RMIT University Vietnam
School of Science, Engineering & Technology
MIET2510 – Mechanical Design

Practical Assessment 2 – Mechanical Engineering Design and Analysis

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I. Introduction

The idea of this Practical Assignment 2 - Mechanical Engineering Design and Analysis is working on an industrial engineering project as an engineer in a company. The task involves designing horizontal axis wind turbines for an offshore wind farm that is around 20 kilometers from the coast. The wind turbine works as the mechanism. Its operation is turning the kinetic energy in wind into mechanical power which is then utilized to generate electricity by turning a generator. Nowadays, the wind turbines come in a variety of sizes and types, but they all have the same basic components.

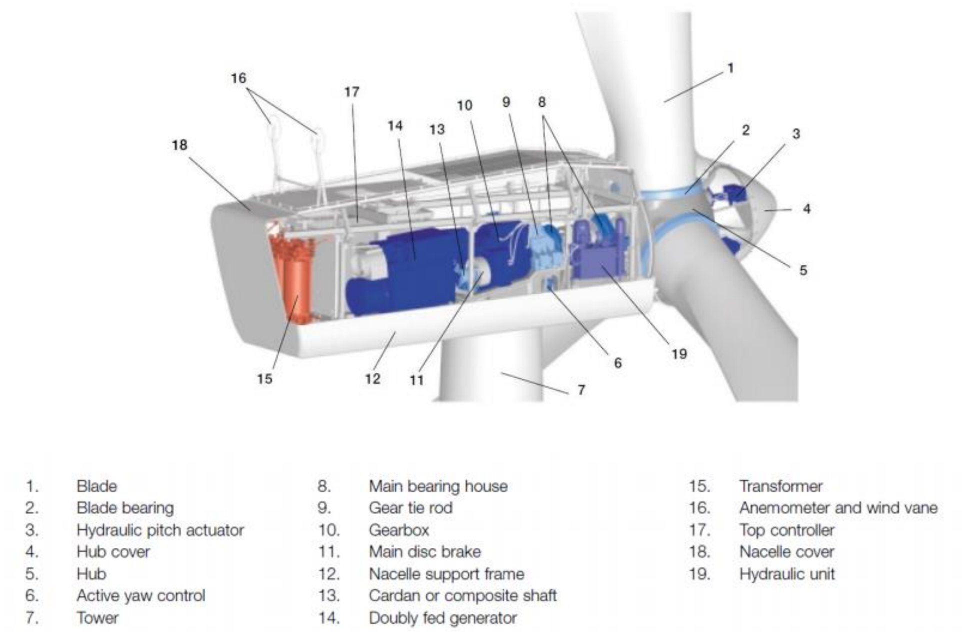

Figure 1: Illustration of wind turbine components

A gearbox is used in a wind turbine to enhance the rotational speed between the main shaft, which is connected to the hub by a bearing, and the high-speed shaft, which is connected to the generator. The gearbox and the high-speed shaft are also connected by a brake which is used for stopping the generation of power when it is necessary or for emergency situations. For the system which I'm going to design, the wind turbine will be the simplified version. The fundamental design parameters of the wind turbine include of bearing, main shaft, high-speed shaft, gear in the gearbox, brake, and key.

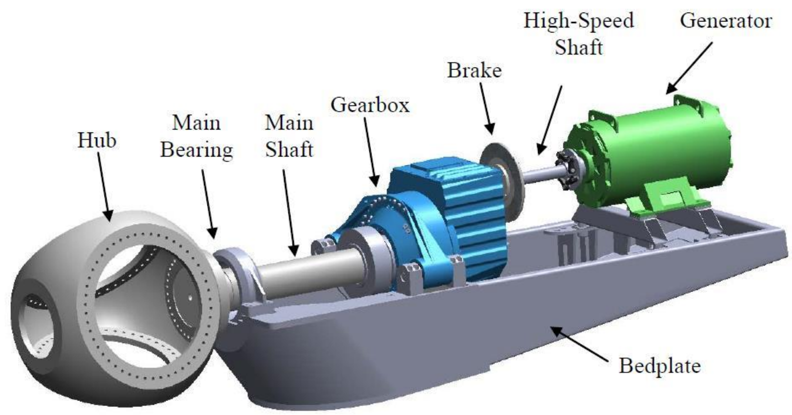

Figure 2: A simplified wind turbine.

II. Problem Statements

There are some primary design specifications for the wind turbine. Firstly, weight and size are optimized to be as small as possible, and material is selected based on operational conditions. For the shaft, the length of main shaft is about 0.15 m and high-speed shaft is 0.1 m and they should be in line. The axial load (thrust) on the bearing from a drag to lift ratio of the blades equal to 0.2. The center of lift of the blades is 10m from the centerline of the hub, bearing selection and shaft key must meet operational conditions. For the gearbox, transmission ratio, gear type and arrangement must meet operational conditions. The operational conditions of the design are the blades will rotate at 60 RPM while the generator will spin at 750 RPM to create 3 MW (Maximum) of useable electricity at maximum operational wind speed. The powertrain, which includes the generator, has an overall efficiency of 80%. Furthermore, the system is planned to run continuously for 8 years, reliability is 99%, and safety factor is 2 for all parts. The other variables and coefficients will be chosen based on our assumption.

III. Methodology

1. Shaft

In term of shaft design, there are two types of shafts to consider: the main shaft and the high-speed shaft. For the main shaft, it is influenced by the wind speed from the blades which will rotate 60 RPM at maximum operating. Similarly, the high-speed shaft is affected by the speed from the generator turns at 750 rpm to provide 3 MW of power at maximum operating. Furthermore, there is a shaft which we need to design. It is the shaft in the gearbox which is used to connect gear 2 and 3.

The three shafts are influenced by different conditions, so the method used to compute the diameter of each shaft will differ. However, the general method is that. Initially, the material will be chosen based on the function of these shaft. Then, we will determine the force, bending moment, torque which apply on the shaft. The minimum diameter of the shaft is found based on Distortion Energy Theory (DET) and Maximum Shear Stress Theory (MSST).

2. Key

In this system, torque must be transferred between shafts and gears, hubs. The key helps in this process by preventing slippage between the two sections. In term of key design in wind turbine, square keys will be selected to link the main shaft with the gearbox and the high-speed shaft with the gearbox.

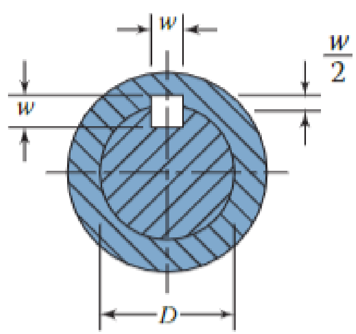

Figure 3: An example of square key in the shaft. [1]

In order to design the key, we need to determine material of the key then find stress in key. And it can be found by the equation of torque $T = Fr\ (N.m)$.

Where:

F is a tangential force located at the shaft surface (N).

r is the radius of the shaft (m).

Then calculating the required length of the square key with the width $w \approx \frac{D}{4}$ [1] by:

$$L = \frac{2Fn}{S_y w} \text{ [1]}$$

Where:

F is the force at the shaft surface (N).

S_y is the yield strength (Pa).

n is the safety factor.

w is the width of square key (m).

3. Gearbox

As the main shaft and high-speed shaft should be in line, the reverted compound gear train is chosen.

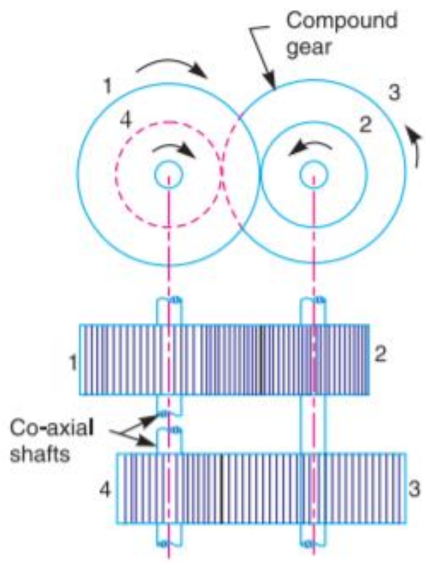

Figure 4: The reverted compound gear train. [2]

Then we got equation of reverted compound gear train:

$$\frac{\omega_{in}}{\omega_{out}} = \frac{N_2 N_4}{N_1 N_3} \text{ [2]}$$

From this equation we can calculate the transmission ratio by angular velocity. Then determining number of teeth of each shaft. Next, we find parameters which are used for gear selection from catalogue: material, module, pitch diameter, bore diameter, allowable torque, pressure angle, etc.

4. Bearing

In the main shaft of a wind turbine, roller bearings are usually used. Since it has a higher capability for combining loads and withstand significantly greater static and dynamic (shock) loads [11]. Therefore, cylindrical roller bearing is selected for the main shaft. According to the provided operational conditions are 8 years lifespan for the system as well as the bearing, 99% of reliability, and safety factor of 2.

Hence, we got equation to calculate the catalog load rating is:

$$C_{10} \approx a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{\frac{1}{b}}} \right]^{\frac{1}{a}} \quad [3]$$

Where:

Design life $x_D = \frac{L_D}{L_R} = \frac{\mathcal{L}_D n_D 60}{L_{10}}$. F_D is the design load (N). a_f is the safety factor. a = 10/3 for roller bearing. R_D is the reliability.	According to Bearing Design – Note 1, assuming the Weibull parameters: $x_0 = 0.02,$ $(\theta - x_0) = 4.439,$ $and\ b = 1.483.$

5. Brake and Clutch

For the brake design, electrical and mechanical brakes are the two most common forms of wind turbine brakes. However, in bigger wind turbines, electrical wind turbine brakes are rarely used. Hence, in this case we decide to design Disk Clutches and Brakes for the brake system of the wind turbine. This type of brake uses frictional contact between two or more surfaces to couple the input and output sides together to slow down or stop the system.

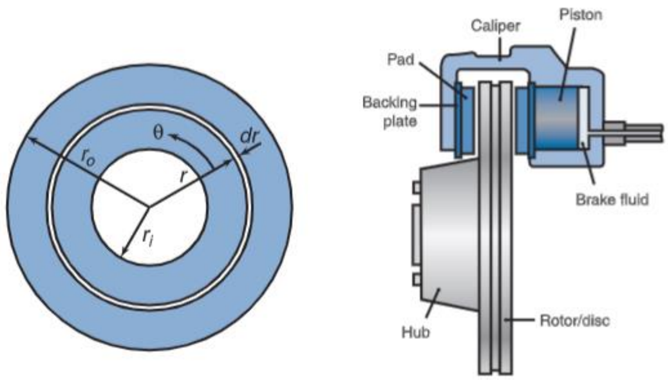

Figure 5: An example of disk clutch. [4]

The brake will be placed at the high-speed shaft, and we need to determine the material of the brake. Then, using disk clutch equation to find the inside and outside radius, the maximum normal force that can be applied to the clutch.

IV. Conceptual Design

For the material of the shafts and gears of the wind turbine, we select EN8 and 080M40 steel. It is a medium carbon engineering steel with good mechanical qualities that is widely used in the construction of mechanical parts including general axles and shafts, gears, bolts, and stud bolts. This material has BS970 standard and the yield strength S_y of 280 MPa, ultimate tensile strength S_u of 550 MPa. In addition, we should select a stronger material for the key, which will prevent it from breaking during system operation. EN8 and 080M40 steel with hardened and tempered + turned or ground condition is selected. It has yield strength S_y of 465 MPa, ultimate tensile strength S_u of 700-850 MPa. [5]

For the wind turbine has power of 3MW, the mass of blade assembly is about 12474 kg [6].
Then the weight of it is $W = m \times g = 12474 \times 9.81 = 122.37$ kN

Since the weight of blade assembly is 122.37 kN and weight is a force which opposes the upward force of lift, so lift equals to -122.37 kN. For the drag to lift ratio of the blades equal to 0.2 $\left({}^{D}/_{L} = 0.2\right)$, then drag equals to $-122.37 \times 0.2 = -24.474$ kN. Moreover, drag is a force which opposes the force of thrust or axial load, hence thrust or axial load equals to 24.474 kN. [7]

From the Mean Wind Speed Site A of Wind Data, eliminating wind speeds above 9.16m/s (storm conditions) and below 5.43m/s (calm). Then Bin Size Interval equal to Wind Speed Range divide Number of frequencies in the range: Bin Size Interval $= (9.16 - 5.43)/10 = 0.373$m/s.

Hence, wind speed at maximum frequency $= 5.43 + (0.373 \times 6) = 7.668$m/s.

This is 0.8 (7.688/9.16) of the maximum wind velocity and with the generator turns at 750 RPM to provide 3 MW (Maximum), therefore rated power (torque) output is $0.8 \times 3 \times 10^6\text{W} = 2.4 \times 10^6\text{W}$. In addition, the powertrain, including the generator, has an overall efficiency of 80%.

Then we got equation to find angular velocity in rad/s and torque of high-speed shaft or output:

$$\omega = \frac{2\pi}{60} \times N = \frac{2\pi}{60} \times 750 = 78.54 \left(\frac{rad}{s}\right); T_{hs} = \frac{H}{\omega} = \frac{2.4 \times 10^6 \times 80\%}{78.54} = 24.45\ kN.m.$$

With gear ratio equals to:

$$\frac{\omega_{in}}{\omega_{out}} = \frac{60}{750} = \frac{1}{12.5} = \frac{T_{out}}{T_{in}} = \frac{T_{hs}}{T_m}. [8]$$

Hence, torque of the main shaft or input is:

$$\Rightarrow T_m = T_{hs} \times 12.5 = 24.45\ kN.m \times 12.5 = 305.63 kN.m.$$

1. Shaft

1.1. Main shaft

Assuming the length of the main shaft is 1.5 m. The main shaft is applied by the radial and axial loading that are 122.37 kN and 24.474 kN respectively as well as torque T_m is 305.63kN.m. Moreover, assuming the mass of the gear 1 of the gearbox is about 50 kg. Comparing with the mass of the blade assembly is about 12474 kg, as we can see load from gear 1 won't affect much to the shaft. Hence, any load produced by the gearbox can be ignored. The free body diagram is:

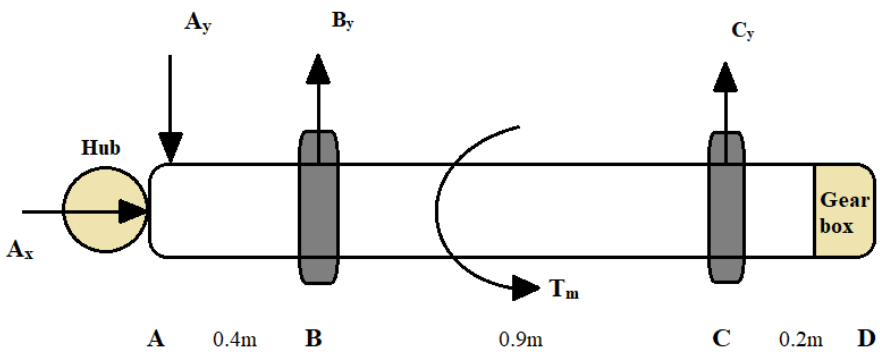

Figure 6: Free body diagram of main shaft.

<u>Calculation:</u>

From the FBD above, A_x and A_y are axial load and radial load respectively. B_y and C_y are the reaction forces of bearing at B and C. Applying equilibrium condition, we got:

$$(\uparrow)\sum F_y = 0 = -A_y + B_y + C_y \Leftrightarrow B_y + C_y = A_y = 122.37 \text{ kN.}$$

Moment at point C:

$$(\circlearrowleft)\sum M_C = 0 = 0 - B_y \times 0.9 + A_y \times 1.3 \Rightarrow B_y = \frac{122.37 \times 1.3}{0.9} = 176.76 \text{ kN.}$$

$$\Rightarrow C_y = 122.37 - 176.76 = -54.39 \text{ kN} = 54.39 \text{ kN}(\downarrow).$$

Then we got the shear force and bending moment diagram:

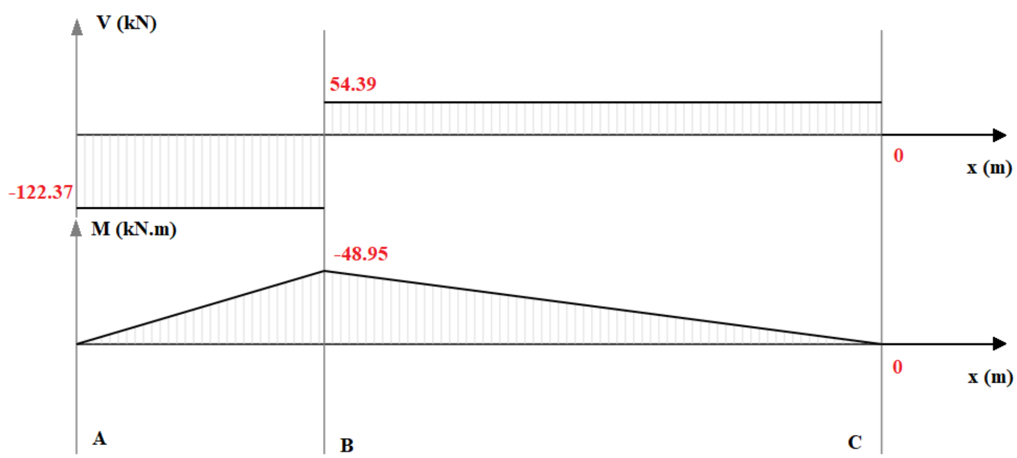

Figure 7: The shear force and bending moment diagram of main shaft.

From the figure above, we can see that the maximum bending moment is equal to 48.95 kN.m. From the information above, the DET and MSST equation to calculate smallest safe diameter are

[9]:

DET	MSST
$\frac{4}{\pi d^3}\sqrt{(8M+Pd)^2+48T^2}=\frac{S_y}{n_s}$	$\frac{4}{\pi d^3}\sqrt{(8M+Pd)^2+64T^2}=\frac{S_y}{n_s}$
Where: M = 48.95 kN.m, P = 24.474 kN, T = 305.63 kN.m, n_s = 2, S_y = 280 MPa = 280*10^6 Pa	
$\frac{4}{\pi d^3}\sqrt{(8\times 48950+24474\times d)^2+48\times \mathbf{305630}^2}=\frac{280\times 10^6}{2}$ $\Rightarrow d=0.\mathbf{2696}\ m=\mathbf{26.96}\ cm$	$\frac{4}{\pi d^3}\sqrt{(8\times 48950+24474\times d)^2+64\times \mathbf{305630}^2}=\frac{280\times 10^6}{2}$ $\Rightarrow d=0.\mathbf{282}\ m=28.2\ cm$

Therefore, the smallest safe diameter is 0.2696m or 26.96cm (DET method). Then the selection for the main shaft's diameter is 0.30m or 30cm.

1.2. High-speed shaft

Assuming the length of the high-speed shaft is 1 m. It is applied by a torque T_{hs} is 24.45 kN.m and a radial load from gear 4 of gearbox. Suppose that gear 4 has a mass of 20 kg ($W = m \times g = 20 \times 9.81 = 196.2$ N. The free body diagram is:

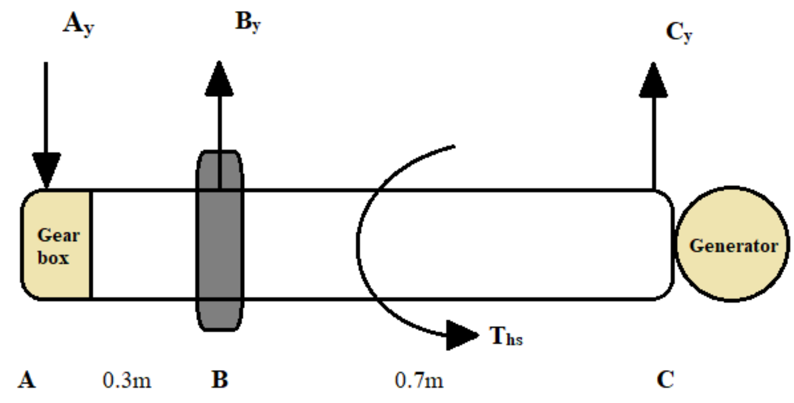

Figure 8: Free body diagram of high-speed shaft.

Calculation:

From the FBD above, A_y is radial load and B_y and C_y is the reaction force of bearing at B and C.

Applying equilibrium condition, we got:

$$(\uparrow)\sum F_y = 0 = -A_y + B_y + C_y \Leftrightarrow B_y + C_y = A_y = 196.2\text{ N}.$$

Moment at point C:

$$(\circlearrowleft)\sum M_C = 0 = 0 - B_y \times 0.7 + A_y \times 1 \Rightarrow B_y = \frac{196.2 \times 1}{0.7} = 280.3\text{ N}.$$

$$\Rightarrow C_y = 196.2 - 280.3 = -84.1\text{ N} = 84.1\text{ N}(\downarrow).$$

Then we got the shear force and bending moment diagram:

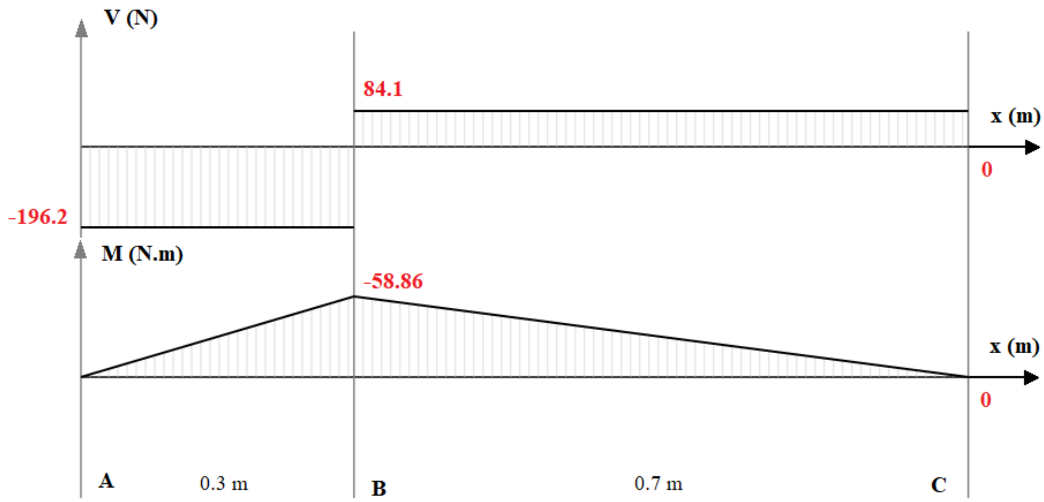

Figure 9: The shear force and bending moment diagram of high-speed shaft.

From the figure above, we can see that the maximum bending moment is equal to 58.68 N.m. From the information above, the DET and MSST equation to calculate smallest safe diameter are [9]:

DET	**MSST**
$d = \left(\frac{32n_s}{\pi S_y}\sqrt{M^2 + 0.75T^2}\right)^{\frac{1}{3}}$	$d = \left(\frac{32n_s}{\pi S_y}\sqrt{M^2 + T^2}\right)^{\frac{1}{3}}$
Where: M = 58.68 N.m, T = 24.45 kN.m, n_s = 2, S_y = 280 MPa = $280*10^6$ Pa	

$\Rightarrow d = \left(\frac{32 \times 2}{\pi \times 280 \times 10^6}\sqrt{58.68^2 + 0.75 \times 24450^2}\right)^{\frac{1}{3}}$ $\Rightarrow d = 0.115\ m = 11.5\ cm$	$\Rightarrow d = \left(\frac{32 \times 2}{\pi \times 280 \times 10^6}\sqrt{58.68^2 + 24450^2}\right)^{\frac{1}{3}}$ $\Rightarrow d = 0.121\ m = 12.1\ cm$

Therefore, the smallest safe diameter is 0.115m or 11.5cm (DET method). Then the selection for the main shaft's diameter is 0.14m or 14cm.

1.3. Shaft in the gearbox

Assuming the length of the high-speed shaft is 0.7 m. It is applied by a torque T, and it equals to

$$\frac{\omega_{in}}{\omega_{out}} = \frac{60}{750} = \frac{1}{12.5} = \frac{T_{out}}{T_{in}} = \frac{T}{T_m} \times \frac{T_{hs}}{T}.$$

$$\Rightarrow \frac{T}{T_m} = \sqrt{\frac{1}{12.5}} = \frac{T}{305.63\ kN.m} \Rightarrow T = 86.44\ kN.m.$$

Furthermore, a radial load from gear 2 and gear 3 of gearbox are also applied to the shaft. Suppose that gear 3 has a mass of 50 kg ($W = m \times g = 50 \times 9.81 = 490.5$ N) and gear 2 has a mass of 20 kg ($W = m \times g = 20 \times 9.81 = 196.2$ N). The free body diagram is:

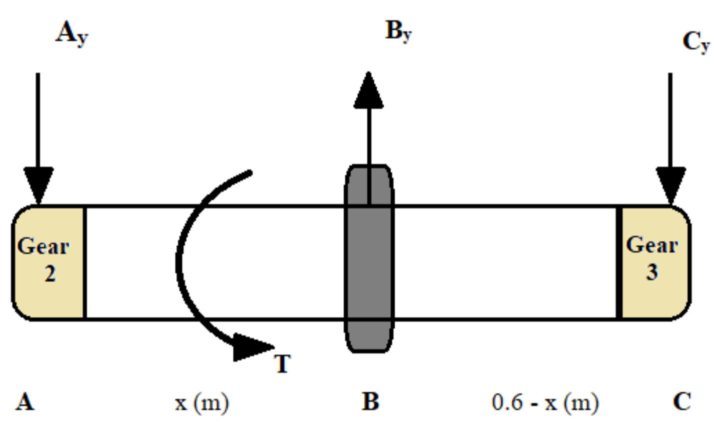

Figure 10: Free body diagram of shaft in the gearbox.

Calculation:

From the FBD above, A_y and C_y is radial load and B_y is the reaction force of bearing at B. Applying equilibrium condition, we got:

$$(\uparrow)\sum F_y = 0 = -A_y + B_y - C_y \Leftrightarrow B_y = A_y + C_y = 686.7\text{ N.}$$

Moment at point C:

$$(\circlearrowleft)\sum M_C = 0 = 0 - B_y \times (0.6 - x) + A_y \times 0.6 \Rightarrow B_y = \frac{196.2 \times 0.6}{0.6 - x} = 686.7\text{ N.}$$

$$\Rightarrow x = {}^{3}/_{7}\text{ m.}$$

Hence, the distance to maintain equilibrium from B to C is ${}^{6}/_{35}$ m and from A to B is ${}^{3}/_{7}$ m. Then we got the shear force and bending moment diagram:

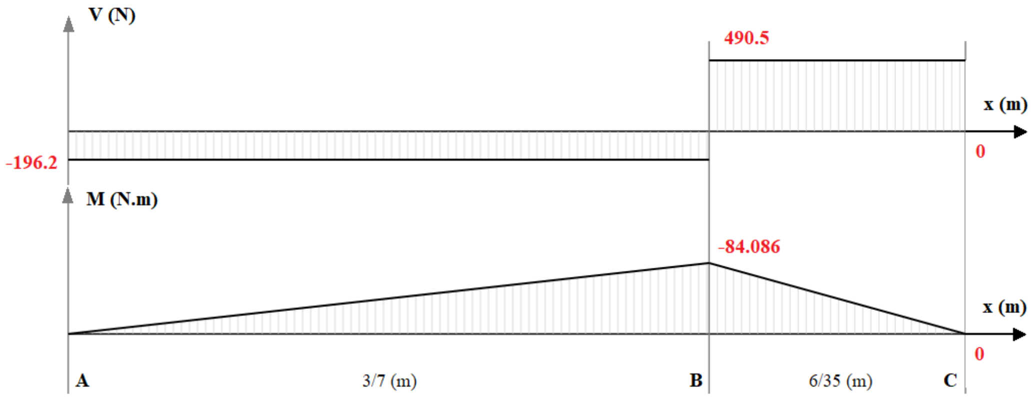

Figure 11: The shear force and bending moment diagram of high-speed shaft.

From the figure above, we can see that the maximum bending moment is equal to 84.086 N.m. From the information above, the DET and MSST equation to calculate smallest safe diameter are [9]:

DET	**MSST**
$d = \left(\frac{32n_s}{\pi S_y}\sqrt{M^2 + 0.75T^2}\right)^{\frac{1}{3}}$	$d = \left(\frac{32n_s}{\pi S_y}\sqrt{M^2 + T^2}\right)^{\frac{1}{3}}$
Where: M = 84.086 N.m, T = 86.44 kN.m, n_s = 2, S_y = 280 MPa = 280*10^6 Pa	
$\Rightarrow d = \left(\frac{32 \times 2}{\pi \times 280 \times 10^6}\sqrt{84.086^2 + 0.75 \times 86440^2}\right)^{\frac{1}{3}}$	$\Rightarrow d = \left(\frac{32 \times 2}{\pi \times 280 \times 10^6}\sqrt{84.086^2 + 86440^2}\right)^{\frac{1}{3}}$

$\Rightarrow d = 0.176\, m = 17.6\, cm$	$\Rightarrow d = 0.1846\, m = 18.46\, cm$

Therefore, the smallest safe diameter is 0.176m or 17.6cm (DET method). Then the selection for the main shaft's diameter is 0.20m or 20cm.

2. Key

Calculation:

- **Main shaft and gear 1**

With the diameter of the main shaft is 30cm, the width of the key is equal to $w \approx \frac{D}{4} \approx \frac{30}{4} \approx 7\, cm$ and r equal to 15 cm. Moreover, T = 305.63 kN.m, $n_s = 2$, S_y = 465MPa = $465*10^6$ Pa.
Then, the force F at the surface of the main shaft is

$$F = \frac{T}{r} = \frac{305630}{0.15} = 2037.5 \text{ kN}$$

The required length of the square key is:

$$L = \frac{2Fn}{S_y w} = \frac{2 \times 2037.5 \times 10^3 \times 2}{465 \times 10^6 \times 0.07} = 0.25m = 25cm.$$

- **High-speed shaft and gear 4**

With the diameter of the main shaft is 14cm, the width of the key is equal to $w \approx \frac{D}{4} \approx \frac{14}{4} \approx 3.5\, cm$ and r equal to 7 cm. Moreover, T = 24.45 kN.m, $n_s = 2$, S_y = 465MPa = $465*10^6$ Pa. The force F at the surface of the main shaft is

$$F = \frac{T}{r} = \frac{24450}{0.07} = 349.3 \text{ kN}$$

Thus, the required length of the square key is:

$$L = \frac{2Fn}{S_y w} = \frac{2 \times 349.3 \times 10^3 \times 2}{465 \times 10^6 \times 0.035} = 0.1m = 10cm.$$

- **Shaft in the gearbox and gear 2 & Shaft in the gearbox and gear 3**

With the diameter of the shaft is 19cm, the width of the key is equal to $w \approx \frac{D}{4} \approx \frac{20}{4} \approx 5\, cm$ and r equal to 10 cm. Moreover, T = 86.44 kN. m, $n_s = 2$, S_y = 465MPa = $465*10^6$ Pa. The force F at the surface of the main shaft is

$$F = \frac{T}{r} = \frac{86440}{0.1} = 864.4 \text{ kN}$$

Thus, the required length of the square key is:

$$L = \frac{2Fn}{S_y w} = \frac{2 \times 864.4 \times 10^3 \times 2}{465 \times 10^6 \times 0.05} = 0.15m = 15cm.$$

3. Gearbox

Calculation:

The reversed gear train's speed ratio is:

$$\frac{\omega_{in}}{\omega_{out}} = \frac{60}{750} = \frac{1}{12.5} = 0.08.$$

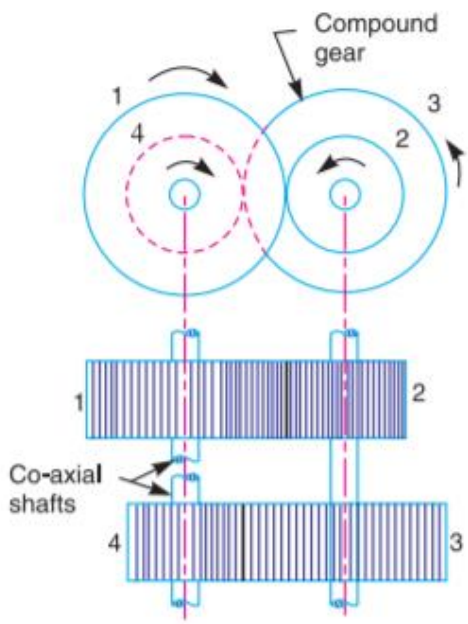

Figure 12: The inverted compound gear train. [2]

Since we are going to use inverted compound gear train for the gearbox, the speed ratio between the gears 1 and 2 as well as the gears 3 and 4 are to be the same. Moreover, any pair of meshing gears has a speed ratio that is inversely proportional to their tooth number, therefore:

$$\frac{\omega_1}{\omega_2} = \frac{\omega_3}{\omega_4} = \sqrt{\frac{1}{12.5}} = 0.283 = \frac{N_2}{N_1} = \frac{N_4}{N_3} (1)\ [2]$$

Since the center distance of the stages must be equal for this inverted compound gear train, we got:

$$r_1 + r_2 = r_3 + r_4;\ N_1 + N_2 = N_3 + N_4\ [2]$$

We have equation of circular pitch:

$$p_c = \frac{\pi d}{N} = \frac{2\pi r}{N} = \pi m \Rightarrow r = \frac{m.N}{2}\ [2]$$

Hence,

$$\frac{m_1.N_1}{2} + \frac{m_2.N_2}{2} = \frac{m_3.N_3}{2} + \frac{m_4.N_4}{2}$$

Assuming $m_1 = m_2 = m_3 = m_4 = 25$ mm or 0.025 m and the center distance is 1.4375 m or 1437.5 mm

Then we got:

$$\frac{m_1.N_1}{2} + \frac{m_2.N_2}{2} = \frac{m_3.N_3}{2} + \frac{m_4.N_4}{2} = 1.4375\ m$$

$$\Leftrightarrow 0.028(N_1 + N_2) = 0.025(N_3 + N_4) = 3$$

$$\Leftrightarrow (N_1 + N_2) = (N_3 + N_4) = \frac{2.875}{0.025} = 115$$

From (1) $\Rightarrow \begin{cases} N_2 = 0.283N_1 \\ N_4 = 0.283N_3 \end{cases}$ then $\Rightarrow \begin{cases} N_1 = {}^{115}/_{1.283} \\ N_3 = {}^{115}/_{1.283} \end{cases} = 89.6$ say 90.

$$\Rightarrow \begin{cases} N_2 = 115 - 90 = 25 \\ N_4 = 115 - 90 = 25 \end{cases}.$$

With the computed values of number of teeth on each gear, the reversed gear train's speed ratio is:

$$\frac{\omega_{in}}{\omega_{out}} = \frac{N_2 N_4}{N_1 N_3} = \frac{25 \times 25}{90 \times 90} = 0.0772.$$

The parameters of the gears are:

Parameters	Gear 1	Gear 2	Gear 3	Gear 4
Module m (m)	0.025	0.025	0.025	0.025
Number of teeth N	90	25	90	25
Pitch diameter or Reference diameter $d = m \times N$ (m)	2.25	0.625	2.25	0.625
Tip diameter $d_a = d + 2m$ (m)	2.3	0.675	2.3	0.675
Root diameter $d_f = d - 2.5m$ (m)	2.1875	0.5625	2.1875	0.5625

Pressure angle ϕ (degree)	20	20	20	20
Circular pitch p = $\pi d/_N$ (m)	0.078	0.078	0.078	0.078
Tooth depth h= 2.25m (m)	0.05625	0.05625	0.05625	0.05625
Addendum h_a = 1.00m (m)	0.025	0.025	0.025	0.025
Dedendum h_f = 1.25m (m)	0.03125	0.03125	0.03125	0.03125
Tooth thickness s = $\pi m/_2$ (m)	0.039	0.039	0.039	0.039
Center distance a = $(d_1 + d_2)/_2$ (m)	1.4375		1.4375	
Backlash c = 0.25m (m)	6.25×10^3	6.25×10^3	6.25×10^3	6.25×10^3

4. Bearing

- **Main shaft**

There is pair of bearings on the main shaft, the first bearing is near the hub, while the other is near the gearbox. With the reliability 99% for all parts, then the individual bearing reliability of them is $\sqrt{0.99} = 0.995$.

Calculation:

- Design load and catalog load rating of the first bearing:

$$F_D = \sqrt{176760^2 + 24474^2} = 178.45 \text{ kN}.$$

$$x_D = \frac{L_D}{L_R} = \frac{\mathcal{L}_D n_D 60}{L_{10}} = \frac{70080 \times 60 \times 60}{10^6} = 252.3$$

Where:

$\mathcal{L}_D$ is the life of bearing which is 8 years.

n_D is the speed of the shaft in RPM.

L_{10} is life in 10^6 revolutions.

Thus, the design life is 252.3 times the L_{10} life.

$$C_{10} \approx a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{\frac{1}{b}}} \right]^{\frac{1}{a}}$$

Where: $a_f = 2, F_D = 178.45$ kN, $x_0 = 0.02, (\theta - x_0) = 4.439, b = 1.483, a = 10/3, x_D =$

252.3 .

$$\Rightarrow C_{10} \approx 2 \times 178450 \left[\frac{252.3}{0.02 + (4.439)(1 - 0.995)^{\frac{1}{1.483}}}\right]^{\frac{1}{\frac{10}{3}}} = 3349.88\ kN$$

With the catalog load rating is calculated above and shaft's diameter is 300mm, the roller bearing with highlighted parameter in the figure below will be selected.

Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designations Bearing with standard cage	Alternative standard cage1)
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
300	460	74	858	1 370	129	1 500	2 000	46	NJ 1060 MA	–
	460	74	858	1 370	129	1 500	2 000	46	▸ NU 1060 MA	–
	460	95	1 510	2 600	245	1 300	2 000	62	NU 2060 ECMA	–
	540	85	1 420	2 120	183	1 300	1 400	89,5	▸ NU 260 M	–
	540	140	2 090	3 450	300	1 200	1 800	145	NU 2260 MA	–
	620	109	2 330	3 350	280	950	1 200	174	NU 360 ECM	–
	620	185	4 020	5 850	480	950	1 600	270	NU 2360 ECMA	–

Figure 13: Catalogue of 300mm inner diameter cylindrical roller bearing from SKF. [10]

- Design load and catalog load rating of the second bearing:

$$F_D = \sqrt{54390^2 + 24474^2} = 59.64 \text{ kN}.$$

$$C_{10} \approx a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{\frac{1}{b}}}\right]^{\frac{1}{a}}$$

Where: $a_f = 2, F_D = 59.65$ kN$, x_0 = 0.02, (\theta - x_0) = 4.439, b = 1.483, a = 10/3, x_D =$ 252.3 .

$$\Rightarrow C_{10} \approx 2 \times 59640 \left[\frac{252.3}{0.02 + (4.439)(1 - 0.995)^{\frac{1}{1.483}}}\right]^{\frac{1}{\frac{10}{3}}} = 1119.57\ kN$$

With the catalog load rating is calculated above, and shaft's diameter is 300mm, the roller bearing with highlighted parameter in the figure below is selected.

Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designations Bearing with standard cage	Alternative standard cage1)
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
300	460	74	858	1 370	129	1 500	2 000	46	NJ 1060 MA	–
	460	74	858	1 370	129	1 500	2 000	46	▸ NU 1060 MA	–
	460	95	1 510	2 600	245	1 300	2 000	62	NU 2060 ECMA	–
	540	85	1 420	2 120	183	1 300	1 400	89,5	▸ NU 260 M	–
	540	140	2 090	3 450	300	1 200	1 800	145	NU 2260 MA	–
	620	109	2 330	3 350	280	950	1 200	174	NU 360 ECM	–
	620	185	4 020	5 850	480	950	1 600	270	NU 2360 ECMA	–

Figure 14: Catalogue of 300mm inner diameter cylindrical roller bearing from SKF. [10]

- **High-speed shaft**

Calculation:

Catalog load rating of the bearing:

$$x_D = \frac{L_D}{L_R} = \frac{\mathcal{L}_D n_D 60}{L_{10}} = \frac{70080 \times 750 \times 60}{10^6} = 3153.6$$

$$C_{10} \approx a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{\frac{1}{b}}}\right]^{\frac{1}{a}}$$

Where: $a_f = 2, F_D = 280.3$ N, $x_0 = 0.02, (\theta - x_0) = 4.439, b = 1.483, a = 10/3, x_D = 3153.6$.

$$\Rightarrow C_{10} \approx 2 \times 280.3 \left[\frac{3153.6}{0.02 + (4.439)(1 - 0.99)^{\frac{1}{1.483}}}\right]^{\frac{1}{\frac{10}{3}}} = 11.23\ kN$$

With the catalog load rating is calculated above and shaft's diameter is 140mm, the roller bearing with highlighted parameter in the figure below will be selected.

Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designations Bearing with standard cage	Alternative standard cage[1]
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
140	210	33	179	255	28	3 600	5 300	4,05	► NU 1028 ML	M
	250	42	450	510	57	2 800	3 200	8,45	► NUP 228 ECJ	M, ML
	250	42	450	510	57	2 800	3 200	8,6	► NJ 228 ECJ	M, ML
	250	42	450	510	57	2 800	3 200	9,4	► NU 228 ECM	J, ML
	250	68	655	830	93	2 800	4 800	15	► NU 2228 ECML	PA
	250	68	655	830	93	2 800	4 800	15,5	► NJ 2228 ECML	PA
	250	68	655	830	93	2 800	4 800	15,5	NUP 2228 ECML	–
	300	62	780	830	88	2 400	2 800	20	► NJ 328 ECJ	M, ML
	300	62	780	830	88	2 400	2 800	22,5	► NU 328 ECM	J, ML

Figure 15: Catalogue of 140mm inner diameter cylindrical roller bearing from SKF. [10]

- **Gearbox shaft**

Calculation:

Catalog load rating of the bearing:

$$\frac{\omega_1}{\omega_2} = \frac{60}{\omega_2} = \sqrt{\frac{1}{12.5}} \Rightarrow n_D = \omega_2 = 212.13 \text{ RPM}$$

$$x_D = \frac{L_D}{L_R} = \frac{\mathcal{L}_D n_D 60}{L_{10}} = \frac{70080 \times 212.13 \times 60}{10^6} = 891.96$$

$$C_{10} \approx a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{\frac{1}{b}}}\right]^{\frac{1}{a}}$$

Where: $a_f = 2, F_D = 686.7$ N, $x_0 = 0.02, (\theta - x_0) = 4.439, b = 1.483, a = 10/3, x_D = 252.3$.

$$\Rightarrow C_{10} \approx 2 \times 686.7 \left[\frac{891.96}{0.02 + (4.439)(1 - 0.99)^{\frac{1}{1.483}}}\right]^{\frac{1}{\frac{10}{3}}} = 16.6\ kN$$

With the catalog load rating is calculated above and shaft's diameter is 200mm, the roller bearing with highlighted parameter in the figure below will be selected.

Principal dimensions			Basic load ratings		Fatigue load limit	Speed ratings		Mass	Designations	
			dynamic	static		Reference speed	Limiting speed		Bearing with standard cage	Alternative standard cage[1)]
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
200	310	51	380	570	58,5	2 400	3 600	14	▸ NU 1040 ML	M
	360	58	880	1 060	106	1 900	3 200	26,5	▸ NU 240 ECML	M
	360	58	880	1 060	106	1 900	3 200	27	▸ NJ 240 ECML	M
	360	98	1 370	1 800	180	1 900	3 200	44	NJ 2240 ECML	–
	360	98	1 370	1 800	180	1 900	3 200	44	▸ NU 2240 ECML	–
	420	80	1 230	1 630	150	1 400	2 800	56,5	NJ 340 ECML	–

Figure 16: Catalogue of 200mm inner diameter cylindrical roller bearing from SKF. [10]

5. Brake and Clutch

The disk brake will be made of sintered metal, and it rubs against sintered metal in dry condition. According to table 2 in Appendix, it has coefficient of friction $\mu = 0.35$, maximum contact pressure $p_{max} = 1800\ kPa$. With safety factor $n_s = 2$, maximum torque $T_w = 24.45\ kN.m$. [4] And we want the disk brake fit to the high-speed shaft, so the inside radius of 0.14m will be chosen.

Calculation:

We have equation to calculate outside radius [4]:

$$r_i({r_o}^2 - {r_i}^2) = \frac{n_s T_w}{\pi \mu p_{max}} = \frac{2 \times 24450}{\pi \times 0.35 \times 1800 \times 10^3} = 0.02$$

$$\Rightarrow r_o = \sqrt{\frac{0.02}{r_i} + {r_i}^2} = \sqrt{\frac{0.02}{0.14} + 0.14^2} = 0.4m.$$

The radius ratio is:

$$\beta = \frac{r_i}{r_o} = \frac{0.14}{0.4} = 0.35.$$

The maximum normal force that can be applied to the clutch without exceeding the pad pressure constraint is [4]:

$$P = \frac{2 n_s T_w}{\mu(r_o + r_i)} = \frac{2 \times 2 \times 24450}{0.35(0.4 + 0.14)} = 517.46 \text{ kN}.$$

Assuming the thickness of the thrust disk clutch is 0.1m and placed 0.4m from the gear 4 in the gearbox.

6. Final design

After we finished all of the components, we put them together as indicated in the diagram below. It also comes with four bearings, four gears, three shafts, and one disk clutch.

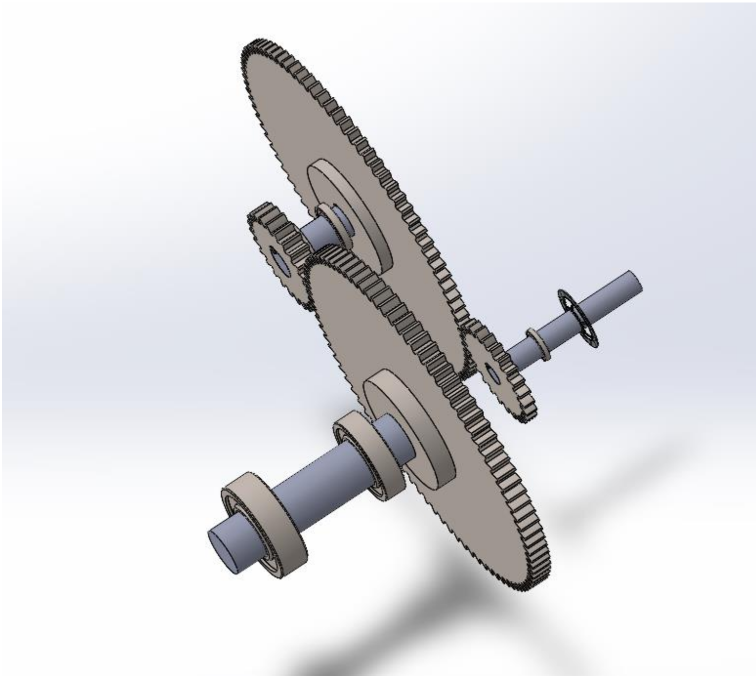

Figure 17: The final design without bedplate.

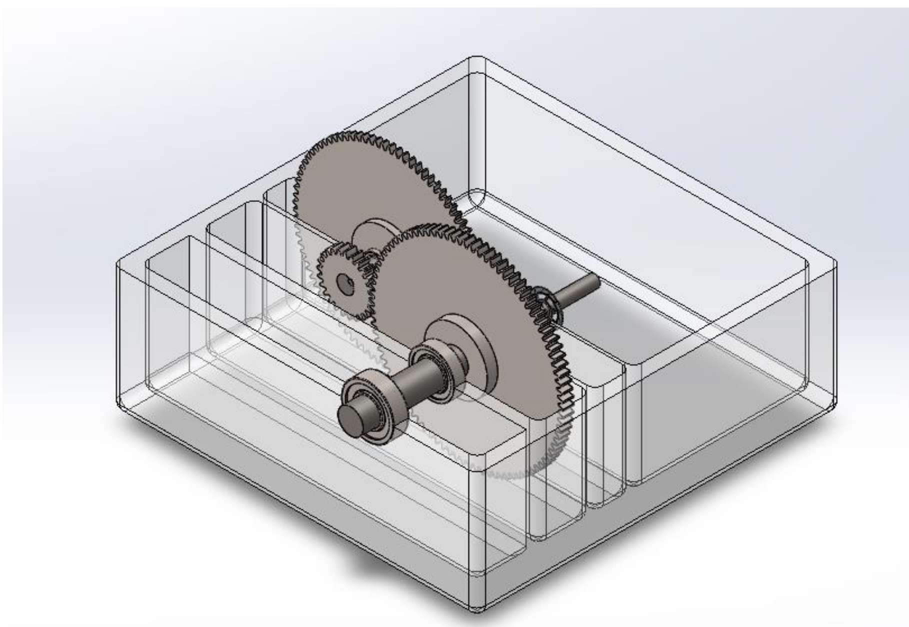

Figure 18: The final design with bedplate.

V. Simulation and System Analysis

1. Motion analysis

As the parts are designed completely on SolidWorks, we connect them together to form a wind turbine. Then by using Motion Analysis on SolidWorks, we can run the whole system and see the angular velocity of the output which is the high-speed shaft. Firstly, we apply a rotor, which has angular velocity of 60 RPM, to the main shaft. The expected angular velocity of the output is 750 RPM or 450 rad/s.

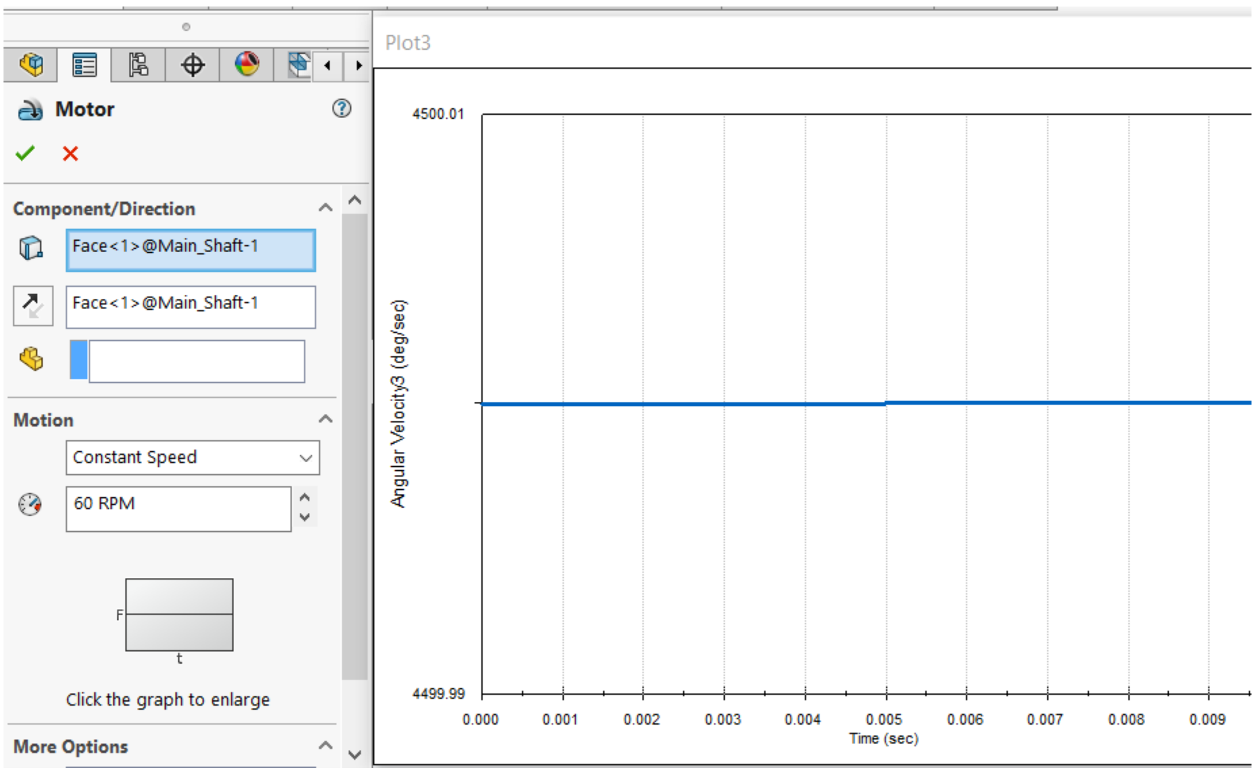

Figure 19: Angular velocity of the high-speed shaft when the main shaft's angular velocity is 60RPM.

According to the figure above, we can see that the angular velocity of the high-speed shaft is 450 rad/s. Since the ratio of the gears worked as expected.

2. FEA simulation

2.1. Shaft

After shafts are designed by using Solidworks, we will analyze the FEA using Simulation function. For the main shaft, it is applied by the radial and axial loading that are 122.37 kN and 24.474 kN respectively as well as torque T_m is 305.63kN.m. The figure below is the mesh, stress, deformation and strain analysis.

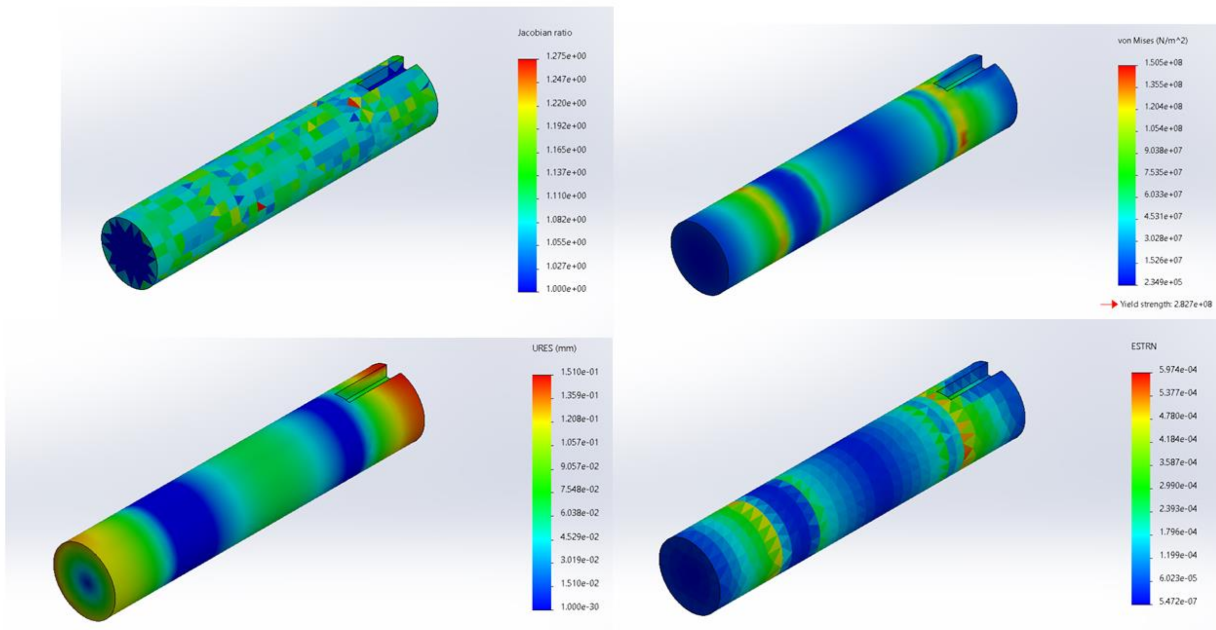

Figure 20: Mesh, Stress, Deformation, and Strain analysis of main shaft.

2.2. Gear

Gear 1 is applied a torque of 305.63 kN.m, Gear 2 and 3 are applied a torque of 86.44 kN.m, and Gear 4 is applied a torque of 24.45 kN.m. The simulation of FEA of them are shown in the appendices part. By looking at these figures, we can see that gears are able to withstand the torque applied to them.

VI. Discussion and Conclusion

As engineer perspective, the design of the wind turbine's components in this project has some good points and problems as well. These good things are the parts are calculated and chosen in the demand of the client and they are available on the market. All components are well linked together and give acceptable result. However, since several design specifications are lacking which force us to make assumptions, the result is not really accurate. For example, initially, we assume the length of shafts is quite short which made the diameter become large compare with the length. This led to the components of the wind turbine are larger. If the design time of this project is longer, we can redesign the components to make them more appropriate.

As student perspective, this project helps us a lot in the future study and career such as how to apply theory to real life's problem, what is the strategy in the design process, and applying software in the design as engineer, etc.

In conclusion, thank to practical assignment 2 - Mechanical Engineering Design and Analysis, we are able to understand the fundamental principles of wind turbines and each component. Moreover, we knew how to use Solidworks not only for drawing and designing components but also analyzing the motion of the wind turbine as well as behavior of the components like stress, strain, and deformation. Although the design is not really perfect, it is in the acceptable range and designed based on client's information. This assignment is helpful and going to support us in future.

VII. Reference

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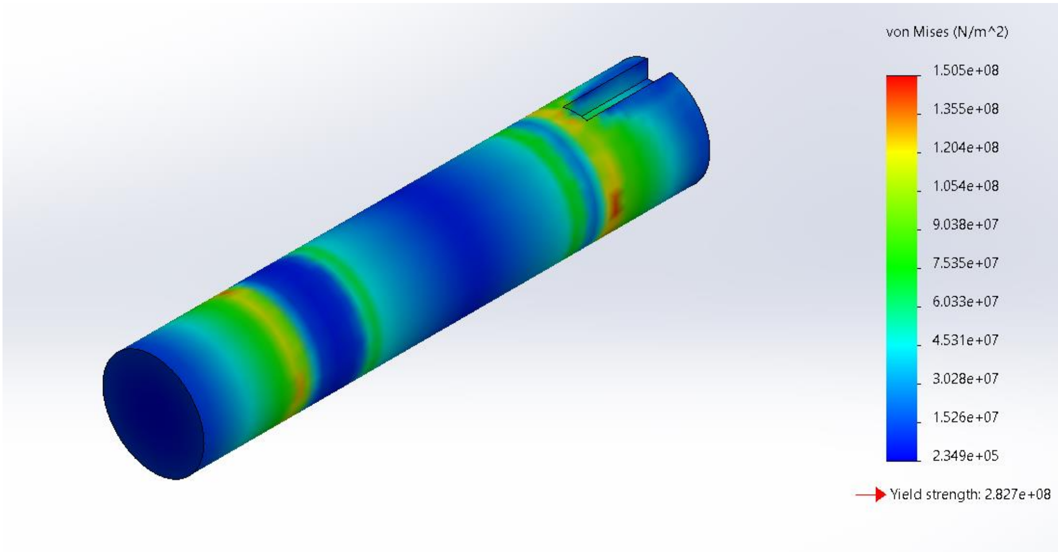
VIII. Appendices

1. Shaft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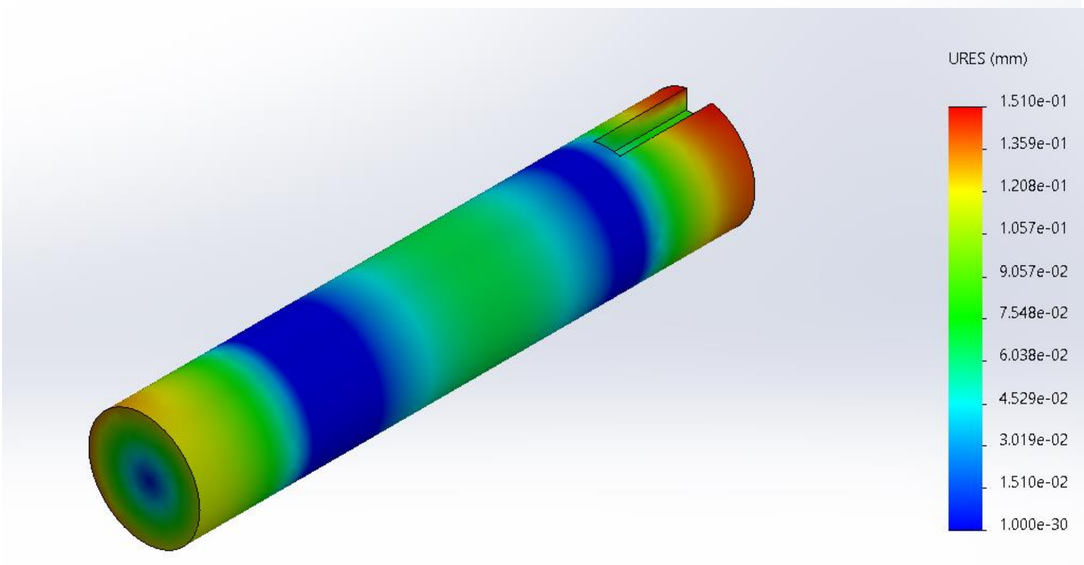
Yield strength: 2.827e+08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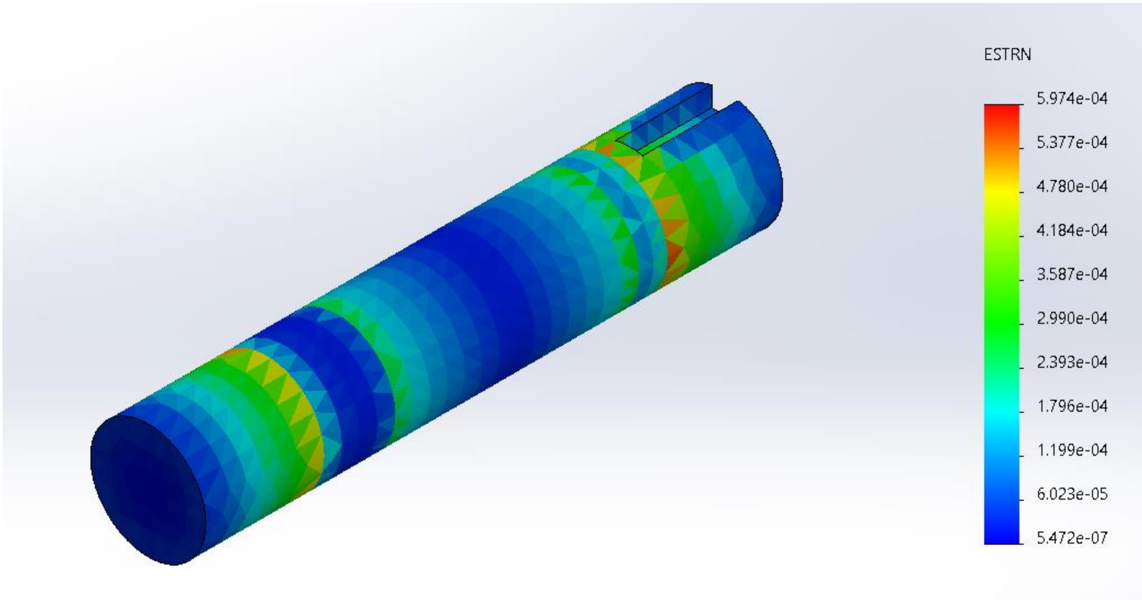
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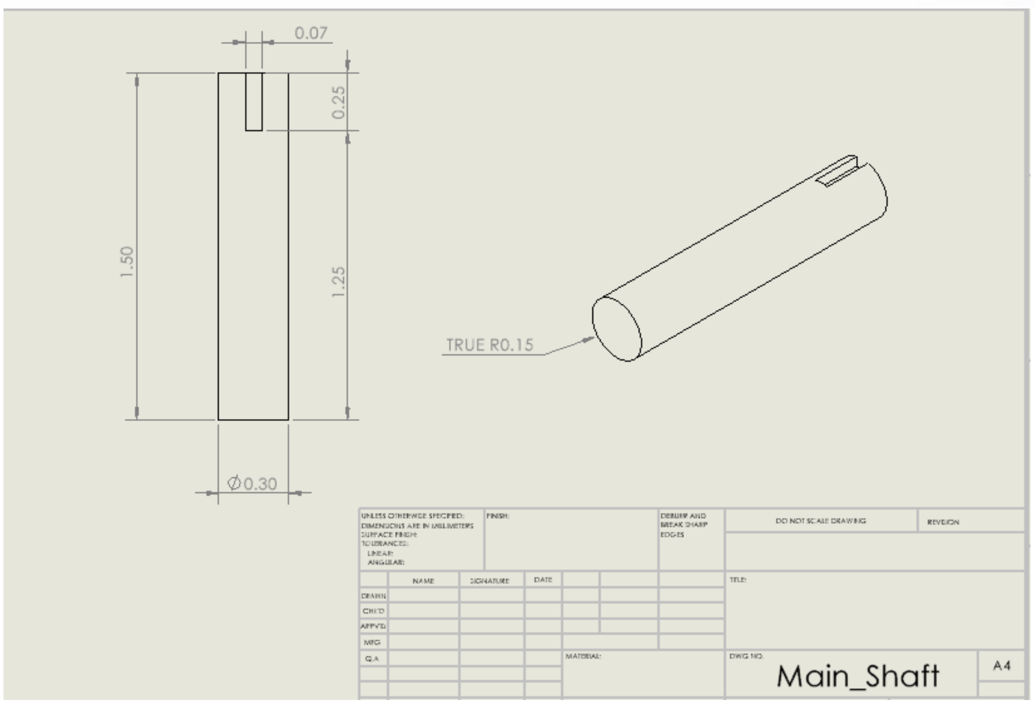
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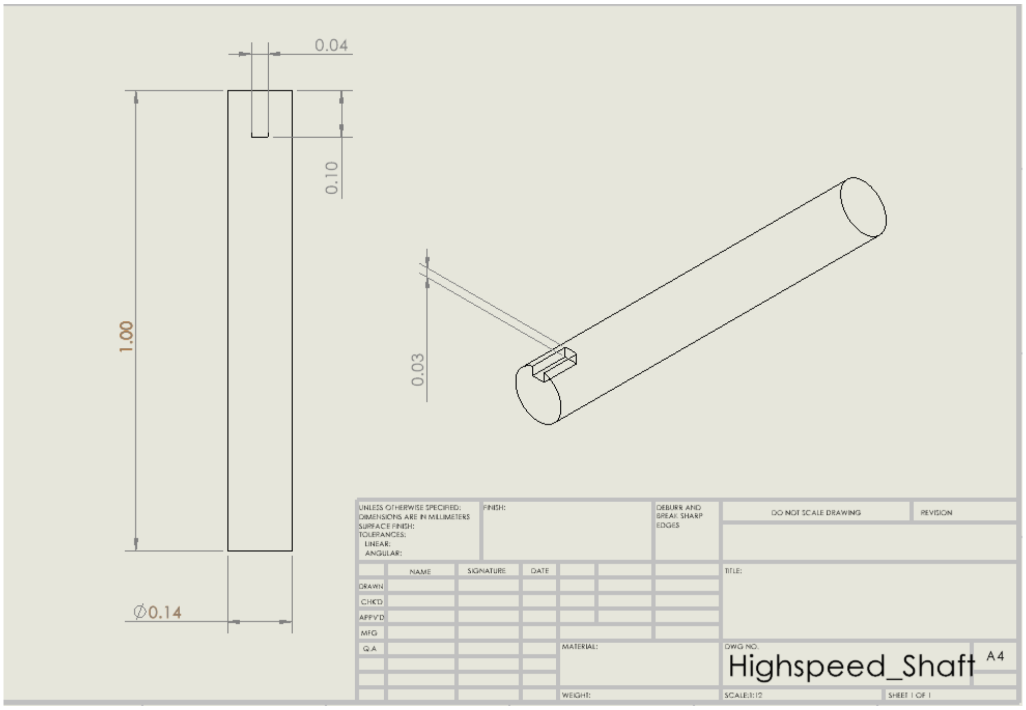
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ANGULAR:
FINISH:
DEBURR AND BREAK SHARP EDGES
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SIGNATURE
DATE
DRAWN
CHK'D
APPV'D
MFG
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Highspeed_Shaft
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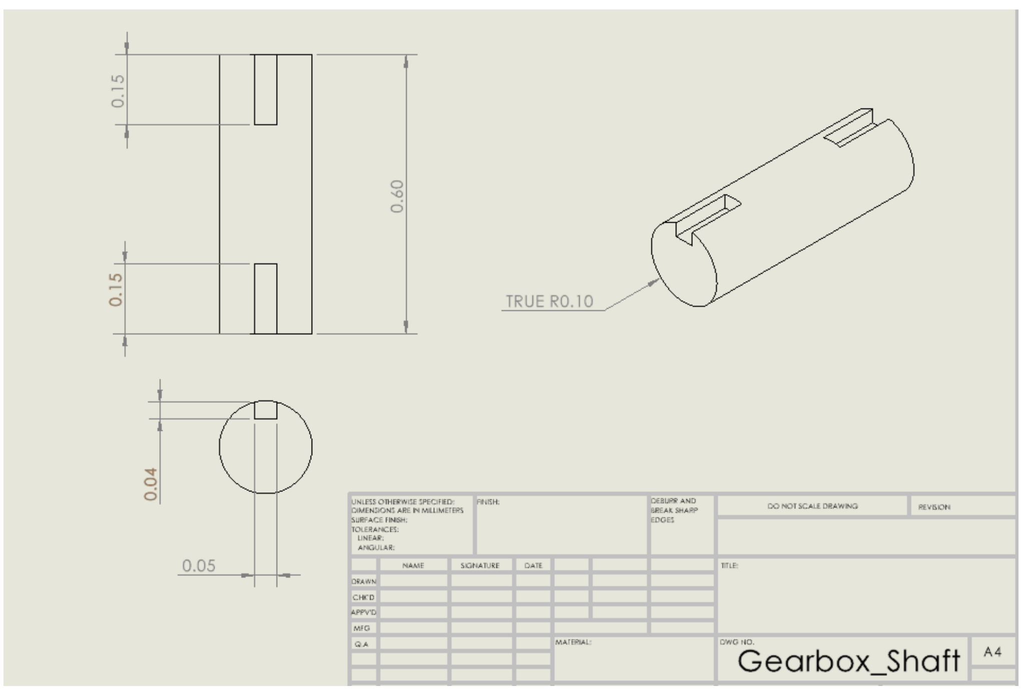
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NAME
SIGNATURE
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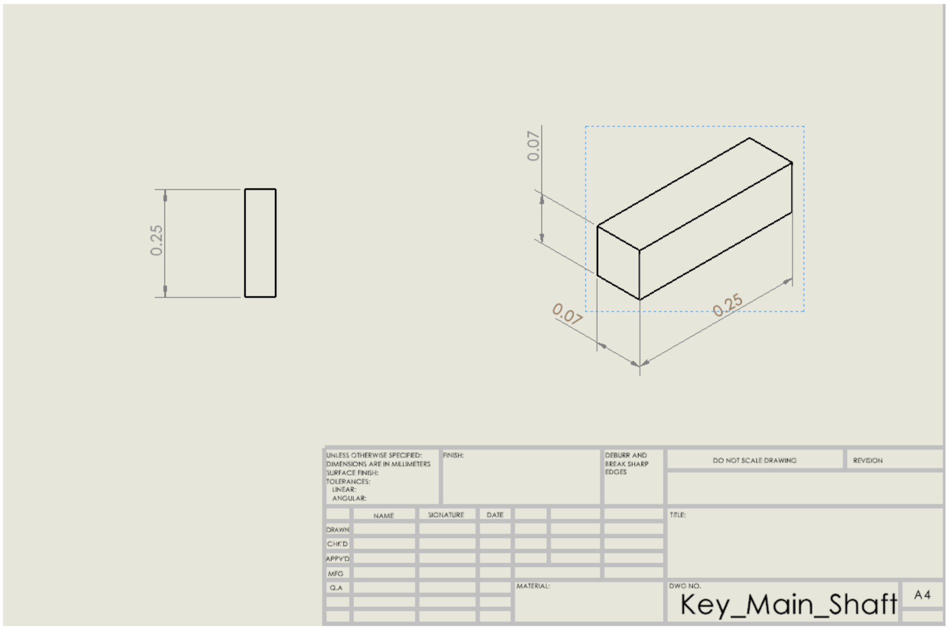
2. Key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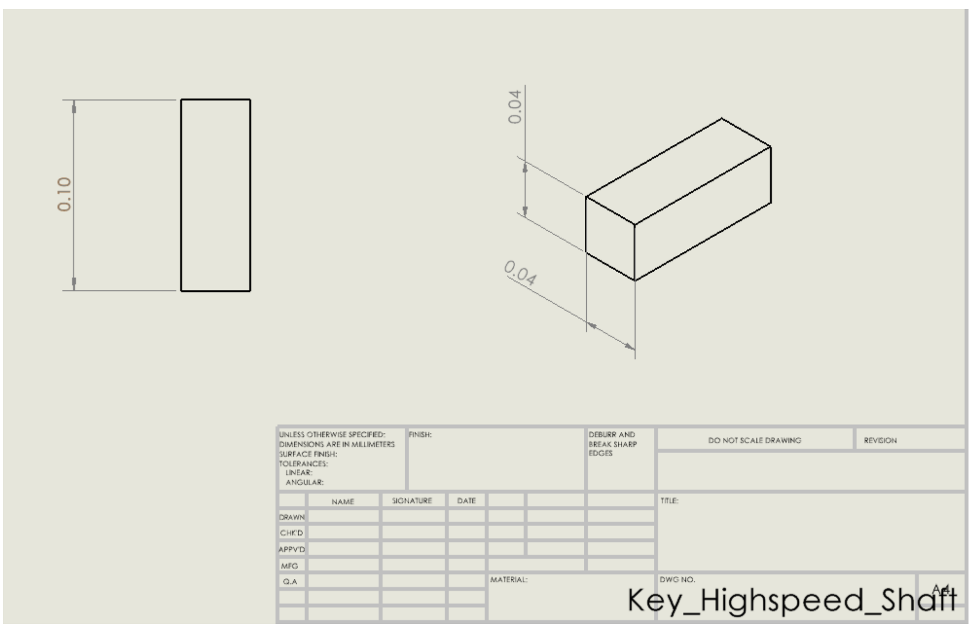
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ANGULAR:
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DEBURR AND BREAK SHARP EDGES
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NAME
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APPV'D
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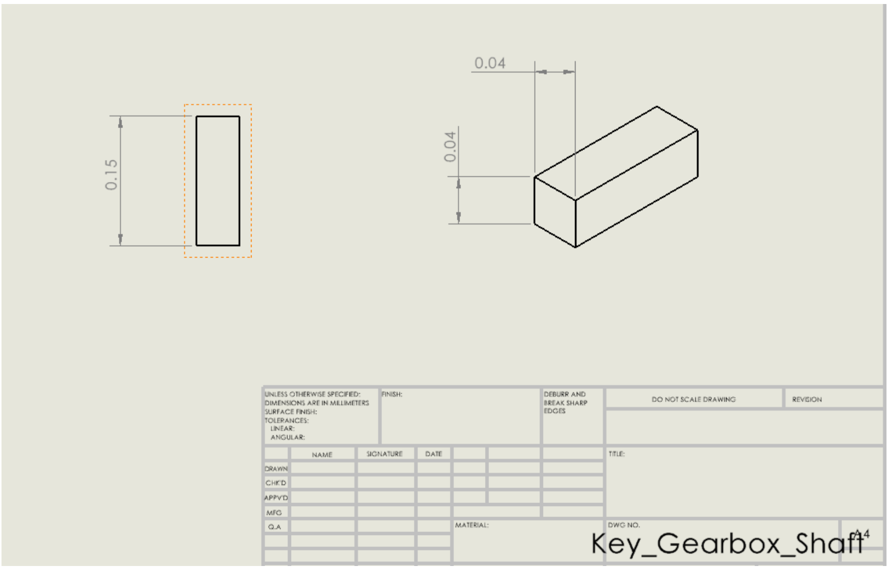
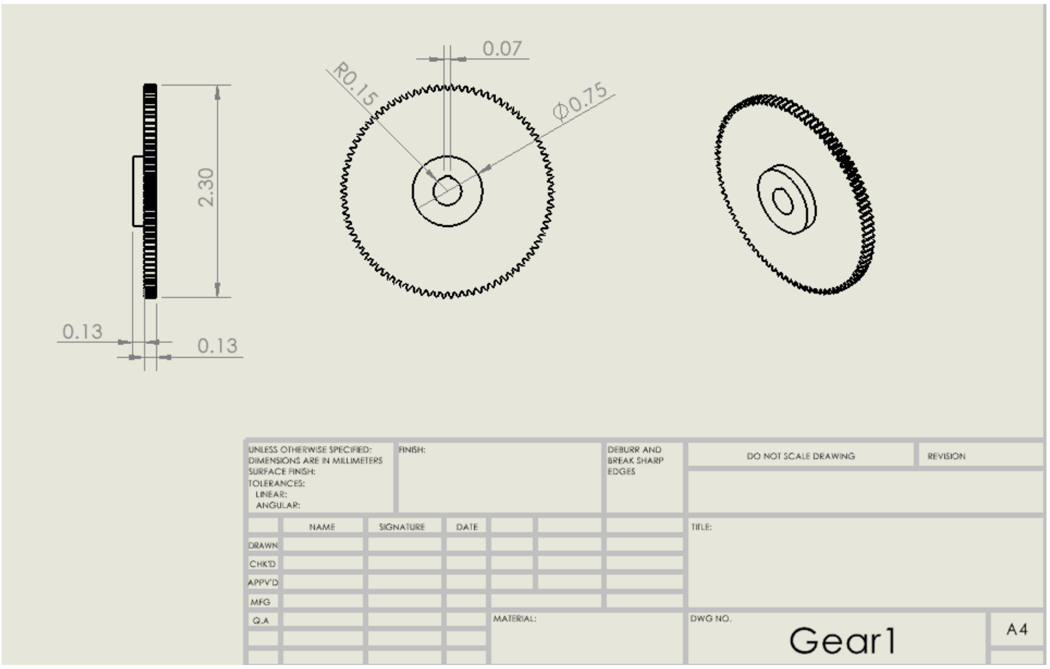
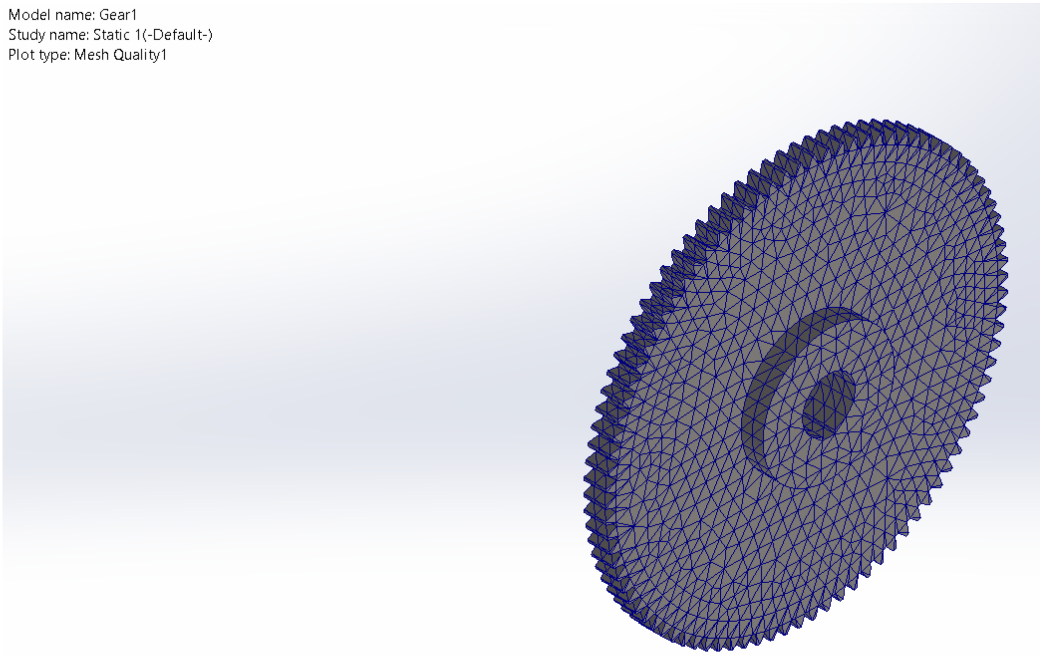


3. Gearbox

Gear 1:

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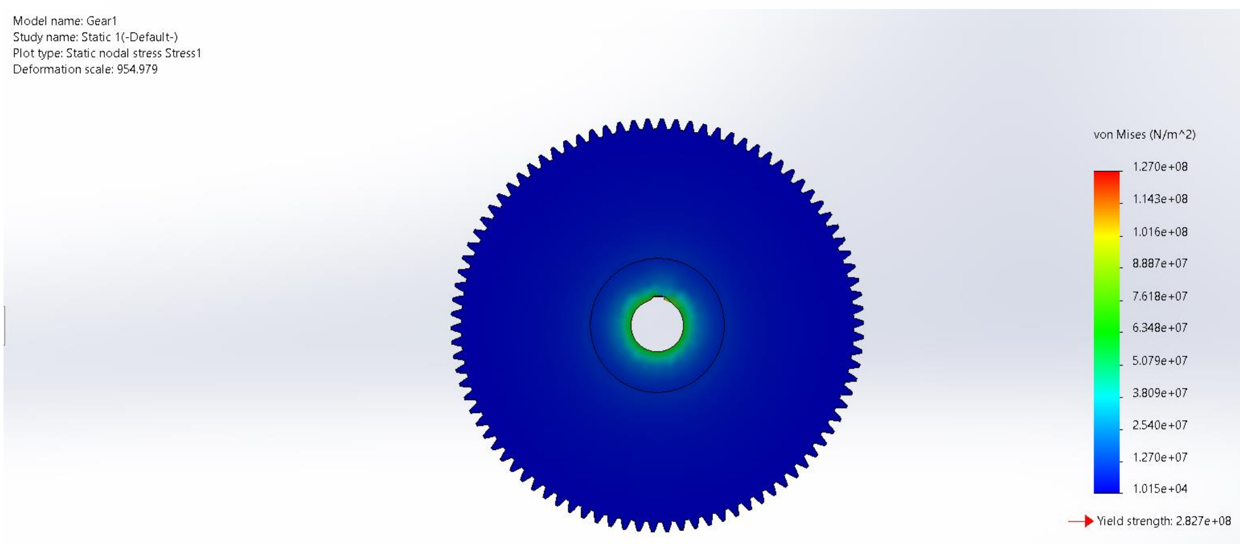
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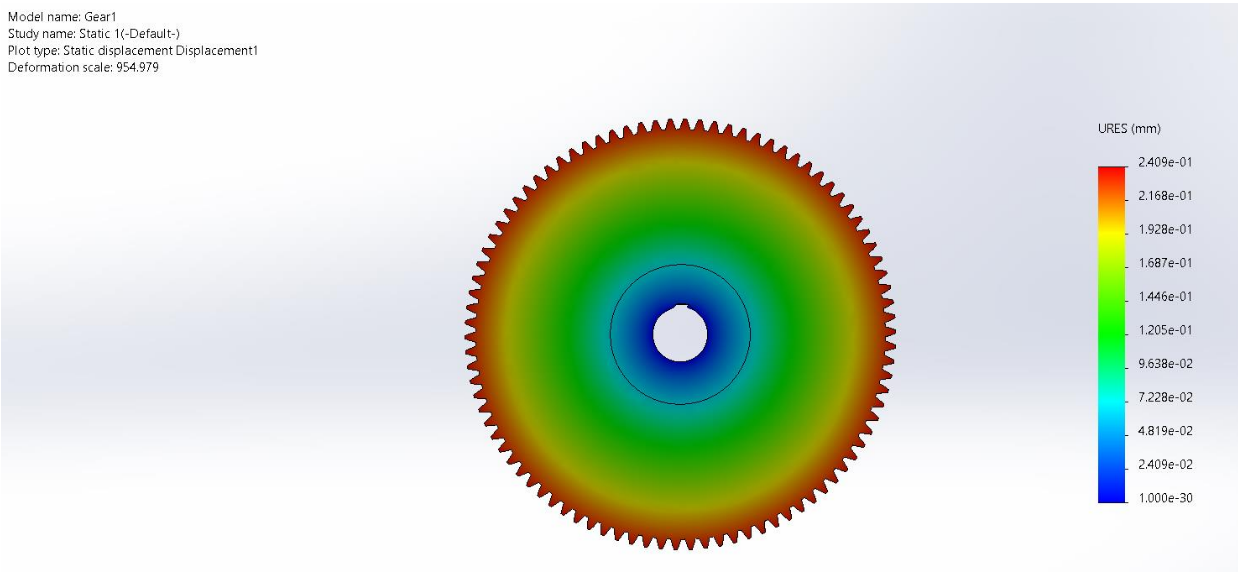
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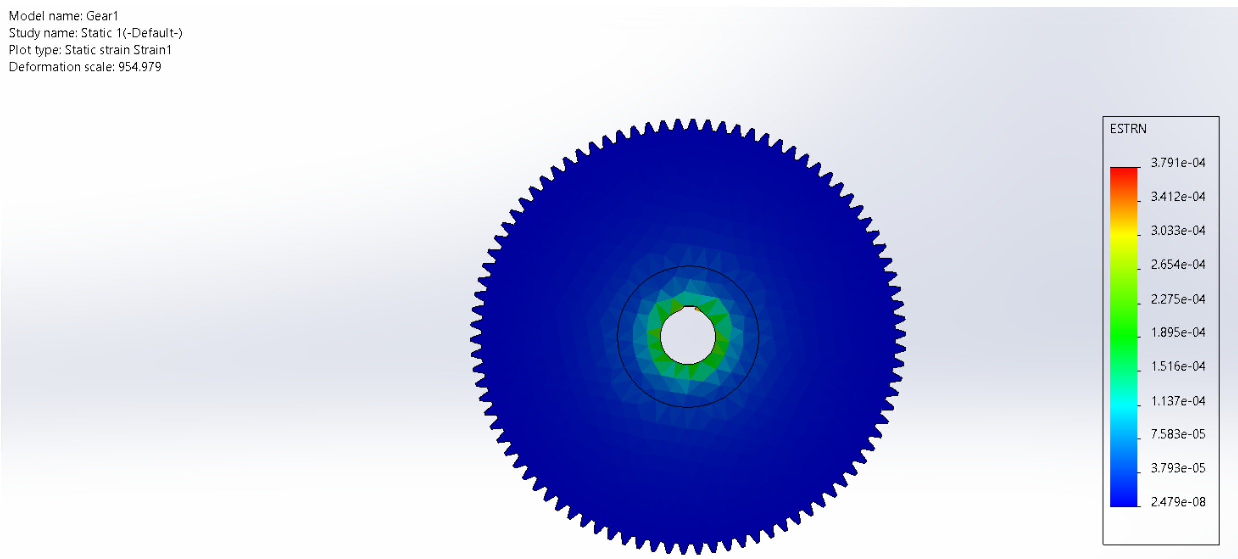
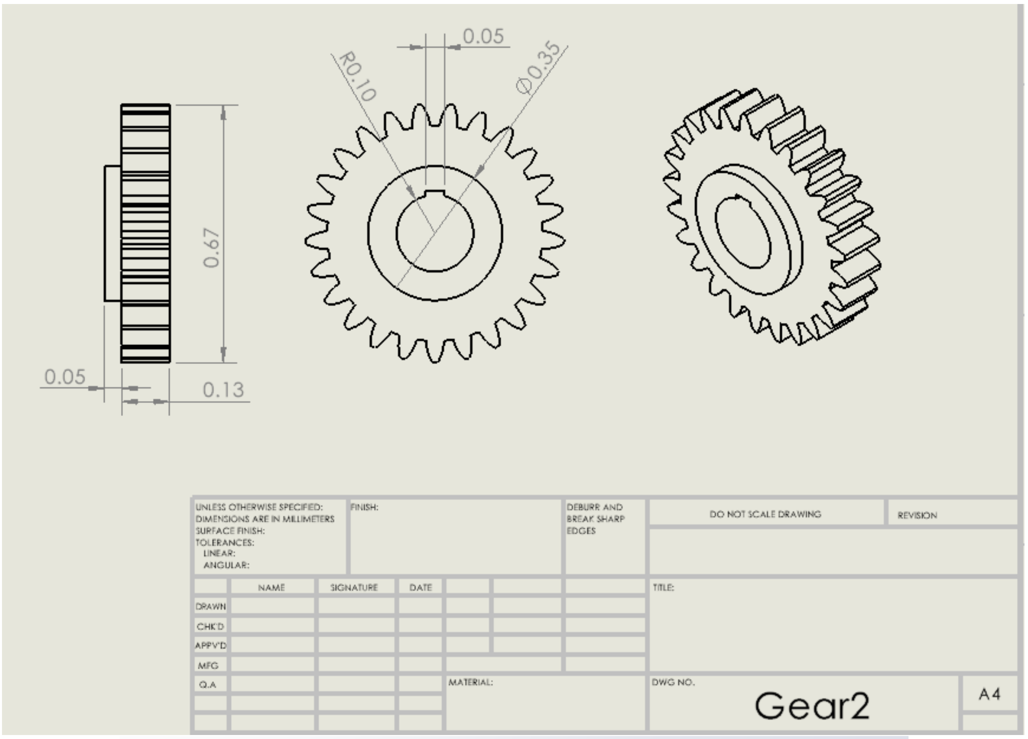
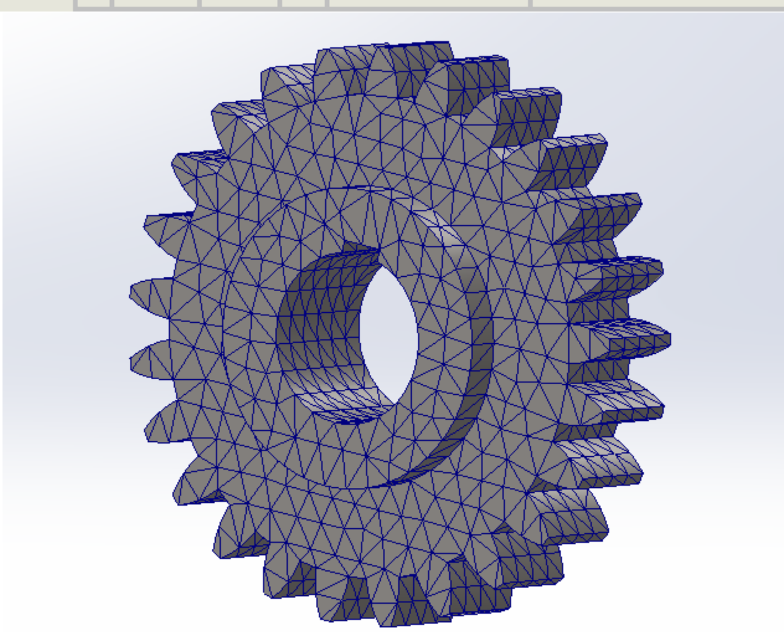
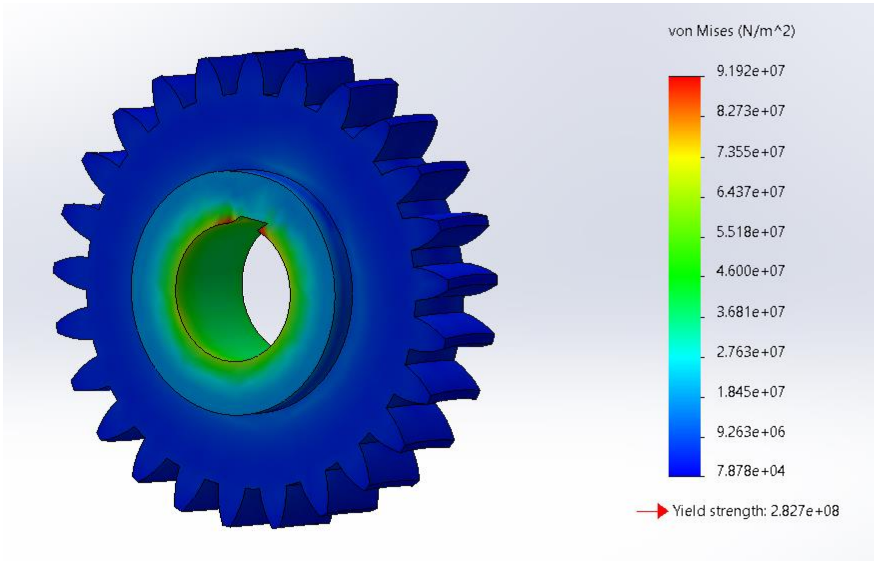
Study name: Static 1(-Default-)
Plot type: Static displacement Displacement1
Deformation scale: 954.979
URES (mm)
2.409e-01
2.168e-01
1.928e-01
1.687e-01
1.446e-01
1.205e-01
9.638e-02
7.228e-02
4.819e-02
2.409e-02
1.000e-30

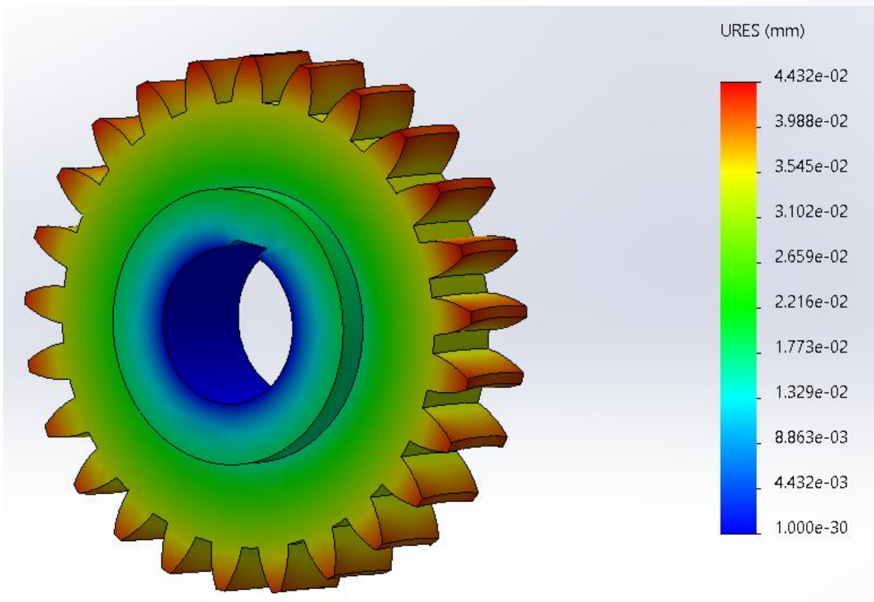

Model name: Gear1
Study name: Static 1(-Default-)
Plot type: Static strain Strain1
Deformation scale: 954.979
ESTRN
3.791e-04
3.412e-04
3.033e-04
2.654e-04
2.275e-04
1.895e-04
1.516e-04
1.137e-04
7.583e-05
3.793e-05
2.479e-08

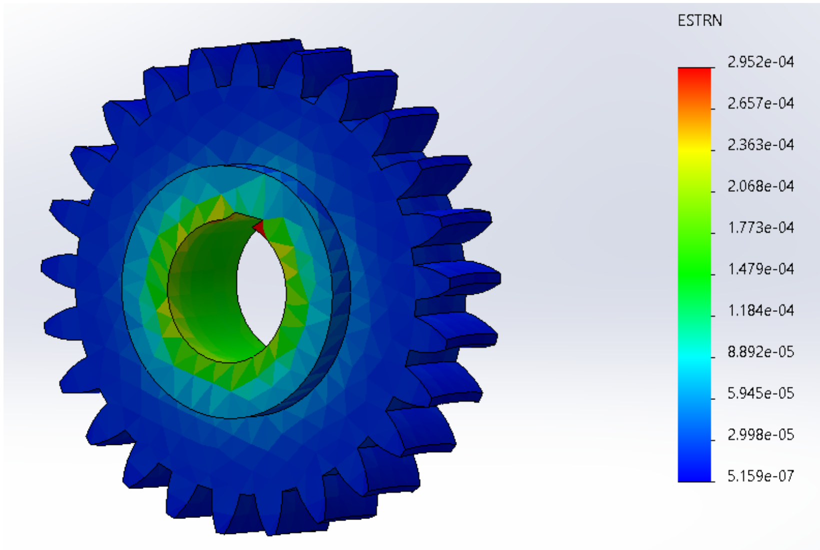
Gear 2:


0.05
R0.10
Ø0.35
0.67
0.05
0.13
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION
NAME
SIGNATURE
DATE
DRAWN
CHK'D
APPV'D
MFG
Q.A
MATERIAL:
TITLE:
DWG NO.
Gear2
A4

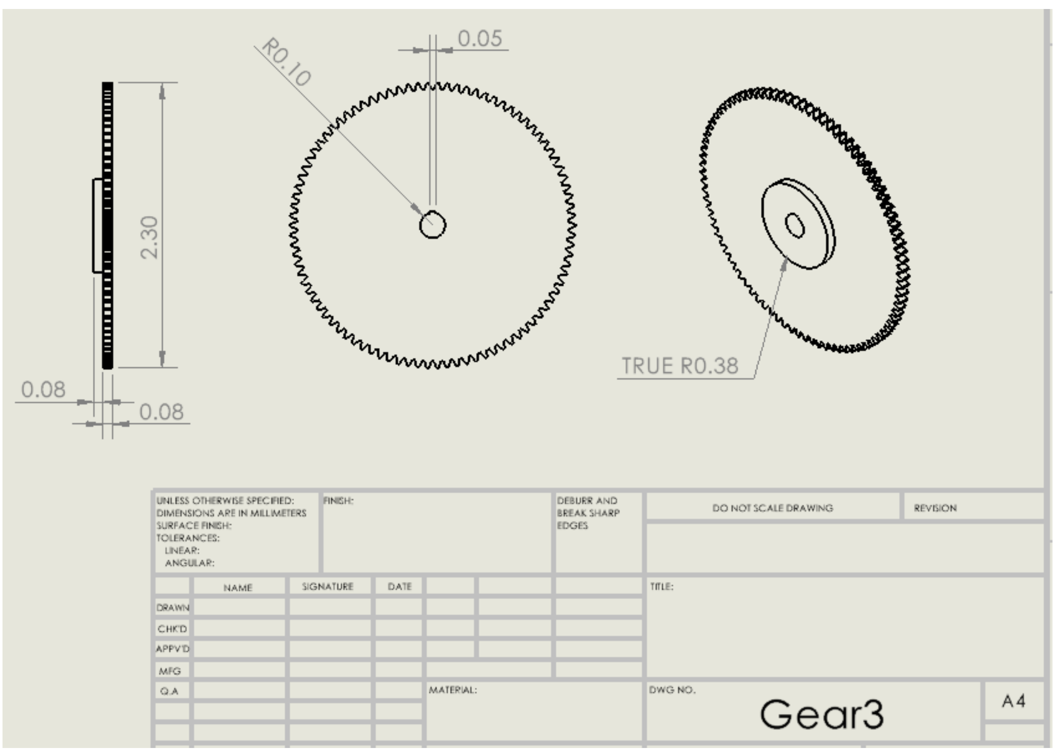


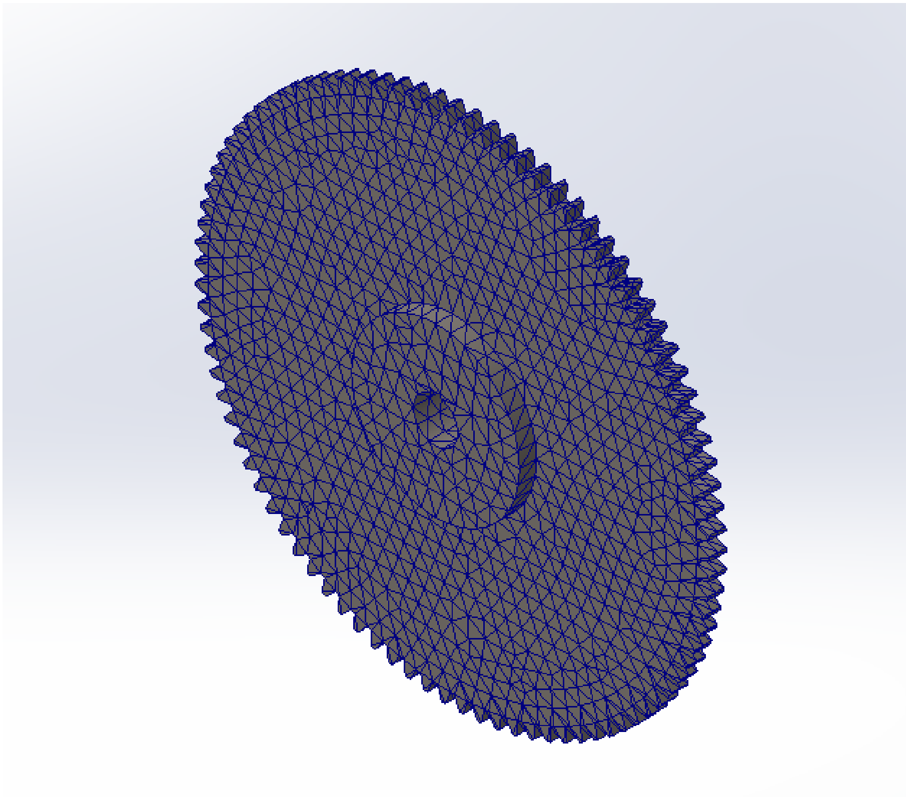

von Mises (N/m^2)
9.192e+07
8.273e+07
7.355e+07
6.437e+07
5.518e+07
4.600e+07
3.681e+07
2.763e+07
1.845e+07
9.263e+06
7.878e+04
Yield strength: 2.827e+08

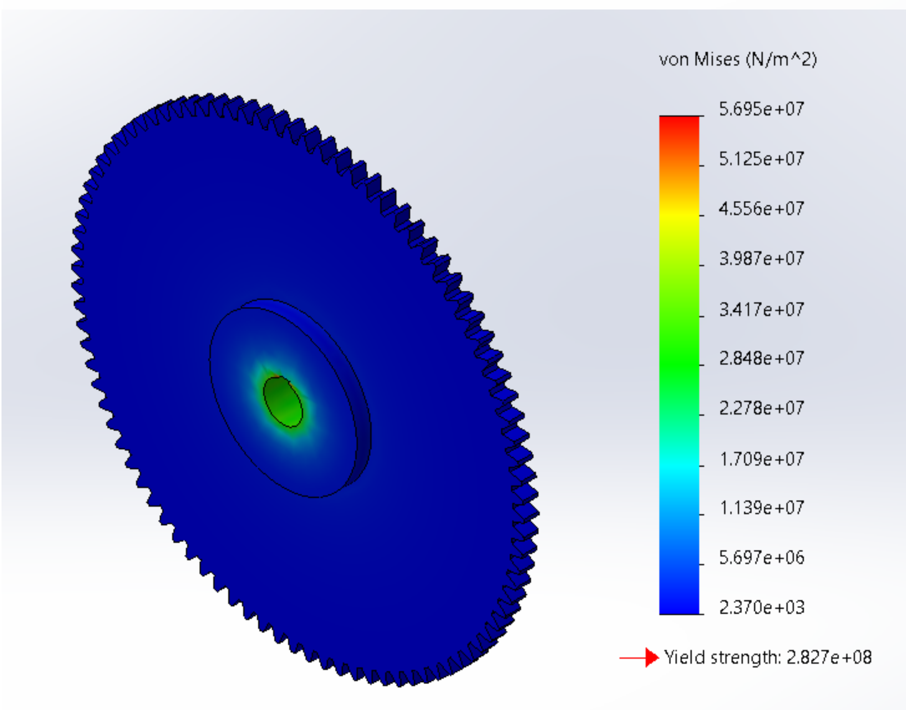

URES (mm)
4.432e-02
3.988e-02
3.545e-02
3.102e-02
2.659e-02
2.216e-02
1.773e-02
1.329e-02
8.863e-03
4.432e-03
1.000e-30

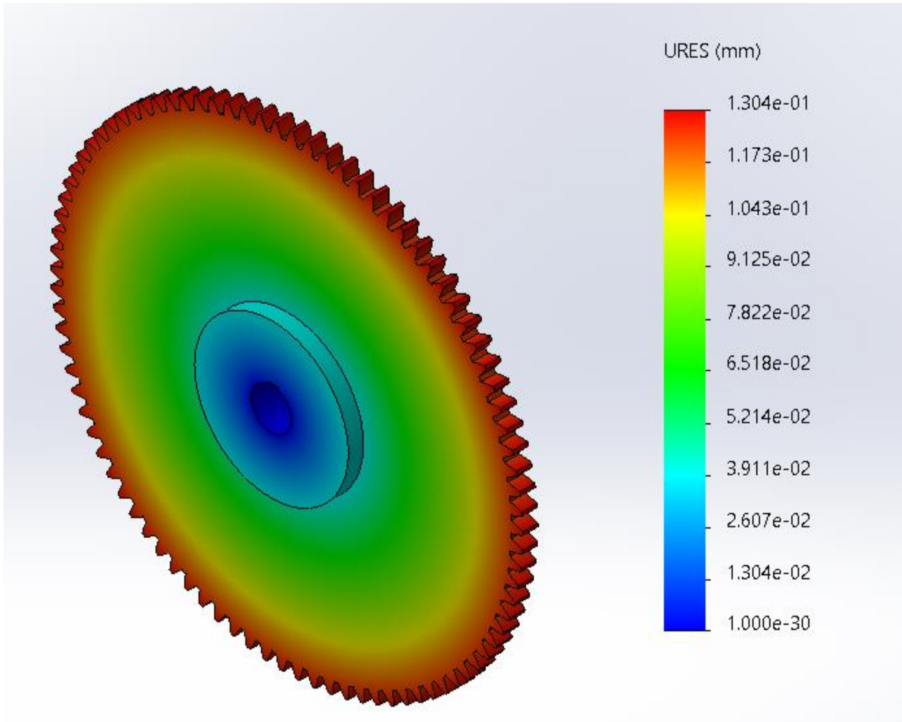

ESTRN
2.952e-04
2.657e-04
2.363e-04
2.068e-04
1.773e-04
1.479e-04
1.184e-04
8.892e-05
5.945e-05
2.998e-05
5.159e-07

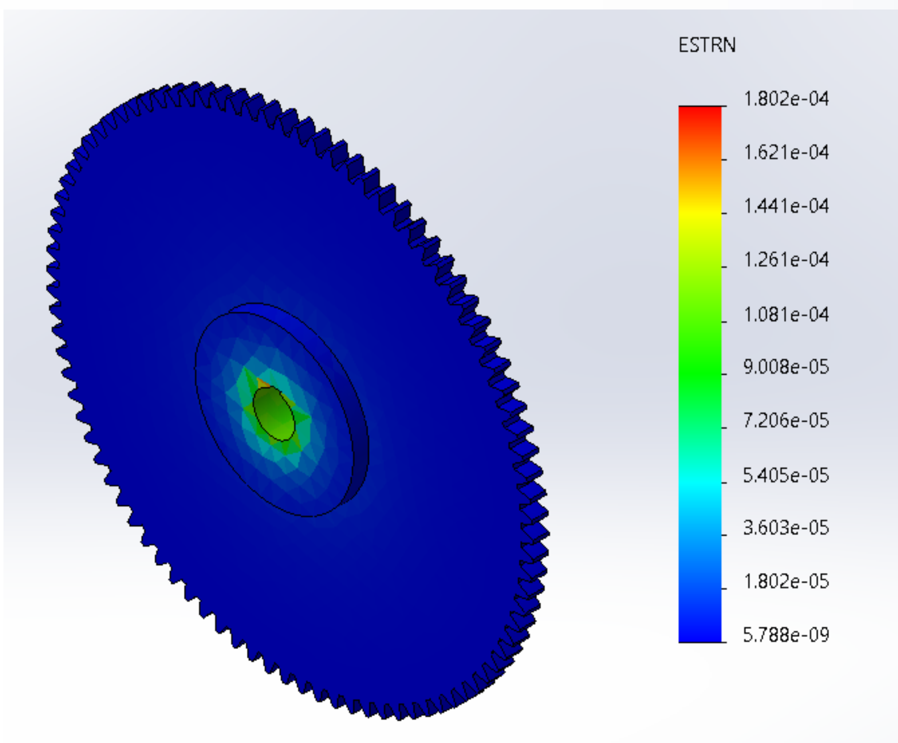
Gear 3:


R0.10
0.05
2.30
TRUE R0.38
0.08
0.08
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION
NAME
SIGNATURE
DATE
TITLE:
DRAWN
CHK'D
APPV'D
MFG
Q.A
MATERIAL:
DWG NO.
Gear3
A4

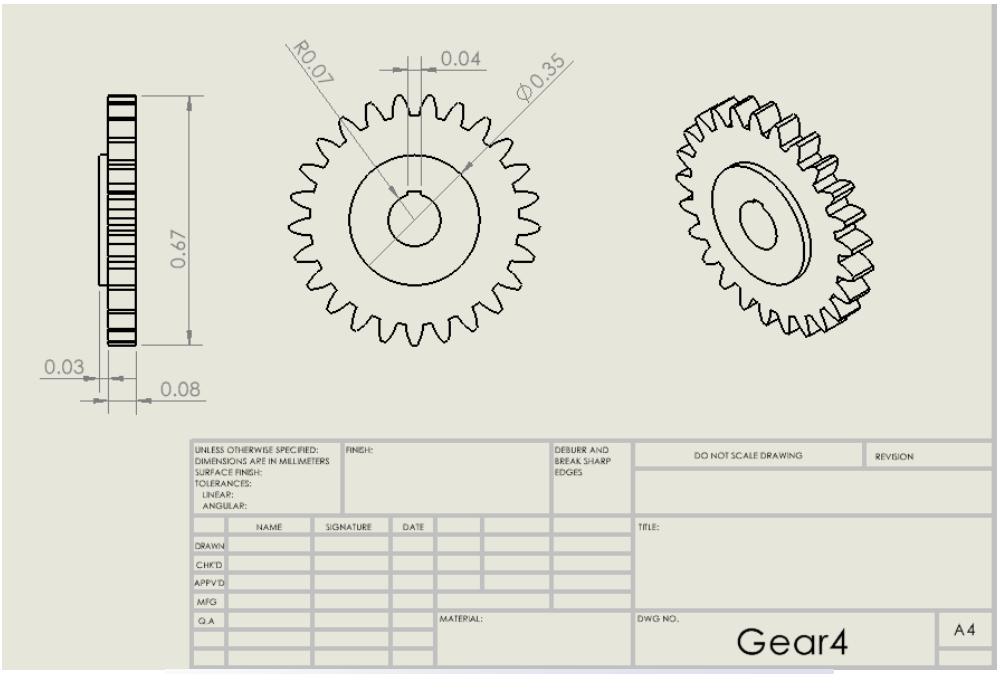


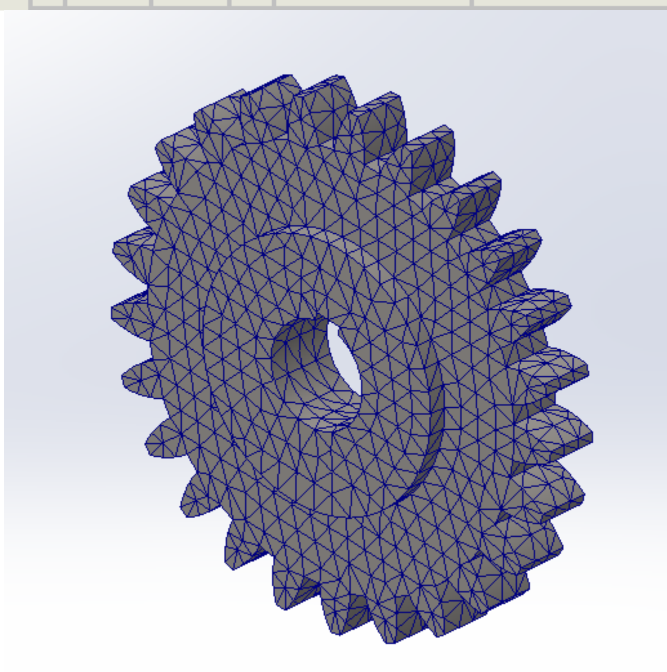

von Mises (N/m^2)
5.695e+07
5.125e+07
4.556e+07
3.987e+07
3.417e+07
2.848e+07
2.278e+07
1.709e+07
1.139e+07
5.697e+06
2.370e+03
Yield strength: 2.827e+08

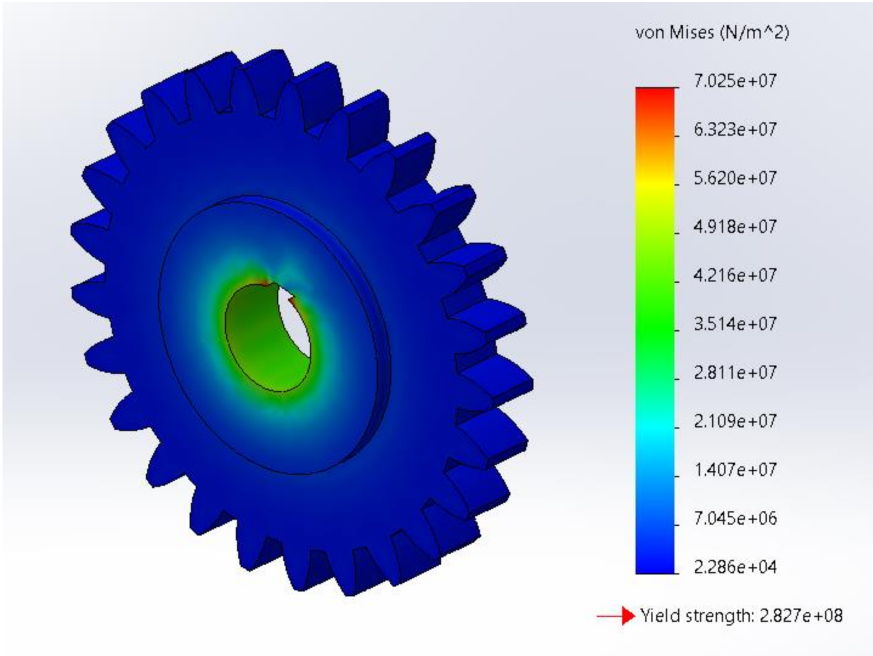

URES (mm)
1.304e-01
1.173e-01
1.043e-01
9.125e-02
7.822e-02
6.518e-02
5.214e-02
3.911e-02
2.607e-02
1.304e-02
1.000e-30

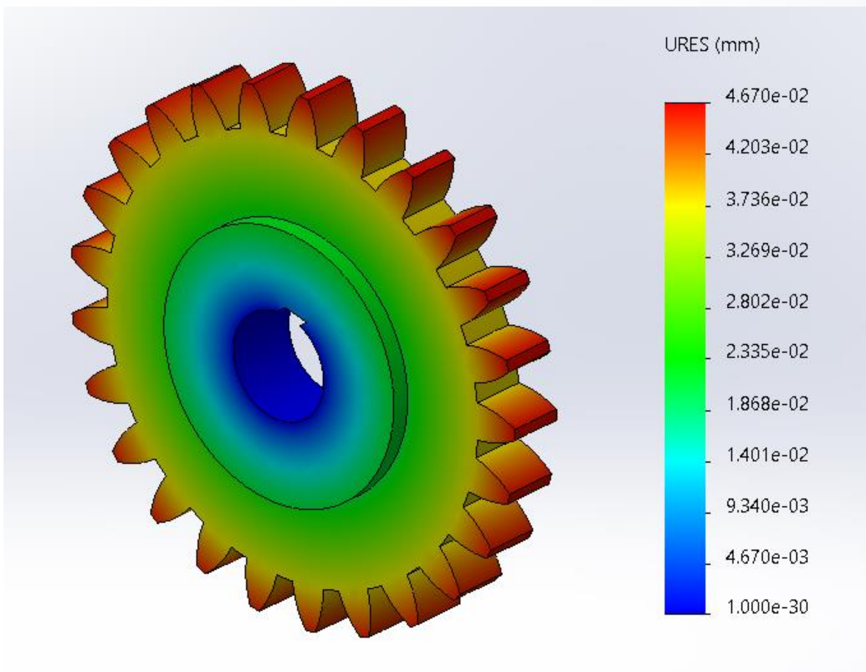

ESTRN
1.802e-04
1.621e-04
1.441e-04
1.261e-04
1.081e-04
9.008e-05
7.206e-05
5.405e-05
3.603e-05
1.802e-05
5.788e-09

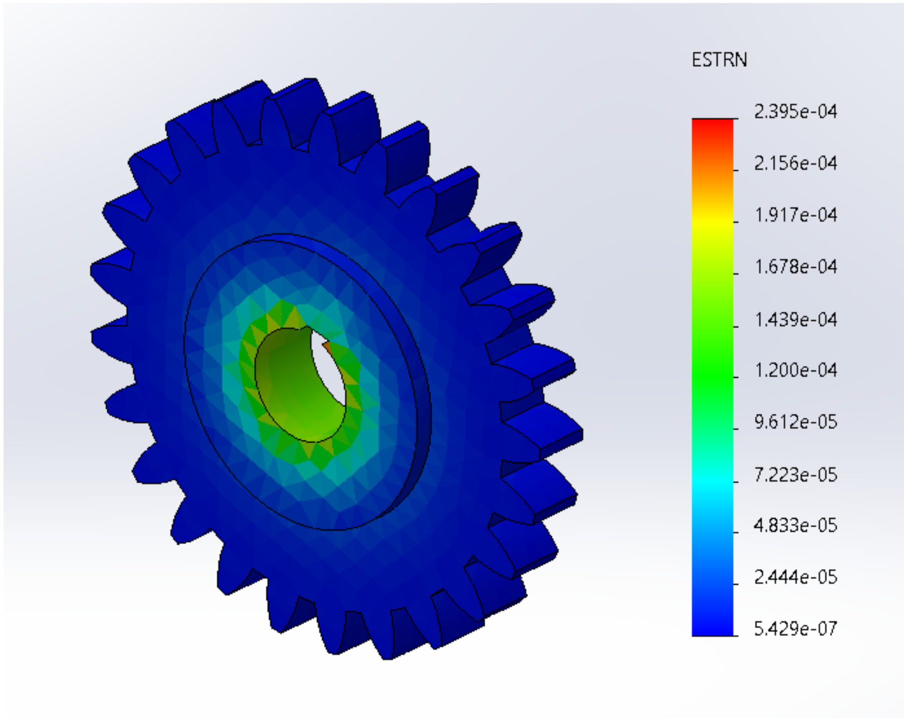
Gear 4


R0.07
0.04
Ø0.35
0.67
0.03
0.08
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION
NAME
SIGNATURE
DATE
DRAWN
CHK'D
APPV'D
MFG
Q.A
MATERIAL:
TITLE:
DWG NO.
Gear4
A4

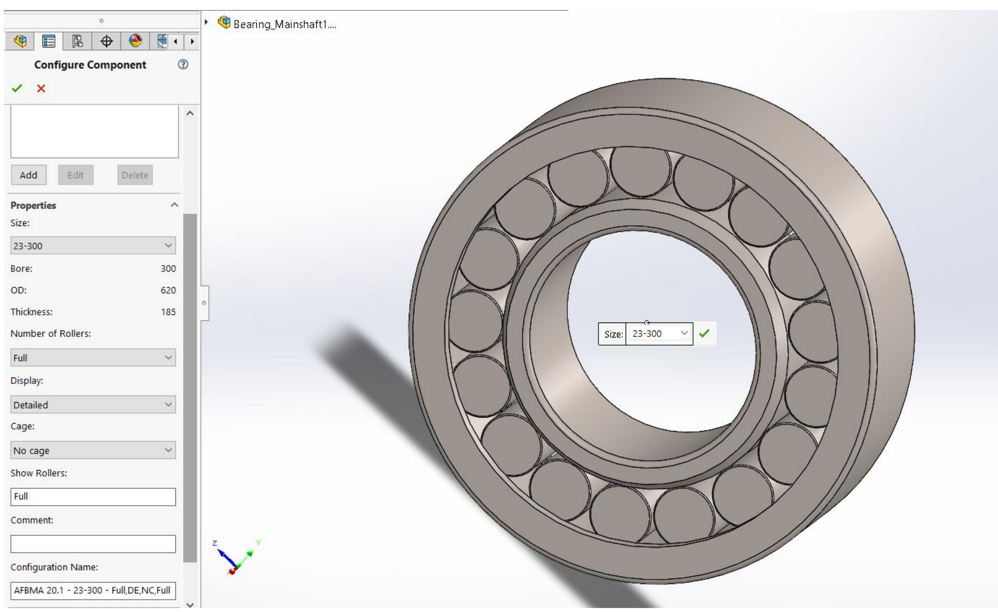


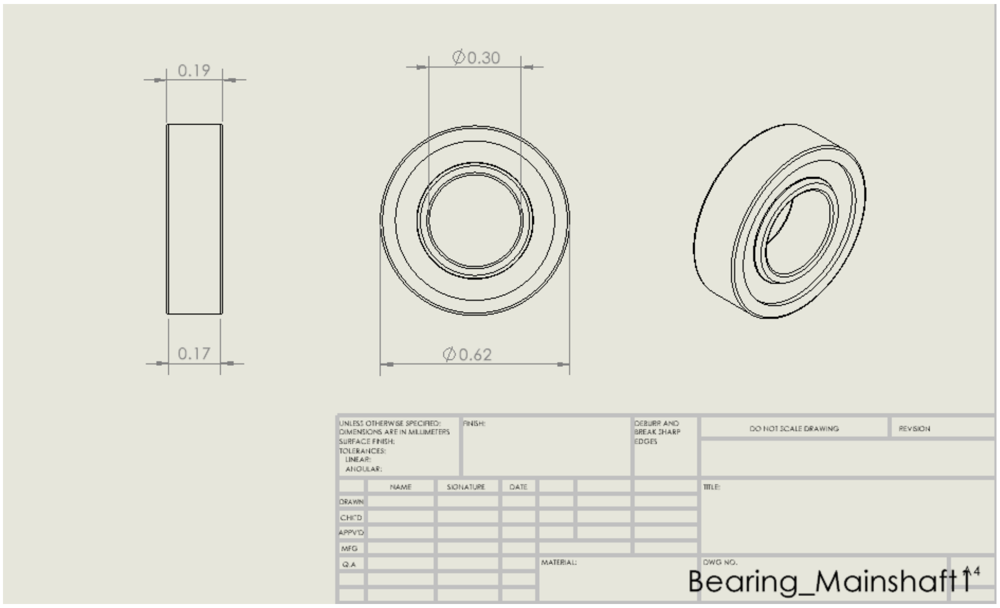

von Mises (N/m^2)
7.025e+07
6.323e+07
5.620e+07
4.918e+07
4.216e+07
3.514e+07
2.811e+07
2.109e+07
1.407e+07
7.045e+06
2.286e+04
Yield strength: 2.827e+08

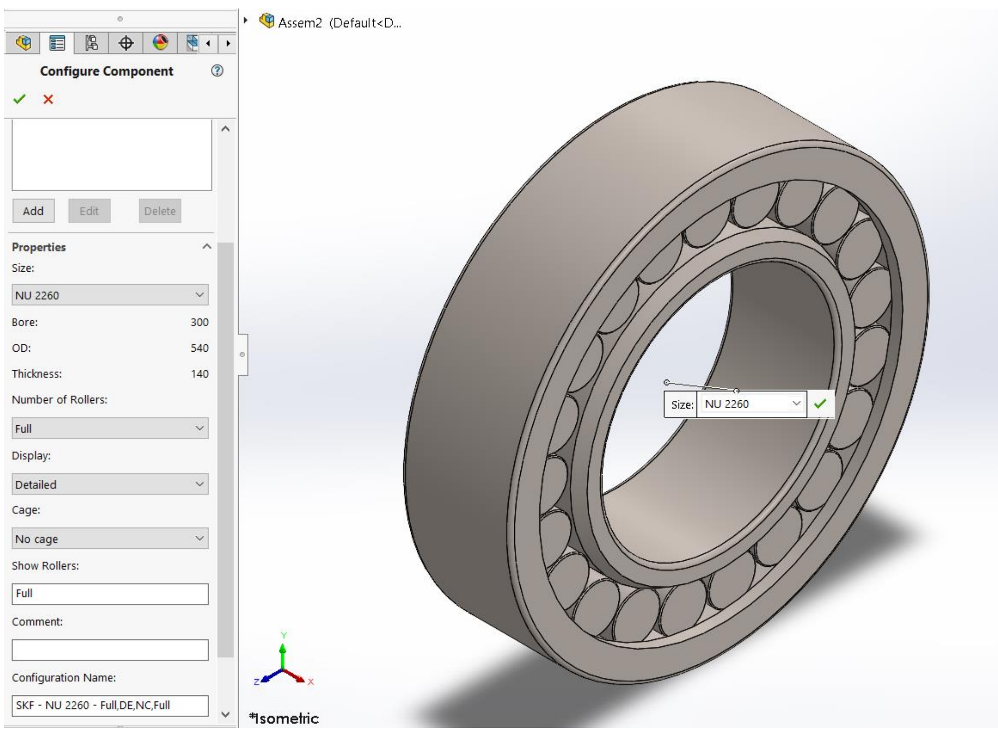

URES (mm)
4.670e-02
4.203e-02
3.736e-02
3.269e-02
2.802e-02
2.335e-02
1.868e-02
1.401e-02
9.340e-03
4.670e-03
1.000e-30

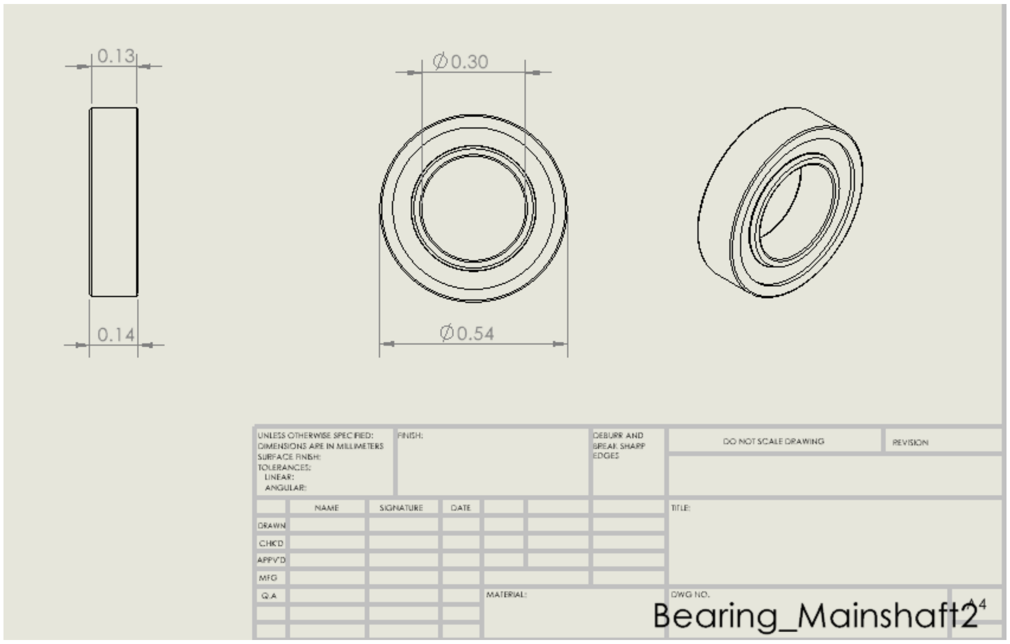

ESTRN
2.395e-04
2.156e-04
1.917e-04
1.678e-04
1.439e-04
1.200e-04
9.612e-05
7.223e-05
4.833e-05
2.444e-05
5.429e-07

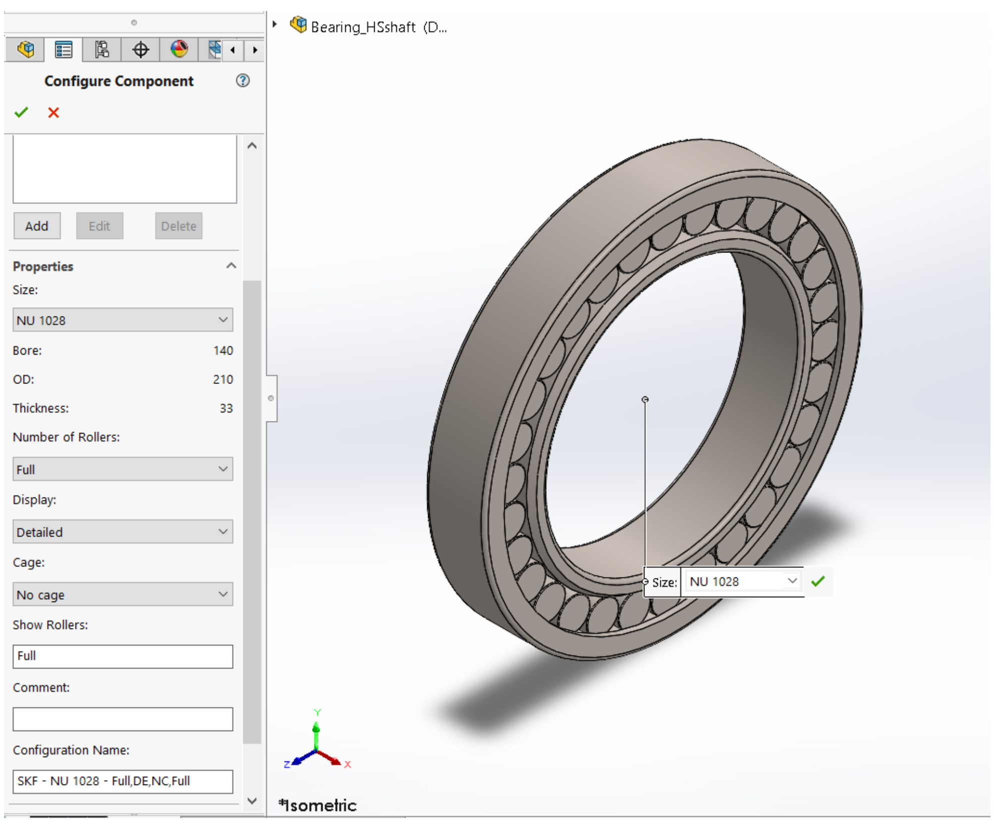
4. Bearing

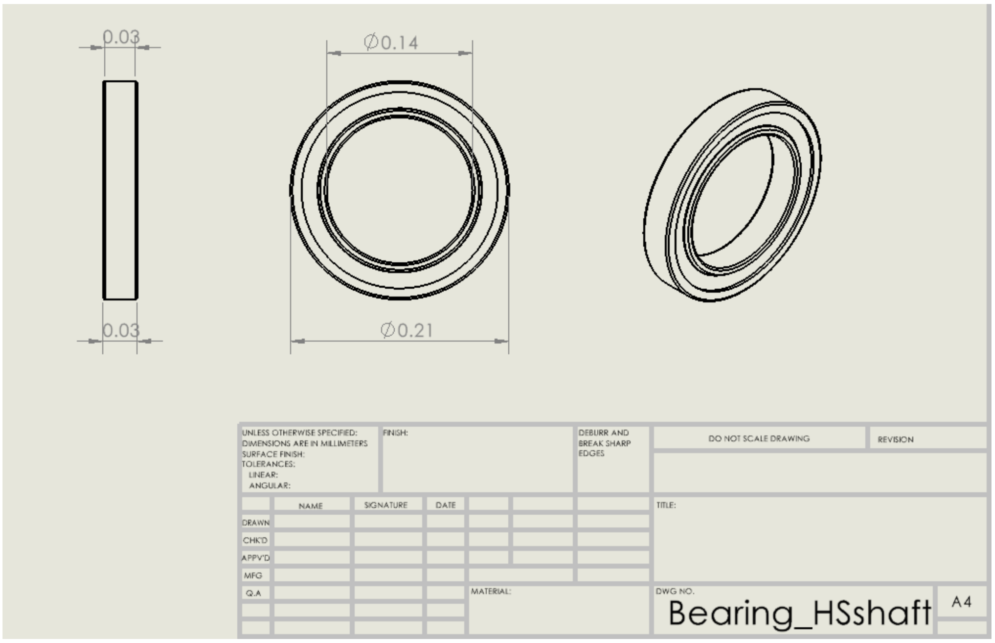

Bearing_Mainshaft1...
Configure Component
Add
Edit
Delete
Properties
Size:
23-300
Bore:
300
OD:
620
Thickness:
185
Number of Rollers:
Full
Display:
Detailed
Cage:
No cage
Show Rollers:
Full
Comment:
Configuration Name:
AFBMA 20.1 - 23-300 - Full,DE,NC,Full
Size:
23-300

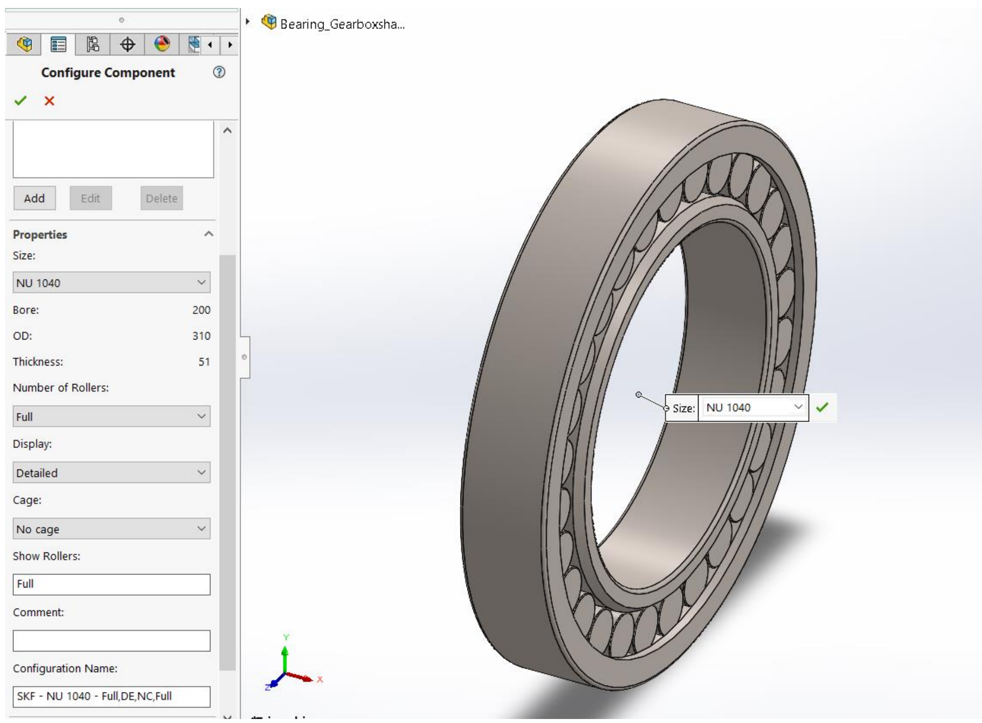

0.19
Ø0.30
0.17
Ø0.62
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND
BREAK SHARP
EDGES
DO NOT SCALE DRAWING
REVISION
NAME
SIGNATURE
DATE
TITLE:
DRAWN
CHK'D
APPV'D
MFG
Q.A
MATERIAL:
DWG NO.
Bearing_Mainshaft1

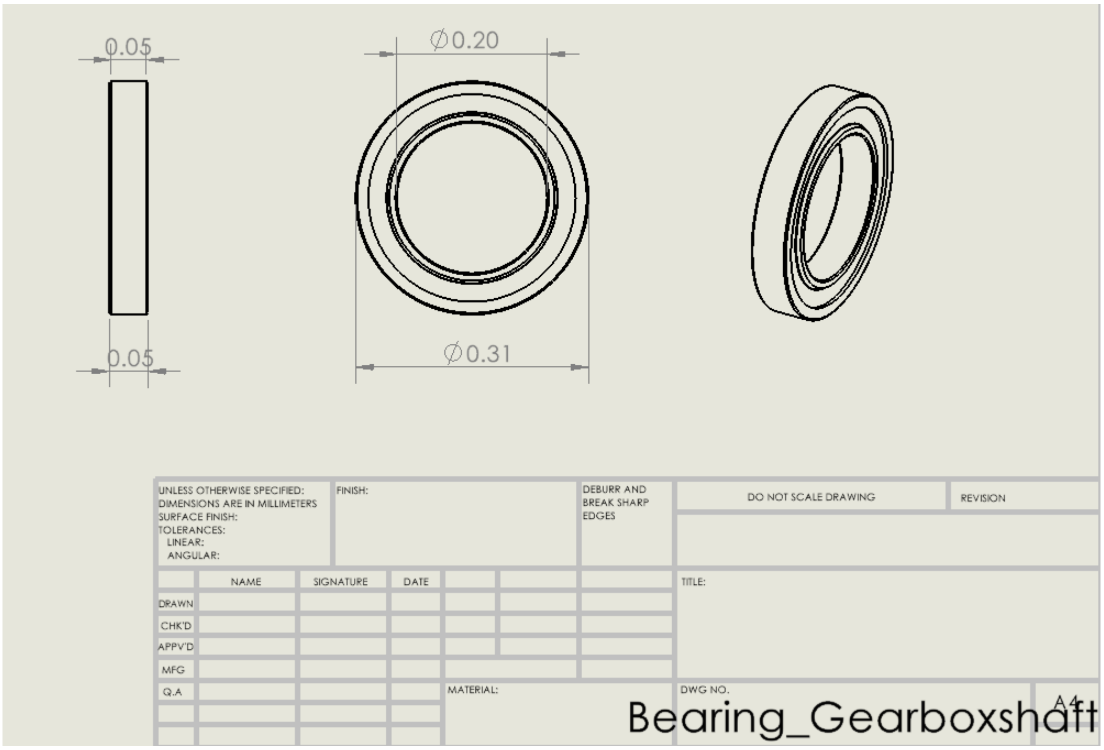

Assem2 (Default<D...
Configure Component
Add
Edit
Delete
Properties
Size:
NU 2260
Bore:
300
OD:
540
Thickness:
140
Number of Rollers:
Full
Display:
Detailed
Cage:
No cage
Show Rollers:
Full
Comment:
Configuration Name:
SKF - NU 2260 - Full,DE,NC,Full
Size:
NU 2260
*Isometric


0.13
Ø0.30
0.14
Ø0.54
Bearing_Mainshaft2
A4


Bearing_HSshaft (D...
Configure Component
Add
Edit
Delete
Properties
Size:
NU 1028
Bore: 140
OD: 210
Thickness: 33
Number of Rollers:
Full
Display:
Detailed
Cage:
No cage
Show Rollers:
Full
Comment:
Configuration Name:
SKF - NU 1028 - Full,DE,NC,Full
Size: NU 1028
*Isometric


0.03
∅0.14
0.03
∅0.21
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION
NAME
SIGNATURE
DATE
DRAWN
CHK'D
APPV'D
MFG
Q.A
MATERIAL:
TITLE:
DWG NO.
Bearing_HSshaft
A4


Bearing_Gearboxsha...
Configure Component
Add
Edit
Delete
Properties
Size:
NU 1040
Bore:
200
OD:
310
Thickness:
51
Number of Rollers:
Full
Display:
Detailed
Cage:
No cage
Show Rollers:
Full
Comment:
Configuration Name:
SKF - NU 1040 - Full,DE,NC,Full
Size:
NU 1040



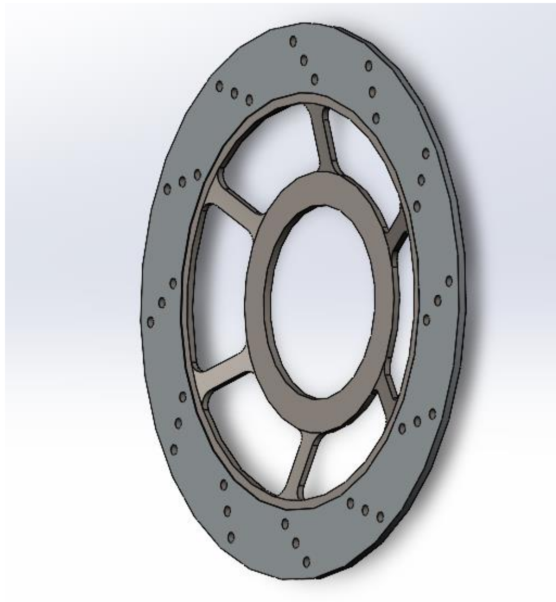
5. Brake and Clutch

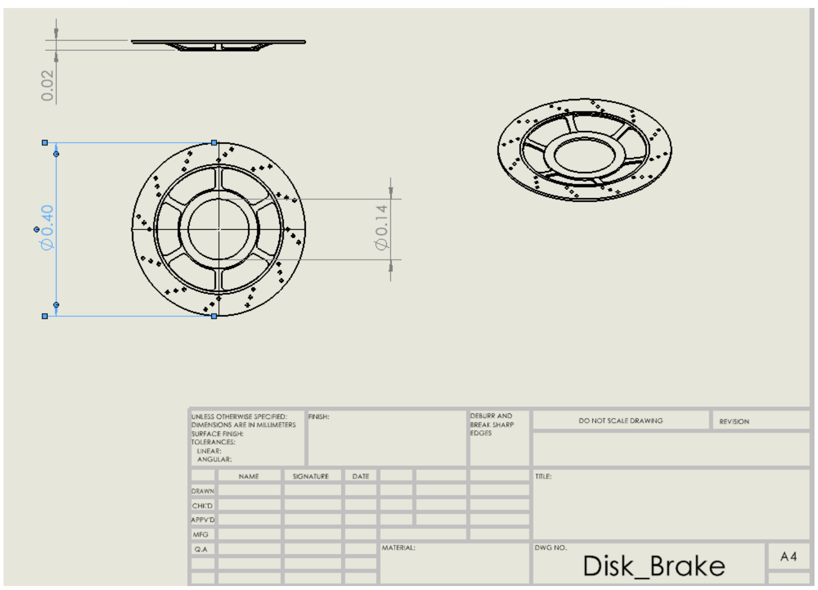
Table 1: Dimensions and Load ratings for Cylindrical Roller Bearings.[10]

Friction material	Coefficient of friction, μ	Maximum contact pressure,[a] p_{max} kPa	Maximum bulk temperature, $t_{m,\ max}$ $^\circ C$
Molded	0.25–0.45	1030–2070	204–260
Woven	0.25–0.45	345–690	204–260
Sintered metal	0.15–0.45	1030–2070	204–677
Cork	0.30–0.50	55–95	82
Wood	0.20–0.30	345–620	93
Cast iron; hard steel	0.15–0.25	690–1720	260

[a] Use of lower values will give longer life.

Table 2: Properties of materials operating dry, when rubbing against cast iron or steel. [4]




0.02
Ø0.40
Ø0.14
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION
NAME
SIGNATURE
DATE
DRAWN
CHK'D
APPV'D
MFG
Q.A
MATERIAL:
TITLE:
DWG NO.
Disk_Brake
A4

6. Final design