ME 324

MACHINE DESIGN I

Power Screw Flap Actuator Design

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Abstract

The report starts with an introduction where we establish our objectives and requirements for the project. We go over the basic design and how it works in basic terms. The assumptions made in the calculations of the project are given. The important parts and the use of these parts are given.

The introduction is followed by the design methodology where we go into the specifics of the calculations, go over the process of calculating the right numbers for our project and designing the 3d model of our project. Design methodology starts with calculating the dimensions of the parts of our project according to our given data. After that, there is the manufacturing processes we would've chosen if we were to make this project into a physical prototype.

After the manufacturing part, we show how we designed the parts and assembled them in Solidworks. There are also detailed drawings of the parts we used in the project. In the conclusion, we wrap up the project and discuss the possible future improvements on the project. We also discuss how this project helped us conceptualize the things we learned in this class.

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1. Introduction

In this project, a power screw flap actuator is designed. The objective in our design was to make a power screw flap actuator that could accommodate the maximum force requirement of 8800 N and move at a speed of 10mm/s. To execute this task, an electric motor, 4 gears, 2 shafts, 4 bearings and a power screw are used. In addition to that, detail parts such as circlips, lock nut/washer, keys, bearing housings, and coverage case are used for fixation and corrosion prevention. The aim was to transform rotational motion, provided by electric motor and transferred via series of shafts and gears, into linear motion via power screw. Gears are utilized to reduce the angular speed and increase torque, whereas power screw is to transform rotational motion into linear motion. Another design objective was to make the actuator as compact and light as possible. Thus, it is aimed not to exceed safety factor of 3. However, in order to be on the safe side, it is also aimed not to fall below safety factor of 2.

Key Assumptions:

- Maximum temperature is 50ºC. (C_T is taken 1 throughout the design)
- Design life is at least 8 years and the actuator operates 10 hours per day at top speed.
- The actuator is assumed to be working only under constant raising torque except for the power screw, and the lowering torque is neglected .
- Reliability factor is %95.

Power Screw Assumptions:

- Surface is commercially polished.
- The screw is fixed at one end due to bearing 4, and pinned at the other end since it is fastened to flap.
- The screw is lubricated with machine oil.
- The material for the screw nut is steel.
- Threads are assumed as grooves with 0.5 mm notch radius under torsion to take into account stress concentration factor and in turn fatigue notch factor.
- The cycle is the full period of the screw nut back and forth. All stresses are alternating. The lowering torque is switched with raising torque and the screw is under completely reversed load.

Gear Assumptions:

- Surface is commercially polished.
- Both the source of power and the driven machinery are working under uniform load.
- Gears are fixed by accurate mountings.
- Gears are manufactured via shaving or grinding with high precision.
- The load is applied at the tip of the tooth. No load sharing occurs.

Shaft Assumptions:

- Shafts are manufactured via hot-rolling.
- Dictating factor of safety for shaft 2 is the lead screw’s factor of safety since they are integrated.

Bearing Assumptions:

- There is no sharing of axial load. Only bearing 3 accommodates axial force.

2. Design Methodology

2.1. Power Screw

Since the linear speed of the power screw movement and the applied force at the tip is given, we began designing the actuator starting from power screw. AISI 4340 Steel was selected as the material for the power screw due to it's high ultimate strength. Since the desired life for the power screw corresponds to infinite life cycle, the selection of material was dictated by ultimate strength value – keeping in mind that we also aim for a small design since we could also select greater diameter. Then, square thread type was selected for the screw. The major diameter and pitch size were selected as 25.4 mm and 6.35 mm by iteration after series of trials. Required angular speed was then calculated as 94.5 rpm since linear speed and lead size is known (Eq.1).

$$rpm = \frac{60v}{l} \tag{1}$$

Normal stress was found at the smallest diameter of the screw which is root diameter and factor of safety for global yielding was determined (Eq. 2).

$$\sigma_z = \frac{F}{A_r} \tag{2}$$

Then, using Johnson's formula, critical force was derived and factor of safety for buckling is determined (Eq. 3).

$$\frac{P_{cr}}{A} = S_y - \left(\frac{S_y}{2\pi}\frac{l}{k}\right)^2 \frac{1}{CE} \tag{3}$$

In determining the end-condition constant required to calculate the critical force, the screw was assumed to be fixed at one end and pinned at the other end. Then, raising and lowering torques were found(Figure 4). The self-locking criterion was met. In determining the coefficient of friction required to calculate raising and lowering torques, the power screw was assumed to be lubricated.

$$T_R = \frac{F d_m}{2}\left(\frac{l + \pi f d_m}{\pi d_m - f l}\right) \quad T_L = \frac{F d_m}{2}\left(\frac{\pi f d_m - l}{\pi d_m + f l}\right) \tag{4}$$

Then, required power and efficiency were found to be 250W and 35% (Figure 5). Accordingly, proper electric motor was selected with the power of 250W and angular speed of 680 rpm.

$$\dot{W} = \frac{T_R \; x \; n \; x \; 2\pi}{60} \qquad e = \frac{T_0}{T_R} \tag{5}$$

Subsequently, bending, bearing and shear stresses were calculated at the root of the threads. By assuming threads as grooves, stress concentration factor was taken into account. Von misses stress was calculated and factor of safety for local yielding was determined (Figure 6).

$$\sigma' = \sqrt{\sigma_y^2 + \sigma_z^2 - \sigma_y\sigma_z + 3\tau_{xz}} \qquad \text{FoS} = \frac{S_n}{\sigma_{ar}} \tag{6}$$

Then, after determining correction factors and fatigue notch sensitivity, endurance limit was set. The cycle was assumed to be the full period of nut movement back and forth and in turn all stresses were assumed to be alternating. Since all stresses were assumed to be alternating, alternating stress became equal to von misses stress (Figure 7). As the final procedure, factor of safety for fatigue was found to be 2.01 – becoming the factor that dictates the design and leading us to make iterations to come up with the proper power screw dimension.

$$\sigma_{ar} = \frac{\bar{\sigma}_a}{1 - \frac{\bar{\sigma}_m}{\sigma_u}}, \quad \bar{\sigma}_m > 0 \qquad\qquad \sigma_{ar} = \bar{\sigma}_a, \quad \bar{\sigma}_m \leq 0$$

EFFECTIVE STRESS AMPLITUDE

$$\bar{\sigma}_a = \bar{\sigma}_{aHMH} = \frac{1}{\sqrt{2}}\sqrt{\left(\sigma_{1a}-\sigma_{2a}\right)^2 + \left(\sigma_{1a}-\sigma_{3a}\right)^2 + \left(\sigma_{2a}-\sigma_{3a}\right)^2} = \tag{7}$$

$$= \frac{1}{\sqrt{2}}\sqrt{\left(\sigma_{xa}-\sigma_{ya}\right)^2 + \left(\sigma_{xa}-\sigma_{za}\right)^2 + \left(\sigma_{ya}-\sigma_{za}\right)^2 + 6\left(\tau_{xya}^2 + \tau_{xza}^2 + \tau_{yza}^2\right)}$$

For the power screw, a process for manufacturing is the thread rolling process where a cylinder which has a diameter between the root and the nominal diameter, is not cut rather rolled to be in the shape of a power screw [1]. In this process no material is cut but displaced to shape the cylinder (Figure 8). Power screws created using this process are stronger than ones created by cutting.

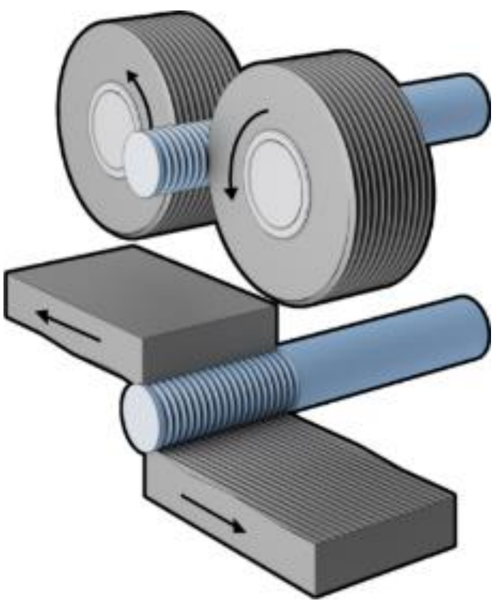

Figure 1: Thread rolling process

In Figure 2, 3D CAD and technical drawing of the power screw are provided.

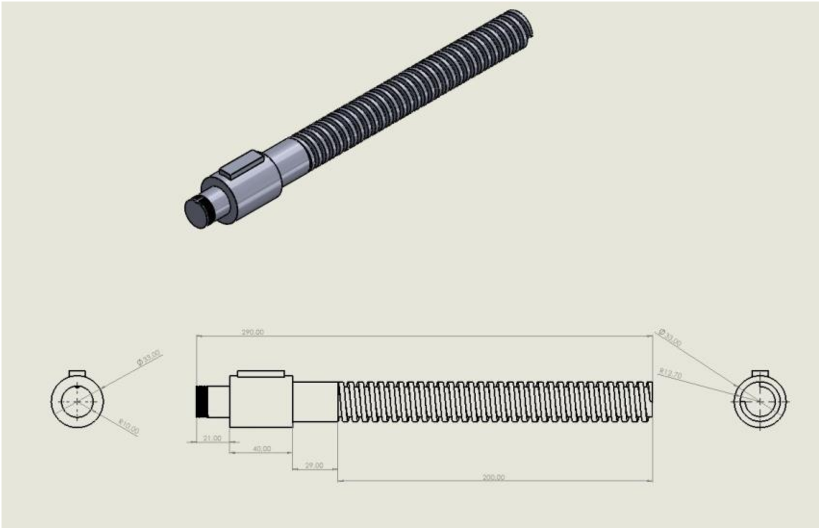

Figure 2: 3D CAD of power screw (shaft 2) and its corresponding technical drawing

2.2. Gears

Upon the completion of power screw design and selection of electric motor, we began designing the gears. AISI 1045 Steel was selected as the material for the gears due to it’s greater ultimate strength even than of AISI 4340 Steel. Since the desired life for all the gears correspond to infinite life cycle, the selection of the material was dictated by ultimate strength value – keeping in mind that we also aim for a small design since we could also select greater diameter. Pressure angle was selected as 25°. Then, module was selected as 3 from Maryland Metrics Standarts. Correction factors (k_O, k_M, C_G, C_S, C_L, k_r, k_T, $k_{ms,}$ C_P, $C_{R,}$ C_{Li}) were selected. In the procedure of determining correction factors, source of power and driven machinery were assumed to be uniform, gears were assumed to be fixed wth accurate mountings and commercially polished. Using the correction factors, endurance limit was derived Figure (10).

$$S_n = S_n' C_L C_G C_S k_r k_t k_{ms} \tag{8}$$

Since the angular speeds of electric motor and power screw were known, train value was calculated. The ratio of teeth numbers between gear 1-2 and gear 3-4 were adjusted in a way that their multiplication would give train value (Figure 11).

$$\frac{\omega_1}{\omega_4} = \frac{\omega_1}{\omega_2} x \frac{\omega_3}{\omega_4} = \frac{N_2}{N_1} x \frac{N_4}{N_3} \tag{9}$$

So, initial values for number of teeth and face width were set. Since the angular speed of the motor is known, the pitch line velocities were calculated accordingly (Figure 12). Then, velocity factors were determined – assuming that the gears are manufactured with high precision.

$$v = \frac{\pi d n}{60000} \qquad K_v = \frac{78 + \sqrt{V}}{78} \tag{10}$$

Since the power of the electric motor is known and the pitch line velocity is found, forces (tangentional & radial) acting on the gears were calculated (Figure 13). Then, geometry factors were set.

$$F_t = \frac{\dot{W}_{motor}}{v} \qquad \mathrm{F_r} = \mathrm{F_t tan\phi} \tag{11}$$

Then, gear tooth bending stresses were found (Figure 14). Subsequently, factor of safety for bending was determined.

$$\sigma = \frac{\mathrm{F_t}}{\mathrm{mbJ}} \mathrm{K_\upsilon K_o K_m} \qquad \mathrm{FOS} = \frac{\mathrm{S_n}}{\sigma} \tag{12}$$

Surface fatigue strength was calculated using hardness value. By multiplying it with life and reliability factors, modified surface fatigue strength was derived (Figure 15).

$$S_H = S_{fe} C_{Li} C_R \tag{13}$$

Geometry factor was calculated for each gear. Upon finding out all unknown variables, hertzian contact stresses were found. Subsequently, safety factors for surface fatigue were calculated – becoming the factor that dictates the design and leading us to make iterations to come up with the proper gear dimensions (Figure 16).

$$\sigma_H = C_P \sqrt{\frac{F_t}{b d_p I} K_v K_o K_m} \qquad FoS = S_H / \sigma_H \tag{14}$$

For the manufacturing, in this project, we only used spur gears. For spur gears the main manufacturing process is hobbing. In this process, a cutting tool called hob (Figure 17) is used to cut out the teeth of the gear as the gear slowly rotates [2]. After our gears are cut, they would be finished using high precision shaving and grinding.

Figure 3: Hob cutting tool

In Figure 4 and Figure 5, both 3D CADs and technical drawings of pinions and of gears are provided.

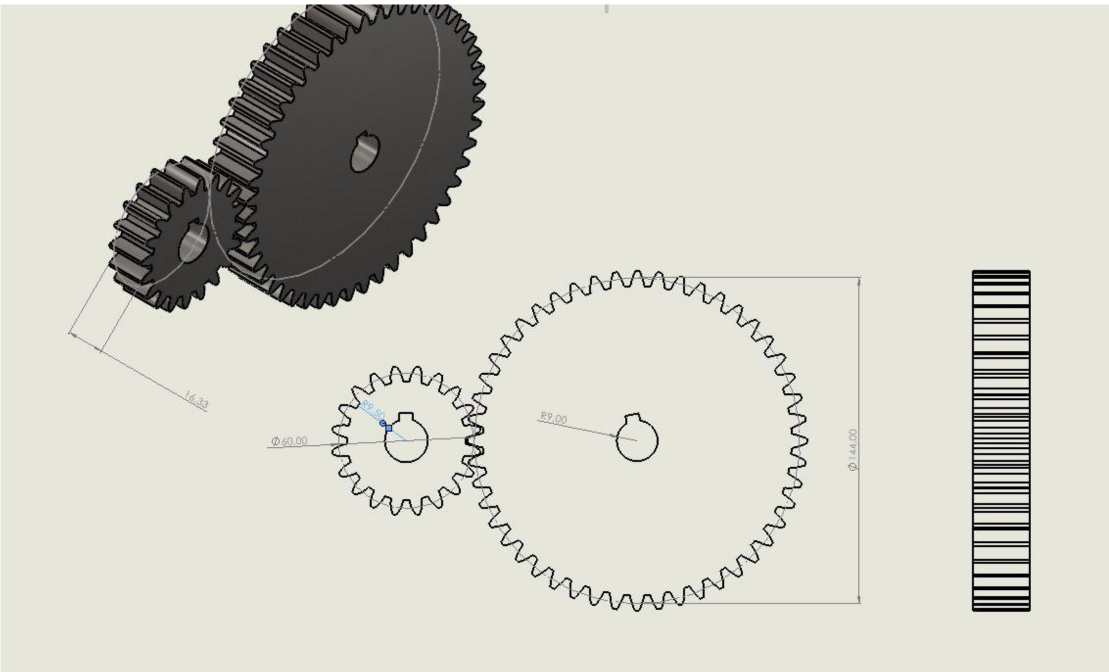

Figure 4: 3D CAD of gear 1&2 and their corresponding technical drawings

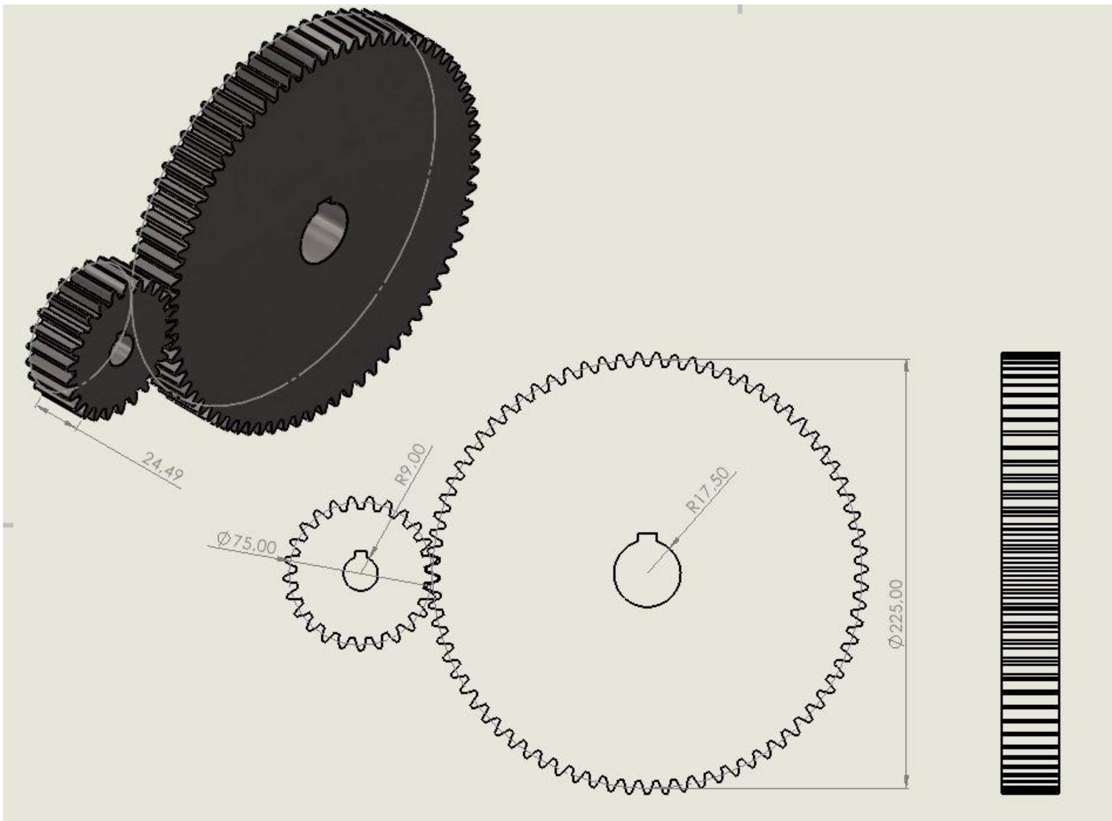

Figure 5: 3D CAD of gear 3&4 and their corresponding technical drawings

2.3. Shafts

Upon the completion of gears, we began designing the shafts. Shaft 2 is integrated with the power screw so half of the work was already done. As the material of the shafts, AISI 1020 steel was selected due to it's lower cost and ease of machining. Since the gears were designed to endure infinite life, their diameters accounted for the size of the system. Thus, it was unnecessary to design shafts with smaller diameters by using strong materials; so, cost-efficient steel having low tensile strength was selected. Then, correction factors (C_G, C_S, C_L, C_R, C_T) were determined. In the procedure of determining correction factors, the surface was assumed to be hot-rolled. Diameters of the shafts for the sections where gears are fastened were set. Using tangential forces applied on the gears, torques that are applied on the shafts were calculated. Subsequently, free body diagrams of the shafts were drawn (Figure 19).

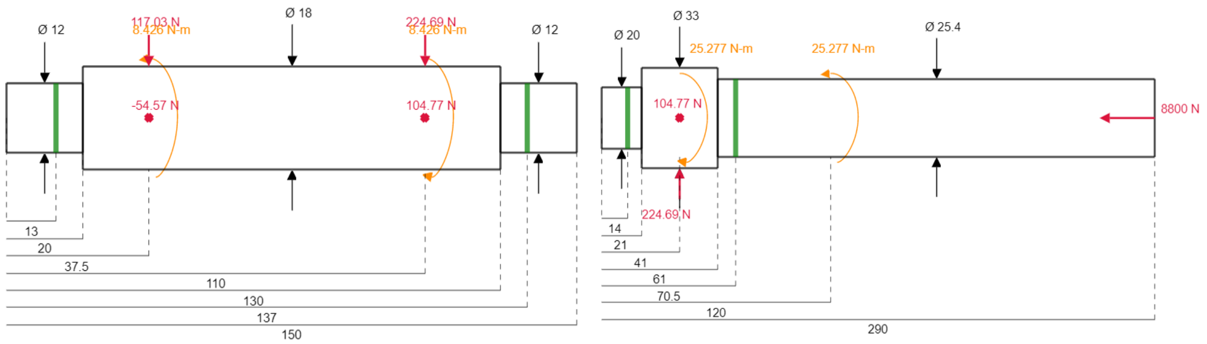

Figure 6: Free body diagrams of shaft 1 and shaft 2 respectively

Only four unknowns were left for each shaft due to bearings. Using 6 equations of equilibrium, bearings forces were also derived (Figure 20) (Figure 21).

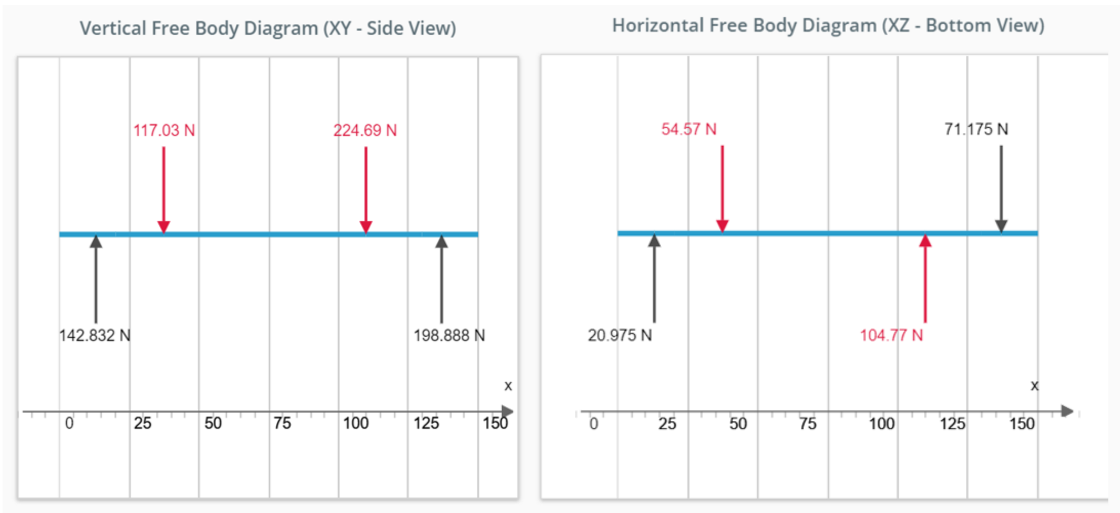

Figure 7: Bearing forces applied on shaft 1 marked with black

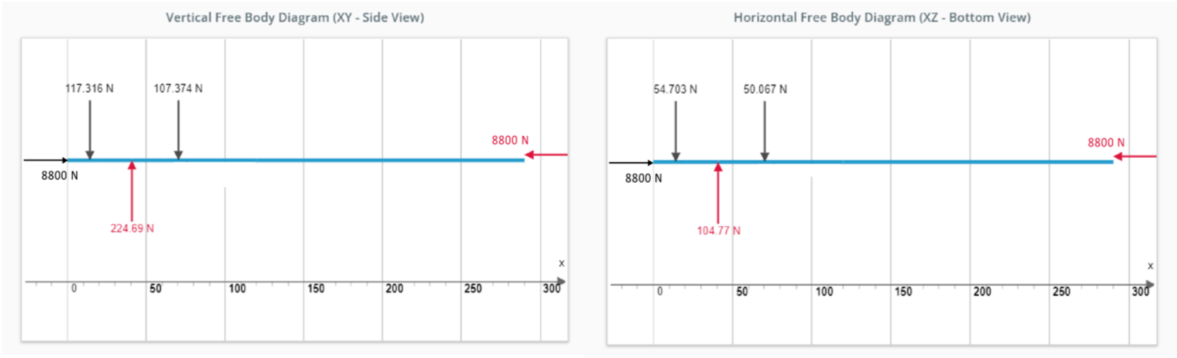

Figure 8: Bearing forces applied on shaft 2 marked with black

Shear and bending diagrams were drawn respectively. The most critical point was found out to be on gear 3 connection point for shaft 1 and gear 4 connection point for shaft 2 (Figure 22) (Figure 23).

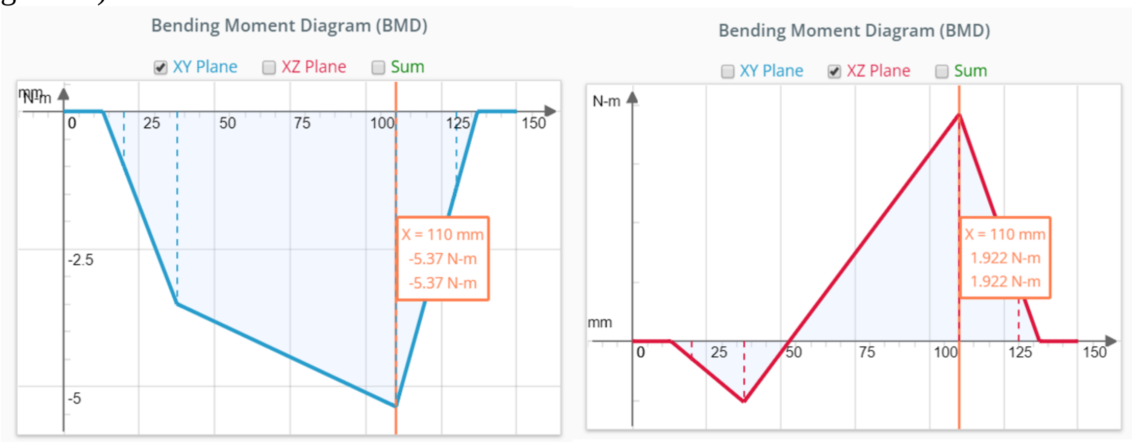

Figure 9: Bending diagrams of shaft 1

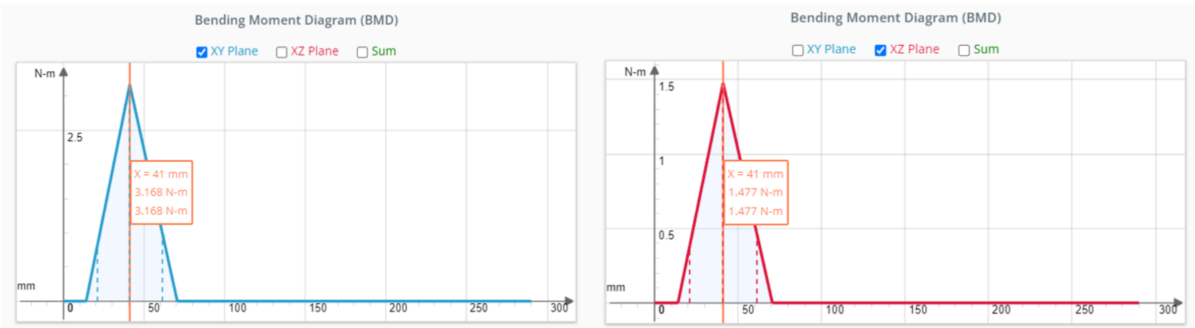

Figure 10: Bending diagrams of shaft 2

The normal and shear stress were calculated at the critical point (Figure 24).

$$\sigma = \frac{32M}{\pi d^3} \qquad \tau = \frac{16T}{\pi d^3} \tag{15}$$

The maximum shear stress was derived (Figure 25). Using Tresca Criterion, factor of safety for yielding was found.

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}{}^2} \qquad \mathrm{FoS} = \frac{\frac{S_y}{2}}{\tau_{\max}} \tag{16}$$

Endurance limit was calculated by multiplying half of the ultimate strength with correction factors. Normal stress stemming from bending was taken as alternating due to rotation of the shaft and in turn shear stress due to torque was taken as mean stress. By substituting load line into goodman line, factor of safety for fatigue was determined (Figure 26).

$$\mathrm{FoS} = \frac{\sqrt{\left(\frac{1}{\frac{1}{\mathrm{Su}} + \frac{\sigma_a}{\sigma_m S_n}}\right)^2 \left(1 + \frac{\sigma_a}{\sigma_m}\right)}}{\sqrt{\sigma_a^2 + \sigma_m^2}} \tag{17}$$

To manufacture the shafts, the most common method is CNC (Computer Numerical Control) turning [3]. In this method, the initial shaft is rotated very fast and cutting tools are used to cut away metal and make a new shaft at the desired diameter.

In Figure 11, 3D CAD and technical drawing of the shaft are provided.

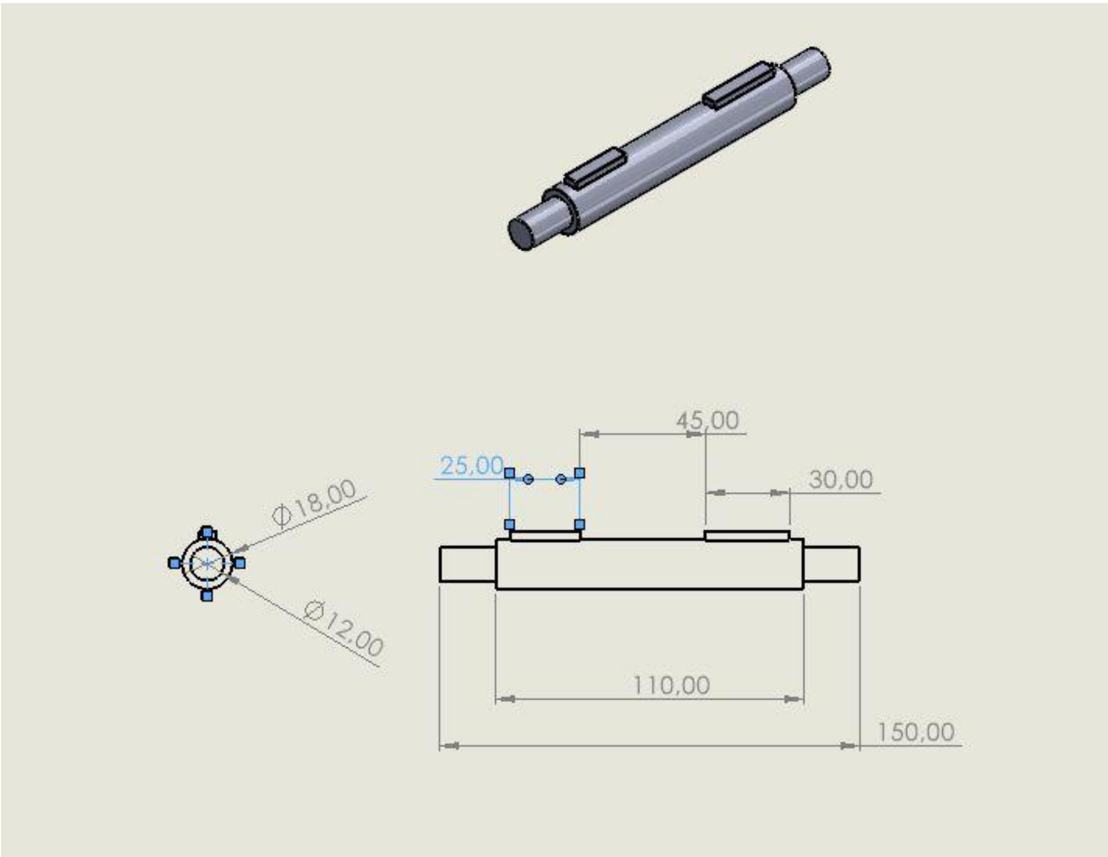

Figure 11: 3D CAD of shaft 1 and its corresponding technical drawing

2.4. Bearings

Using SKF bearing selector tool, proper bearings that are able to accommodate applied forces and endure desired life were selected. Exponent of the life equation and reliability correction were determined. Inner diameters were selected. Since there is no axial force applied on shaft 1, bearing 1&2 were selected as self aligning ball bearings; whereas, due to presence of axial force in shaft 2, bearing 3&4 were selected as deep groove ball bearings. Basic dynamic load ratings and basic rating lifes were derived from SKF website. Basic rating life was in terms of hours so it was converted into revolution form (Figure 28).

$$L_{10} = L_{10h}.60.n \tag{18}$$

Radial forces acting on the bearings were calculated from y and z components (Figure 20) (Figure 21) (Figure 29).

$$F_r = \sqrt{F_y + F_z} \tag{19}$$

For bearing 1-2-4, radial forces became equal to equivalent dynamic load; whereas, equivalent dynamic load was greater than radial force for bearing 3 since it accomodates axial force (Figure 30).

$$\begin{aligned} &For \quad 0 < \frac{F_t}{F_r} < 0.35, \;\; F_e = F_r \\ &For \quad \frac{F_t}{F_r} > 10, \quad F_e = 1.176F_t \end{aligned} \tag{20}$$

Then, desired life was calculated for each bearing in terms of revolution due to different angular speed between shafts (Figure 31).

$$N = 8\ x\ 365\ x\ 10\ x\ 60\ x\ n \tag{21}$$

As the final procedure, the design life was calculated and checked to see whether it surpasses the desired life or not (Figure 32).

$$L = K_r L_{10}(C/P)^{\,p} \tag{22}$$

For the bearings, the first step of production is heating the material to a necessary temperature, then forging it into the outer and the inner shells. After forging comes turning, where the shells and the raceway are cut. Then these shells go through heat treatment. Then grinding is done to precisely finish the details of the shells. After that the steel balls are made and they are all assembled [4].

Finally, for after the manufacturing process, there is the mounting and dismounting of the bearings. We can find the instructions for these in the skf website (mount.skf.com). To mount the bearing, an skf fitting tool can be used to push the bearing into place after aligning with the shaft. A locking device is then used to lock the bearing in place free to rotate. Similarly, to dismount, an sfk puller (Figure 33) can be used to pull the bearing out.

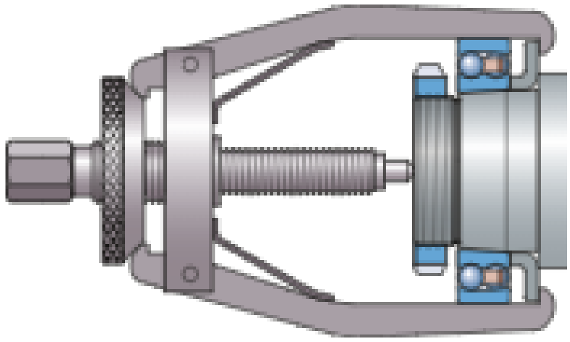

Figure 12: Illustration of a puller

In Figure 13 and Figure 14, 3D CAD of bearings and of the electric motor are provided, respectively.

Figure 13: 3D CAD of bearings

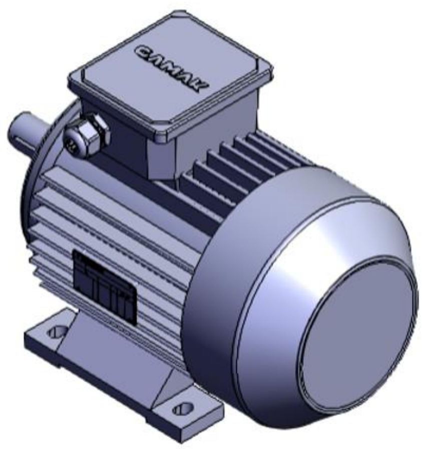

Figure 14: 3D CAD of electric motor

In Figure 15, 3D CAD and technical drawing of the power screw casing are provided.

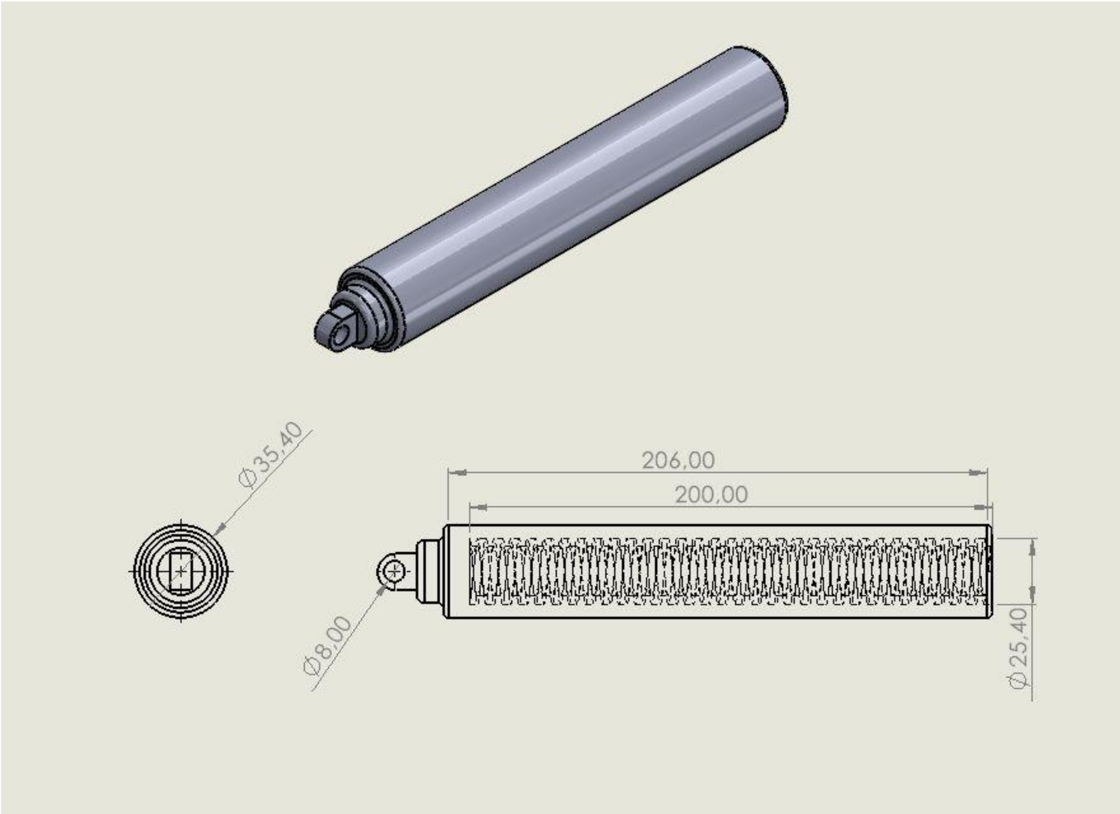

Figure 15: 3D CAD of power screw case and its corresponding technical drawing

3. Conclusion

In conclusion, this project has helped us better understand the concepts and the processes of machine design. We experienced designing a piece of machinery firsthand. From the calculations to the modelling. Without this project, what we learned in this class would just be left as equations and we probably wouldn't be able to apricate the importance of the real-life use of those equations. This project was also challenging enough to get us ready for the world of mechanical engineering and the projects that we will have to do in the future.

Considering the vast scope of machine design, this was a basic project. We used an electric motor and gears to move an arm linearly at a certain speed. So future improvements on this project are limited.

References

[1] CJWinter, "What is thread rolling?," 4 May 2020. [Online]. Available: https://www.cjwinter.com/blog/what-is-thread-rolling.

[2] MechanichTips, "Gears manufacturing methods," 13 11 2012. [Online]. Available: https://www.youtube.com/watch?v=B0XSsa79Y1w.

[3] L. Wood, "What Is CNC Turning? – Simply Explained," 2 5 2020. [Online]. Available: https://all3dp.com/2/what-is-cnc-turning-simply-explained/.

[4] "Ball Bearing and Roller Bearing Manufacturing Process and Technology," 23 12 2017. [Online]. Available: https://uxgnep.jimdofree.com/2017/12/23/ball-bearing-and-roller-bearing-manufacturing-process-and-technology/.