Power Transmission Design

MSE3380B - Prof. Price

Group 7

Student Name	**Student Number**
Allen Sun	251037497
Courtney Lacroix	251125131
Dongjun Kim	250727774
JD Herlehy	251089056
Nathan Smith	251070515

Table of Contents

Introduction

To transmit the same power across a mechanism while having the ability to change the ratio of speed and torque is the motivating factor for power transmission devices. Our device will have two stages: two sets of internal gear pairs. This design is focused on providing 30 horsepower (hp) at the output with an input speed of 1850 rotations per minute (rpm) — see Appendix A for how the values were obtained. For this to be achieved, various internal components must be designed and selected where appropriate. These internal components include gears, bearings, retaining rings, keys, and the intermediate shaft. Considerations for designing and selecting these components include part life, ease of assembly/disassembly, and cost. General calculations will provide the stress, displacement, and forces that determine the material and geometric properties of all components to be designed. Another consideration regarding the life of components is the method of lubrication. The required steps to obtain the properties of the two-stage power transmission device are outlined in this report.

Specifications and Performance Criteria

With our group number (7), we used the formulas given in the project manual to find the power to be delivered and the input speeds where the calculations for the same are in Appendix A.

Several assumptions were made during the design of the gear. Assuming the input speed is mostly staying on the nominal speed, calculations for the gear specifications were done based on the nominal input speed. For the dimension of the gearbox, height, clearance from the gear and the gearbox's inner wall, and the wall thickness were assumed to calculate the gear diametral pitch. The quality number for the tolerance was set to be 7 to have the most precise gear out of the commercial quality gear. When calculating the load distribution factor, several variables were calculated based on the assumption that the gear is a straddle-mounted pinion, has uncrowned teeth, and the product is a commercial enclosed unit. Also, the surface condition factor, overload factor, and size factor were assumed to be 1 when the contact stress was calculated. Moreover, for the calculation of the allowable contact stress, the temperature factor and the hardness ratio factors were assumed to be 1. Also, the design factor was assumed to be 1.2.

Since advanced shaft analysis software is not readily available for use, the integration methods from Mechanics of Materials will be used for deflection analysis. If the true intermediate shaft geometry involving seven different shaft sections were used for these calculations, the problem would produce 14 integration constants and 14 required boundary conditions to solve for in each of the two planes (xy and xz) — a total of 28 equations. To simplify, the shaft will be reduced to three sections with diameters D_1, D_3, and $D_7 = D_1$ for the purposes of deflection analysis (see Figure 1). This reduces the complexity to a total of 12 equations. For any axial location, this simplification either maintains or reduces the diameter. Therefore, the deflection analysis will yield conservative results (higher slopes and deflections than would be obtained if the original diameters were considered).

Assume that the case material has sufficient thermal properties to handle any thermal loads from the lubricant selected. Assume for the first iteration that where needed $D/d \cong 1.2$, $r/d = 0.02$, $K_f = 2.7$, $K_{fs} = 2.2$, and Marin factor k_b based on the varying initial diameter. Also, for the Marin factors, take the temperature factor to be 0.997 as the average operating temperature range is 12.5°C from Appendix A, which is reasonably close to room temperature compared to the melting point of steel. Also, since cold-drawn steel has been selected, which has “so-called directional characteristics of the [cold drawing] operation,” the miscellaneous Marin factor is made to be $k_f = 1$ [1].

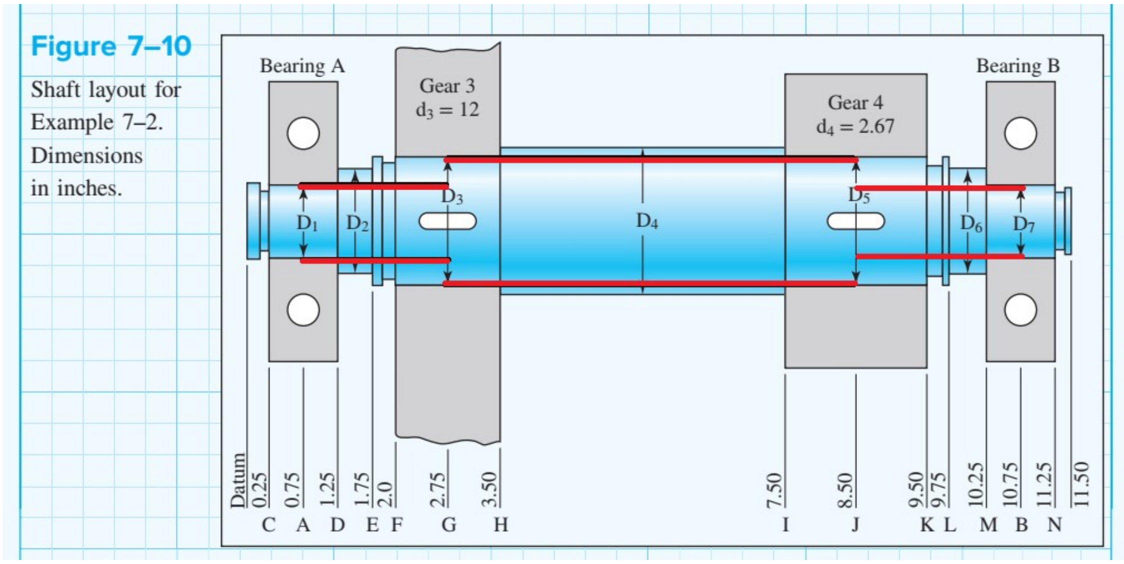


Figure 1: Diameters used for deflection analysis in red. Adapted from [1, p. 386].

Design Process

Kinematic Requirements and Factors of Safety

The kinematic requirements are solved partially in Appendix A, and the necessary gear ratios for such kinematics are calculated in the gear section. It can be seen that the input speed will nominally be 1850 rpm, and the output speed will nominally be 92.5 rpm with the minimum speed reduction given in the project manual. As mentioned as an assumption, the maximum input speed will be neglected for calculations, but it is essential to remember this and adjust the factor of safety with that variance in speed in mind. Multiple other parts will also have variances in the production quality and the amount of stresses and deflections they can undergo. Some of the variances of the intermediate shaft is taken into account from the miscellaneous Marin factor and thus doesn't have to be accounted for again in a factor of safety. The variance in material and possible defects from the supplier (likely present as we wish to make this product cost-competitive) can be large. Our factor of safety estimate begins at 1.2. Also, the influence of the temperature variance throughout the part is an additional factor. Our final factor of safety, with the background thought that we need to stay cost-competitive, is decided to be greater than 1.5. However, for the gears, the factor of safety over 1 was considered to be enough since the gears are replaceable. Note that if the resulting factor of safety for any part is greater than our desired factor of safety, we will keep this instead of reducing material or geometric properties, as we want to stay reliable to our clients even if it ends up costing more; We want to be cost-competitive, but we also want to provide reliable service to our clients… We may get continued business if they like us.

Material

The intermediate shaft must have infinite life, and from this, aluminum is ruled out. A goal of this design study is to have as little cost as possible. This translates into having high tensile strength and a good surface finish that comes with no added procedures. A unique outcome of increasing the strength is that a smaller diameter is then required, but as a result, the deflections would become larger as the diameter is decreased [1]. Notice as well how the amount of material, and thus the cost, decreases with decreasing diameter, but failure due to deflection certainly increases, and failure to cyclic loading may increase. For these reasons, steel is chosen as steel is the best tradeoff for strength and cost. Within the steels, a cold-drawn is selected as this surface finish is better than hot-rolled, and the yield and tensile strengths are better by about 10%. Higher strength in choosing a higher carbon steel is usually mitigated by the cost of such a steel and as such, a medium carbon steel is chosen [1]. The final material chosen for the intermediate shaft is AISI 1035 CD steel.

Gear Ratios and Reduction Stages

The number of gear teeth on each gear had to be calculated based on the specifications that had to be followed. Since the production rate is expected to be 1000 units per year, it was considered that the cost of the customized dimensions is not high enough to put the preferred size option into consideration. So, the calculation was done based on the number of gear teeth instead of the available preferred size. The number of teeth, angular velocity, and the torque on each gear were calculated to meet the design specification and for the follow-up design process on the intermediate shaft and the AGMA stress analysis for gears.

In addition, spur gear has been chosen for all 4 gears. The main reasons for the spur gear choice are the followings: 1) Since there is no requirement for noise reduction, the most efficient and reliable spur gear was selected; 2) Spur gear has good efficiency and low power loss; 3) Easy teeth alignment; 4) the design that allows gear replacement reduces the risk of using spur gear that is likely to have less strength than other types of gears.

Free-Body, Shear Force, Bending Moment, Axial Load, and Torsion Diagrams

To see all the layouts, see Appendix B. When reading the graphs, note that the datum being referenced is the center point of bearing A. The graphs finish at the center of bearing B as afterwards the bending moments are symmetric until the right end of the intermediate shaft. Also see Appendix C for the engineering drawings of the intermediate shaft.

AGMA Stress Analysis of Gears

From the estimation of the gearbox dimensions, the gear's diametral pitch was first calculated. Then, using the diametral pitch and the number of teeth required for each gear, the diameters of each gear were obtained. Face widths of each gear were found according to the diametral pitch (from knowing it is usually 3 to 5 times the circular pitch).

Once the face width of each gear has been decided, the contact factor of safety is found. Contact stress and allowable contact stress could be calculated based on some assumptions about the gear and the product unit. The contact strength of the heat-treated or case hardened steel had to be higher than the calculated required contact strength. The allowable contact strength with the choice of the heat treatment could then be paired up with the stress-cycle factor and the contact stress to find the contact factor of safety. The contact factor of safety determines how safe the gears are when they are contacting each other to transmit power without wearing out. The contact factor of safety has to be larger than 1 to not fail.

There is another factor of safety that shows how safe the gears are in terms of bending. It is another important factor that has to be larger than 1 because a factor of safety less than 1 will likely cause the gear to deform and bend.

Combining the contact factor of safety and the bending factor of safety, the correct and safe heat treatment method could be selected with the corresponding gear diameter and the face width.

Intermediate Shaft Design for Stress

In the design of the shaft, two failure modes were considered, failure due to fatigue and first cycle failure. To satisfy safety criteria, a factor of safety of 1.5 had to be met for the fatigue criterion, and a factor of safety over 2 was desired for first cycle failure. The equations for testing failure in fatigue and first cycle failure were implemented into Matlab, then tested using an established example [1, p. 387].

To solve for failure, first, initial conditions had to be determined. AISI 1035 CD steel was determined as a desirable material for the shaft, as outlined in the Material section of the report, so the initial design began with AISI 1035 CD steel. AISI 1035 CD steel has an ultimate tensile strength of 79.77 kpsi and a yield strength of 66.72 kpsi. Qualitatively, AISI 1035 CD is ductile, not conservative and has equivalent strength in tension and compression, meaning it falls under distortion-energy (DE) theory [1, p. 276]. The cold-drawing process was used in the Marin factor analysis. An environmental factor, working temperature, was assumed to be 12.5°C (from Appendix A). The required reliability was over 99.5%, so a greater reliability of 99.9% was set as the design target. Additionally, the desired number of cycles was calculated from the steady-state speed of the input shaft and the gear ratios. The main shaft had a desired number of cycles of 2.96×10^8.

The geometric coefficients K_f and K_{fs} are derived from K_t and K_{ts} values found in Table A-15-7 and Table A-15-8 [1, p. 1044]. However, to give a generous estimate, K_t and K_{ts} were assumed to be 2.7 and 2.2, respectively, on the first pass analysis. For following iterations, the tables were used to manually estimate K_t and K_{ts} factors for each new section of analysis. When iterating on the 6.75 in critical location, the 0.02 inch radius/diameter ratio was providing exceedingly high shape factors, which lead to alarmingly low factors of safety. To account for this, the r/d ratio was slightly increased to 0.04, which allowed for more predictable failure for the shaft. In general, a smaller diameter/ large diameter ratio of 1.2 was aimed for, but this was not strictly followed. The ratios between the final diameters of 2.3, 1.9, 1.6 and 1.2 are 1.21, 1.19 and 1.33 respectively.

Stresses in the shaft were derived from applied bending moment and torque; these values were taken at the location of critical points along the shaft. These important locations were at each of the corners where the diameter changed and at the middle of a gear, where the bending moment was greatest while torque was still applied. Due to the balanced nature of the shaft, the bending moment fluctuates around the neutral axis and has a mean bending moment of zero. Additionally, the torque provided by the gears is assumed to be constant, meaning the alternating torque in the system is zero. From these values, K_f/K_{fs} coefficients and the assumed diameters of the shear and bending stresses were calculated. Then, using the combination of loading modes stress equations [1, p. 362], the alternating and mean stresses were calculated.

Marin Factor analysis was implemented in a general capacity to allow for changes in material, surface finish, loading, operating temperature, reliability and miscellaneous other factors. The surface factor can be set by changing the initial conditions between ground, machined/cold-drawn, hot rolled and forged, allowing for different processes to be used to create the shaft. This is important because the shaft has been specified to be produced in bulk, up to 1000 units per year. The size factor automatically takes diameter into account, allowing for dynamic analysis. Loading factor is set as 1 because the shaft is only ever in two loading modes, bending or combined, and both have a loading factor of 1. The temperature factor uses an equation unique to steel, so further analysis would be needed here if a non-steel material was chosen. However, steel was determined as the desired material early on, and the temperature factor is hence always close to 1. The reliability factor decreases the endurance strength; meanwhile, the miscellaneous factor has no effect. Finally, the temporary endurance strength is half of the ultimate tensile strength as outlined in Equation 6-10 [1, p. 359]

DE-Goodman, DE-Gerber and DE-SWT analysis was performed on critical locations to provide a broad idea of the factor of safety at any given location. DE-Goodman was used to make final design decisions because it is more conservative than DE-Gerber and had better analysis at high alternating, low mean stress than DE-SWT, which underestimates this area. The final factors of safety estimates are from DE-Goodman, although other calculations are present in the work.

First cycle yielding was checked in two ways, producing a regular and conservative factor of safety. A good factor of safety was considered 2; the lowest was 3.9.

Finally, the calculated 2.96×10^8 cycles the shaft is expected to undergo in its 12 khr, 100% duty cycle life is designed for in the last calculations. The number of cycles is estimated using the complete reverse simple loading conditions [1, p. 361]. The shaft undergoes a constant torque, which applies a constant mean stress on the shaft. This does not exist in completely reversing simple loading, so, instead, the mean stress was added to the alternating stress to give a conservative estimate of the number of cycles. The lowest number of cycles was 3.88×10^9, which is more than ten times the necessary number.

Datum Distance [in]	6.75	8.75	9.50	7.75
Diameter (Small) [in]	1.90	1.60	1.20	1.90
Diameter (Large) [in]	2.30	1.90	1.60	1.90
Radius [in]	0.076	0.064	0.048	Infinite
Se [kpsi]	19.00	19.36	19.96	19.00
Stress Alternating [psi]	1064.5	1183.1	1168.1	746.1
Stress Mean [psi]	655.8	0	0	452.1
FOS: DE-Goodman	1.56	1.64	1.71	2.23
FOS: DE-Gerber	1.75	1.64	1.71	2.50
FOS: DE-SWT	1.40	1.64	1.71	2.01
FOS: Yielding	5.34	5.64	5.71	7.65
FOS: Yielding (Conservative)	3.88	5.64	5.71	5.57
Number of Cycles	3.88E+09	1.22E+10	2.06E+10	8.90E+09

Intermediate Shaft Design for Deflection

As explained in the Specifications and Performance section, the intermediate shaft geometry considered in deflection analysis is a simplified model intended to reduce the complexity of the system of equations to solve for. Figure 1 shows the simplification, with three diameters to be considered: D_1, D_3, and $D_7 = D_1$. From the design for stress analysis, the nominal diameters were determined as $D_1 = D_7 = 1.2$ in, $D_3 = 1.9$ in.

Following the methods from Mechanics of Materials, the two bending moment equations in the xy- and xz-planes were each integrated once to determine the slope equations (in radians) and once more to determine the deflection equations (in inches). The Young's modulus used for the shaft is the value for carbon steel (E = 30.0 Mpsi) from Table A–5 [1, p.

1023]. The moment of inertias at each shaft section was calculated and used for the respective piece of the bending moment equation. To solve for the integration constants, six boundary conditions (per plane) were required. Two of these boundary conditions came from known deflection values: zero at both of the bearing locations. Another two arose from the necessity for deflection to remain continuous along the shaft, particularly at locations G and J. The final two boundary conditions came from continuity of slope at locations G and J. The slopes and deflections were plotted (Figures 2 and 3), and the resultant values at key locations were calculated (Table 1).

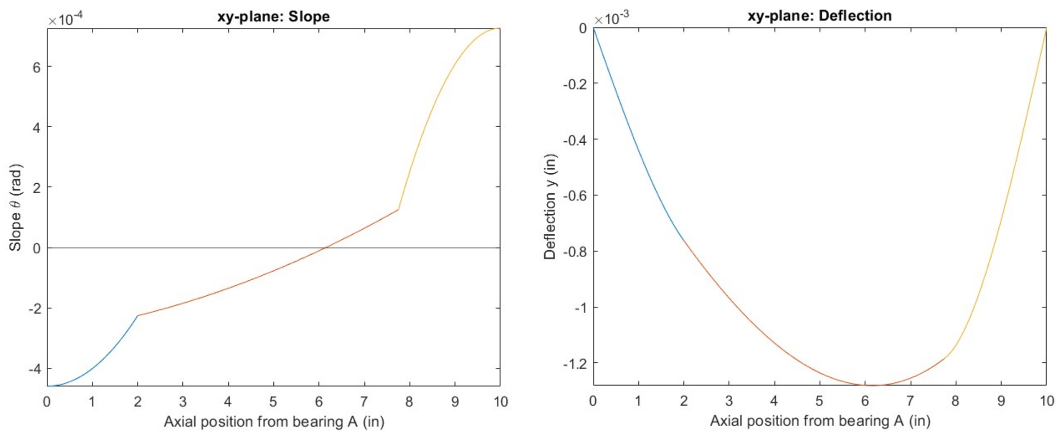


Figure 2: Slopes and deflections of the shaft in the xy-plane.

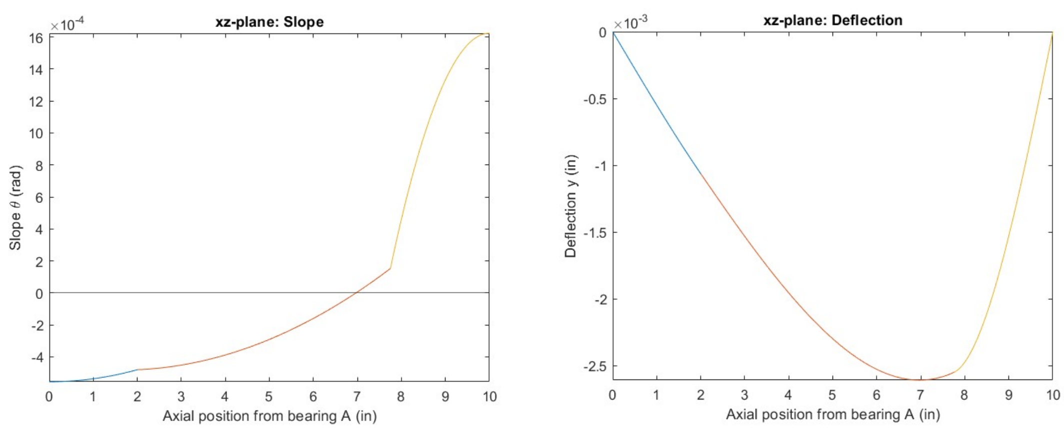


Figure 3: Slopes and deflections of the shaft in the xz-plane.

Location	Slope [rad]	Deflection [in]
Bearing A	0.0007215	Not relevant for this analysis
Bearing B	0.001779	Not relevant for this analysis
Gear G	0.0005315	0.001308
Gear J	0.0001969	0.002811

Table 1: Resultant slopes and deflections at key locations.

The calculation process for this deflection analysis was compared against Shigley's Example 7–3 [1, p. 392]. It showed that the diameter simplification resulted in slopes and deflections that are about twice as high as what is predicted with the shaft analysis software used in the text. Therefore, it is safe to assume that the above results are very conservative. According to Table 7–2 in Shigley's [1, p. 391], the calculated slopes at location A fall within the typical ranges for any type of bearing. For location B, the calculations agree with the typical range for a deep-groove ball bearing. However, the conservativeness of these calculations due to the need for simplification means that a cylindrical roller at B would also fit within the acceptable range in reality (since the real slope at Bearing B is likely to be up to twice as small as the calculated 0.001779 rad, easily putting the value under 0.0012 rad). The deflections can safely accommodate spur gears with any number of teeth up to 50 per inch. The slope at the location of Gear G is calculated to be slightly greater than the typical maximum value of 0.0005 rad, but again the real slope certainly falls within the acceptable limits for the reasons mentioned previously.

Intermediate Shaft Design Drawing

To perform a free-body force analysis and to obtain shear-moment diagrams, it is important to specify early in the design process the general layout of a shaft to accommodate shaft elements, such as gears, bearings, and pulleys. Typically, for easy assembly and disassembly, the largest diameter should be in the center, making shafts geometrically similar to stepped cylinders. Since the shaft required for the gearbox doesn't need to be super long, only two bearings will be used. To reduce bending moments and deflections, shafts should be kept as short as possible. Still, some axial space between components is needed to allow for lubricant and to provide access space for disassembly. A load-bearing component should be placed close to the bearings to minimize bending moments at locations that are likely to experience stress concentrations, as well as to minimize deflections at the load-bearing components. To minimize vibrations and deflections of the component, the shoulder is used to provide a solid support; therefore components should be placed against them. A key that fits into the groove in a shaft and gear is one of the most cost-effective and efficient methods for transmitting moderate to high levels of torque. With a slip-fit design, keyed components are easy to assemble and disassemble, and in cases where phase angle timing is essential, the key provides positive angular orientation of the component. In most cases, a shaft's geometric configuration can be derived by modifying existing models and making a limited number of modifications. The intermediate shaft design was based on the drawing provided in Shigley's Figure 18-2 [1, p. 946] since it met all of the requirements we needed, such as limiting the axial distance between components (minimizing the shaft length), components against shoulders, largest diameter in the center of the shaft (easy assembly and disassembly). A sketch of the layout is provided below (Figure 4).

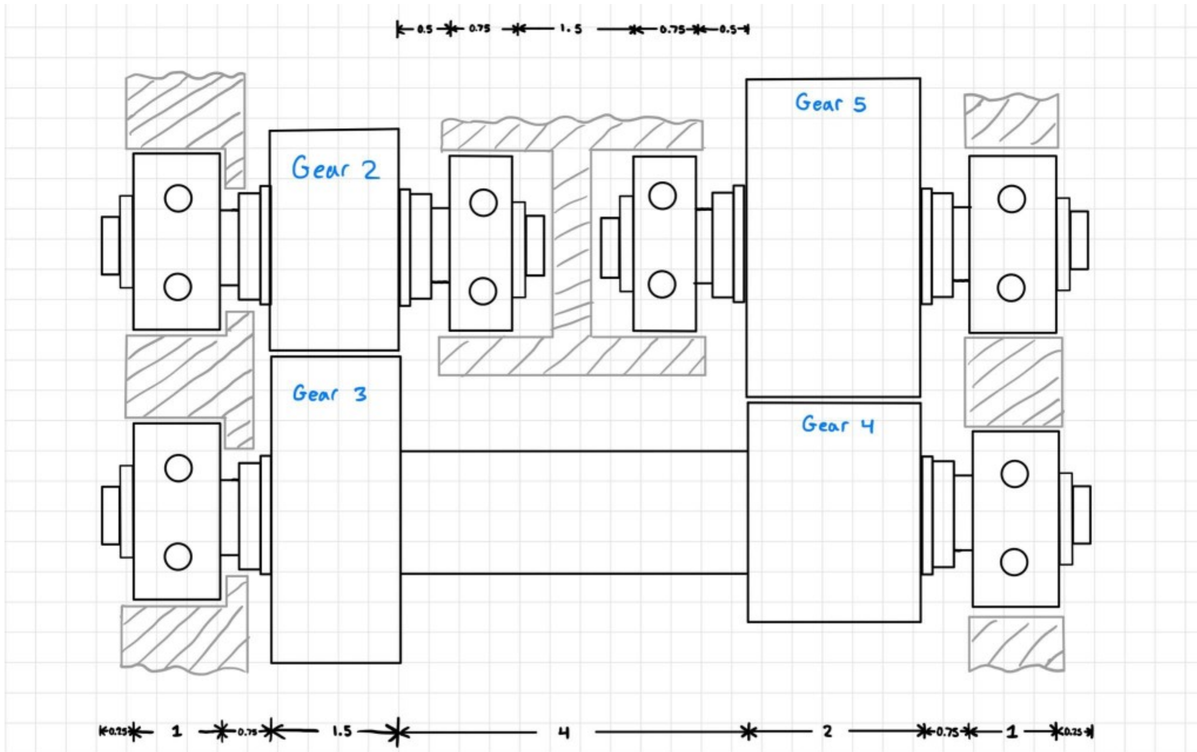


Figure 4: Shaft layout.

Bearing Specification

For proper bearing selection, both the Basic Dynamic Load Rating value (C_{10}) and the shaft diameter must be taken into consideration. From Shigley's Table 11-6 [1, p. 613], a rating life of 10^6 was chosen since it is used for Manufacturer 2 (for ball and straight rollers). From Shigley's Table 11-6 [1, p. 613], the guaranteed value of x, 63.2121 percentile value of x (x_0) and the shape parameter (θ) that controls the skewness (b) were also determined to be 0.02, 4.459 and 1.483. From Shigley's Table 11-5 [1, p. 589], 'a' was found to be 1.0 since the gearbox has precision gearing. As options for supporting the shafts, ball bearings and roller bearings were considered, resulting in $'a'$ values of 3 for ball bearings and $\frac{10}{3}$ for roller bearings. Using Shigley's Equation 11-10 [1, p. 584], C_{10} values can be determined for each bearing, depending on whether it is a ball bearing or roller bearing. For simplification, the bearing on the left side of Gear 3 will be referred to as Bearing A, and the bearing on the right side of Gear 4 will be referred to as Bearing B (Figure 4). The MATLAB file "Bearing_Selection.mlx" was used for these calculations. The results can be seen below (Table 2).

Bearing	Bearing Life (rev)	x_0	θ	b	'a' for ball bearings	'a' for roller bearings	C_{10} for ball bearings	C_{10} for roller bearings
A	2.8×10^8	0.02	4.459	1.483	3	$\frac{10}{3}$	4.75×10^3	3.69×10^3
B	2.8×10^8	0.02	4.459	1.483	3	$\frac{10}{3}$	2.44×10^4	1.89×10^4

Table 2: Basic dynamic load rating results for both bearings on the intermediate shaft.

The first step toward selecting the right bearing was determining whether roller bearings or ball bearings were better suited to the intended application. The benefits of roller bearings include low friction and a long lifespan. Moreover, they are easy to replace, and they support axial and radial loads. A roller element can support a greater load than a ball bearing, but they are also bigger and more expensive.

Through calculations, it was determined that for Bearing B, the basic dynamic loading rating (C_{10}) would need to be at least 24,354 lbf for it to be a ball bearing, which is high for a ball bearing. The basic dynamic load rating was recalculated using an 'a' value of $\frac{10}{3}$ for cylindrical roller bearings, and it was determined that the dynamic loading rating would have to be at least 18,889 lbf, which is a reasonable value for a cylindrical roller bearing. NU2207 ET Cylindrical roller bearing, single row from NSK met all of the constraints and was chosen for Bearing B [4]. The specifications for the bearing can be seen below (Table 3).

Bearing B (Cylindrical Bearing)	
Bore Diameter	1.378 in
Outside Diameter	2.83 in
Width	0.906 in

Basic Dynamic Load Rating (C)	61,500 lbf

Table 3: Bearing specifications for Bearing B on the intermediate shaft. [4]

Through calculations, it was determined that for Bearing A, the basic dynamic loading rating (C) would need to be at least 4753 lbf for it to be a ball bearing, which is a reasonable value for a ball bearing. 6307 Ball Bearing from McMaster-Carr met all of the constraints and was chosen for Bearing A [5]. The specifications for the bearing can be seen below (Table 4).

Bearing A (Ball Bearing)	
Bore Diameter	1.378 in
Outside Diameter	3.15 in
Width	0.827 in
Basic Dynamic Load Rating (C)	7500 lbf

Table 4: Bearing specifications for Bearing A on the intermediate shaft. [5]

Both bearings fit our requirements; however, due to the limitations of available bearings and needing to meet all of our constraints, the closet bore size found was 1.378 inches. Therefore, the shaft diameter will have to be increased to 1.378, which is a little bit bigger than our original shaft diameter of 1.2 inches but is still a reasonable size.. The new diameter also improves our factor of safety.

One of the requirements for our design is that the inside of the gearbox should be sealed against particulates typical of industrial settings. In both non-contact seals and shielded bearings, rotational torque is reduced since there is no contact with the inner ring. Non-contact enclosures, however, offer less protection against contamination than contact seals. Despite their better protection from contamination, contact seals generate a higher rotational torque due to friction between the seal lip and the inner ring of the bearing. Choosing between torque and protection is important when choosing how it will be sealed. Our team decided to use open bearings and add non-contact seals to the housing. The bearings will be supported by the shoulder on one side and the house on the other.

Lubrication Strategy

Since we are using open bearings, we are going to ask that the housing has labyrinth seals on either side of the shafts to protect from debris typical of industrial settings getting into the mechanism. For labyrinth seals, contaminants that try to get past the seal must navigate a maze-like combination of angles and turns that have been designed to prevent entry. A significant amount of flow friction and turbulence must be overcome for contamination to pass through the seal barrier. Additionally, because they are non-contact, they are frictionless, except for the damping effect of the lubricant, which leads to the elimination of stick-slip and starting torque, meaning it requires no special lubrication. It's an excellent alternative to traditional seals because labyrinth seals are reliable, long-lasting, frictionless, more efficient, and easy to install. This seal will also help us keep the lubrication inside the device. Lubrication with grease is simpler and less costly than lubrication with oil. Furthermore, it can remain in equipment for longer periods of time, it can provide better seals against contaminants, and it is a better choice when a continuous supply of oil cannot be maintained. Since the operating temperature is low, a grease is leaned towards; however, the speed can be high along the shaft. For these reasons, a low viscosity grease is used, which also accounts for stiffening in the colder operating temperatures.

Key and Retaining Ring Specification

With a nominal shaft diameter of 1.9", a square key with a width of 0.5 inches was chosen. In order to choose the square key that is already available, plain carbon steel was chosen as a material of the square key. [2] To achieve the factor of safety of 2 with the key specifications above, the length of the key only needed to be 0.6148 inches, but with the currently available square key, a length of 1 inch was chosen. Therefore, with the final choice of the key, it has the factor of safety of 3.2533, which indicates it is safe to use.

Since our nominal shaft diameter is 1.9”, it was required to find the retaining ring with a smaller groove diameter to hold the gear in place. So an external retaining ring with a groove diameter of 1.886” was chosen. (see Table 5) [3] These retaining rings will hold the spur gears, which are not likely to generate an axial load, so the heavy-duty retaining ring was not chosen to save cost.

Retaining Ring Specifications for both gears	
Nominal Shaft Diameter	1.9”
Groove Diameter	1.886” (–0.005”, +0.005”)
Groove Width	0.068” (+0.004”)
Nominal Groove Depth	0.057”
Thrust load capacity	14800 lbs

Table 5: Retaining ring specifications for both gears on the intermediate shaft. [3]

Vibration Analysis

This analysis was done with SOLIDWORKS frequency analysis on the intermediate shaft, and it was found that the critical speed was: 21032 rpm after converting the frequency from Hz displayed in Figure 5; see Appendix A for the calculation. This critical speed is larger than the maximum speed along the intermediate shaft: 494 rpm, and thus our design will not fail due to vibration.

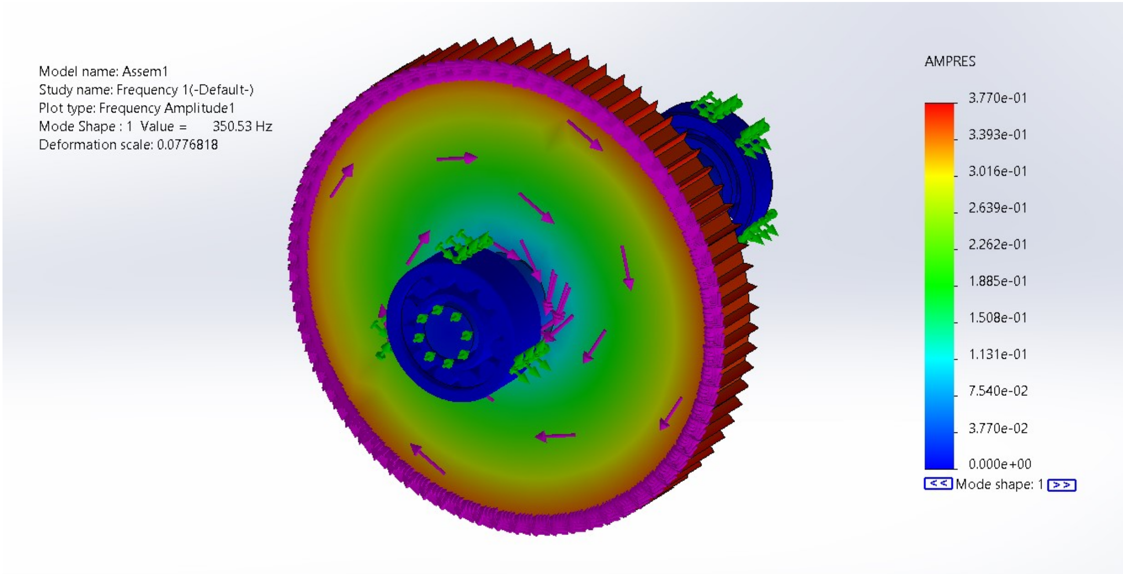


Figure 5: Frequency analysis results from SOLIDWORKS.

Conclusions

The layout of the shaft was set in a way to minimize the axial distance between components so that the bending of the shaft could be minimized. The middle of the intermediate shaft was determined to be the location of the biggest diameter for ease of assembly and disassembly, and components were placed against the shoulders for better support. It was important that the material and size of the gears, bearings and the shaft be chosen in such a way that they could withstand stress produced during operation. A conservative deflection analysis involving simplifications determined suitable bearing types based on slopes and transverse deflections. It was necessary to select proper bearings so as to minimize losses in shafts caused by friction and wear on components. Through analysis, it was determined the ET Cylindrical roller bearing was a good choice for Bearing B, and the 6307 Ball Bearing from McMaster-Carr was a good choice for Bearing A. In order to minimize maintenance required, a low viscosity grease was chosen as the method of lubrication for the bearings due to the low-speed nature of the gearbox.

Overall, the design of the power transmission unit required almost all knowledge obtained throughout the course, and we learned that designing a mechanical component is a very well-structured and delicate process. It was found that when the project specifications are given in one unit system, it is better to stick with that unit system all the way through, as switching back and forth can become confusing. Almost every selection process had its own detailed formulas and variables that reflected the real world to build the safest and the most efficient possible component. Regarding gear specification design, it was important to look at the entire design process to obtain the desired factor of safety instead of focusing only on choosing the heat treatment method. Thus, a major lesson from the project was that following the right steps with reasonable assumptions could lead to a successful component design.

References

[1] R. G. Budynas and J. K. Nisbett, *Shigley's Mechanical Engineering Design*, 11th ed. New York, NY: McGraw-Hill Education, 2020.

[2] *McMaster*. [Online]. Available: https://www.mcmaster.com/98870A480/. [Accessed: 07-Apr-2022].

[3] *McMaster. [Online]. Available: https://www.mcmaster.com/98585A141/. [Accessed: 07-Apr-2022].*

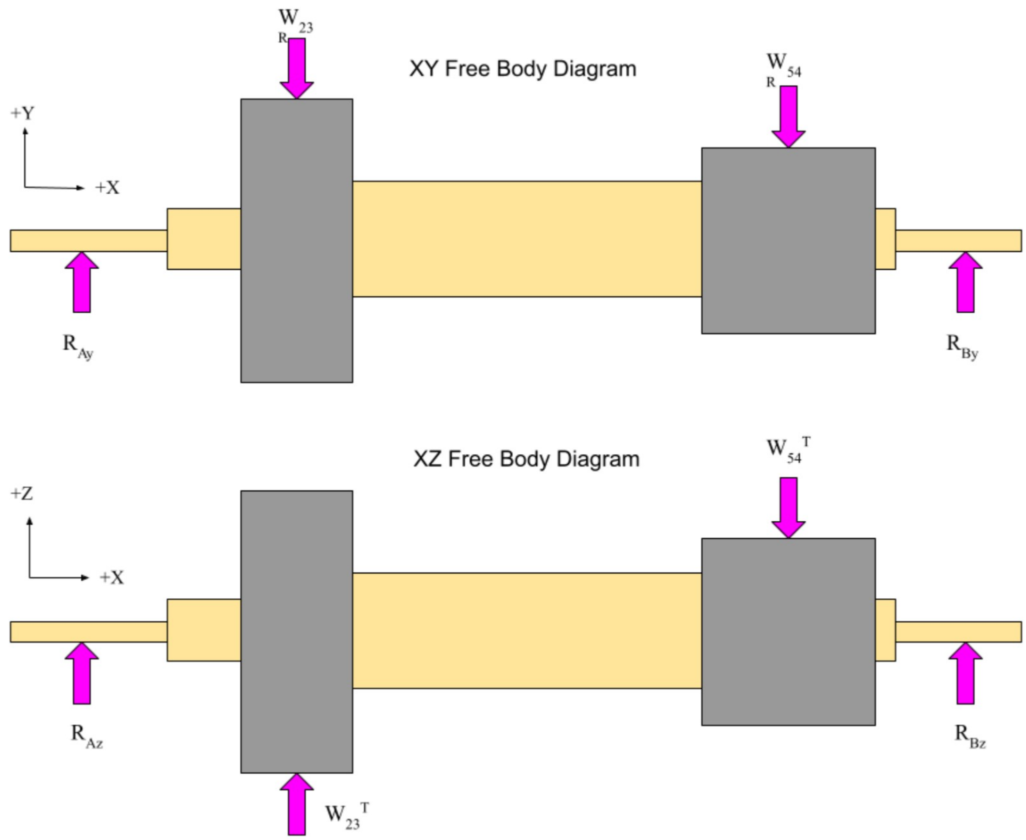
[4] *NSK. [Online]. Available: https://www.nsk.com/common/data/ctrgPdf/bearings/split/e1102/nsk_cat_e1102m_b84-109.pdf. [Accessed: 07-Apr-2022].*

[5] *McMaster. [Online]. Available: https://www.mcmaster.com/5972K503/. [Accessed: 07-Apr-2022].*

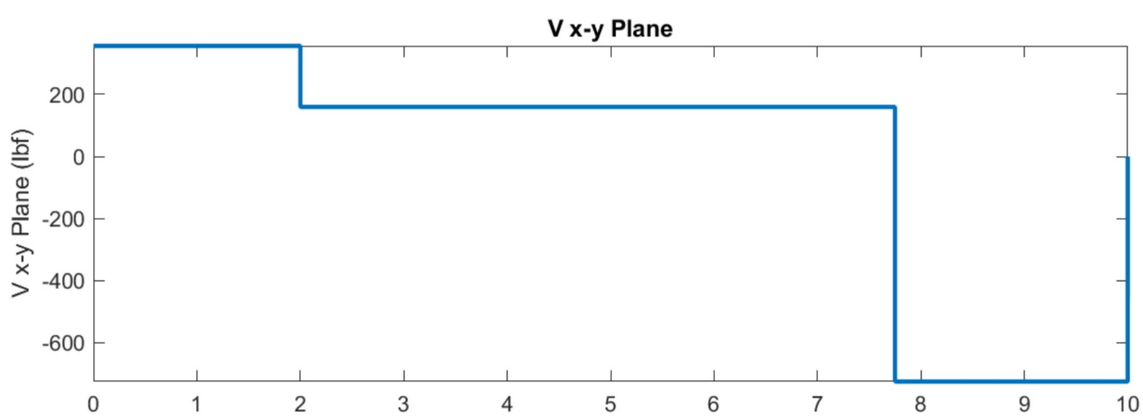
Appendix A

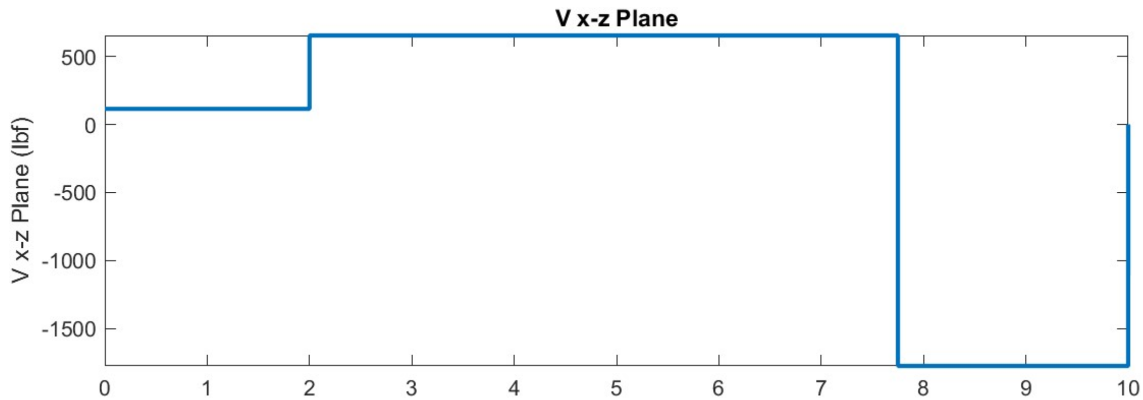
Parameter	Equation	Input Values	Output Value
Power to be delivered	((group No) mod 5) * 5 hp + 20 hp	7	30 hp
Nominal Input Speed	((group No) mod 6) * 100 rpm + 1750 rpm	7	1850 rpm
Max Input Speed	Nominal Input Speed * 1.2	1850 * 1.2 rpm	2220 rpm
Output Speed	Nominal Input Speed / Minimum Speed Reduction	1850 rpm, 20:1	92.5 rpm
Max Speed Intermediate Shaft	Max Input Speed * gear ratio input:intermediate	2220 rpm, 2.5:11.25	494 rpm
Critical Shaft Speed	Frequency in Hz * 60	350.53 Hz	21032 rpm
Avg. Operating Temp.	(Min Temp + Max Temp) / 2	–25°C, 50°C	12.5°C

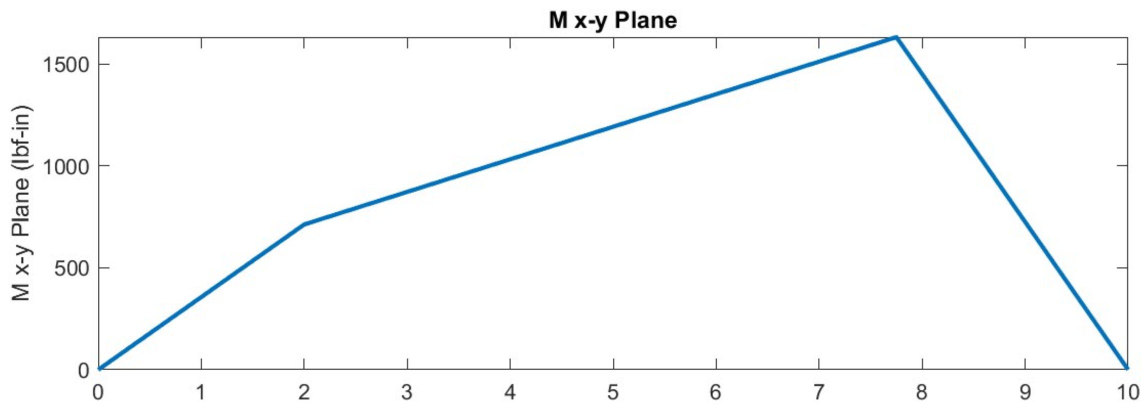
Appendix B

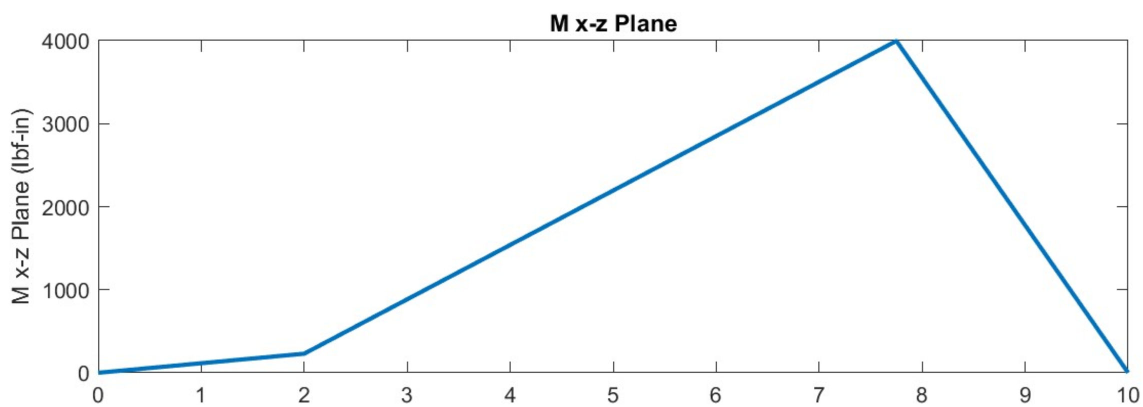


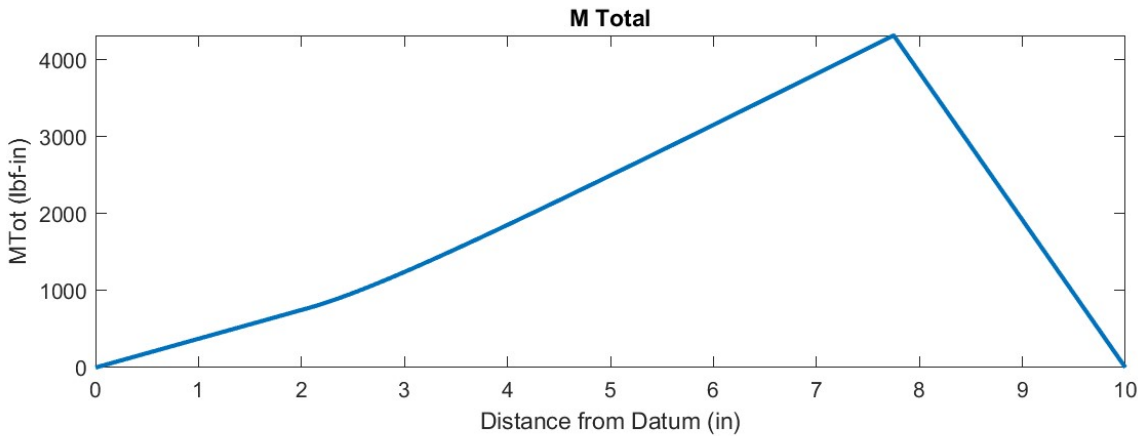
Free body diagrams of intermediate shaft in different planes

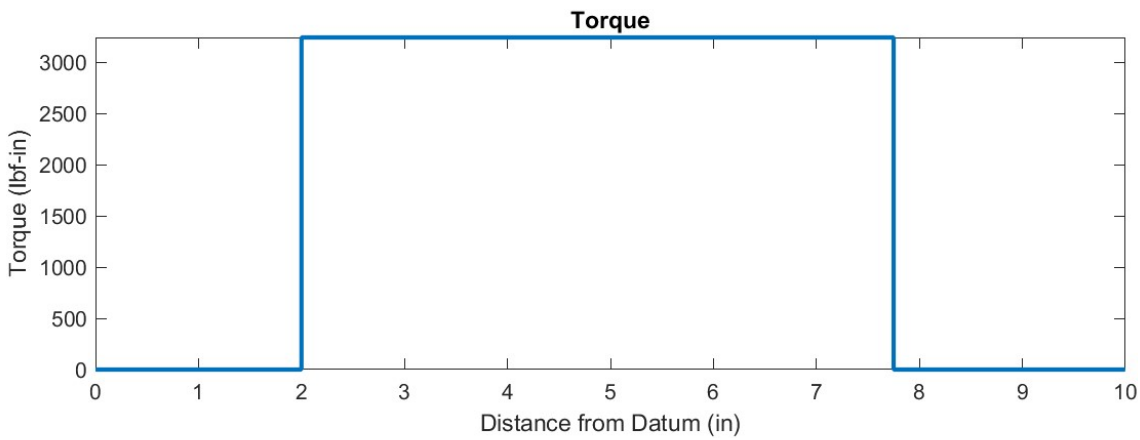



V x-z Plane
V x-z Plane (lbf)
500
0
-500
-1000
-1500
0 1 2 3 4 5 6 7 8 9 10

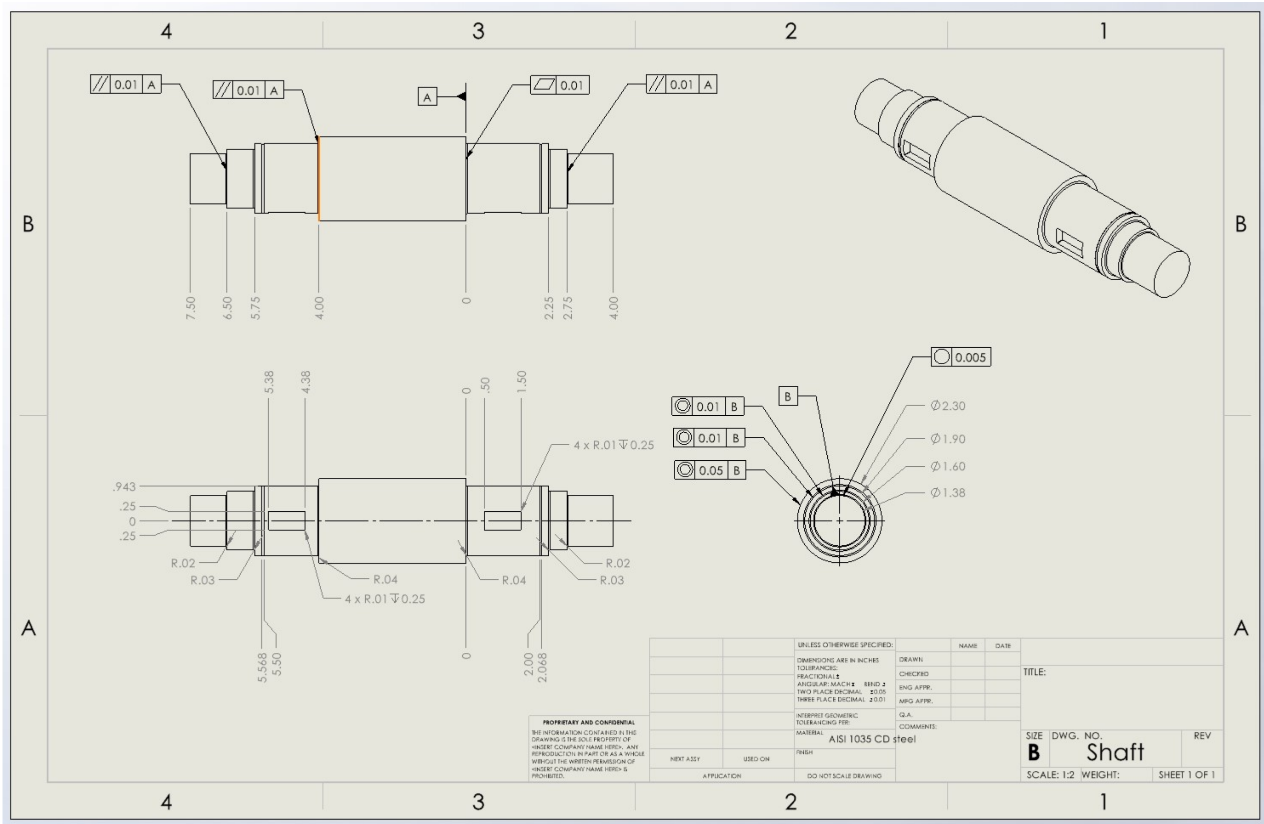

M x-y Plane
M x-y Plane (lbf-in)
1500
1000
500
0
0 1 2 3 4 5 6 7 8 9 10


M x-z Plane
M x-z Plane (lbf-in)
4000
3000
2000
1000
0
0 1 2 3 4 5 6 7 8 9 10


M Total
MTot (lbf-in)
Distance from Datum (in)
0
1000
2000
3000
4000
0
1
2
3
4
5
6
7
8
9
10


Torque
Torque (lbf-in)
Distance from Datum (in)
0
500
1000
1500
2000
2500
3000
0
1
2
3
4
5
6
7
8
9
10

Appendix C


0.01 A
0.01 A
A
0.01
0.01 A
7.50
6.50
5.75
4.00
0
2.25
2.75
4.00
0.005
0.01 B
0.01 B
0.05 B
B
Ø2.30
Ø1.90
Ø1.60
Ø1.38
5.38
4.38
0
.50
1.50
4 x R.01 ↧ 0.25
.943
.25
0
.25
R.02
R.03
R.04
R.04
R.02
R.03
4 x R.01 ↧ 0.25
5.568
5.50
0
2.00
2.068
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
MATERIAL
AISI 1035 CD steel
TITLE:
SIZE B
DWG. NO.
Shaft
REV
SCALE: 1:2
WEIGHT:
SHEET 1 OF 1
PROPRIETARY AND CONFIDENTIAL
DO NOT SCALE DRAWING