ASSEMBLY	
		NON-ACCUMULATIVE			

Figure 16: Pinion Shaft Sub Assembly

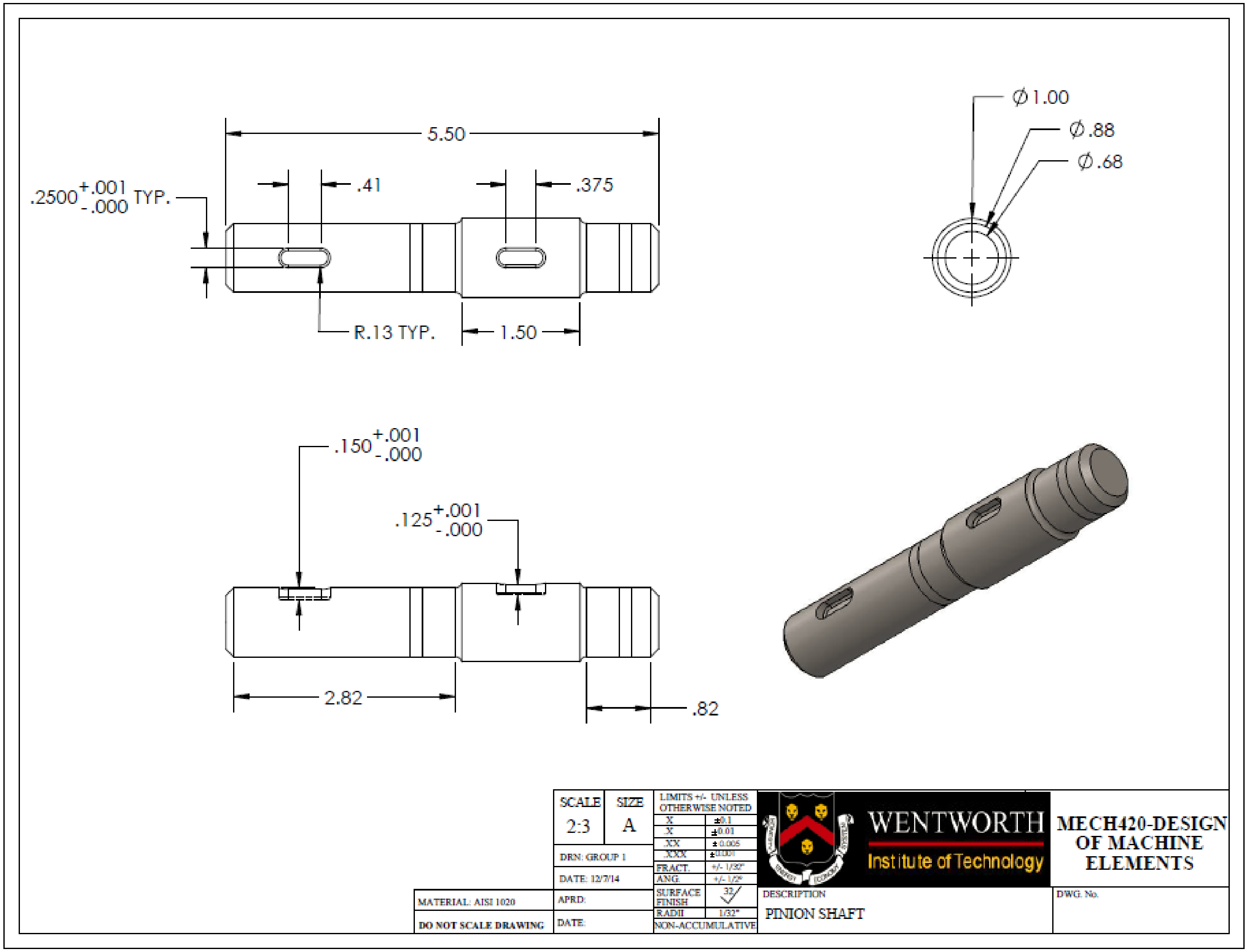


Figure 17: Pinion Shaft

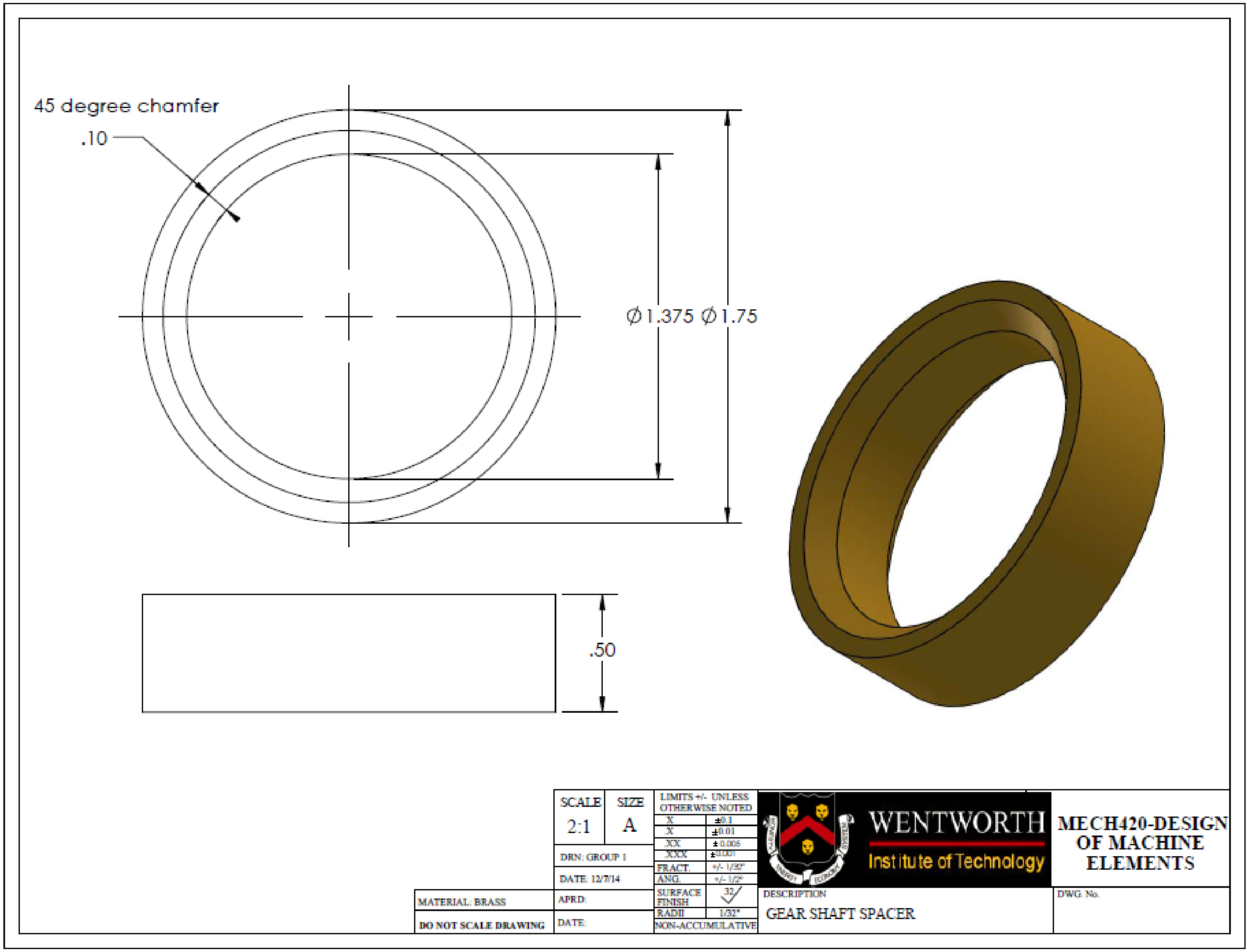


Figure 18: Gear Shaft Spacer

Discussions and Conclusions

After creating a design interface program with Microsoft Excel, choosing component materials, calculating dimensions, and modeling the assembly within SolidWorks, it is safe to say that the design intent was met. The design specifications called for:

- ✓ WIT-MEGB-01 Model Gearbox
  - ❖ 10 horsepower input
  - ❖ 3.5 gear ratio with 1% Variation
- ✓ A delivered power of 15~30 horsepower with an input speed of 1800 rpm.
- ✓ The design life of the gears and bearings will be 4000 hours.
- ✓ A factor of safety of 3.25 for shaft static design and 1.25 for shaft fatigue design.
- ✓ The reliability index for the gear, bearing, and shaft will be 0.90.
- ✓ Mechanical keys will be used for the connection between the power source and the driven device

These specifications were met by designing the individual components and implementing them into a single gear box. Even though the components were designed individually, it is important to note that the dimensions of the parts were dependent on one another. This dependence allows for the entire system to be used efficiently in order to deliver the required power from the input.

The gear and the pinion were among the first component designs to be conducted. Ultimately, the pinion was designed to have 24 teeth while the gear was designed to have 84 teeth; with both having a face width of 1.2 inches and a diametral pitch of 10 inches. By calculating the required brinnell hardness of the gears during the gear strength design process, AISI 4130 steel was selected for the material of the gear and AISI 4340 steel was selected for the material of the pinion. It can be noted that the pinion material is stronger than the gear material, this is the case to prevent the pinion from being broken by the larger gear.

Once the gear dimensions were finalized, the design of the shafts were initiated. The shafts had to be long enough to allow enough space for the width of the gear, spacers, and bearing. The shafts also had to have a large enough diameter to prevent the shafts from bending as a result of the resultant force generated by the gears. The final diameters chosen for the shafts were dependent on the available sizes for ball bearings offered by McMaster-Carr. Ultimately, in addition to the required diameters for the resultant force, two bearings were selected to drive the diameters of the shafts which were R14 and R22. The final diameter dimensions for the shaft can be seen in figure 17. The mechanical key design allowed for the implementation of mechanical keys which transmitted the power to and from the shafts to the gears as well as performing the duty of a sacrificial “fuse” to break before the shaft or gear in case the system is subject to sudden excessive stress. For this reason, the fuse was made of AISI 1010 steel because it is weaker than the materials used for the shafts as well as the gear and pinion. It was also important to ensure that the gears and shafts were designed to be able to accommodate the length and width of the key seats.

The housing was the final component designed for this gear box system. The housing needed to provide support for bearing holders, space for an input and output shaft, and it needed enough space to accommodate the gear and pinion shaft assemblies. The ultimate design called for 4 individual parts to complete the housing assembly. There was a top and bottom portion which acted as covers. Top and bottom bearing holders were also designed to ensure the bearing and shaft was supported and that the non-input or output end of the shaft was sealed off from the outside. This final component design resulted in a fully operational gear box which satisfies the design specifications and can be seen in the assembly drawing, figure 15.

This project reinforced the students understanding of the concept of gearbox design learned in class by applying the principles in order to design a fully functional gearbox. This excellent opportunity provides members of the group with experience with a topic more applicable to a problem they may be faced with while working in the engineering industry. It reinforced the understanding of the importance, manipulation, and implementation of the design-driving equations learned from this class. The students were able to create a safe and cost effective solution to this product by minimizing the use of material, as well as by selecting the appropriate material suited to this application. This material selection provided students with more experience to understand which materials would be reasonable in an application like this. Overall, this project provided the students with the opportunity of applying the theory learned in class to design a fully functional gearbox all the while gaining experience in gearbox design and component design in general.

Appendix

Mechanical Key Design

TABLE 11–1 Key size vs. shaft diameter

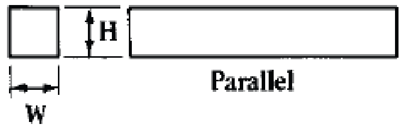
Nominal shaft diameter		Nominal key size		
			Height, *H*	
Over	To (incl.)	Width, *W*	Square	Rectangular
5/16	7/16	3/32	3/32	
7/16	9/16	1/8	1/8	3/32
9/16	7/8	3/16	3/16	1/8
7/8	$1\frac{1}{4}$	1/4	1/4	3/16
$1\frac{1}{4}$	$1\frac{3}{8}$	5/16	5/16	1/4
$1\frac{3}{8}$	$1\frac{3}{4}$	3/8	3/8	1/4
$1\frac{3}{4}$	$2\frac{1}{4}$	1/2	1/2	3/8
$2\frac{1}{4}$	$2\frac{3}{4}$	5/8	5/8	7/16
$2\frac{3}{4}$	$3\frac{1}{4}$	3/4	3/4	1/2
$3\frac{1}{4}$	$3\frac{3}{4}$	7/8	7/8	5/8
$3\frac{3}{4}$	$4\frac{1}{2}$	1	1	3/4
$4\frac{1}{2}$	$5\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	7/8
$5\frac{1}{2}$	$6\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	1
$6\frac{1}{2}$	$7\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{1}{2}$
$7\frac{1}{2}$	9	2	2	$1\frac{1}{2}$
9	11	$2\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{3}{4}$
11	13	3	3	2
13	15	$3\frac{1}{2}$	$3\frac{1}{2}$	$2\frac{1}{2}$
15	18	4		3
18	22	5		$3\frac{1}{2}$
22	26	6		4
26	30	7		5

Source: Reprinted from ANSI Standard B17.1-1967(R98), by permission of the American Society of Mechanical Engineers. All rights reserved.

Note: Values in nonshaded areas are preferred. Dimensions are in inches.

Table 11: Key Size vs. Shaft Diameter [1]

Table 3. ANSI Standard Plain and Gib Head Keys *ANSI B17.1-1967 (R2003)*



Key			Nominal Key Size Width *W* Over	To (Incl.)	Tolerance Width, *W*		Height, *H*	
Parallel	Square	Keystock	…	1¼	+0.001	–0.000	+0.001	–0.000
			1¼	3	+0.002	–0.000	+0.002	–0.000
			3	3½	+0.003	–0.000	+0.003	–0.000
		Bar Stock	…	¾	+0.000	–0.002	+0.000	–0.002
			¾	1½	+0.000	–0.003	+0.000	–0.003
			1½	2½	+0.000	–0.004	+0.000	–0.004
			2½	3½	+0.000	–0.006	+0.000	–0.006
	Rectangular	Keystock	…	1¼	+0.001	–0.000	+0.005	–0.005
			1¼	3	+0.002	–0.000	+0.005	–0.005
			3	7	+0.003	–0.000	+0.005	–0.005
		Bar Stock	…	¾	+0.000	–0.003	+0.000	–0.003
			¾	1½	+0.000	–0.004	+0.000	–0.004
			1½	3	+0.000	–0.005	+0.000	–0.005
			3	4	+0.000	–0.006	+0.000	–0.006
			4	6	+0.000	–0.008	+0.000	–0.008
			6	7	+0.000	–0.013	+0.000	–0.013

Table 12: ANSI Standard Plain and Gib Head Keys[2]

Table 11-2 Suggested Keyseat Fillet Radius and Key Chamfer
ANSI B17.1-1967 (R2003)

Keyseat Depth, *H*/2 Over	To (Incl.)	Fillet Radius	45 deg. Chamfer	Keyseat Depth, *H*/2 Over	To (Incl.)	Fillet Radius	45 deg. Chamfer
⅛	¼	1/32	3/64	⅞	1¼	3/16	7/32
¼	½	1/16	5/64	1¼	1¾	¼	9/32
½	⅞	⅛	5/32	1¾	2½	⅜	13/32

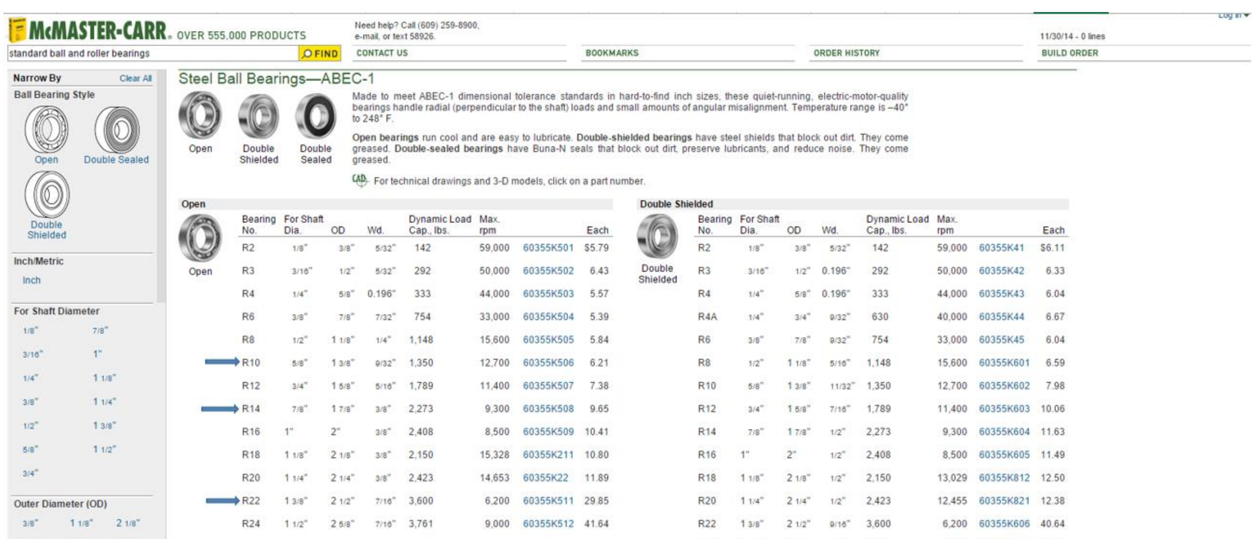
Table 13: Suggested Key seat Fillet Radius and Key Chamfer [2]

Table 4. ANSI Standard Fits for Parallel and Taper Keys *ANSI B17.1-1967 (R2003)*

Type of Key	Key Width		Side Fit			Top and Bottom Fit			
			Width Tolerance			Depth Tolerance			
	Over	To (Incl.)	Key	Key-Seat	Fit Range[a]	Key	Shaft Key-Seat	Hub Key-Seat	Fit Range[a]
Class 1 Fit for Parallel Keys							S	T	
Square	…	½	+0.000	+0.002	0.004 CL	+0.000	+0.000	+0.010	0.032 CL
			−0.002	−0.000	0.000	−0.002	−0.015	−0.000	0.005 CL
	½	¾	+0.000	+0.003	0.005 CL	+0.000	+0.000	+0.010	0.032 CL
			−0.002	−0.000	0.000	−0.002	−0.015	−0.000	0.005 CL
	¾	1	+0.000	+0.003	0.006 CL	+0.000	+0.000	+0.010	0.033 CL
			−0.003	−0.000	0.000	−0.003	−0.015	−0.000	0.005 CL
	1	1½	+0.000	+0.004	0.007 CL	+0.000	+0.000	+0.010	0.033 CL
			−0.003	−0.000	0.000	−0.003	−0.015	−0.000	0.005 CL
	1½	2½	+0.000	+0.004	0.008 CL	+0.000	+0.000	+0.010	0.034 CL
			−0.004	−0.000	0.000	−0.004	−0.015	−0.000	0.005 CL
	2½	3½	+0.000	+0.004	0.010 CL	+0.000	+0.000	+0.010	0.036 CL
			−0.006	−0.000	0.000	−0.006	−0.015	−0.000	0.005 CL

Table 14: ANSI Standard Fits for Parallel and Taper Keys [2]

Bearing Design



Open

Bearing No.	For Shaft Dia.	OD	Wd.	Dynamic Load Cap., lbs.	Max. rpm		Each
R2	1/8"	3/8"	5/32"	142	59,000	60355K501	$5.79
R3	3/16"	1/2"	5/32"	292	50,000	60355K502	6.43
R4	1/4"	5/8"	0.196"	333	44,000	60355K503	5.57
R6	3/8"	7/8"	7/32"	754	33,000	60355K504	5.39
R8	1/2"	1 1/8"	1/4"	1,148	15,600	60355K505	5.84
R10	5/8"	1 3/8"	9/32"	1,350	12,700	60355K506	6.21
R12	3/4"	1 5/8"	5/16"	1,789	11,400	60355K507	7.38
R14	7/8"	1 7/8"	3/8"	2,273	9,300	60355K508	9.65
R16	1"	2"	3/8"	2,408	8,500	60355K509	10.41
R18	1 1/8"	2 1/8"	3/8"	2,150	15,328	60355K211	10.80
R20	1 1/4"	2 1/4"	3/8"	2,423	14,653	60355K22	11.89
R22	1 3/8"	2 1/2"	7/16"	3,600	6,200	60355K511	29.85
R24	1 1/2"	2 5/8"	7/16"	3,761	9,000	60355K512	41.64

Double Shielded

Bearing No.	For Shaft Dia.	OD	Wd.	Dynamic Load Cap., lbs.	Max. rpm		Each
R2	1/8"	3/8"	5/32"	142	59,000	60355K41	$6.11
R3	3/16"	1/2"	0.196"	292	50,000	60355K42	6.33
R4	1/4"	5/8"	0.196"	333	44,000	60355K43	6.04
R4A	1/4"	3/4"	9/32"	630	40,000	60355K44	6.67
R6	3/8"	7/8"	9/32"	754	33,000	60355K45	6.04
R8	1/2"	1 1/8"	5/16"	1,148	15,600	60355K601	6.59
R10	5/8"	1 3/8"	11/32"	1,350	12,700	60355K602	7.98
R12	3/4"	1 5/8"	7/16"	1,789	11,400	60355K603	10.06
R14	7/8"	1 7/8"	1/2"	2,273	9,300	60355K604	11.63
R16	1"	2"	1/2"	2,408	8,500	60355K605	11.49
R18	1 1/8"	2 1/8"	1/2"	2,150	13,029	60355K812	12.50
R20	1 1/4"	2 1/4"	1/2"	2,423	12,455	60355K821	12.38
R22	1 3/8"	2 1/2"	9/16"	3,600	6,200	60355K606	40.64
R24	1 1/2"	2 5/8"	9/16"	3,761	9,000	60355K607	46.53

Figure 19: McMaster-Carr Bearing Catalog [3]

Type of Application	Life, kh
Instruments and apparatus for infrequent use	Up to 0.5
Aircraft engines	0.5–2
Machines for short or intermittent operation where service interruption is of minor importance	4–8
Machines for intermittent service where reliable operation is of great importance	8–14
Machines for 8-h service that are not always fully utilized	14–20
Machines for 8-h service that are fully utilized	20–30
Machines for continuous 24-h service	50–60
Machines for continuous 24-h service where reliability is of extreme importance	100–200

Table 15: Design Life Table [1]

Type of Application	Load Factor
Precision gearing	1.0–1.1
Commercial gearing	1.1–1.3
Applications with poor bearing seals	1.2
Machinery with no impact	1.0–1.2
Machinery with light impact	1.2–1.5
Machinery with moderate impact	1.5–3.0

Table 16: Design Load Factor [1]

Department of Mechanical Engineering and Technology

Project Assignment Document

GEAR BOX DESIGN

MECH 420 MAJOR PROJECT

Single-stage Gear Box

Project Leader: Kyle Trube Project Members: Alex Potwardowski, Austin Breadmore, Scott Rogers, Nick Naimie

12/9/2014

Department of Mechanical Engineering and Technology

College of Engineering and Technology

Wentworth Institute of Technology

550 Huntington Ave, Boston, MA 02115

Introduction

A gearbox is a mechanical method of transferring energy from one device to another and is generally used to increase torque while reducing speed. Following picture is an example of the gear box, which is the main gearbox and rotor of a Bristol Sycamore helicopter.

Figure 1: The main gearbox and rotor of a Bristol Sycamore helicopter (http://en.wikipedia.org/wiki/Gear_box)

Lots of machinery or devices consist generally of a power source and a power transmission system, which provides controlled application of the power. Through the transmission, the power is transmitted from an engine to a live axle. Often transmission refers simply to the gearbox that uses gears and gear trains to provide speed and torque conversions from a rotating power source to another device. Therefore, the gearbox is a kind of semi-standard sub-unit or sub-system for lot of machinery.

This MECH420 class project is to design a single-reduction spur gearbox, which including, housing (case), ball bearings, spur gears, mechanical keys and shafts. You will be asked to provide the engineering documents including sketch layout, detail calculation, component drawing, and assemblies and necessary documents for this single-reduction spur gearbox. According to the engineering documents, the singe-reduction gearbox can be implemented. The detail requirements are specified in the next section.

The single-reduction spur gearbox reducer

A company plans to develop a series of single-reduction spur gearbox reducers. Followings are some design requirements (design specifications):

The delivered power will be in the range of 15 ~ 30 hp. The power source and driven device can be treated as "light-shock" loadings. The input speed from the power source is 1800 rpm.

The gearboxes will be single-reduction spur gearbox with the 3 choices of the gear ratio 3.5, 5.2 and 7.2 with 1% variation. For the spur gear, the pressure angle is 20^0, and the recommended AGMA quality number will be 5.

It is recommended that the single-row ball bearing will be used for the single-reduction spur gearbox.

The design life for the gears and bearing will be 4, 000 hours.

For the shaft, the factor of safety is 3.25 for the static design and 1.25 for the fatigue design. For the fatigue design of the shaft, it is assumed that endurance limit will be used.

The reliability index for gear, bearing and shaft will be 0.90.

The connection between the power source/ the driven device and the designed single-reduction spur gearbox will be through mechanical keys.

The single stage gearbox will have 10 different models from WIT-MEGB-01 to WIT-MEGB-10. Each design team will choose one of following model for your design.

WIT-MEGB-01: 10 hp; gear ratio 3.5 with 1% variation,

WIT-MEGB-02: 10 hp; gear ratio 5.2 with 1% variation

WIT-MEGB-03: 10 hp; gear ratio 7.2 with 1% variation

WIT-MEGB-04: 15 hp; gear ratio 3.5 with 1% variation,

WIT-MEGB-05: 15 hp; gear ratio 5.2 with 1% variation

WIT-MEGB-06: 15 hp; gear ratio 7.2 with 1% variation

WIT-MEGB-07: 20 hp; gear ratio 3.5 with 1% variation

WIT-MEGB-08: 20 hp; gear ratio 5.2 with 1% variation

WIT-MEGB-09: 20 hp; gear ratio 7.2 with 1% variation

WIT-MEGB-10: 30 hp; gear ratio 3.5 with 1% variation

WIT-MEGB-11: 30 hp; gear ratio 5.2 with 1% variation

WIT-MEGB-12: 30 hp; gear ratio 7.2 with 1% variation

Some additional information:

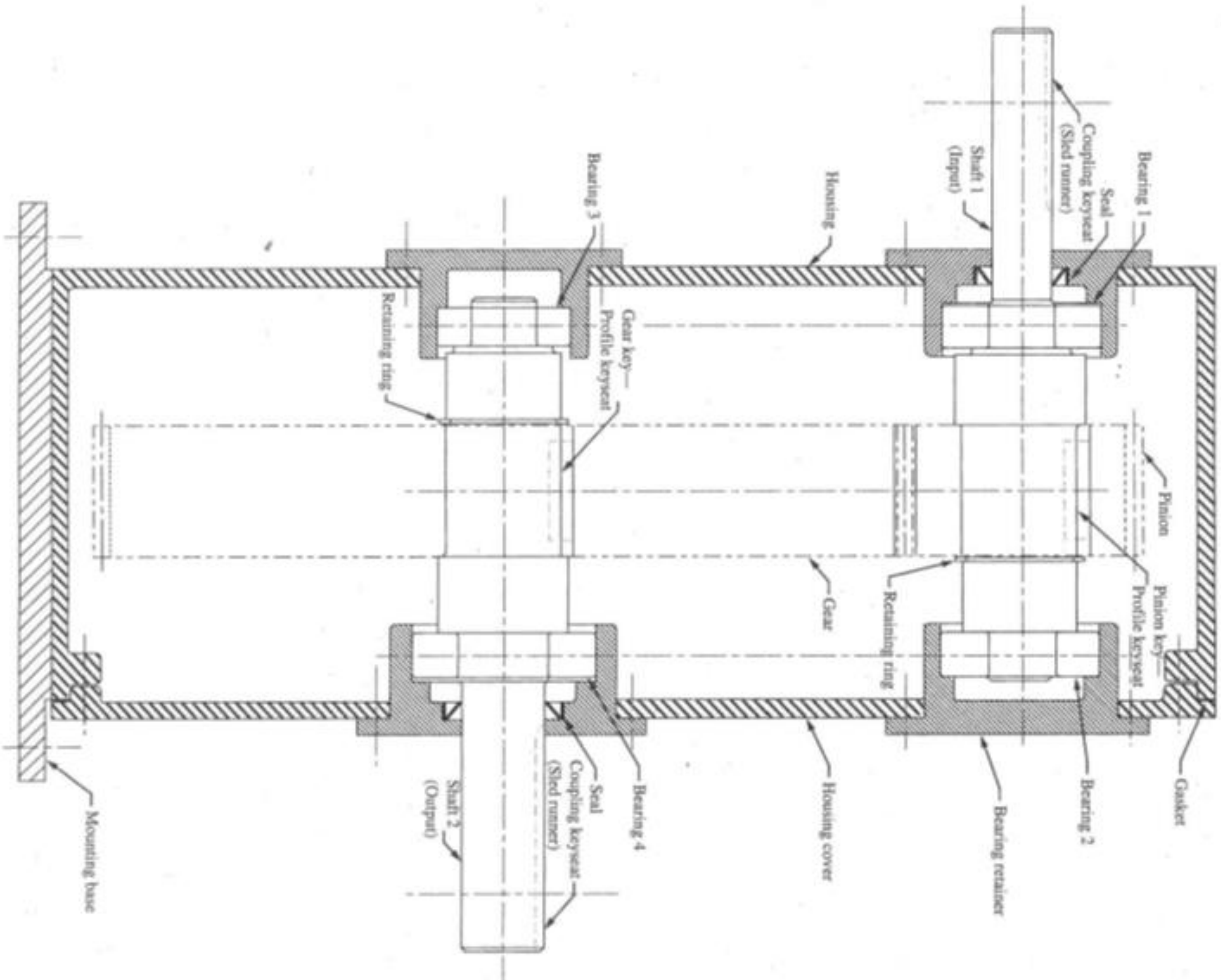
Notes: you can ignore following additional information and use your own decision.

The hub length for the input shaft for the engine might be at least 1.5”

The hub length for the output shaft for the working device might be at least 2”

The gaps between the rotating gears and the stationary part (such as housing, bearings) might be at least 0.5”

The housing might be made of plates and bearing retainers as shown in the follow layout for a single stage gear box [Robert L. Mott, Machine Elements in Mechanical Design, figure 15-6 in page 656 of the textbook, fourth Edition, Pearson Prentice Hall, 2004].



A sample of the layout for a single stage gearbox

Important dates

At the end of the class design project, each group will submit the project report and deliver the project presentations.

The due date for the report: 11 pm of 12/09/2014

The presentation video due date: 12/08/2014.

Notes: You don't need to submit the hardcopy of the report and presentation file. You will submit a USB disk which includes all information such as email communication, document, report, simulation and teaming working photos.

Report format

Cover page

Executive summary (5 pts)

The final submission shall be a PDF file which will contain the following sections and items listed therein.

Note: every team member must sign on the page of the executive summary.

Table of contents with page numbers (5 pts)

Introduction (10 pts)

Design specifications (5 pts)

Design and Analysis (40 pts)

Gear design

Shaft design

Mechanical key design

Bearing design

The housing design

Notes: Detail calculation for shaft, gear, key, and bearing must be provided. For the selection of some related parameters during calculation, the related source (curves and tables) must be provided in the report. The FEA simulation is recommended for the design of some components such as shaft and the housing.

Drawings for components and assemblies (25 pts)

Notes: the part drawing of the non-standard components, assembly drawing of sub-systems such as the shaft with gear and key; the assembly drawing of the whole gearbox. In this section, the manufacturing routine of components and assembly must be provided.

Discussions and conclusions (10 pts)

Appendix

References

Presentation of the major design project

The presentation date will be 12/08/2014 during our normal meeting time. In this presentation, you will show the presentation video and answer some questions if there is any. For this presentation of the major design project:

Each team must prepare a PowerPoint presentation file which is the text form of your video presentation.

Each team must prepare an around 15-minute video presentation which will be shown during our presentation date.

Each team will have around 5 minute for Question and Answers

Appendix A: Team Contract

Team Contract for the major project

Course: MECH420-Design of Machine element

Department of Mechanical Engineering and Technology

Wentworth Institute of Technology

As a member of project team #____, I agree that coordinated teamwork is essential for successfully completing assigned projects. I understand my individual performance will affect the success of the entire project. I also understand that all members of successful teams need to exhibit the following behaviors during the execution of the project:

Fulfills duties of team role

Attend all team meetings

Actively participate in team meetings

Be prepared to present work assigned from previous meeting

Researches and gathers pertinent information

Listens to other teammates

Shares work evenly

Submit meeting minutes and weekly report by the project manager

Regularly scheduled meetings will meet (at least total two hours per week):

Time/Day: Place:

Time/Day: Place:

Additional Requirements Identified By Team Consensus:

____

____

Printed Team Member Name	Team Member Signature	Date:
1.		
2.		
3.		
4.		
5.		
Project manager:	Signature:	

Appendix B: weekly report

WENTWORTH Institute of Technology

Department of Mechanical Engineering and Technology

MECH420-Design of Machine Elements

Weekly report of the major design project

Group number #:

Week number #:

Submitted date:

Department of Electronics and mechanical

Wentworth Institute of Technology

550 Huntington Ave, Boston, MA 02115

Fall 2014

WENTWORTH Institute of Technology

Department of Mechanical Engineering and Technology

Week No:

Meeting minutes

Meeting time:		Location:	
Attendance:			
Activities:			

(Notes: For each meeting, fill the above table. For the weekly report, you need clearly show where, when you have meeting and who attend meeting. Every team must work physically together at least 2 hours.)

Tasks worked during this week

(Notes: You need to briefly describe what you have been working on and what kind of conclusion you have.)

Tasks scheduled for next week

(Notes: You need to describe what will be next contents for the design team.)

Appendix C: Expected progress

Expected progress

Week#	Expected progress
Week#8	Form the design team. Each team will have 3~4 team members Elect project manager and negotiate the weekly meeting time Sign and submit the team contact Submit the weekly report No1
Week#9	Study the project task menu Collect information about the gearbox and its typical applications Find at least two gearbox manufactures. List and show the commercial available gearbox with detail design specifications Search for the typical structures or layout of the gearbox reducers Work on the final report Submit weekly report No2
Week#10	Search for the typical structures or layout of the gearbox reducers Search the website to see the typical styles (structures) for gears Conduct gear design Create models (some dimensions might be modified for shaft design and the mechanical key design) Work on the final report Submit the weekly report No3
Week#11	Conduct the gearbox layout Work on the gearbox case design Conduct the shaft layout Conduct the shaft design including the mechanical keys Create the shaft models Work on the final report Work on the video presentation Submit the weekly report No4
Week#12	Conduct the bearing design Finalize the shafts Conduct the gearbox case design Create models and drawing Conduct the FEA simulation for stress, deflections and the natural frequency Work on the final report Work on the video presentation Submit the weekly report No5

Expected progress

Week#13	Create models and drawing Work on the report Work on and finalize & record the video presentation Submit the weekly report No6
Week#14	Finalize the report

	Video presentation on 12/8/2014 Submit the report on 12/9/2014 Submit the weekly report No7

References

[1] Budynas, R., and Nisbett, K., 2010, "Shigley's Mechanical Engineering Design + Connect Access Card to accompany Mechanical Engineering Design," McGraw-Hill Education, .

[2] Oberg, E., Horton, H.L., Jones, F.D., 2008, "Machinery's Handbook: A Reference Book for the Mechanical Engineer, Designer, Manufacturing Engineer, Draftsman, Toolmaker, and Machinist," Industrial Press,

[3] McMaster-Carr, 2014, http://www.mcmaster.com/#standard-ball-and-roller-bearings/=uxq9kv

[4] My-Acoustic, Rahmat, 2006, "How gearbox work (manual Toyota T50), http://www.my-acoustic.com/Car/gearbox/working/working.htm