ME-465: Design Project
Final Report

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1. Background

This project involves the design of a proposed winch system which must meet some given minimum requirements. From the provided project description:

> *A winch, schematically shown in the figure below* (Figure 1)*, is operated by a worm-gear mesh. The input torque to the worm is provided by an electric motor that rotates at 1,500 rpm. The expected speed of the worm gear is between 30 and 35 rpm. The peak torque requirement for the winch is about 4,000 lbf·in, the operating ambient temperature is 120 °F, and the average output power doesn't have to exceed 1.2 hp. Knowing that the winch drum radius is 8 inch and that the winch operates 4 to 5 hours a day, perform the following tasks, which combined lead to the overall design project.*

2: Problem Statement

The second component of designing the winch is to select an appropriate shaft which will support the worm gear. From the provided project description:

> *Select the bearings (1 & 2) required to support the shaft holding the worm gear. Your report should incorporate, but not be limited to, the following: The overall project design, including the deliverables that were submitted in the progress reports. Updated deliverables from Progress Reports 1 & 2 based on the feedback you received on the correctness of your initial submissions. An overall summary (no longer than two pages) detailing the assumptions made during the design process, the outcomes of the design, and the lessons learned from this project.*

3. Objectives

The bearings supporting the gear shaft should meet the following requirements:

- Support the combination of radial and axial forces from the shaft
- Provide sufficiently long life and reliability

Bearings should also be designed with the objective of minimizing cost of parts, manufacturing, and assembly of the design.

4. Approach

Bearing selection will start with an analysis of forces on the bearings from the gear. Varieties of bearings can then be analyzed to determine which are feasible for the design based on forces, shaft speed, cost, durability, and reliability. Finally, specific bearings may be selected which meet requirements and provide sufficient performance for the winch design.

5. Assumptions

Forces acting on the bearing will be taken from the free body analysis of the shaft from a previous report, shown in Figures 2 and 3. To summarize, Bearing 1 must support a combined radial load of $F_{1,r} = \sqrt{175.2^2 + 288.2^2} = 377.3\ lbf$, while Bearing 2 must support a combined radial load of $F_{2,r} = \sqrt{45.2^2 + 211.8^2} = 216.6\ lbf$. The bearings must also support an axial load of F_a = 53.3 lbf.

A previous report found that the angular deflection at Bearing 1 was greater than the allowable deflection for tapered roller bearings. As such, this report will move forward with the assumption that a tapered roller bearing cannot be safely used at Bearing 1. While a tapered roller bearing may be appropriate for Bearing 2 which would accept the 53.3 lbf axial load during lifting operation of the winch, another method would be required to accept any axial loading in the opposite direction. The feasibility of implementing a tapered roller bearing at Bearing 2 will be examined in the sections below. As a result, it is assumed that Bearing 1 accepts negligible axial load.

The same report found that the angular deflection at Bearing 1 is approximately 0.003 rad for a shaft diameter of 2.6 in. at the gear and 1.875 in. at the bearings. Angular deflection at Bearing 2 is less than that at Bearing 1 by about an order of magnitude, and transverse deflection at the center of the gear face is about 0.0017 in. Acceptable deflection tolerances for various shaft components are listed in Table 1. For these shaft dimensions, the angular deflection at Bearing 1 is within tolerance for spherical ball bearings, and is very near the maximum for deep-groove ball bearings. With the abundance of bearings offered in metric dimensions, it will be assumed that the shaft diameter at the bearings can be made to accommodate bearings with bore diameters of at least $1.875\ in. \cdot 25.4 \frac{mm}{in.} \approx 48\ mm$.

The problem statement lists a required winch drum speed of 30−35 rpm. However, findings from a previous report concerning gear selection identified that this output speed was incompatible with other requirements. The maximum theoretical output speed given the 1.2 hp motor and 4,000 lbf·in output torque is 18.75 rpm, while the maximum output speed providing the maximum torque for the gears selected is 12 rpm. This report will move forward under the assumption, where appropriate, that the average speed of the gear shaft is approximately 15 rpm (considering that the winch may not always need to produce the maximum torque).

Desired combined reliability of both bearings will be assumed to be 99%. Application load factor a_f will be assumed to be about 1.3, which is within recommendations for "Machinery with light impact" [1, pp. 576]. Designed bearing life $\mathscr{L}_D$ will be taken as approximately 10 kh, per recommendations for "Machines for intermittent service where reliable operation is of great importance" due to the 4-5 hours of operation per day and the safety risks associated with lifting heavy loads overhead [1, pp. 575]. At this lifetime, bearings would be expected to operate daily for approximately six years before requiring maintenance or replacement.

Bearing calculations will use life ratings and Weibull parameters from manufacturer 2, which are similar to those used by SKF [1, pp. 601].

6. Calculations

The rating life from the manufacturer L_R is given as 10^6 revolutions. The design life required for the gear shaft is:

$$L_D = 60\, \mathcal{L}_D\, n = 60\ \tfrac{min}{hr} \cdot 10\ kh \cdot 15\ rpm = 9 \times 10^6\ \text{ revolutions}$$

This indicates a multiple of rating life of:

$$x_D = \tfrac{9 \times 10^6}{1 \times 10^6} = 9$$

The combined reliability of both bearings should be at least 99%. If both bearings are to have equal reliability, individual reliability must be $R_D \geq \sqrt{0.99} \approx 0.995$ for each bearing.

6.1 Bearing 1 calculations

Due to the high shaft angular deflection at Bearing 1, it will be designed to assume negligible axial load, which will instead be accepted by Bearing 2. As a result, Bearing 1 must only support the radial load of the shaft at this location which was found above to be 377.3 lbf.

Computing catalog load rating for Bearing 1 gives:

$$C_{10} \approx a_f F_D \left[\tfrac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{1/b}} \right]^{1/a} = 1.3 \cdot 377.3\ lbf \left[\tfrac{9}{0.02 + (4.459 - 0.02)(1 - 0.995)^{1/1.483}} \right]^{1/3} = 1943.6\ lbf$$

Thus, ball bearings with a catalog load rating of approximately 1950 lbf or, equivalently, about 8675 kN should provide the necessary performance, lifetime, and reliability. This equates to single-row 02-series deep-groove or angular-contact ball bearings with bores of at least 17 mm [1, pp. 573]. The shaft design from a previous report uses a diameter at Bearing 1 of about 48 mm. This large diameter was used to reduce angular deflection at Bearing 1, but necessitates selecting bearings with catalog load ratings about four times what was found above, and at about four times the price. The diameter of the shaft at the bearings could be reduced to permit selection of smaller, more appropriate bearings, but this reduction would push the angular deflection of Bearing 1 above acceptable limits. The shaft diameter at the gear location could be increased to compensate for the reduced bearing location diameter, but with a hub outer diameter of only 3.0 in., the thickness of the gear hub is already reduced to a potentially significant degree and increasing the shaft diameter beyond the current 2.6 in. would reduce it further.

Alternatives to roller bearings should be considered. Journal bearings are one possibility. Assuming lubrication by SAE-30 oil and a sump temperature of about 175°F, oil viscosity will be approximately 2.17×10^{-6} reyn. Further assuming a bore of 1.875 in. and a bearing length of 1.0 in., the bearing modulus for Bearing 1 can be found:

$$BM = \frac{\mu N}{P} = \frac{2.17\times10^{-6} reyn \cdot 15\ rpm}{201.23\ psi} \approx 1.62 \times 10^{-7}$$

Design recommendations suggest that the bearing modulus exceed 1.7×10^{-6} for stable, thick film lubrication. Achieving this value at such a low shaft speed would require a bearing length of over 10 in., which, needless to say, is not feasible for mounting the shaft to the gear case. Due to this calculation as well as the high angular deflection of the shaft at Bearing 1, thick film journal bearings do not seem to be any more appropriate than ball bearings.

Lastly, boundary-lubricated bushings can be considered. Bushings excel at high load, low speed applications. Bushing wear can be calculated by the following:

$$P = \frac{F}{DL} = \frac{377.3\ lbf}{1.875 in. \cdot 1.0 in.} = 201.23\ psi$$

$$V = \frac{\pi DN}{12} = \frac{\pi \cdot 1.875 in. \cdot 15 rpm}{12\ \frac{in}{ft}} = 7.363\ \frac{ft}{min}$$

$$f_1 \approx 1.15 \qquad \textit{(Rotary, P<720psi, 3.3<V<33)}$$

$$f_2 \approx 4.5 \qquad (140° < T_a < 210°, \textit{ no foreign matter})$$

$$K = 0.6 \times 10^{-10} \frac{in^3 \cdot min}{lbf \cdot ft \cdot hr} \qquad \textit{(Oiles 500 bushing)}$$

$$w = \frac{f_1 f_2 K F N t}{3L} \Rightarrow \frac{w}{t} = \frac{1.15 \cdot 4.5 \cdot (0.6\times10^{-10}) \cdot 377.3 lbf \cdot 15 rpm}{3 \cdot 1.0 in.} = 5.86 \times 10^{-7} \frac{in.}{hr.}$$

The operation time required for such a bushing to lose 0.01 in. of material in diameter (the amount of allowable transverse deflection at the center of the gear face) is approximately 17,000 hours, or about 10 years of 4−5 hours of daily use. Bushings can also be purchased or designed with flanges to accept significant thrust loads. Angular shaft deflection limits for bushings were not immediately available, but their lack of moving parts and thick-film lubrication alleviates many of the tolerance problems associated with roller and journal bearings. This makes them ideal candidates for Bearing 1, and will likely provide a suitable solution for Bearing 2, as will be calculated next.

6.2 Bearing 2 calculations

Bearing 2 must support both a combined radial load of 216.6 lbf (0.963 kN) and thrust load of 53.3 lbf (0.237 kN). Calculating the catalog load rating will be performed iteratively. An initial guess of $F_a/C_0 = 0.070$ provides an effective radial load of:

$$F_e = X_i F_r + Y_i F_a = 0.56 \cdot 0.963 kN + 1.63 \cdot 0.237 kN = 0.926\ kN$$

This is less than the combined radial load, so the original radial load will be used instead. The initial catalog load rating is then:

$$C_{10} \approx 1.3 \cdot 0.963\ kN \left[\frac{9}{0.02+(4.459-0.02)(1-0.995)^{1/1.483}}\right]^{1/3} = 4.961\ kN$$

The smallest angular contact bearing which meets this requirement has a catalog load rating C_{10} of 7.02 kN and a basic static load rating C_0 of 3.05 kN [1, pp. 572]. This allows the computation of the ratio:

$$F_a/C_0 = 0.237 / 3.05 = 0.0777$$

This generates a new guess for calculations:

$$e \approx 0.276$$

$$\frac{F_a}{VF_r} = \frac{0.237}{1 \cdot 0.963} = 0.246 \leq e \;\Rightarrow\; X_1 = 1.00,\ Y_1 = 0$$

The combined radial load is still greater than the effective radial load, so the catalog load rating has not changed:

$$C_{10} \approx 4.961\ kN$$

This is less than the catalog load rating for the bearing, so this bearing will sustain the combined radial and axial loading. However, with a bore diameter of only 12 mm, the same issue encountered for Bearing 1 is revisited here for Bearing 2. The shaft diameter cannot be reduced to accommodate smaller bearings without increasing already problematic deflections, and larger bearings are significantly stronger and more expensive than needed for this design. Instead, bushings will again be investigated for effectiveness.

The nominal pressure at Bearing 2 is:

$$P = \frac{F}{DL} = \frac{216.6\ lbf}{1.875 in. \cdot 1.0 in.} = 115.5\ psi$$

All other values are identical to those calculated for Bearing 1. This results in a wear of:

$$\frac{w}{t} = \frac{1.15 \cdot 4.5 \cdot (0.6 \times 10^{-10}) \cdot 216.6 lbf \cdot 15 rpm}{3 \cdot 1.0 in.} = 3.36 \times 10^{-7}\ \frac{in.}{hr.}$$

The lower radial load on Bearing 2 results in a longer lifespan, losing 0.01 in. in diameter after about 18 years of 4−5 hours of daily use.

7. Results

Due to the shaft diameter required to mitigate angular deflection of the shaft, ball bearings prove to be a poor choice for the winch design. The gear shaft will instead be supported by flanged bushings at Bearing 1 and Bearing 2 locations. These bushings are perfectly suited to the low-speed operation of the winch and are more affordable than roller bearings with similar bore diameters. The lifetime of the bushing will depend on the material, but can exceed the rated life of bearings and can be replaced quickly and easily when wear becomes significant.

8. Discussion

Selecting bearings for the gear shaft involves navigating a curious dilemma: bearings with bore diameters as small as 12 mm are strong enough to withstand the forces from the shaft, but the shaft diameter must be significantly larger to prevent excessive deflection of the shaft. Maintaining the shaft diameter would require the use of bearings four times as large as needed, with an accompanying price increase for additional material and strength which is entirely unneeded. Journal bearings would be a suitable alternative at higher shaft speeds, but the sluggish 15 rpm output of the winch is not sufficient to maintain a minimum lubrication film thickness. Instead, bronze bushings will be used due to their low cost and ease of replacement.

9. Summary of Project Results

With the design of the individual winch components complete, a complete picture of the design is available. Figure 4 shows an assembly of the gear and bushings mounted on the shaft, enclosed in a gear case with the worm and worm shaft.

9.1 Gear selection

The initial problem stated that we should convert the 1500 rpm, 1.2 hp output of the motor to the required 4,000 in·lbf torque to the winch drum, which should have an output speed of 30–35 rpm. The design needed to be able to operate in 120°F temperatures without producing excessive heat and operate under a given load 4–5 hours per day without excessive wear or damage. A focus on minimizing cost and size was emphasised when approaching the design of this system. However, after an initial analysis, it became clear that the input speed, output speed, torque, and power requirements could not all be met simultaneously. If the first three were met, then the power requirements became too high, and similarly if the power or speed requirements were met, then the torque fell too low. To resolve this, we proposed reducing the input and output speeds to 1,200 rpm and 15–20 rpm, respectively. This allowed for the torque, stress, and sump temperature requirements to be met by two worm gears: WB6100 and WB696. The WB696 (6 tpi diametral pitch, 96 teeth) gear was selected, along with the matching WHG6 worm, as they were slightly smaller and cheaper than the other option and still maintained a low sump temperature.

9.2 Shaft Design

To design the shaft, it was stated that it would need to deliver 4,000 in·lbf torque from the worm gear to the winch drum, and be able to do so without excessive bending or torsion, and without failing. Using the gears selected in the previous report, diameters for sections of the shaft were selected, along with lengths that would meet the needs outlined for this project. After the initial design was created using the assumptions outlined above, a simple force analysis of the free body diagram was performed, which allowed for a fully detailed shaft to be created. Analyses were then performed on each point of interest, to determine the factors of safety at each point, and if any of these points would fail due to fatigue or yielding. After determining that they would not, a deflection analysis was performed, which revealed an issue with the thickness of the shaft. It was decided that the issue could be potentially resolved with the selection of bearings, and so was put on hold until the bearings were selected. The team then moved to the final critical speed analysis, which showed that there were no further issues with the design. This concluded the design of the shaft, with a potential for the diameter to be altered in the bearing selection stage of the design, if they could not resolve the diameter issue alone. The final shaft design is shown in Figure 5.

9.3 Bearing Selection

Bearing selection started with an analysis of forces on the bearings from the gear. With this information, several bearing options were identified and expanded upon. Analysis started with the assumption of using a ball bearing, but data from the shaft analysis showed that the shaft deflection would be too much using a ball bearing. Due to this, several other types of bearings, such as roller bearings were considered. After the analysis was complete, it was decided that the gear shaft should be supported by flanged bushings at Bearing 1 and Bearing 2 locations. These bushings were perfectly suited to the low-speed operation of the winch and are more affordable than roller bearings with similar bore diameters. The lifetime of the bushing would depend on the material, but could exceed the rated life of bearings and could be replaced quickly and easily when wear becomes significant.

9.4 Conclusions

Figure 6 shows the completed gear shaft assembly with gear, bushings, key, and retaining rings installed. Mechanical drawings illustrating the complete assembly are submitted alongside this report.

Meeting all design objectives for this project was ultimately not possible, which made some aspects of the project difficult. Having no client to consult regarding what direction to proceed once an objective could not be met meant making these decisions independently with little to no knowledge of what design objectives could be compromised while preserving the utility of the winch. However, this problem-solving was a helpful exercise in engineering design. The strategy used in these situations was to identify and quantify the problem, consider as many solutions as possible, and weigh the advantages and disadvantages of each. Often, only one or two of the several possible solutions are reasonable. In a true design project, these solutions would be presented to the client, addressing how they solve the problem and what compromises would be made implementing them. The client would then indicate which, if any, of these compromises are acceptable and the design can move forward with that solution.

In the case of this report, however, assumptions must be made regarding the use of the device and what functions and design objectives are critical and which can be adjusted. In this case, it was assumed in the gear selection process that the torque output of the winch was more important than the speed of the winch drum shaft. This was assumed due to the fact that a slower speed can lift any load up to the required 500 lbf albeit at a reduced speed, while meeting the output speed limitation would reduce the maximum load capacity of the winch below the stated requirement. In essence, any load within stated limits can be lifted with a slower output with only the inconvenience of longer time required to lift, while maintaining the faster output speed permits the *convenience* of faster lifting time but with the *impossibility* of lifting some loads. This maximizes the functionality of the device within the design objectives. If this was proposed and the client emphasized the importance of meeting the output speed, the reduced capacity could be implemented, or a more powerful motor could be used to drive the winch at both the rated output torque and speed.

This final design is calculated to be sufficient to handle the stresses of operating the winch. Testing should be done on a prototype of the design to ensure expected operation and durability. The design may be updated with the findings from this testing, but this is outside the scope of this report. The design of the gears, shaft, and bearings are considered sufficient and completed at this point.

10. Appendices

A1. Figures

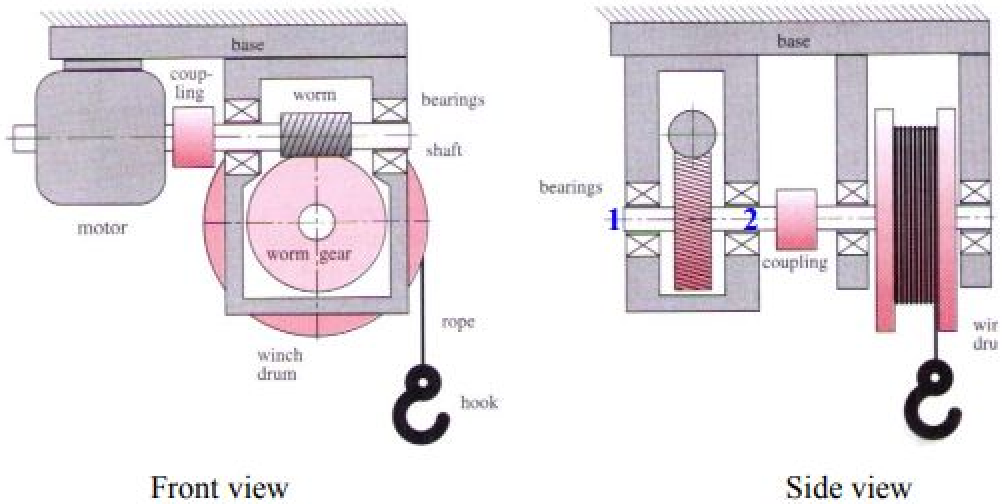

Figure 1: Diagram of the basic winch setup, provided in the project description

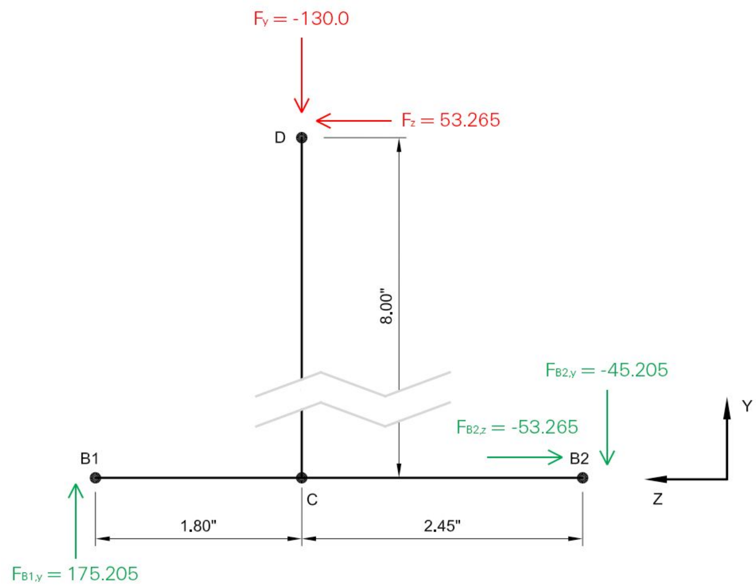

Figure 2: Right-view shaft free body diagram

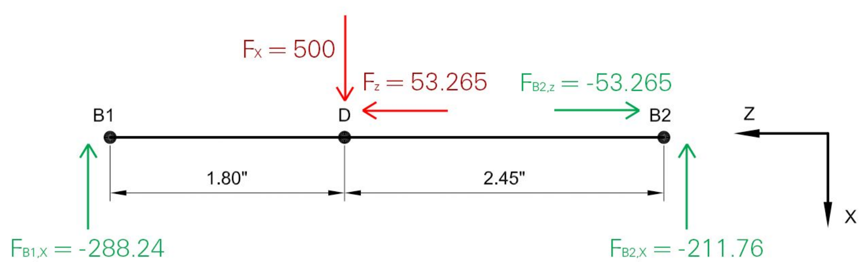

Figure 3: Top-view shaft free body diagram

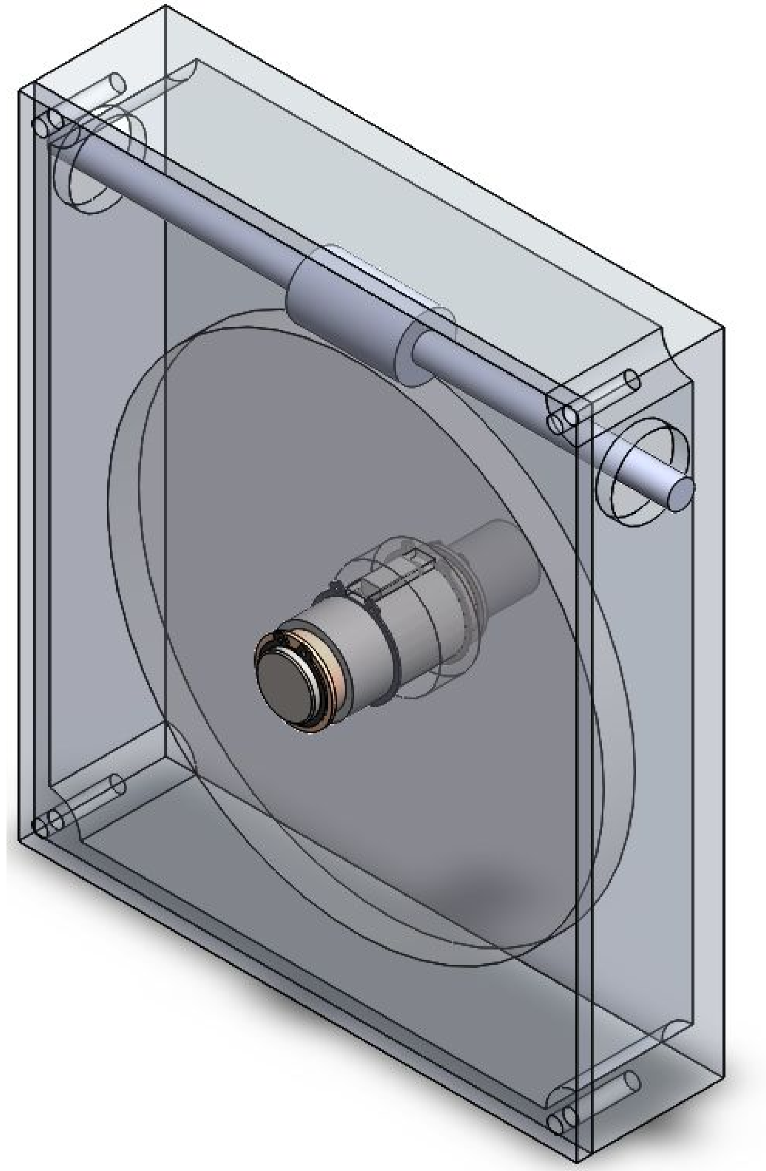

Figure 4: Winch gearbox assembly

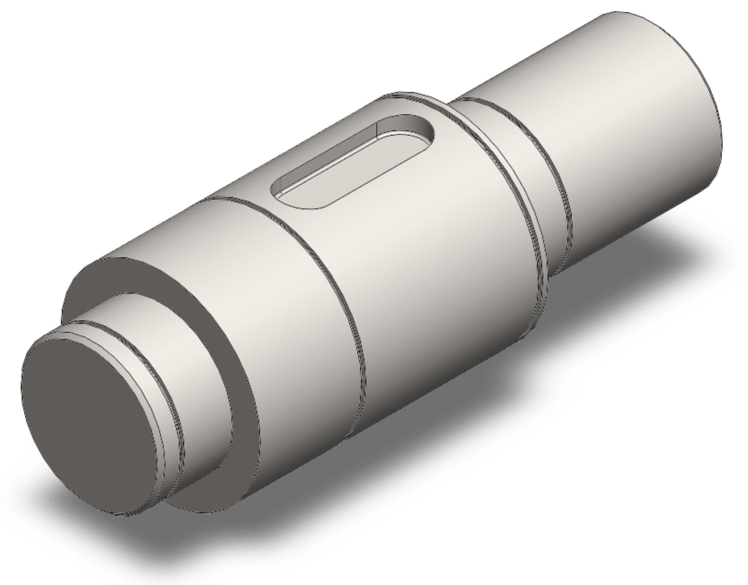

Figure 5: Final shaft design

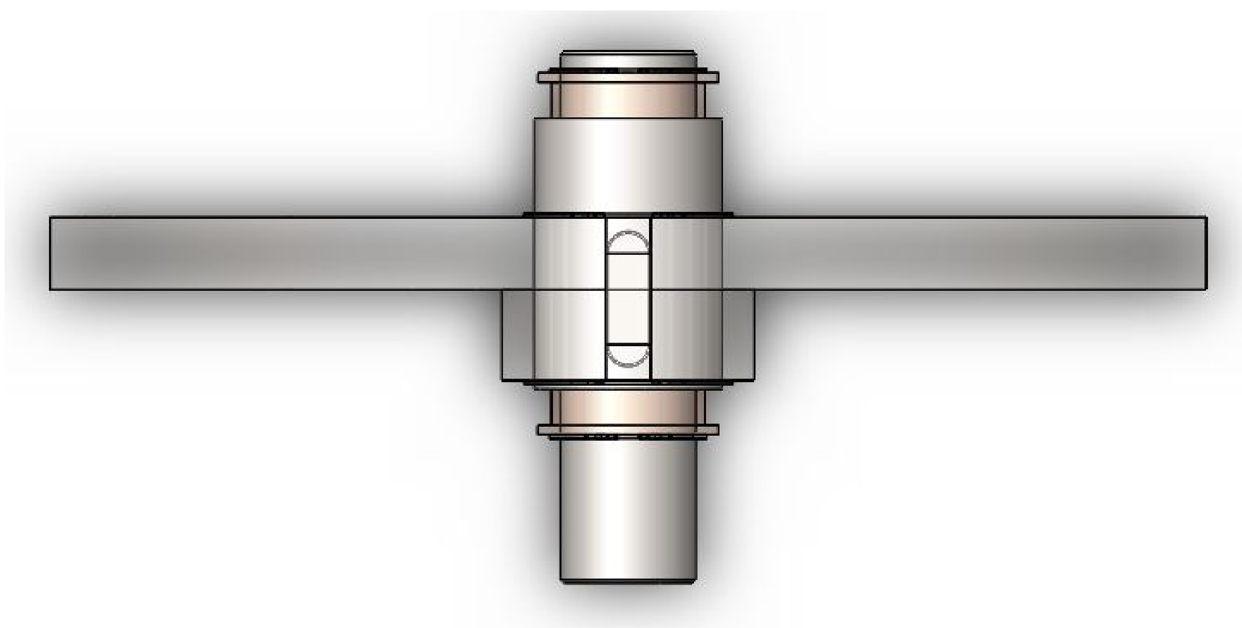

Figure 6: Top-view of gear shaft assembly

A2. Tables

Table 1: Maximum deflection for various shaft components [1, pp. 371]

Slopes (rad)	
Tapered roller bearings	0.005−0.0012
Cylindrical roller bearings	0.0008−0.0012
Deep-groove ball bearings	0.001−0.003
Spherical ball bearings	0.026−0.052
Transverse deflection (in.)	
Spur gears with $P < 10$ tpi	0.010

A2. References

[1] R. G. Budynas and J. K. Nisbett, Shigley's Mechanical Engineering Design, 5th ed. New York, NY, US: McGraw-Hill, 2015.