Introduction

This report will specify the design of a drive system for a large commercial band saw. The saw is to be designed to transmit 12.0 hp. The saw will be used to cut steel tubing for an automotive exhaust pipes. The pinion rotates at 3450 rpm, while the gear must rotate between 725 and 735 rpm. It has been specified that the gears are to be made from SAE 4340 steel, oil quenched and tempered. Case hardening is not to be used. This report will detail a comprehensive design of all the parts within this system. The parts that will be designed are the following: two spur gears, two parallel shafts, axial locating elements, and four bearings. Spur gears produce only radial loads which simplify selection of the bearings that support the shafts. Spur gears are relatively inexpensive to produce. Shafts will be parallel which should be simple to align with the power transmitting device as well as the drive shaft for the saw. Please note that unless otherwise specified, all tables, figures, and charts, are in reference to *Machine Elements in Mechanical Design* fifth edition. Drawings were conducted using CREO parametric 4.0.

Design of Spur Gears

Given:

P = 12 hp

$n_p = 3450\ rpm$

$n_G = 725 - 735\ rpm$

SAE 4340 steel, oil quenched and tempered

In order to determine the overload design factor, the application and power source must be taken into account. An electric motor input is said to be uniform while the output of a band saw is said to have a moderate shock. Based off the previously stated, the suggested value of the overload factor is:

$$K_o = 1.5$$

Therefore the design power transmitted can be calculated by:

$$P_{des} = PK_o$$

Therefore:

$$P_{des} = 18\ hp$$

Based off figure 9-27, for the desired power and the pinion speed the suggested diametral pitch is:

$$P_d = 10$$

For this pitch and the application the value of the number of teeth for the pinion of 18 is sufficient.

$$N_p = 18\ teeth$$

In order to calculate the velocity ratio (VR) the average of the range of acceptable output speed of the gear will be used.

$$VR = \frac{n_p}{n_G}$$

Therefore:

VR = 4.73

With the velocity ratio and the number of teeth for the pinion known, the number of teeth for the gear can be calculated with the following:

$$N_G = N_p VR$$

$$N_G = 85.14\ teeth$$

The number of teeth must be rounded to the nearest integer value:

$$N_G = 85 teeth$$

The actual output speed can be calculated with the following:

$$n_G = \frac{n_p}{VR}$$

$$n_G = 731\ rpm$$

This value falls within the acceptable range.

The pitch diameters for both the pinion and the gear can now be calculated

$$D_P = \frac{N_P}{P_d}$$

$$D_P = 1.8\ in$$

$$D_G = \frac{N_G}{P_d}$$

$$D_G = 8.5\ in$$

The center distance and face width can now be calculated.

The center distance can be calculated with the following:

$$C = \frac{(N_G + N_p)}{P_d}$$

$$C = 5.15\ in$$

The lower limit of the face width is:

$$F_{lower\ limit} = \frac{8}{P_d}$$

$$F_{lower\ limit} = 0.8\ in$$

The Upper limit:

$$F_{upper\ limit} = \frac{16}{P_d}$$

$$F_{upper\ limit} = 1.6\ in$$

The nominal value:

$$F_{nom} = \frac{12}{P_d}$$

$$F_{nom} = 1.2\ in$$

In this design the use of steel has already been specified. Based off this predetermined material:

$$C_p = 27.7kpsi$$

The pitch line velocity can be calculated with the following:

$$v_t = R_p\omega_p$$

$$v_t = \frac{D_p}{2P_d}\omega_p$$

$$v_t = 27.1\ ft/s$$

The transmitted load can then be calculated with the following:

$$W_t = \frac{P}{v_t}$$

$$W_t = 243\ lb$$

Referring to table 9-2 the recommended quality numbers can be estimated. Based off the values given:

$$Q = 8$$

From figure 9-21, using the already determined quality number as well as the pitch line speed the recommended value for the dynamic factor for bending strength is given by:

$$K_v = 1.33$$

In order to find the bending geometry factors an angle for the gear teeth must be chosen. 20 degrees is the standard value to use in this type of application. With that choice being made the geometry factors can be found as:

$$J_p = 0.3$$

$$J_G = 0.4$$

From the figure 9-23 the pitting factor can be determined as:

$$I = 0.106$$

In order to find the pinion proportion factor, the face-to-pinion diameter ratio can be found to be:

$$\frac{F}{Dp}$$

$$= 0.69$$

With this ratio known, referring to figure 9-18 the pinion proportion factor is found to be:

$$C_{pf} = 0.048$$

For this application it was given that this was a commercial gear unit. Based off that information the mesh alignment factor is found from figure 9-19 is:

$$C_{ma} = 0.147$$

The pinion proportion factor is then fond by the following equation:

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

$$K_m = 1.2$$

Referring to table 9-6, for a diametral pitch greater than 5 the size factor is given by:

$$K_s = 1.0$$

In this application the gears are solid gear blanks therefore from figure 9-20:

$$K_B = 1.0$$

Since no unusual conditions exist a service factor of 1.0 will be selected.

The reliability for this application can be safely chosen as R=0.99, therefore the reliability factor can be found from table 9-8 and is given as:

$$K_R = 1.0$$

The next step is to calculate the design life. It is recommended that gearing should be designed with a life between 8,000 and 15,000 hours. Since this is a commercial saw that will not be constantly running the lower limit of that factor and assign L=8,000 hours. With this the number of duty cycles for each gear can be found with:

For the pinion:

$$N_{cP} = (60)Ln_Pq$$

$$N_{cP} = 1.7 * 10^9 \; load\ cycles$$

For the gear

$$N_{cg} = (60)Ln_gq$$

$$N_{cg} = 3.5 * 10^8 \; load\ cycles$$

From figure 9-22 the bending stress factors for both the pinion and the gear can be determined to be:

$$Y_{Np} = 0.93$$

$$Y_{Np} = 0.96$$

From figure 9-24 the values for Z are as follows:

$$Z_{Np} = 0.89$$

$$Z_{Ng} = 0.92$$

There is now sufficient information to calculate the expected bending stress for both the gear and the pinion.

The pinion:

$$s_{tp} = \frac{W_t P_d}{F J_p} K_o K_s K_m K_B K_v$$

$$s_{tp} = 15.06\ kpsi$$

The gear:

$$s_{tG} = \frac{W_t P_d}{F J_G} K_o K_s K_m K_B K_v$$

$$s_{tG} = 11.68\ kpsi$$

Therefore the allowable bending stress can be calculated.

The pinion:

$$s_{atp} > \frac{K_R (SF)}{Y_{Np}} S_{tp}$$

$$s_{atp} > 16.19\ kpsi$$

For the gear:

$$s_{atG} > \frac{K_R(SF)}{Y_{NG}} S_{tG}$$

$$s_{atG} > 12.15\ kpsi$$

The contact stress number can be calculated by:

$$s_c = C_p \left[\frac{W_t K_o K_s K_m K_v}{F D_p I}\right]^{\frac{1}{2}}$$

$$s_c = 113.75\ kpsi$$

Therefore the pitting stress number can now be calculated.

For the pinion:

$$s_{acp} > s_c \frac{K_R(SF)}{Z_{Np} C_H}$$

$$s_{acp} > 127.81\ kpsi$$

For the gear:

$$s_{acG} > s_c \frac{K_R(SF)}{Z_{NG} C_H}$$

$$s_{acG} > 123.78\ kpsi$$

It has been specified that the gears are to be made from SAE 4340 steel, oil quenched and tempered. The required hardness must still be found and can be done so with the AGMA 2001-

C95 charts. Using the contact stress and comparing it to the required Brinell hardness number the result is:

Pinion = 300 HB

Gear = 290 HB

Looking through the steel tables, with all the information known it is recommended to use the following material:

Pinion = AISI 4340 OQT 1100, with a hardness of 321 HB and 19% elongation.

Gear = AISI 4340 OQT 1200, with a hardness of 293 HB and 21% elongation.

Quenching and tempering are process that strengthen and harden materials. Oil quenching involves heating the material and then rapidly cooling it in oil. After this process tempering is achieved by heating the material below the critical point and then allowing it to cool in air.

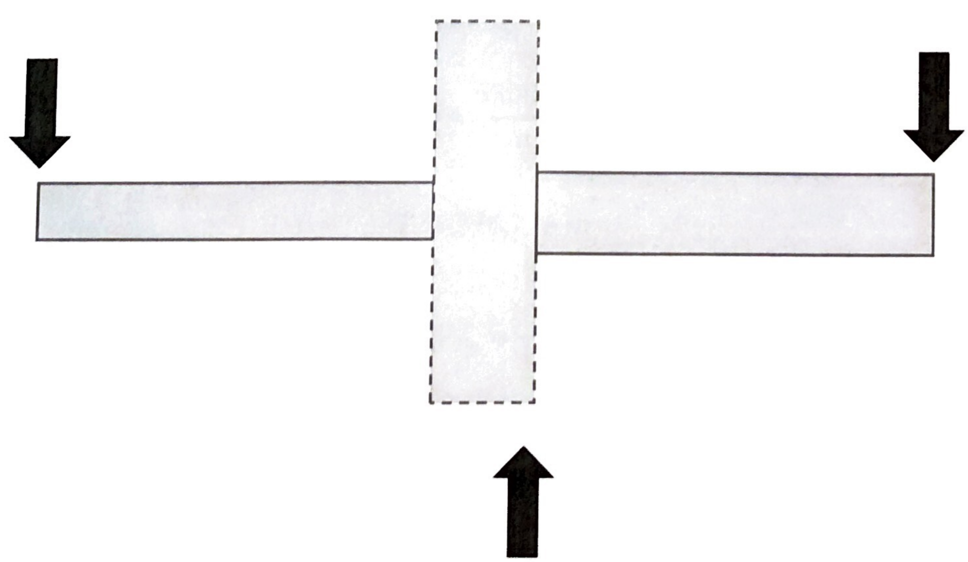
With the gears designed the design of the shafts may now be designed.

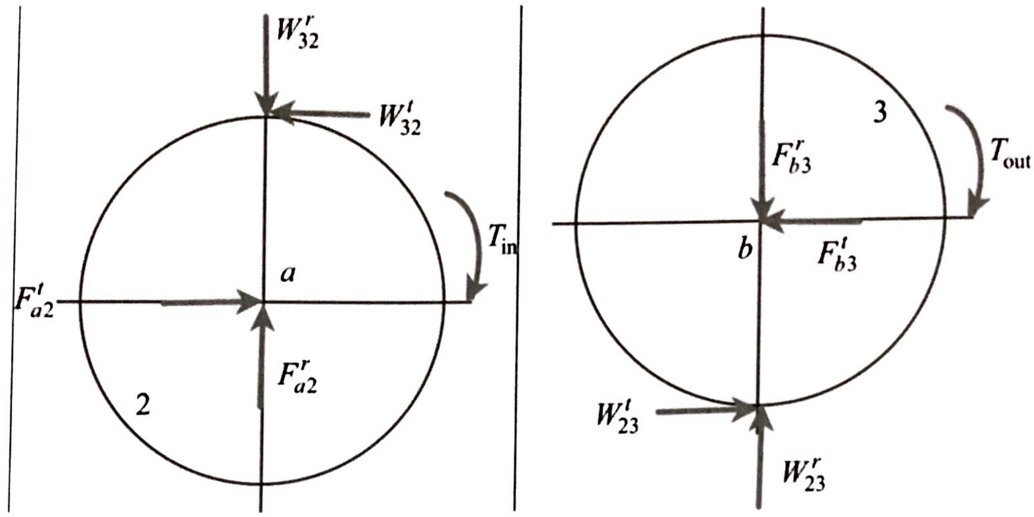
Design of the Shafts

Before the beginning of the shaft design a couple of decisions must be made. There will be two parallel shafts that will be made from the same material to cut down costs. Both shafts will utilize a key in order to keep the gear and the shaft connected. This will result in a stress concentration that will need to be considered from the beginning of the design. The length of each shaft will be equal and initially designed to be 10 inches. Both the gear and the pinion will be set to the middle of their respective shaft. Each shaft will have two bearings on both sides of the shaft of equal distances away. In addition to the key both a spacer and an external collar will be used to limit any axial movement. Since both these elements are external and will be designed to be lightweight; there effect on stress concentration as well as for normal and tangential forces on the shaft as they will be assumed negligible. Shaft A will refer to the shaft with the pinion and power input while shaft B will refer to the shaft with the gear and the resulting power output.

Shaft A:

For the FBD let 2=pinion and 3=gear.





All the forces and Torques must be found. This will also result in the reaction forces that will act on the bearings. The maximum bending stress will also be calculated and with it be able to calculate the required diameter for the application.

$$T = \frac{63000P}{n}$$

$$T = 328.7\ lb\ in$$

$$W_{Tp} = \frac{T}{\frac{D_p}{2}}$$

$$W_{Tp} = 365.2\ lbf$$

$$W_{rp} = W_{Tp} tan\phi$$

$$W_{rp} = 132.92\ lbf$$

$$F_{atp} = W_{Tp} = 365.2\ lbf \rightarrow$$

$$F_{arp} = W_{rp} = 132.92\ lbf \uparrow$$

With the forces acting on the shaft now realized and the distance of the gear on the shaft known, calculate the maximum bending moment on the shaft.

$$M_{ax} = F_{atp} * 5 - T$$

$$M_{ax} = 1826\ lbf\ in$$

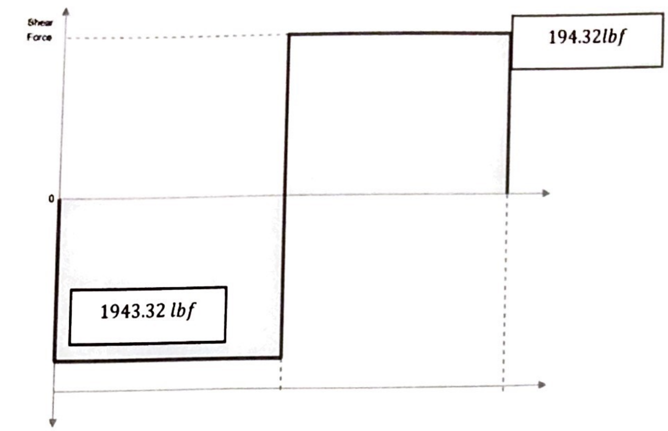
$$M_{ay} = F_{arp} * 5 - T$$

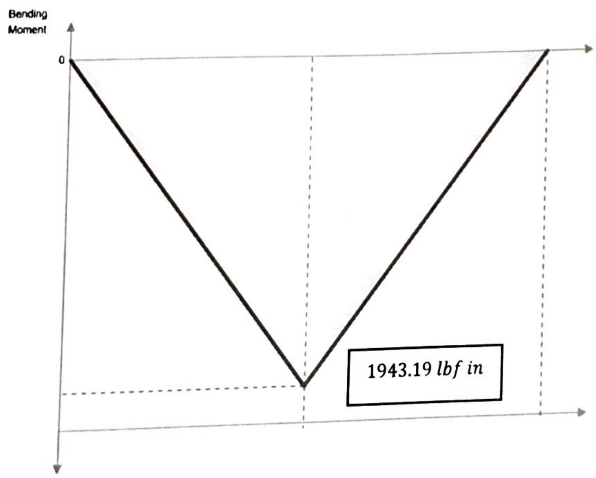
$$M_{ay} = 664.6$$

$$M = \left[(M_{ax})^2 + \left(M_{ay}\right)^2\right]^{\frac{1}{2}}$$

$$M = 1943.19\ lbf\ in$$

The shear moment diagram and the bending moment diagram are provided below. Because of the symmetry both diagrams for both respective shafts are equal and opposite. The nominal values used are the resultant forces as well as the maximum bending moment. The free body diagrams on the gears and the shafts are provided as well.





The conditions of this application call for a steel that has a moderately high strength, good fatigue resistance, good ductility, as well as machinability. These properties can be found from medium-carbon-alloy steel that has been cold-drawn or oil-quenched and tempered. Because the gears that have already been designed went through an oil-quenched and tempered process the same will apply to the shafts. For the design of the shaft AISI 1144 OQT 1000 steel with 19% elongation will be used. This material should be the right blend of strength and ductility in order to adequately perform the task. This type of steel is sometimes referred to as a *re-sulfurized, free- machining* grade of steel. The high Sulfur content makes this grade of steel machinable.

The chosen material has the following values:

$$s_y = 83000\ psi$$

$$s_u = 118000\ psi$$

$$s_n = 42000\ psi$$

Both a size factor and a reliability factor must be specified. Although the diameter of the shaft has not yet be determined, a diameter of 2 inches will be chosen attempting to slightly over-design. This diameter has a resulting size factor, from figure 5-9, of:

$$C_s = 0.8$$

Like the previous gear design, select a reliability of 0.99 which will result in the following:

$$C_R = 0.81$$

The estimated actual endurance strength can now be calculated with:

$$s'_n = s_n C_s C_R$$

$$s'_n = 27.216\ kpsi$$

Like the previous gear design since this application does not result in any unusual impact or shock the design factor N will be:

$$N = 2$$

With the bending moment known the diameter of the shaft can now be calculated. The last piece of information that needs to be taken into consideration is the correction factor due to the stress concentration caused by the keyhole. For a profile key, which is what will be used in this application, the factor is:

$$K_t = 2.0$$

The diameter can be calculated with the following equation:

$$D_a = \left[\left(\frac{32N}{\pi}\right)\sqrt{\left(\frac{K_t M}{s'_n}\right)^2 + \frac{3}{4}\left(\frac{T}{s_y}\right)^2}\right]^{\frac{1}{3}}$$

$$D_a = 0.79\ in$$

This Diameter is the minimum diameter that can carry the load as well as withstand the stress concentrations of the key. The same process will apply for shaft B.

Bearing selection

Spur gears produce only radial loads which simplify selection of the bearings that support the shafts. Selecting commercially available, single-row, deep-groove ball bearings for this application. The design load is equal to the radial load making theses types of bearings adequate for this design. Because of the simplicity, and symmetry, of the design the radial loads on the shaft are produced only at the sight of the gears and the bearings. This means that the four bearings in this design will experience the same forces and thus will all be the same.

First the load on the bearings must be calculated:

$$F_R = [(F_t)^2 + (F_r)^2]^{\frac{1}{2}}$$

$$F_R = 388.64$$

Because of the symmetry in this design the Reaction forces at each bearing in the radial direction and thus the load subjected on each bearing can be calculated by:

$$P = \frac{F_R}{2}$$

$$P = 194.32\ lbf$$

The design life of the bearings must now be calculated. This is the total number of revolutions expected in service. When designing the gears an estimation of 8000 hours was used and that will be continued here.

$$L_d = L * n * 60$$

Shaft A:

$$L_d = 1.65x10^9 rev$$

Shaft B:

$$L_d = 3.5x10^8 rev$$

The required basic load rating C can be calculated for the bearings relative to their shaft. Note that since single-row deep-groove ball bearings are being used a value of k=3 will be used.

Shaft A:

$$C = P_d \left(\frac{L_d}{10^6}\right)^{\frac{1}{k}}$$

$$C = 2296.2\ lbf$$

Shaft B:

$$C = P_d \left(\frac{L_d}{10^6}\right)^{\frac{1}{k}}$$

$$C = 1369.43\ lbf$$

A bearing must be selected with a dynamic load rating at least as high as the values computed. This means that shaft A is the limiting factor and that value will be used in order to make the decision. Consulting both tables 14-3 and 14-2 and comparing the dynamic loading as well as the minimum diameter of each bearing a selection can be made. For this design the 6205 bearing will be used. It has the following dimensions:

d = 0.9843 in

D = 2.0472 in

B = 0.05906 in

Preferred shoulder shaft d = 1.172 in

Min shoulder S = 1.14 in

Max shoulder H = 1.81 in

Weight = 0.29 lb

C = 2430 lbf

For the scope of this design the housing of the bearings will be ignored. The limits of the filet radius at the shoulder is 0.039 – 0.035. It is important to note that for this use the tolerances associated with these bearings are subject to the ABEC 1 tolerances. The limits of fit for the shaft to bearing is 0.0001 t0 0.0008. Because of this the shaft will be outfitted with a 0.0001 tolerance at all bearing coupling sites. Also, lubrication is something that needs to be addressed. Ball bearings need some sort of oil or grease in order to remain lubricated. It is suggested for this application, based of regular temperatures and the rpms that the elements will be exposed to, to use a Group III medium-temperature grease. With the Bearings selected the shaft design will briefly be revisited. In order to fit the bearings that have been chosen, both shaft A and shaft B will have a new diameter of:

$$Shaft\ diameter = 0.9840\ in$$

The shoulder will have a diameter:

$$Shaft\ shoulder\ diameter = 1.25\ in$$

This will add a new stress concentration factor of:

$$K_t = 3.0$$

Redoing the minimum diameter of the shaft calculation with the additional stress concentration factor still yields a minimum shaft diameter that falls under the new diameter size of 0.98 in. The final piece of the design will be the two keys.

Design of the Keys

With the dimensions of the shaft known the key and key-seat can be designed. This application will see the use of two square keys. The only calculation needed is the length of the key and key seat. It is recommended to use AISI 1020 cold drawn steel for most key selections. This is a low-carbon, cold-drawn steel that is both strong and has good ductility, 15% elongation. This material has the following properties:

$$s_y = 51\ kspi$$

From table 11-1, shafts that have a diameter between 0.875 and 1.25 inches should use a key of:

$$W = \frac{1}{4} in$$

$$H = \frac{1}{4} in$$

For this application a factor of safety N=3 will be used. To calculate the required length of the square keys with the following formula:

$$L = \frac{4TN}{DW s_y}$$

Shaft A:

$$L = 0.13\ inches$$

Shaft B:

$$L = 0.62\ inches$$

Since both shafts will be made identical, the length calculated using the Torque on shaft B is the controlling factor. This length is well below the width of the hub on the gear. Because of this, the length of the key and key seat equal to:

$$L = 0.75\ inches$$

The 2 keys will be:

0.75 X 0.25 X0.25 inches of AISI 1020 cold drawn steel 15% elongation.

The recommended tolerances and fits for the keys and key seats for a Width less than 0.5 is:

Tolerance on key = -0.002

Tolerance on keyseat+ +0.002

Total fit range = -0.000-0.004

With the keys designed the final thing to do is recheck the shaft for any additional stress concentration factors that have been added along the design process. After quick calculations everything falls within reasonable distance away from maximum allowable stresses. This will conclude the design of all the components.

CAD and Other Considerations

All CAD drawings were conducted by Christopher Riso using Creo 4.0 student version. The 3-D parts images provided are screen grabs from the Creo software. The .prt files for each part are available upon request. The final assembly is taken from a side 3-D view that shows all features accept for the keys which are hidden from that view. The following notes pertain to each of the drawings and annotations.

NOTES:

All dimensions provided are in inches.

Unless otherwise specified in the report all tolerances follow the same guidelines as provided here:

X.X = +/- 0.050

X.XX = +/- 0.010

X.XXX = +/- 0.005

X.XXXX = +/- 0.0008

All filets radii, unless otherwise specified within the report, are equal to:

Radii = 0.050

The key holes were designed and can be observed on both the shaft and the gear. Only the main features have been annotated with their corresponding dimensions. All other dimensions can be found within the report or can be made available upon request. Since the shafts and bearings are of identical design and dimensions the images provided, in order are:

Pinion

Gear

Shaft

Bearing

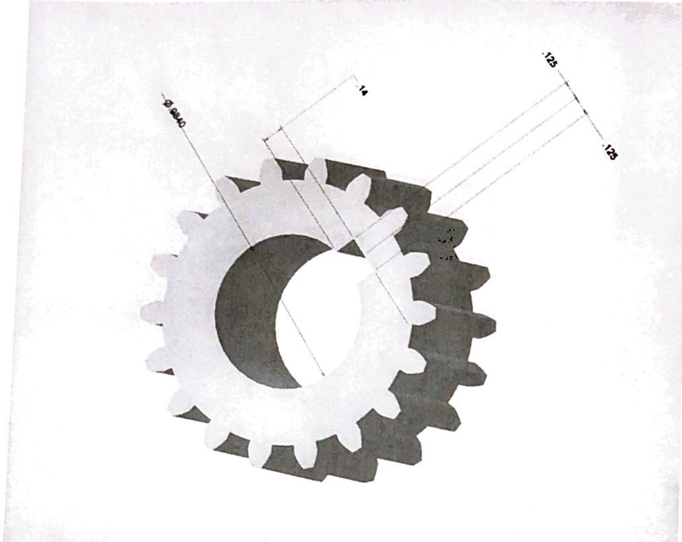
Shaft A assembled

Shaft B assembled

Final Assembly

Note:

The bowed external 5101 ring collar is not depicted in 3-D drawings. This will be applied to the shaft before the bearing is attached to the shaft. The shaft extends out past the bearing on opposite side of each shaft. This is where the coupling to the power supply as well as output would be. For the scope of this design those coupling features as well as the housing have been neglected.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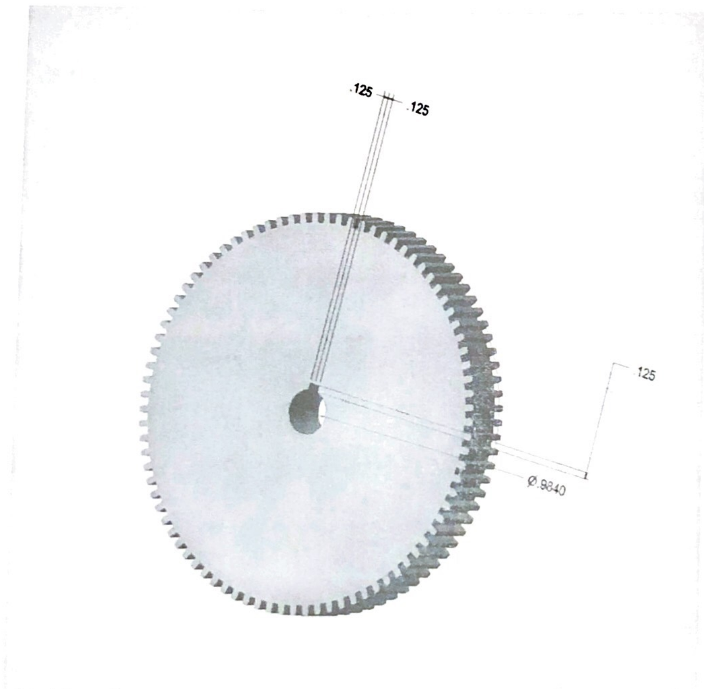
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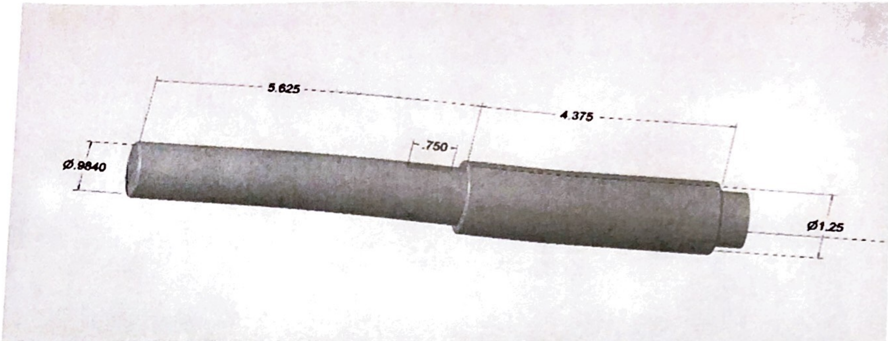
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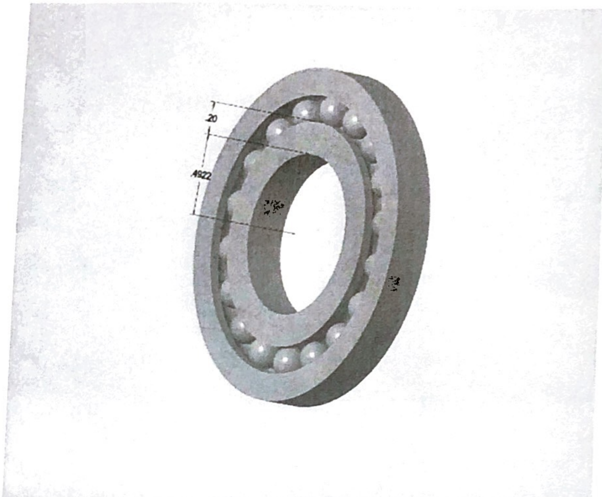

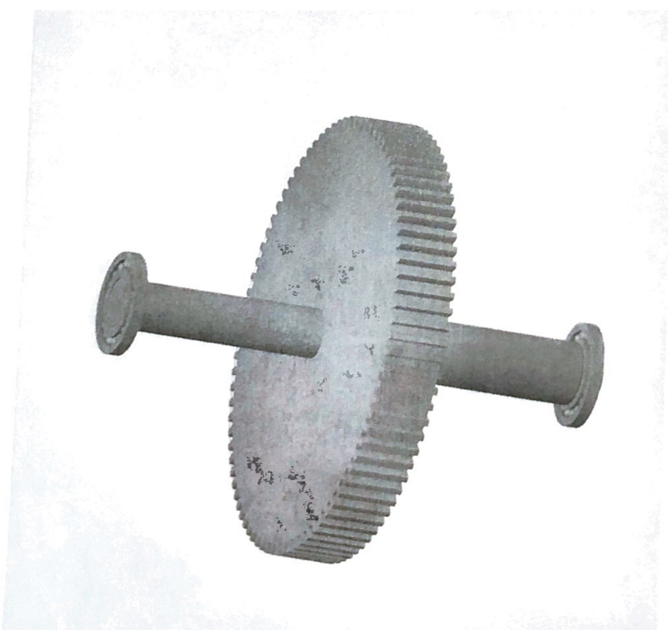

Conclusion

This report details the comprehensive design of a two gear single speed-reducer for a commercial band saw. With the values given, a simple two gear parallel shaft system can adequately perform the task at hand. Four bearings also went through a design process. Axial orientating devices were also designed and use of a key in each shaft and gear bore as well as utilized a shaft shoulder and a collar ring. These design decisions were critically analyzed to determine that the additions of those stress concentrations were still within the reasonable range and would not result in failure from the shaft. After finishing the design a critique would be that the shaft lengths could have been shortened by a large percent to decrease the total space taken up in the gear box. The value was arbitrarily chosen and although the device still will function well it could have been designed to be more compact. After completing the design all drawings were done utilizing Creo parametric 4.0. Throughout the report there are numerous notes for why each material is chosen, as well as, tolerance considerations, lubrication, and other stress factors. This was a simple design but still required every detail to be critically analyzed and considered.