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FACULTY OF ENGINEERING

FINITE ELEMENT STRESS ANALYSIS OF THREE STAGE GEAR BOX

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FINITE ELEMENT STRESS ANALYSIS OF THREE STAGE GEAR BOX

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ABSTRACT

Stress Analysis and Modal Analysis of Three Stage Gear Box

The stress analysis of the three-stage gearbox was carried out by finite element analysis method using the Ansys program. The triple reduction helical gearbox was made of AISI 5115 Steel(16MnCr5) and AISI 8620. Static structural and rigid dynamic analyzes were performed in this project. The three-stage helical gearbox was a three stage gearbox transmitting 0.5 kW at 1390 rpm with a reduction ratio of 127.99:1. The boundary conditions for helical gear was performed using the Ansys Program. Rotational velocity is given to rotating gear for static structural analysis. By analyzing these types of gearboxes to be produced, it enables the product to be produced more consciously with optimum data with its safety coefficients. In addition to the analysis, the application areas of this gearbox have been examined. Therefore, stress analysis should be used to find an optimum production.

After the static analysis, dynamic analysis was performed using the modal analysis option of Ansys software. After the mode shape analysis and natural frequency analysis are performed, the operation frequency values of the gearbox are calculated by calculating the input and output rpm values in the gearbox. The fact that these calculated values are much smaller than the values obtained from the modal analysis simulation shows us that there will be no resonance in the gear box design.

INTRODUCTION

In this thesis, it is aimed to analyze the three-stage gearbox and find its parametric strain and stress analysis**.** The components of gear box are gears, shafts, bearings, pins, screws and case. Three stage means, power and speed are reducing at each stage so you get higher torque at each stage, because gear reduction has the opposite effect on torque.

These stresses analyze are important in order to find out which gears and shafts can withstand how much stress and where the weakest area is and to recommend strengthening this area or using stronger strength materials. Because, as everyone knows, a chain is as strong as its weakest ring. At the same time, as well as static analysis, dynamic analysis is important in terms of seeing how much deformity this gear system is at which frequencies. By using modal analysis, it can be observed whether there is resonance in the calculated speed ranges and where the resonance occurs in mode shape analysis.

These analyzes were performed in ANSYS program using finite element method. Static Structural part in the program was selected for the static stress part of the analysis and modal part was selected for the dynamic analysis. The reason for using ANSYS in this work is to find the desired analysis types in a single program and that it is the one of the most reliable and most widely used CAE (computer aided engineering) program.

1.GEAR BOX

1.1 WHAT IS THE GEAR BOX?

Gear box is a closed torque, power and speed transmission system. Gear boxes can be used in many different fields. For example, by connecting to an electric motor or a mechanical motor, it allows us to use the speed, torque and power obtained by reducing or subtracting to the desired level. The following describes various industrial gearboxes and how they are commonly used.

1.2 TYPES OF GEAR BOX

1.2.1 Helical Gear Box

Helical gearboxes have low power consumption and compact structures. This equipment is used for a wide variety of industrial applications, but is typically used in heavy duty operations. Helical gear box is popular in plastic, cement, rubber and other heavy industrial environments. Useful for crushers, extruders, coolers and conveyors, all with low power applications. The helical gearbox is unique in that it is fixed at an angle which allows more teeth to interact in the same direction while on the move. This ensures continuous contact within a given period of time. These boxes are used when torsional stiffness needs to be maximized and for low noise applications.

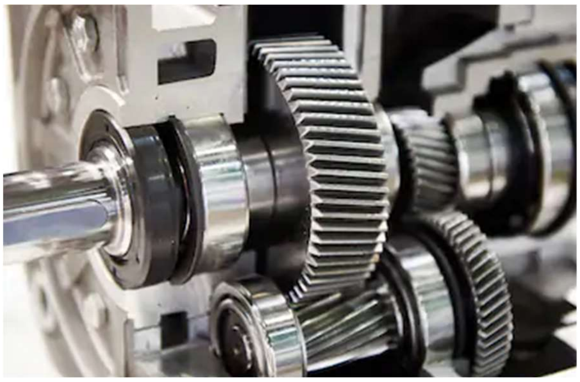

Figure 1.2.1 Helical Gear Box [11]

1.2.2 Bevel Helical Gear Box

The critical feature of this type of transmission is a set of curved teeth located on the cone-shaped surface near the edge of the unit. This box is used to provide rotary motion between non-parallel shafts. They are typically used in the mining industry and conveyors.

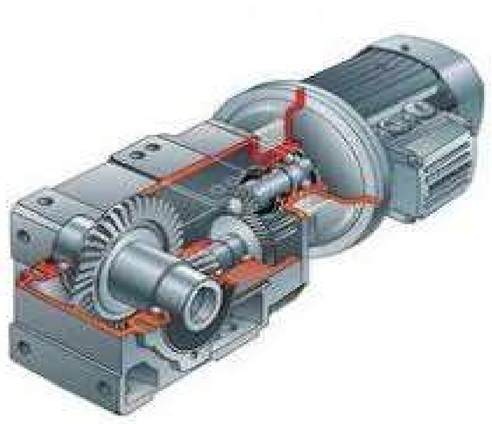

Figure 1.2.2 Bevel Helical Gear Box [12]

1.2.3 Spur Gear Box

Spur gears have straight teeth and are mounted on parallel shafts. Many spur gears are used at the same time to form very large gear reducers. Used in many areas such as screwdriver, washing machine and clock. This gear type makes a lot of noise. Each time a toothed tooth attaches a tooth to the other gear, the teeth collide and the effect makes a sound

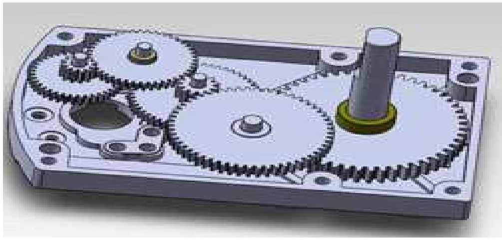

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Figure 1.2.3 Spur Gear Box [13]

1.2.4 Worm Gear Box

Worm gearboxes are used in heavy duty and used to increase speed reduction. The worm or thread connects with the threads in the peripheral area of the gear unit. The rotational movement of the worm causes the wheel to move similarly due to the screw-like motion. These gearboxes are mainly used for heavy duty work such as chemical or gun industry.

Figure 1.2.4 Worm Gear Box [14]

1.2.5 Planetary Gear Box

Planetary gearboxes are durable and sensitive. They are used in areas where durability and precision are important. The use of this type of transmission increases the life of the equipment and keeps its performance at an optimum level

Figure 1.2.5 Planetary Gear Box [15]

1.3 APPLICATION AREAS OF GEAR BOXES

They are used to change speed and torque in mechanical systems with a large number of gears. The gear unit changes the torque ratio of the rotational movement by means of gears. They increase the torque by reducing the engine speed according to the needs of the machines to be used. We can see gear boxes in every area of our lives. If there is machine use in a location and this machine is transmitting power, gearboxes must be used.

Gear wheels, gearboxes, are the most widely used, both in terms of transmittable power and achievable environmental speeds are also mechanisms that have a special place in the mechanisms. In the mechanism consisting of two cogwheels, the gear which is small, that is, the number of teeth, is called pinion and the larger one is called the wheel.

Gearboxes consisting of gear wheels are important components used in a wide range of industries from small scale production factories to elevators, robots, crane systems and automation systems, as well as heavy industrial organizations such as cement-concrete and automotive.

Figure 1.3.1 Automobile Transmission [16]

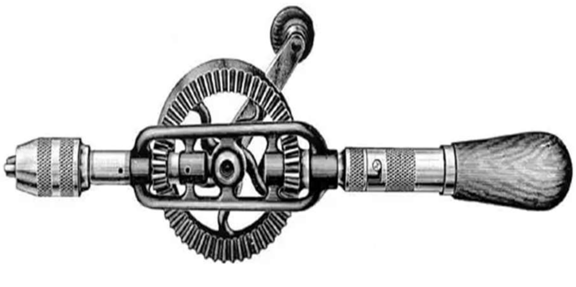

Figure 1.3.2 Hand Drill [17]

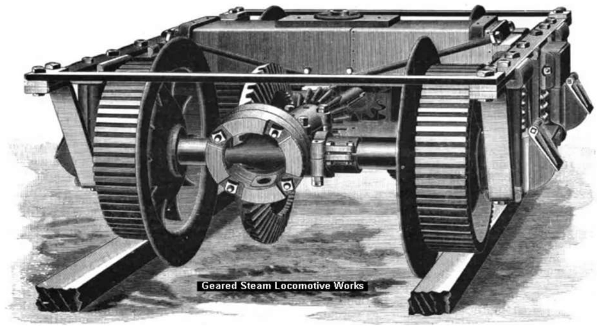

Figure 1.3.3 Locomotive Transmission [17]

1.4 ANALYTICAL CALCULATIONS IN GEAR BOX

In this part, analytical calculations of 3-stage gear box design will be explained. The speed, power and torque transfer calculations of the 6 helical gears given in the table below will be made analytically.

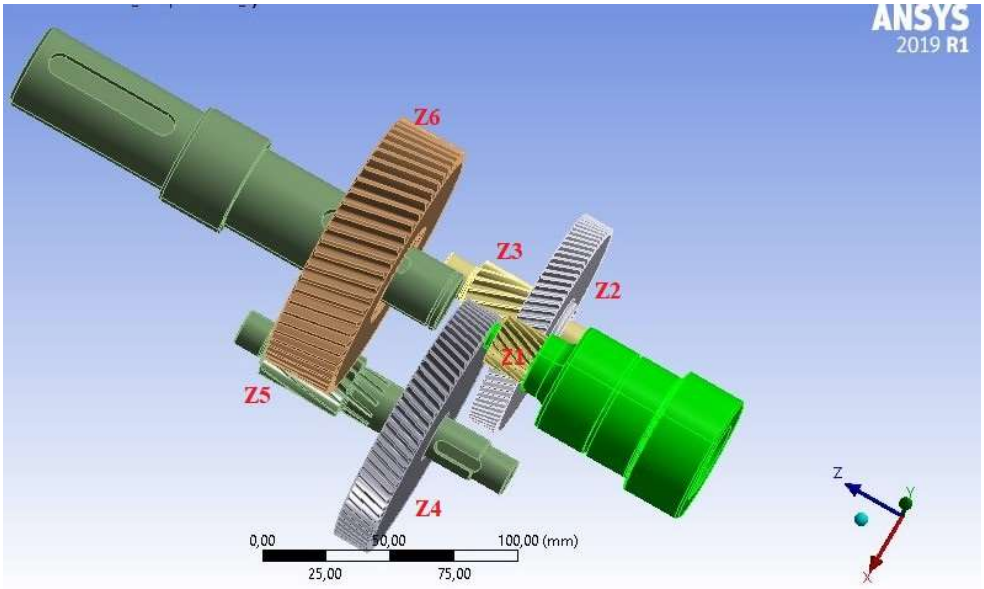

Figure 1.4.1 Gear Box Cad Drawing

Table 1.4.1 Gears' Properties

Drawings of this gearbox taken from IMAK Reducer Variator Industry and Trade Inc.

Gear	Number of Teeth	Normal Module(m)	Pressure Angle	Helix Angle	Helix	Status
Z1	17	1	20°	22°	Right	Returns
Z2	84	1	20°	22°	Left	Returned
Z3	15	1.25	20°	18°	Left	Returns
Z4	79	1.25	20°	18°	Right	Returned
Z5	12	1.75	20°	12°	Right	Returns
Z6	59	1.75	20°	12°	Left	Returned

1.4.1 Speed Calculations

Rotation speed of the motor and gear Z1 is 1390 rpm. Rotational speed varies depending on the number of teeth. Z1 has 17 teeth and Z2 has 84 teeth. Therefore,

N_i=number of teeth in Gear (i)

W_i=Rotational speed in Gear(i)

W_1=1390 rpm

N_1=17

N2=84

$$\mathbf{W_2 = \frac{W_1}{N_2} * N1 \rightarrow W_2 = \frac{1390}{84} * 17 = 281.3\ rev/min}$$

And obtained **W_2=281.3 rpm**

W_3= W_2 (Because Gear 2 and Gear 3 on the same shaft) =281.3 rpm

N_3=15

N_4=79

$$\mathbf{W_4 = \frac{W_3}{N_4} * N3 \Rightarrow W_4 = \frac{281.3}{79} * 15 = 53.41\ rev/min}$$

And obtained **W_4=53.41 rpm**

W_4= W_5 (Because Gear 4 and Gear 5 on the same shaft) =53.41 rpm

N_5=12

N_6=59

$$\mathbf{W_6 = \frac{W_5}{N_6} * N5 \Rightarrow W_6 = \frac{53.41}{59} * 12 = 10.86\ rev/min}$$

And obtained **W_6=10.86 rpm**

Table 1.4.2 Gears' Speed

Gear	Rotational Speed(rpm)	Rotational Speed(rad/s)
Z1	1390	145.56
Z2	281.3	29.45
Z3	281.3	29.45
Z4	53.41	5.59
Z5	53.41	5.59
Z6	10.86	1.13

1.4.2 Power Calculations

Gear Efficiency

For the value of pressure angle α_0=20° (see from Table 1.4.1 Gears' Properties) the efficiency is approximately $\mu_1 = \mu_2 = \mu_3 = 0.89$ taken from references [1]

To calculate power for each gear, the gear efficiency for each stage is assumed as

$\mu_1 = \mu_2 = \mu_3 = 0.89$.

Then $\mu_{total} = \mu_1 \times \mu_2 \times \mu_3 = 0.89 \times 0.89 \times 0.89 = 0.705$ kW

$P_1 = P_{input} = 0.5$ kW

$P_2 = P_3 = P_1 \times \mu_1 = 0.5 \times 0.89 = 0.445$ kW

$P_4 = P_5 = P_2 \times \mu_2 = 0.445 \times 0.89 = 0.396$ kW

$P_6 = P_{output} = P_1 \times \mu_{total} = 0.5 \times 0.705 = 0.352$ kW

Total power losses in reducer = $P_{input} - P_{output} = 0.5 - 0.352 = 0.148$ kW

1.4.3 Torque Calculations

$$I_{12} = \frac{N_2}{N_1} \Rightarrow I_{12} = \frac{82}{17} = 4.82$$

$$W_1 = \frac{2 \times \pi \times n_1}{60} = \frac{2 \times \pi \times 1390}{60} = 145.56 \ \frac{1}{\text{sec}} \quad \text{(Angular velocity of Gear 1)}$$

$$\tau_{Z1} = \frac{P_1}{W_1} = \frac{500}{145.56} = 3.43 \text{ N.m} \quad \text{(Torque for Gear 1)}$$

$$\tau_{Z2} = I_{12} \times \tau_{Z1} \times \mu_1 = 4.82 \times 3.43 \times 0.89 = 14.71 \text{ N.m} \quad \text{(Torque for Gear 2)}$$

$$\tau_{Z2} = \tau_{Z3} = 14.71 \text{ N.m} \quad \text{(Gear 2 and Gear 3 is on same shaft)}$$

$$I_{34} = \frac{N_4}{N_3} \Rightarrow I_{34} = \frac{79}{15} = 5.26$$

$\tau_{Z4} = I_{34} \times \tau_{Z3} \times \mu_2 = 5.26 \times 14.71 \times 0.89 = 68.88\ \mathrm{N.m}$ (Torque for Gear 4)

$\tau_{Z4} = \tau_{Z5} = 68.88\ \mathrm{N.m}$ (Gear 4 and Gear 5 is on same shaft)

$I_{56} = \frac{N_6}{N_5}$ ➔ $I_{56} = \frac{59}{12} = 4.91$

$\tau_{Z6} = I_{56} \times \tau_{Z5} \times \mu_3 = 4.91 \times 68.88 \times 0.89 = 301.00\ \mathrm{N.m}$ (Torque for Gear 6)

$\tau_{input} = 3.43$ Nm

$\tau_{input} = 301.00$ Nm

1.4.4 Stress Control in Stages:

Agma Stress Equation:

The formula used in the calculations section is the agma stress equation. And examined the bending stress analysis in gear.

$$\boldsymbol{\sigma_{max} = W_t K_o K_v K_s \frac{1}{b \times m} \frac{K_H K_B}{Y}}$$

This formula taken from Shigley's Mechanical Engineering Design Book [10]

W_t is the tangential transmitted load (N)

K_o is the overload factor (Taken as 1 because our material is uniform)

K_v is the dynamic factor (Taken as 1 because we examine the statistical analysis)

K_s is the size factor (AGMA has identified and provided a symbol for size factor. Also AGMA suggests K_s =1)

b is the face width of the narrower member, in (mm)

m is the metric module, in (m)

K_H is the load-distribution factor (Taken as 1 our material is uniform)

K_B is the rim-thickness factor (Taken as 1 because our gear backup ratio bigger than 1.2)

Y is the geometry factor for bending stress (which includes root fillet

stress-concentration factor K_f)

Use figure 1.4.4 Forming Factor Table for Y

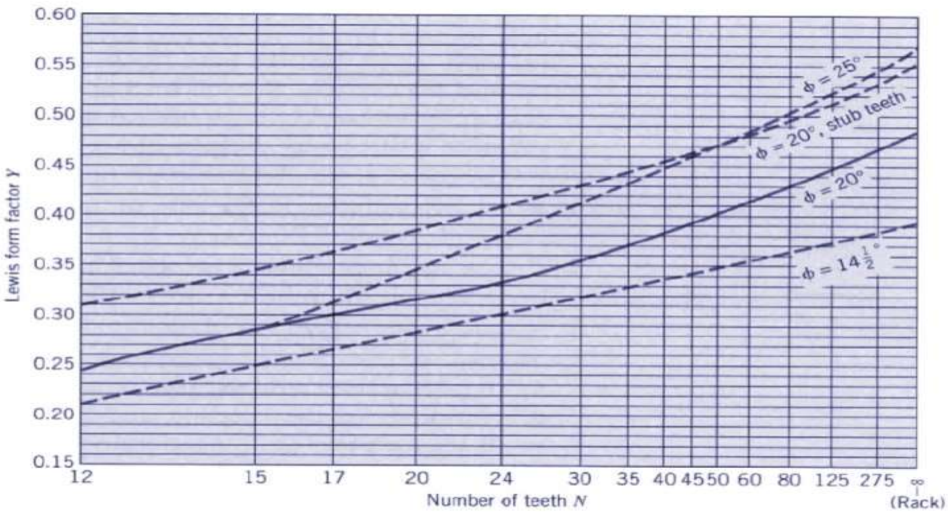

Figure 1.4.4 Forming Factor Table [10]

Therefore, The AGMA stress equation used is;

$$\boldsymbol{\sigma_{max}} = \boldsymbol{W_t} \times \frac{\mathbf{1}}{\mathbf{b} \times \mathbf{m}} \times \frac{\mathbf{1}}{\boldsymbol{Y}}$$

This formula taken from Shigley's Mechanical Engineering Design Book [10]

Table 1.4.4.1

These properties taken from references [18][19]

Properties	AISI 5110(16MnCr5) [18]	AISI 8620[19]
Tensile Strength	880 MPa	1157 MPa
Yield Strength	484 MPa	833 MPa
Modulus of Elasticity	210 GPa	250 GPa
Poisson's Ratio	0.27-0.3	0.29
Hardness Rockwell C	57±2	57±2
Shear Modulus	80 GPa	80 GPa
Density	7850 kg/m^{-3}	7850 kg/m^{-3}

Stage 1 between Z1-Z2

For Gear 1:

$$\sigma_{\max} = W_t \times \frac{1}{\mathrm{b} \times \mathrm{m}} \times \frac{1}{Y}$$

Safety factor is assumed as S=1.5.

Material for gears is selected as AISI 8620.

Properties of AISI 8620 from table 1.4.4.1

σ_t = 1157 N/mm^2 (Tensile Strength)

E = 2.5×10^5 N/mm^2 (Modulus of Elasticity)

σ_{sy} = 833 N/mm^2 (Yield Strength)

$$\sigma_{\mathrm{allow}} = \frac{\sigma_{\mathrm{sy}}}{\mathrm{S}} = \frac{833}{1.5} = 555.33\mathrm{N/mm^2}$$

Pressure angle is selected from table 1.4.1 φ=20°.

Forming factor for φ=20° and N_1= 17 from Figure 1.4.4

$Y=k_{f1} = 0.30$

$\mathrm{d_1 = m \times N_1 = 1 \times 17 = 17}$ mm

$$W_t = \frac{2 \times Torque}{diameter}$$

$$\mathrm{W_t} = \frac{2 \times \tau_{Z1}}{\mathrm{d_1}} = \frac{2 \times 3.43}{17 \times 10^{-3}} = 403.53 \text{ N}$$

$$\sigma_{\max} = W_t \times \frac{1}{\mathrm{b \times m}} \times \frac{1}{Y}$$

Where b= $\varphi \times \mathrm{m} = 20 \times 1 = 20$ mm

$$\sigma_{\max} = 403.53 \times \frac{1}{20 \times 1} \times \frac{1}{0.3} = 67.255 \text{ N/mm}^2$$

$67.255 \leq 555.33$ ➔ $\sigma_{\max} \leq \sigma_{\text{allow}}$ **It's OK.**

For Gear 2:

Safety factor is assumed as S=1.5.

Material for gears is selected as AISI 5110 16MnCr5.

Properties of AISI 5110(MnCr5) from table 1.4.4.1

σ_t = 880 N/mm^2 (Tensile Strength)

E = 2.1×10^5 N/mm^2 (Modulus of Elasticity)

σ_{sy} = 484 N/mm^2 (Yield Strength)

$$\sigma_{\text{allow}} = \frac{\sigma_{\text{sy}}}{\mathrm{S}} = \frac{484}{1.5} = 322.66 \text{ N/mm}^2$$

Pressure angle is selected from table 1.4.1 φ=20°.

Forming factor for φ=20° and N_2= 80 from Figure 1.4.4

$Y = k_{f2} = 0.43$

S=1.5 (Safety Factor)

$d_2 = m \times N_2 = 1 \times 84 = 84$ mm

$$W_t = \frac{2 \times \tau_{Z2}}{d_2} = \frac{2 \times 14.71}{84 \times 10^{-3}} = 350.24 \text{ N}$$

$$\sigma_{max} = W_t \times \frac{1}{b \times m} \times \frac{1}{Y}$$

Where b= $\varphi \times m = 20 \times 1 = 20$ mm

$$\sigma_{max} = 350.24 \times \frac{1}{20 \times 1} \times \frac{1}{0.43} = 40.72 \text{ N/mm}^2$$

$40.72 \leq 322.66 \Rightarrow \sigma_{max} \leq \sigma_{allow}$ **It's OK.**

Stage 2 between Z3-Z4

For Gear 3:

$$\sigma_{max} = W_t \times \frac{1}{b \times m} \times \frac{1}{Y}$$

Safety factor is assumed as S=1.5.

Material for gears is selected as AISI 8620.

Properties of AISI 8620 from table 1.4.4.1

σ_t = 1157 N/mm^2 (Tensile Strength)

E = 2.5×10^5 N/mm^2 (Modulus of Elasticity)

σ_{sy} = 833 N/mm^2 (Yield Strength)

$$\sigma_{\text{allow}} = \frac{\sigma_{\text{sy}}}{S} = \frac{833}{1.5} = 555.33\text{N/mm}^2$$

Pressure angle is selected from table 1.4.1 φ=20°.

Forming factor for φ=20° and N_3= 15 from Figure 1.4.4

$Y=k_{f3} = 0.27$

$$d_3 = m \times N_3 = 1.25 \times 15 = 18.75 \text{ mm}$$

$$W_t = \frac{2 \times Torque}{diameter}$$

$$W_t = \frac{2 \times \tau_{Z3}}{d_3} = \frac{2 \times 14.71}{18.75 \times 10^{-3}} = 1569.1 \text{ N}$$

$$\sigma_{\max} = W_t \times \frac{1}{b \times m} \times \frac{1}{Y}$$

Where b= $\varphi \times m = 20 \times 1.25 = 25$ mm

$$\sigma_{\max} = 1569.1 \times \frac{1}{25 \times 1.25} \times \frac{1}{0.27} = 185.96 \text{ N/mm}^2$$

$185.96 \leq 555.33 \Rightarrow \sigma_{\max} \leq \sigma_{\text{allow}}$ **It's OK.**

For Gear 4:

Safety factor is assumed as S=1.5.

Material for gears is selected as AISI 5110 16MnCr5.

Properties of AISI 5110(MnCr5) from table 1.4.4.1

σ_t = 880 N/mm^2 (Tensile Strength)

E = 2.1×10^5 N/mm^2 (Modulus of Elasticity)

σ_{sy} = 484 N/mm^2 (Yield Strength)

$$\sigma_{\mathrm{allow}} = \frac{\sigma_{\mathrm{sy}}}{\mathrm{S}} = \frac{484}{1.5} = 322.66\ \mathrm{N/mm^2}$$

Pressure angle is selected from table 1.4.1 φ=20°.

Forming factor for φ=20° and N_4= 79 from Figure 1.4.4

$Y=k_{f2}$ = 0.43

S=1.5 (Safety Factor)

$$\mathrm{d_4} = \mathrm{m} \times \mathrm{N_4} = 1.25 \times 79 = 98.75\ \mathrm{mm}$$

$$\mathrm{W_t} = \frac{2 \times \tau_{Z4}}{\mathrm{d_4}} = \frac{2 \times 68.88}{98.75 \times 10^{-3}} = 1395\ \mathrm{N}$$

$$\sigma_{\max} = W_t \times \frac{1}{\mathrm{b} \times \mathrm{m}} \times \frac{1}{Y}$$

Where b= $\varphi \times \mathrm{m} = 20 \times 1.25 = 25\ \mathrm{mm}$

$$\sigma_{\max} = 1395 \times \frac{1}{25 \times 1.25} \times \frac{1}{0.43} = 103.81\ \mathrm{N/mm^2}$$

$103.81 \leq 322.66$ ➔ $\sigma_{\max} \leq \sigma_{\mathrm{allow}}$ **It’s OK.**

Stage 3 between Z5-Z6

For Gear 5:

$$\sigma_{\max} = W_t \times \frac{1}{\mathrm{b} \times \mathrm{m}} \times \frac{1}{Y}$$

Safety factor is assumed as S=1.5.

Material for gears is selected as AISI 8620.

Properties of AISI 8620 from table 1.4.4.1

σ_t = 1157 $\mathrm{N/mm^2}$ (Tensile Strength)

$E = 2.5\times10^5$ N/mm^2 (Modulus of Elasticity)

σ_{sy} = 833 N/mm^2 (Yield Strength)

$$\sigma_{allow} = \frac{\sigma_{sy}}{S} = \frac{833}{1.5} = 555.33\mathrm{N/mm^2}$$

Pressure angle is selected from table 1.4.1 φ=20°.

Forming factor for φ=20° and N_5= 12 from Figure 1.4.4

$Y=k_{f5} = 0.25$

$$d_5 = m \times N_5 = 1.75 \times 12 = 21 \text{ mm}$$

$$W_t = \frac{2 \times Torque}{diameter}$$

$$W_t = \frac{2 \times \tau_{Z5}}{d_5} = \frac{2 \times 68.88}{21 \times 10^{-3}} = 6560 \text{ N}$$

$$\sigma_{max} = W_t \times \frac{1}{b \times m} \times \frac{1}{Y}$$

Where $b= \varphi \times m = 20 \times 1.75 = 35$ mm

$$\sigma_{max} = 6560 \times \frac{1}{35 \times 1.75} \times \frac{1}{0.25} = 428.41\mathrm{N/mm^2}$$

$428.41 \leq 555.33 \Rightarrow \sigma_{max} \leq \sigma_{allow}$ **It's OK.**

For Gear 6:

Safety factor is assumed as S=1.5.

Material for gears is selected as AISI 5110 16MnCr5.

Properties of AISI 5110(MnCr5) from table 1.4.4.1

σ_t = 880 N/mm^2 (Tensile Strength)

$E = 2.1 \times 10^5$ N/mm^2 (Modulus of Elasticity)

σ_{sy} = 484 N/mm^2 (Yield Strength)

$$\sigma_{\text{allow}} = \frac{\sigma_{\text{sy}}}{\text{S}} = \frac{484}{1.5} = 322.66 \text{ N/mm}^2$$

Pressure angle is selected from table 1.4.1 φ=20°.

Forming factor for φ=20° and N_6= 59 from Figure 1.4.4

$Y = k_{f6} = 0.42$

S=1.5 (Safety Factor)

$$\text{d}_6 = \text{m} \times \text{N}_6 = 1.75 \times 59 = 103.25 \text{ mm}$$

$$\text{W}_\text{t} = \frac{2 \times \tau_{\text{Z6}}}{\text{d}_6} = \frac{2 \times 301}{103.25 \times 10^{-3}} = 5830.51\text{N}$$

$$\sigma_{\text{max}} = W_t \times \frac{1}{\text{b} \times \text{m}} \times \frac{1}{Y}$$

Where b= $\varphi \times \text{m} = 20 \times 1.75 = 35$ mm

$$\sigma_{\text{max}} = 5830.51 \times \frac{1}{35 \times 1.75} \times \frac{1}{0.42} = 226.64 \text{ N/mm}^2$$

$226.64 \leq 322.66 \Rightarrow \sigma_{\text{max}} \leq \sigma_{\text{allow}}$ **It's OK.**

2.FINITE ELEMENT METHOD

2.1 WHAT IS FINITE ELEMENT METHOD?

In this section, it is aimed to review the Finite Element Method and explain in detail. The finite element method is explained in more detail in this section, analogical studies, thesis and articles using this method are compiled below.

The finite element method is a numerical technique for dividing any physical. phenomenon into pieces and analyzing those pieces separately and simultaneously. Engineers use it to reduce the number of experiments, improve products faster and analyze them faster. One of the most important benefits of using the finite element method is that it offers great freedom in the choice of appreciation for elements that can be used to separate both the space and the basic functions.

2.2 LITERATURE SURVEY

Klaus-Jurgen Bathe [2] described that the Finite Element Procedures. FEA is now an important part of engineering analysis and design. Finite element analysis is practical application for the analysis all of physical phenomenon. Designers use it to minimize the number of physical prototypes so it can be easier to solve and to improve the design in minimum time by optimizing the components.

Dan Yang et. al [3] using the finite element method in ANSYS software studied the dynamic characteristics of the brake drum and theoretically calculated. After the analysis and calculation of the model, the natural frequencies and vibration shapes are computed.

Balasaheb Sahebrao Vikhe [4] improved and optimized the efficiency of the design process based on static and dynamic modal analysis results. recommends modifying and developing the geometric model and maintaining the optimization until satisfactory results are obtained. This helps to find an optimized design for the transmission case, where it performs best with minimal loads on the housing.

Vijaykumar, Mr. Shivaraju, [5] by using finite element analysis in ANSYS software, determined the vibration analysis of the transmission case using a harmonic frequency response for the case to prevent resonance. In order to prevent resonance, the frequency ratio should be set to 0.25 from the first modal natural frequency.

Ramamurti V, [6] made comparison of stress results obtained from classical method and FEM method and dynamic analysis of the model. The results of this analysis determined the deflection of the shaft under the influence of gear forces by using FEM and classical methods.

Devan P D, V.R. Muruganantham, [7] stated in his article that we assume that the resonance will not take place on gear train if the natural frequencies found as a result of modal analysis are much higher than the operation frequency calculated using the input and output rpm values of the gear system.

Isa Yesilyurt et. al, [8] measured the reduction of stiffness of gear teeth with the help of modal analysis. In addition, with the help of Modal analysis, he determined that the gear tooth can be detected as damage and wear damage. In order to prove these analyzes in a physical environment, an experiment was made to obtain the FRF of the gear tooth. The steel-tipped impact hammer used as a stimulator and accelerometer acts as a response detector.

Peter Weis [9] et.al tried to calculate the first 20 natural frequencies and corresponding mode shapes in the modal analysis in the Ansys workbench.

3-FEM ANALYSIS OF THREE STAGE GEARBOX

3.1 CAD DESIGN OF THREE STAGE GEAR BOX

The drawing of the 3-stage gearbox designed within the scope of this project in computer environment. The design has 6 gears, 3 shafts ,6 keys and rotor engine. For static analysis it is examined the tensile forces between the two gears in the stage parts and also key, gear and shaft located on the same shaft.

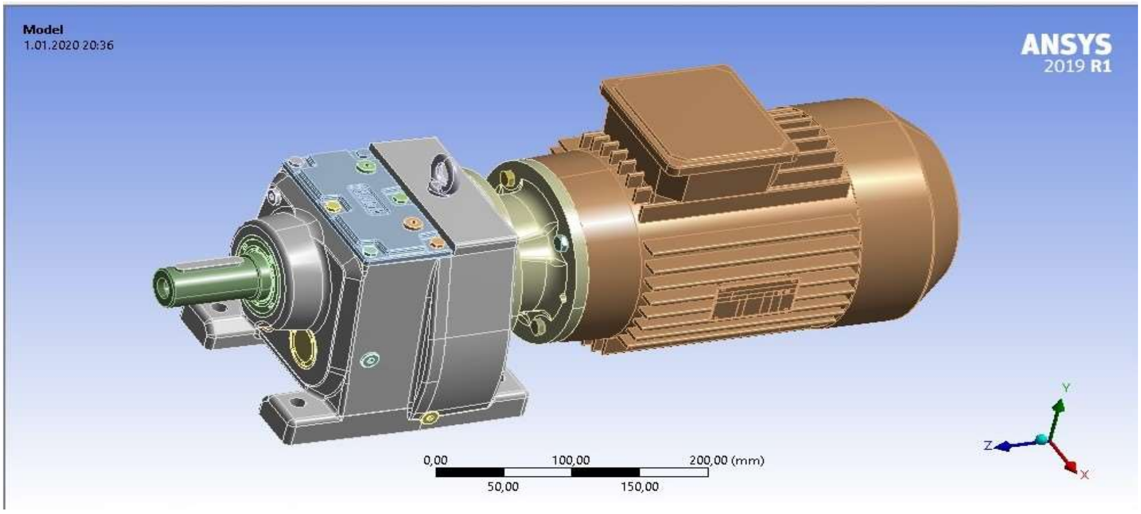

Figure 3.1.1 Whole Design of Three Stage Gear Box

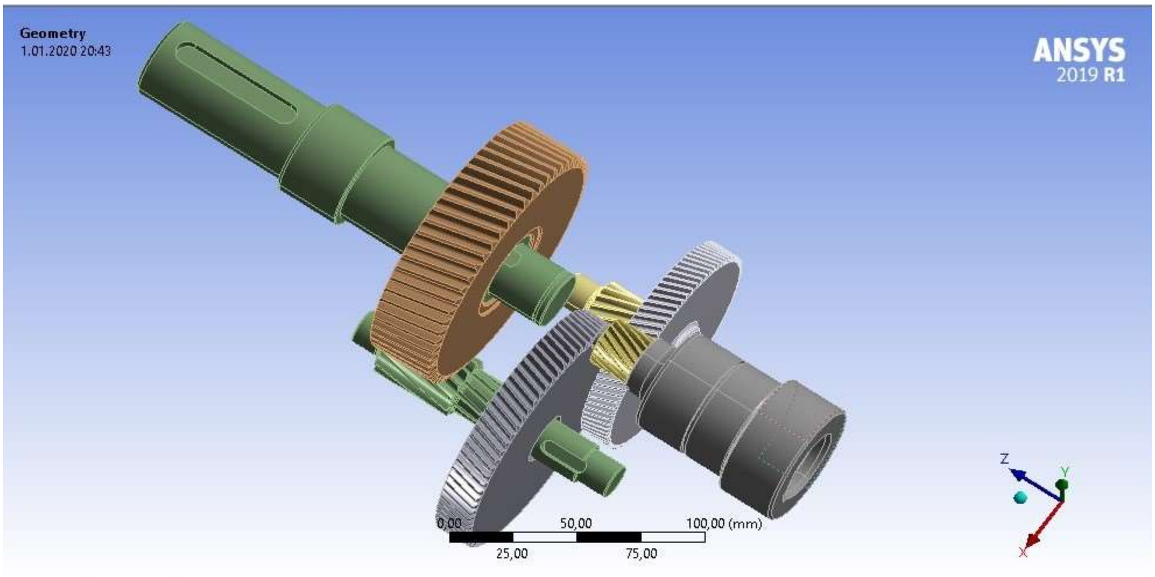

Figure 3.1.2 Interest Parts of Whole Design

Input Shaft-Z1 Gear Relationship

In this section, torque transfer occurs between the electric motor and the Z1 gear by using of a key. Input shaft and Z1 gear have a power of 0.5 kW. And these two parts rotate together at 1390 rpm.

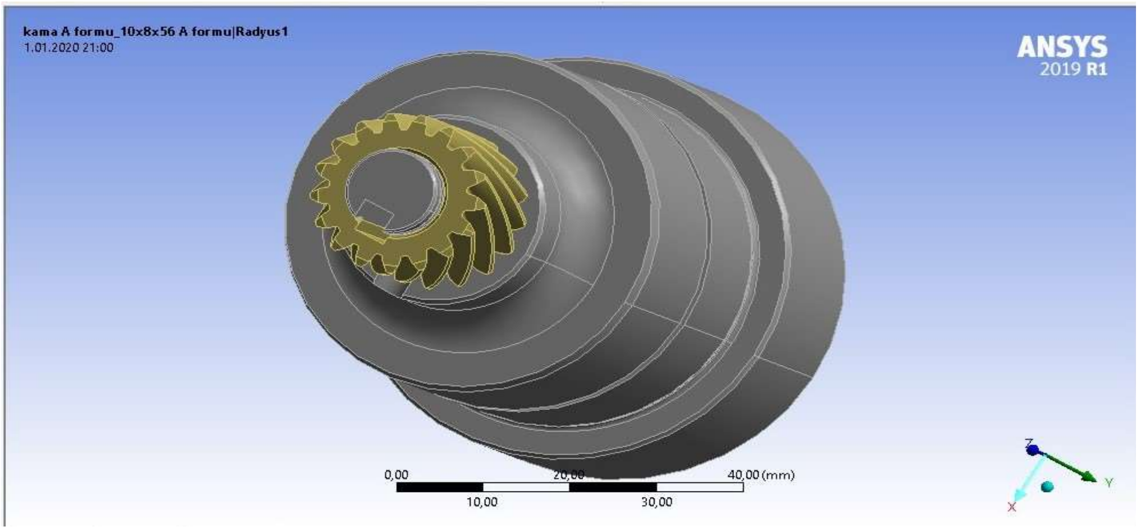

Figure 3.1.3 Rotor Engine Input Shaft-Z1

Z1-Z2 Gear Relationship (First Reduction Stage)

Z1 gear 17 and Z2 gear have 84 teeth. These part is the first reduction stage. Reduction rate found in the ratio of the number of teeth. Reduction rate is $\frac{17}{84} = 0.202$.Therefore, Gear Z2 rotates at 281.3 rpm while gear Z1 rotates at 1390 rpm

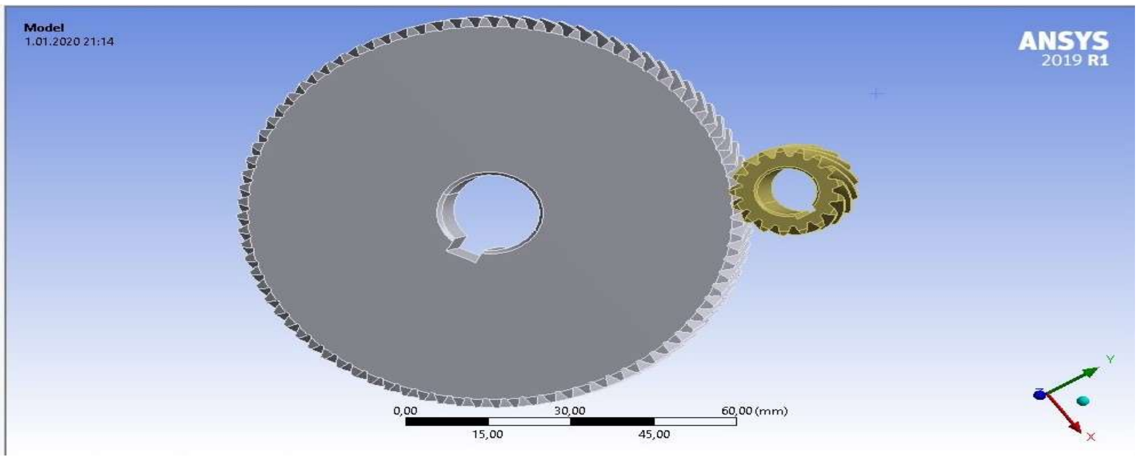

Figure 3.1.4 Z1 and Z2 gears

Z2-Z3 Gears and Key

Gear Z3 rotates by taking the torque generated by the Z2 using a key. Rpm is the same as Z2 because they are in the same shaft. They rotate with 281.3 rpm.

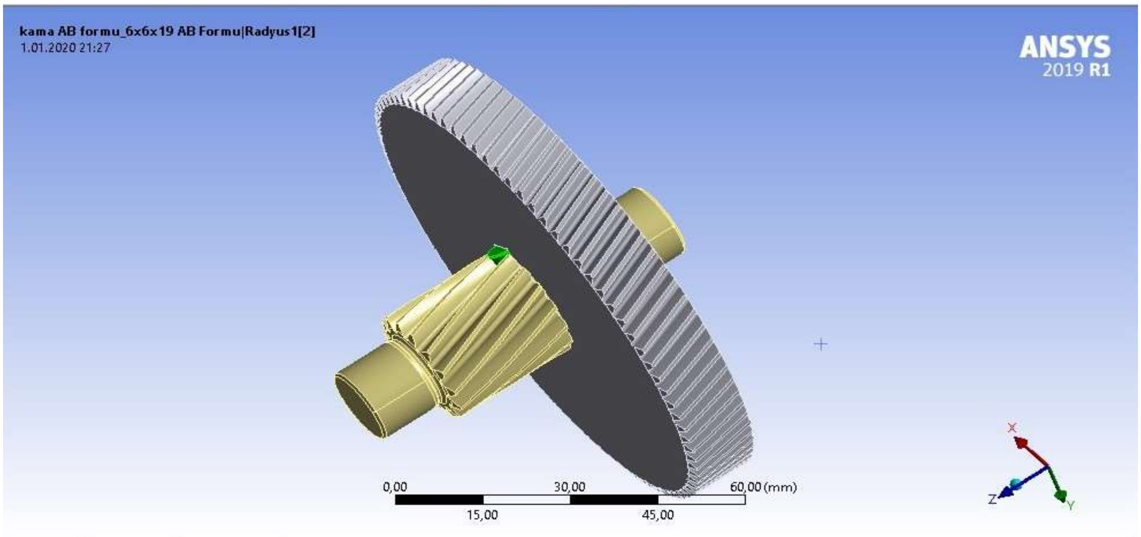

Figure 3.1.5 Z2-Z3-Key

Z3-Z4 Gears Relationship (Second Reduction Stage)

This stage is the second reduction stage. Z3 gear 15 and Z4 gear have 79 teeth. Reduction rate found in the ratio of the number of teeth. Reduction rate is $\frac{15}{79} = 0.19$. Therefore, Gear Z4 rotates at 53.46 rpm while gear Z3 rotates at 281.3 rpm.

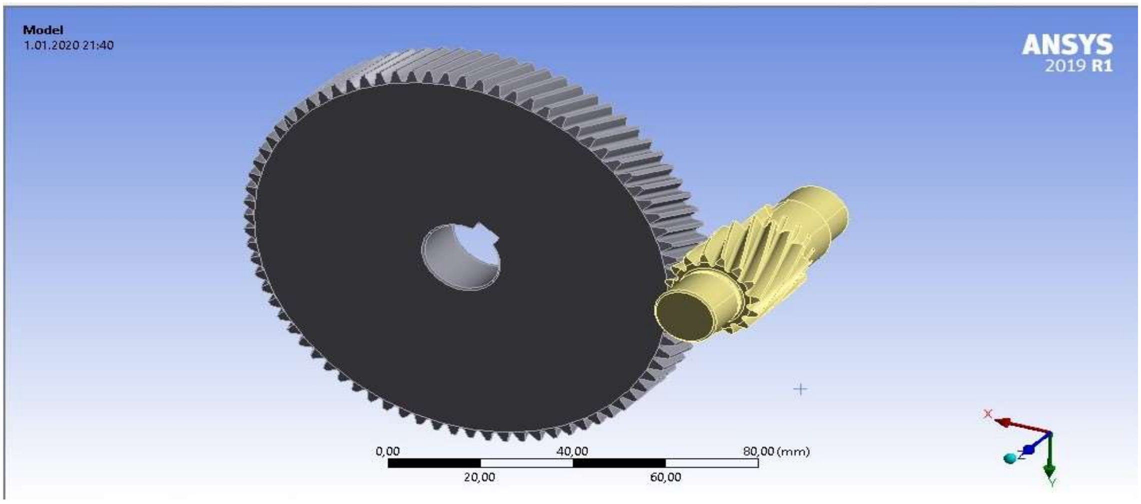

Figure 3.1.6 Z3-Z4 Gears

Z4-Z5 Gears and Key

Gear Z5 rotates by taking the torque generated by the Z4 using a key. Rpm is the same as Z4 because they are in the same shaft. They rotate with 53.46 rpm.

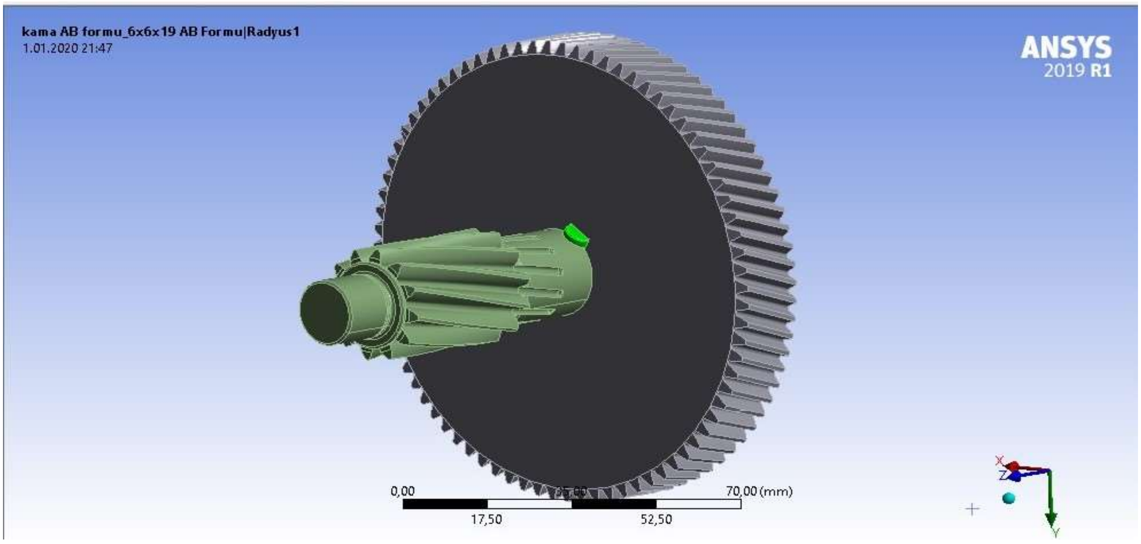

Figure 3.1.7 Z4-Z5 Gears and Key

Z5-Z6 Gears Relationship (Third Reduction Stage)

This stage is the second reduction stage. Z5 gear 12 and Z6 gear have 59 teeth. Reduction rate found in the ratio of the number of teeth. Reduction rate is $\frac{12}{59} = 0.203$. Therefore, Gear Z6 rotates at 10.87 rpm while gear Z5 rotates at 53.46 rpm.

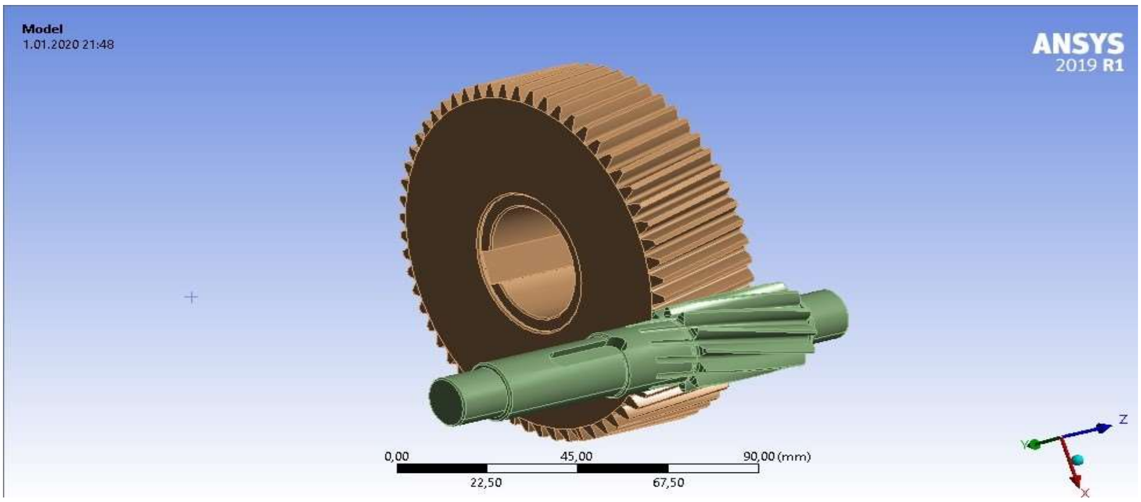

Figure 3.1.8 Z5-Z6 Gears

Z6-Output Shaft and Key

Torque transfer occurs between the Z6 and output shaft by using of a key. Output shaft and Z6 gear have a power of 0.352 kW as mentioned power calculations part. And these two parts rotate together at 10.87rpm. Engine produced 1390 rpm and 10.87 rpm turns are taken from the output shaft so total reduction rate is 127.99:1.

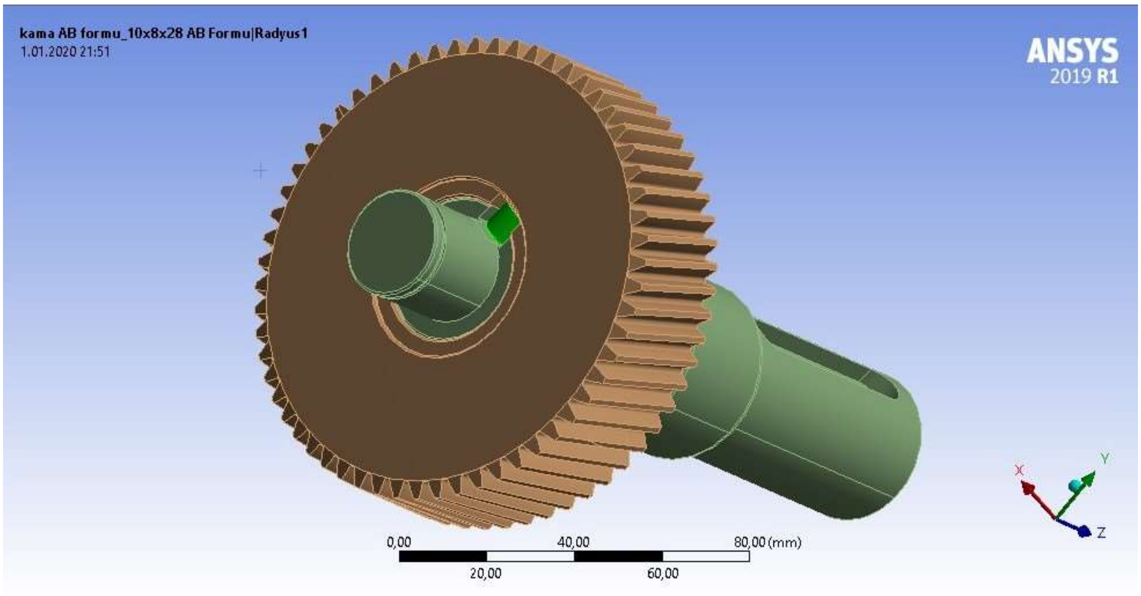

Figure 3.1.9 Z6-Output Shaft

3.2 BOUNDARY CONDITIONS

3.2.1 Materials

The Ansys program begins by defining materials. First the materials to be used must be written in the engineering data section. 3 materials were used in this project. These materials are AISI 1050 Steel for input and output shaft, AISI 8620 for gears with pitch diameter less than 50 mm (Gear 1,3 and 5) and AISI 5115 MnCr5 for gears with pitch diameter bigger than 50 mm (Gear 2,4 and 6). Structural steel is used for keys, which is the default option of the program. After entering engineering data tab, add material and change the data of structural steel option and write the properties of the desired material according to the material in the design.

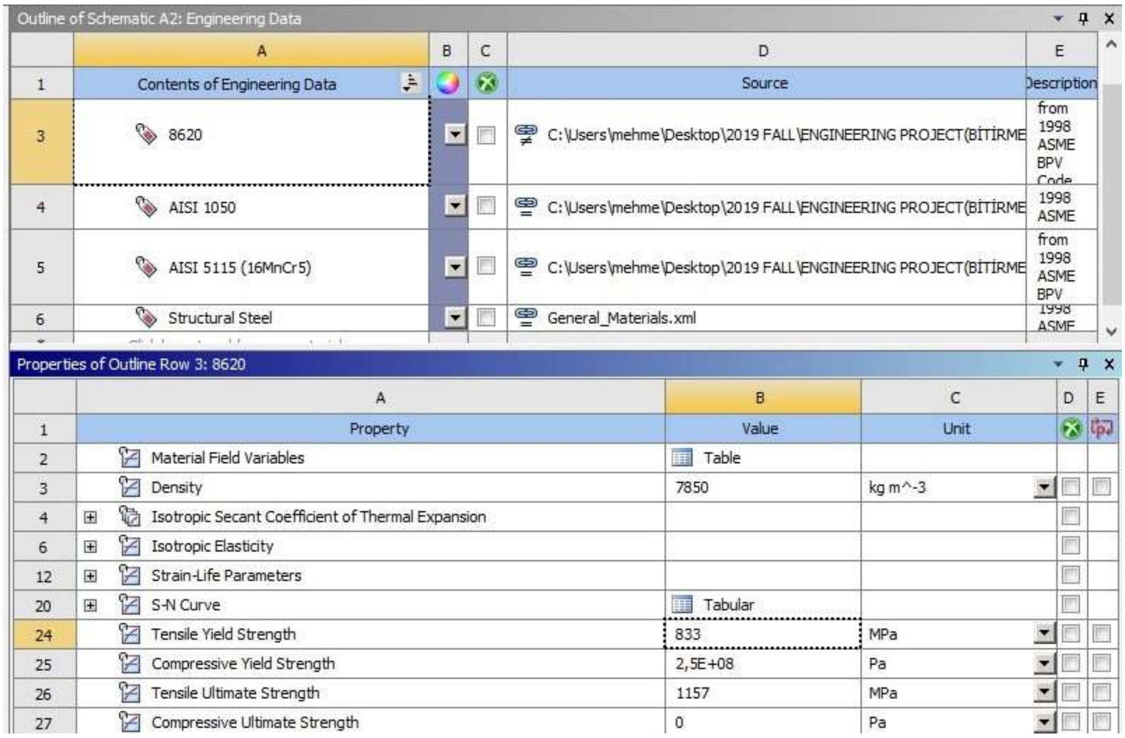

Figure 3.2.1.1 Materials Assigned in Engineering Data

After the material assignment is made, the model option is entered and the boundary conditions begin to be determined.

Figure 3.2.1.2

The boundary conditions are determined in the order given in the figure. After assigning the material, coordinate system is created. You must create coordinate system according to the process you will use. In this project, spherical mesh body sizing has been used as spherical shape in order to make more mesh in specific regions. The coordinate system in which the spherical shape will be formed is entered in the boundary conditions section. This process will be explained in detail in the mesh section. After this, the contacts between the gear, key and shaft will be determined. The part that should be considered while giving contact is whether the parts will move together or separately.

3.2.2 Contacts Boundary Conditions

There is no separation contact between the rotor shaft and gear Z1 because these parts will move separately. Besides that, the contact between key and shaft must be bonded because these parts will move together.

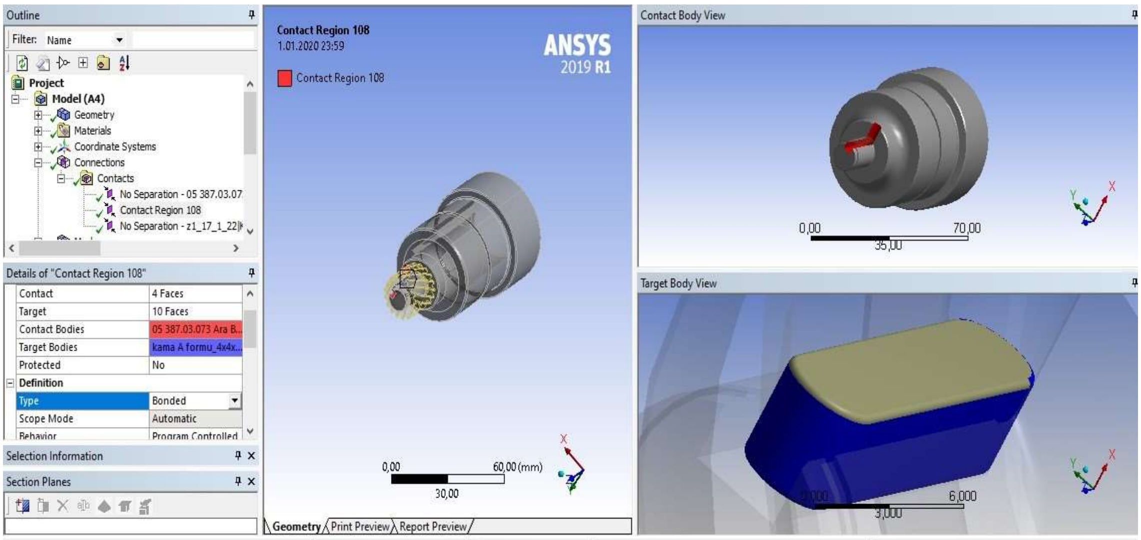

Figure 3.2.2.1 Bonded Contact Between Key and Shaft

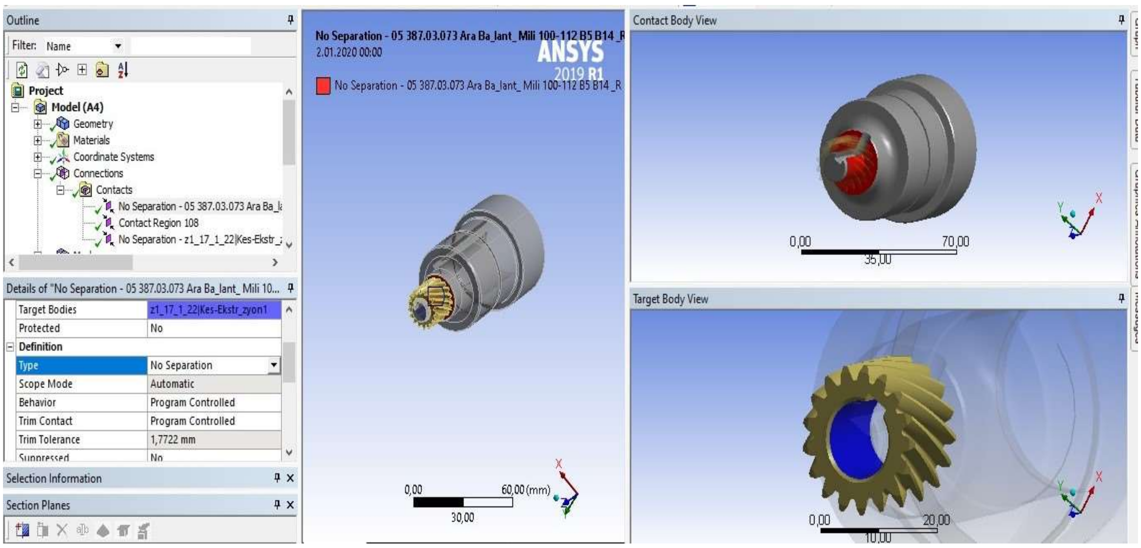

Figure 3.2.2.2 No Separation Contact Between Gear and Shaft

As stated above, contact boundary conditions can be given in Shaft, key and gear analyzes. In the analysis of stresses between 2 gears, boundary conditions are given as follows.

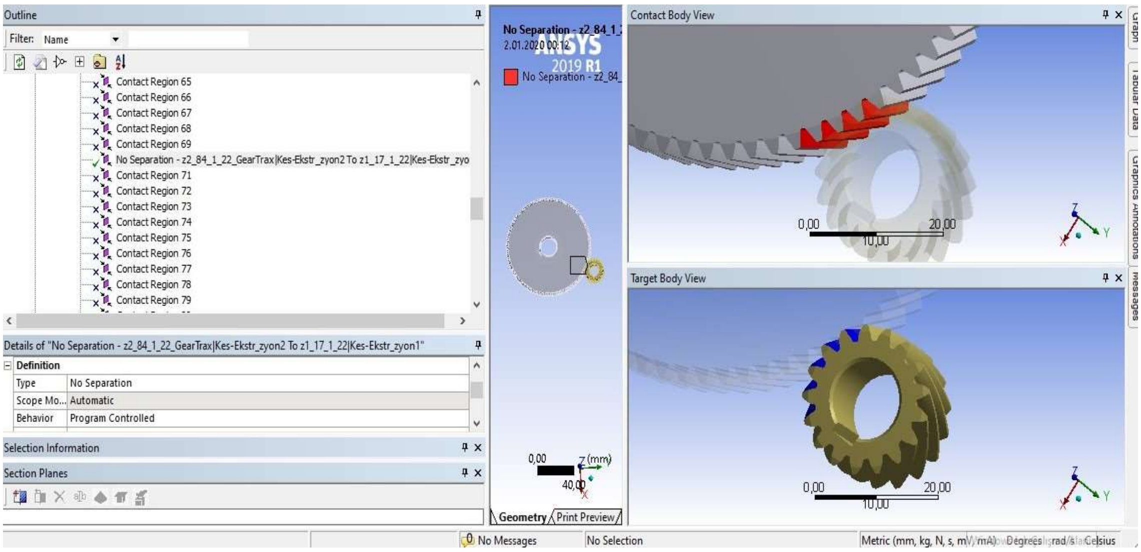

Figure 3.2.2.3 No Separation Contact Between Gears

No separation option should be selected since gears cannot rotate same direction.

3.3 MESH

Mesh generation is one of the most important part in Finite Element Method. Mesh generation can be defined as the process of dividing a physical definition range into smaller definition ranges (elements). The aim is to simplify the solution of a differential equation. After the element properties are determined, the model is divided into small elements. So the model is meshed. The important thing here is how to better split the model into smaller pieces using the selected element.

At the beginning, default mesh is performed for gears. But the small number of elements limited the accuracy of the results. So, the accuracy of the solution was increased by increasing the number of meshes and nodes in the sensible areas where the stress is increased.

3.3.1 Creating Body Sizing

First, sphere-shaped body sizing was determined to the region where we expected the most stress. Outside of the spherical region that we determined to be seen in the figure, mesh is less fine. In addition, because the mesh element in the spherical region is more, it has become a finer mesh so that it can obtain the results more accurately.

To create a body sizing mesh, a new coordinate was created from the coordinate system and its spherical shape was created according to that coordinate system. considering this shape, spherical body radius is 10 mm and mesh element size is 1 mm. Mesh element size may be taken smaller but it may take more time to solve the process according to the quality of the computer.

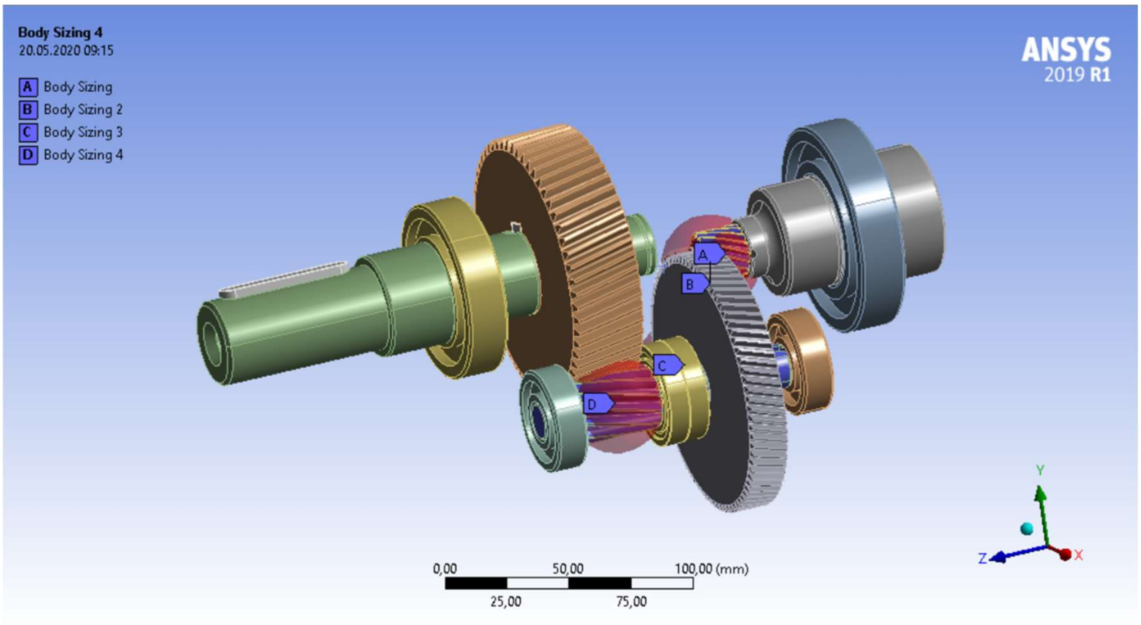

Figure 3.3.1 Body Sizing Mesh

As a result, these operations were repeated in all areas between gear-gear and gear-wedge stress. If you think where the stress may be more, making a finer mesh in those areas will not only tire the computer as a solution speed, and will also provide you with more accurate results.

3.4 FEM ANALYSIS LOAD CAD DESIGN ARRANGEMENT

After contacts boundary conditions are defined and mesh is generated, load application(moment) is made.

3.4.1 Load Application Respectively

In order to see better results, moments are applied to each gear and the other parts of model is suppressed.

To be able to do this properly, firstly one of the gear or shaft is fixed and then the other gear is given moment. All moments are applied on the Z direction.

Input Shaft-Z1 Gear Relationship

As stated on the section (**1.4.3** Torque Calculations), τ_{Z1}=3,43 N.m is applied to the faces of (see red zone) gear Z1 and input shaft is fixed.

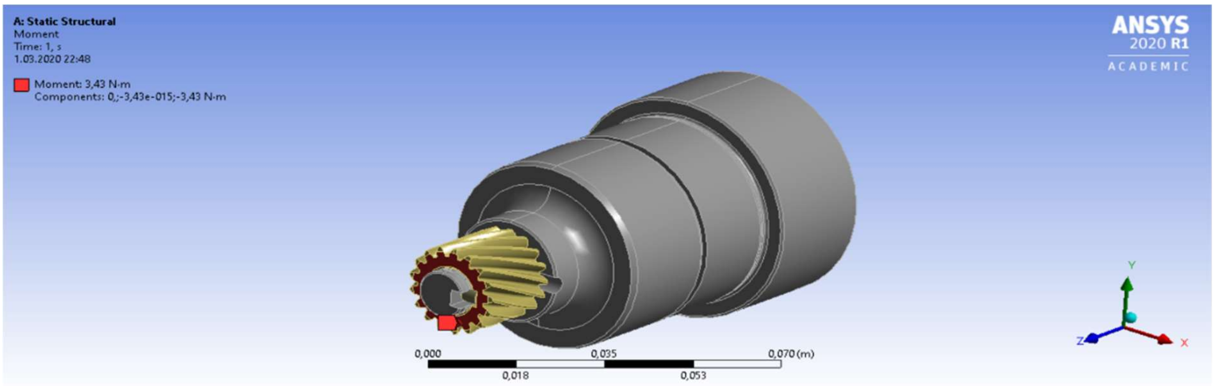

Figure 3.4.1 Rotor Engine Input Shaft-Z1

Z1-Z2 Gear Relationship (First Reduction Stage)

For gears Z1 and Z2, τ_{Z1}=3,43 N.m is applied to the inner faces of gear Z1 and gear Z2 is fixed. After that, to see better results, gears Z1 and Z2 are switched and τ_{Z2}= 14,71 N.m is applied to faces of gear Z2 and Z1 is fixed.

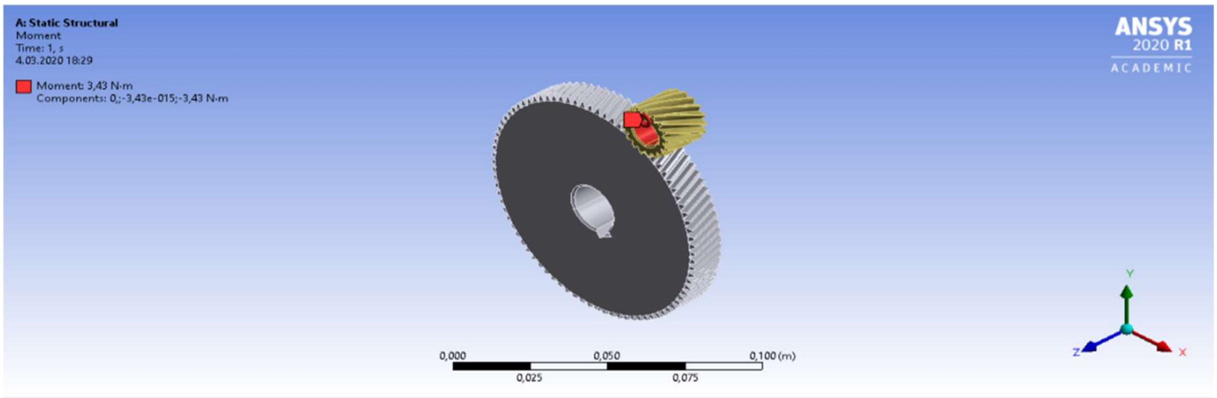

Figure 3.4.2 Z1 and Z2 gears (z2 fixed)

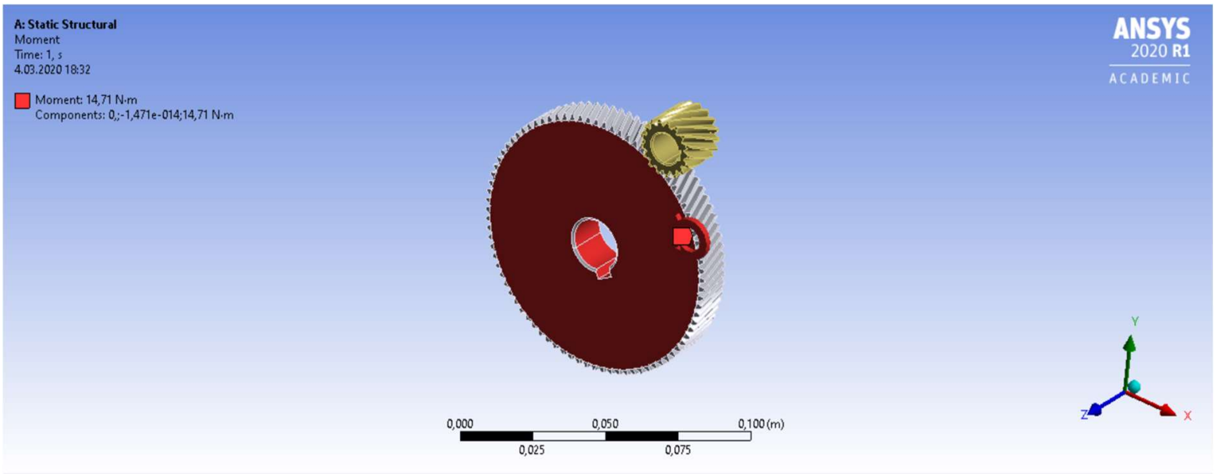

Figure 3.4.3 Z1 and Z2 gears (z1 fixed)

Z2-Z3 Gears and Key

τ_{Z2}=14,71 N.m is applied to the gear Z2 and gear Z3 is fixed.

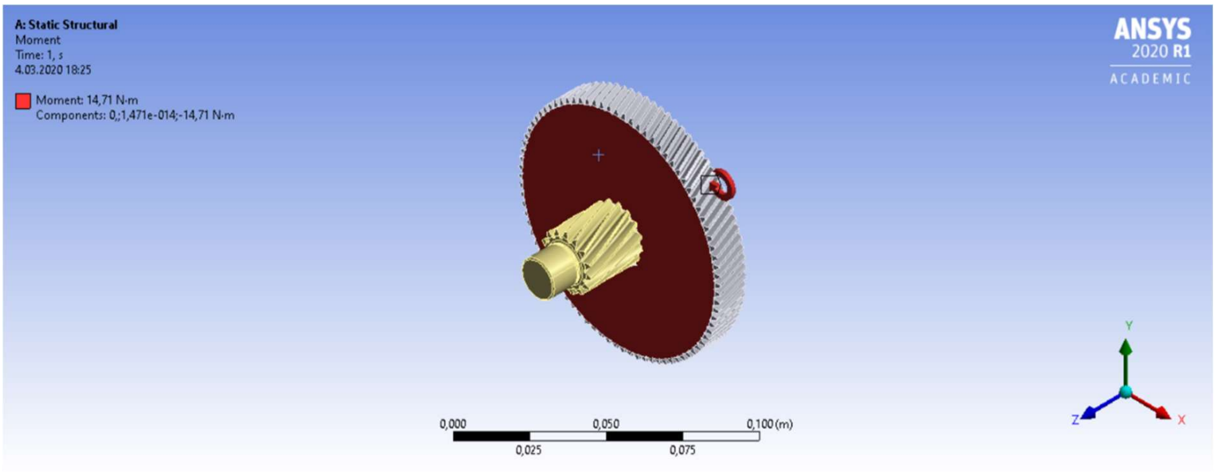

Figure 3.4.4 Z2-Z3 gears-key

Z3-Z4 Gears Relationship (Second Reduction Stage)

To see the better deformations on gears Z3 and Z4, first τ_{Z3}=14,71 N.m is applied to gear Z3 and gear Z4 is fixed. Then τ_{Z4}=68,88 N.m is applied to gear Z4 and gear Z3 is fixed.

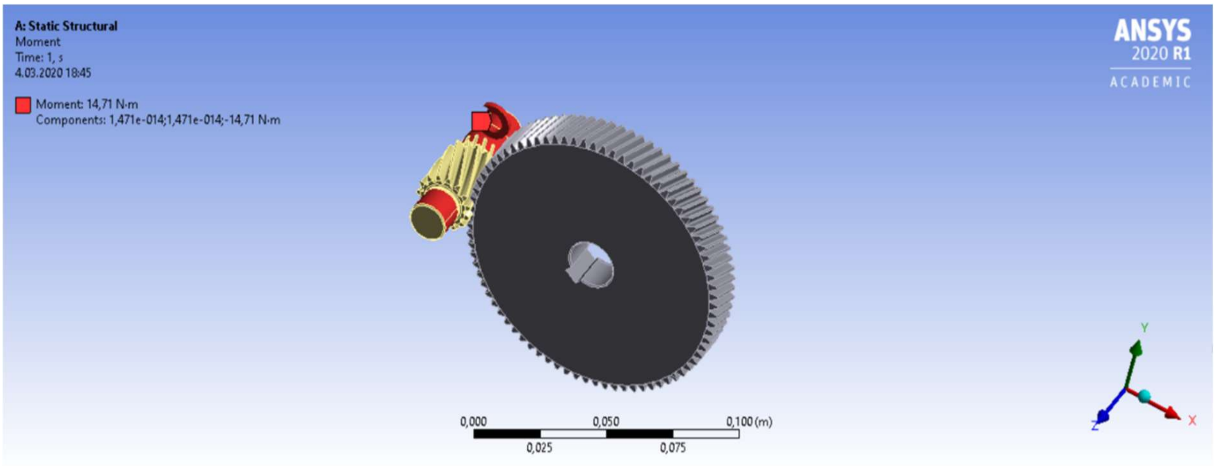

Figure 3.4.5 Z3-Z4 gears (z4 fixed)

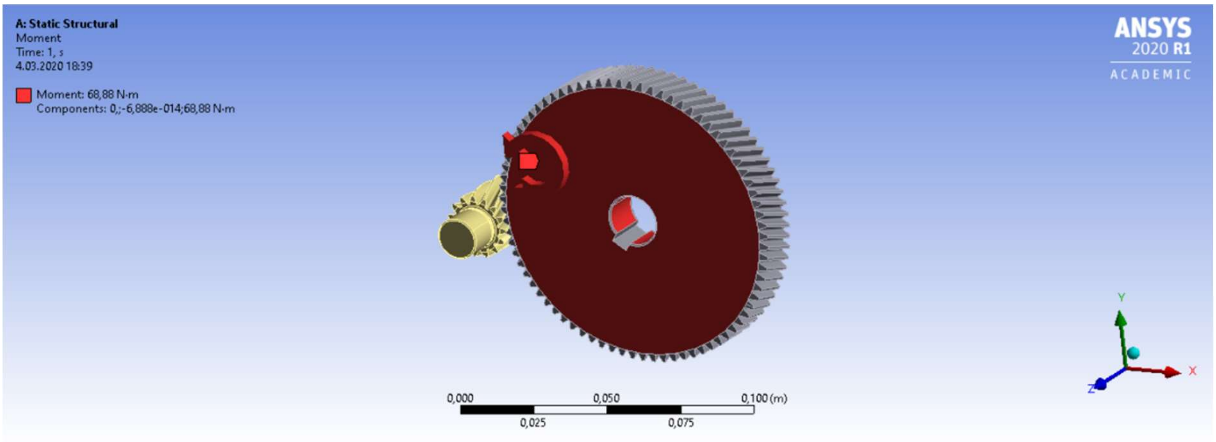

Figure 3.4.6 Z3-Z4 gears (z3 fixed)

Z4-Z5 Gears and Key

τ_{Z4}=68,88 N.m is applied to the gear Z4 and gear Z5 is fixed.

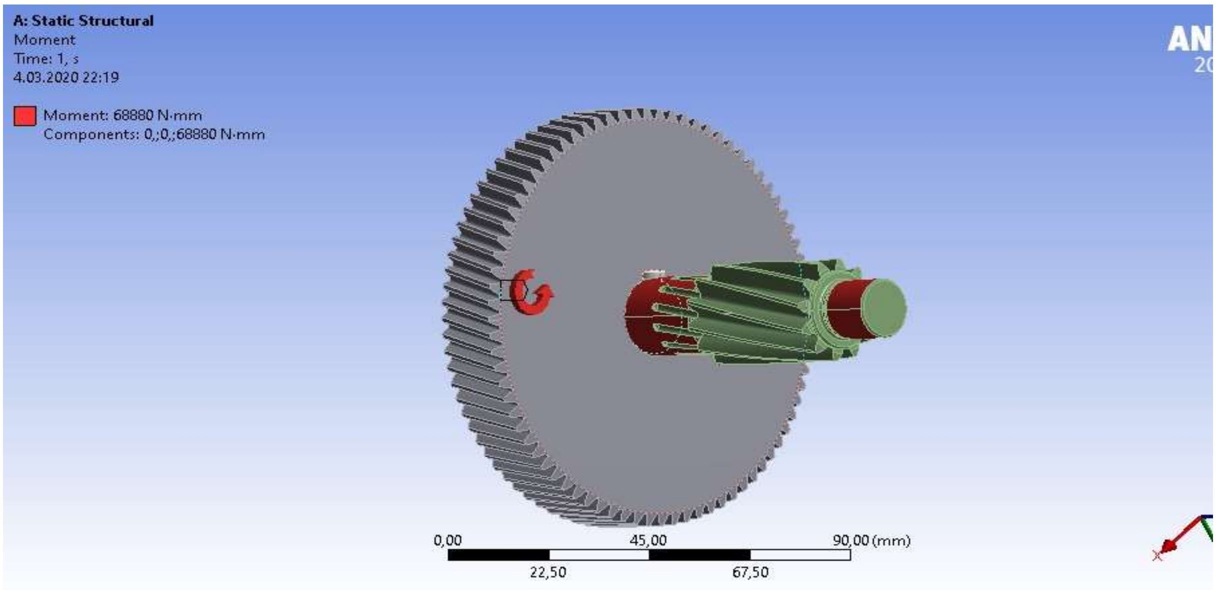

Figure 3.4.7 Z4-Z5 gears

Z5-Z6 Gears Relationship (Third Reduction Stage)

Similar to Z5-Z6, to see the better deformations on gears Z5 and Z6, first τ_{Z5}=68,88 N.m is applied to gear Z5 and gear Z6 is fixed. Then τ_{Z6}=301 N.m is applied to gear Z6 and gear Z5 is fixed.

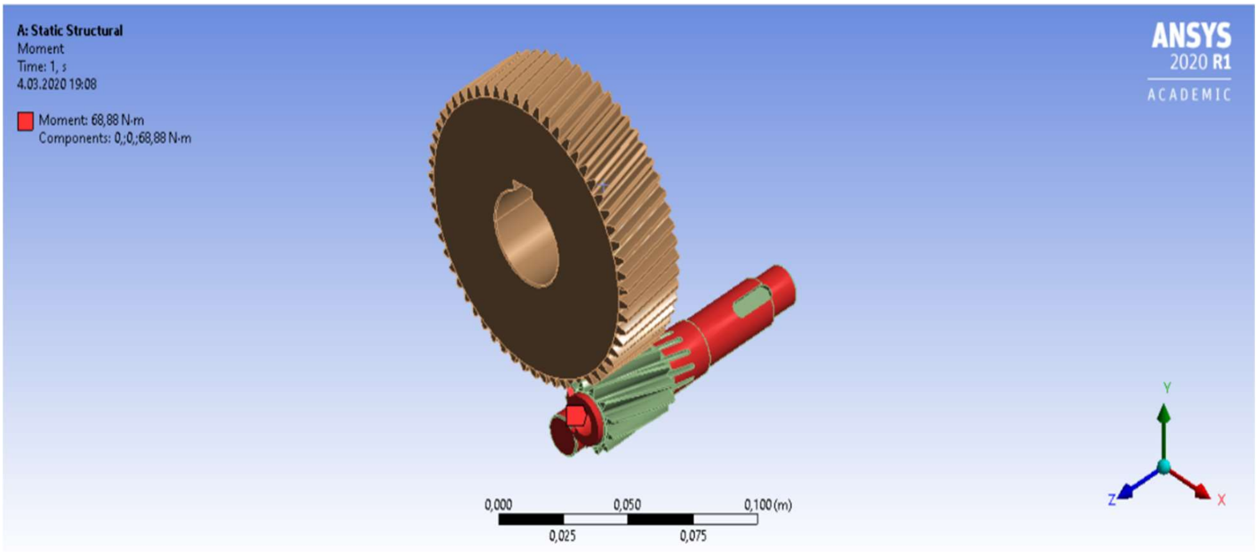

Figure 3.4.8 Z5-Z6 gears (Z6 fixed)

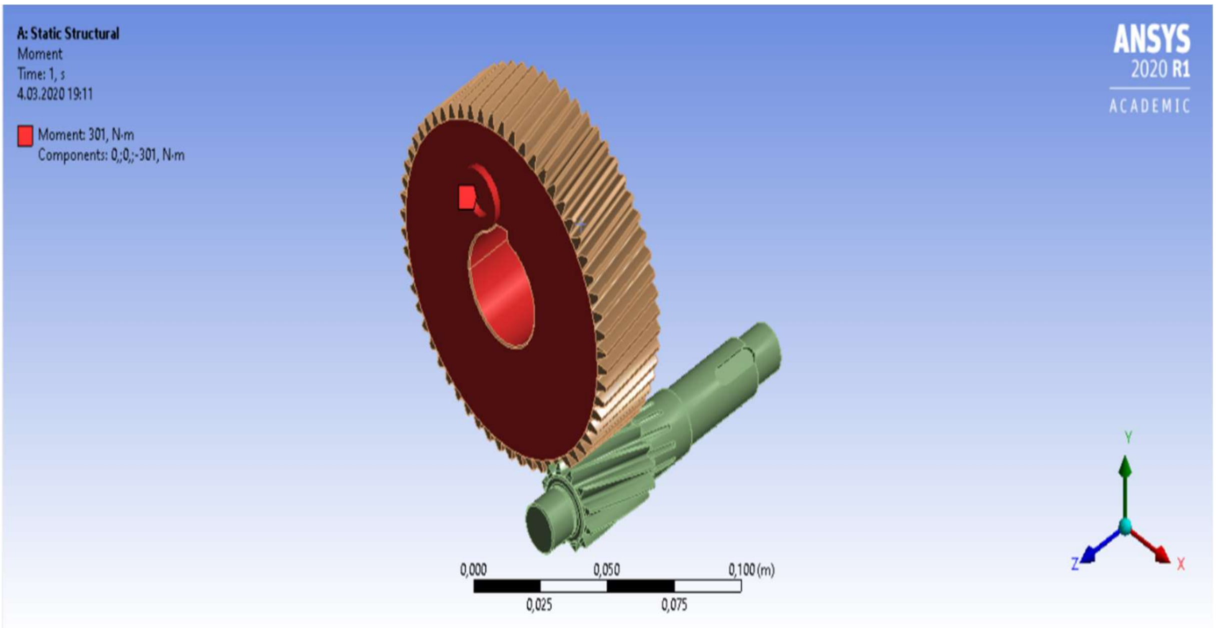

Figure 3.4.9 Z5-Z6 gears (Z5 fixed)

Z6-Output Shaft and Key

Finally, τ_{Z6}=301 N.m is applied to gear Z6 and output shaft is fixed.

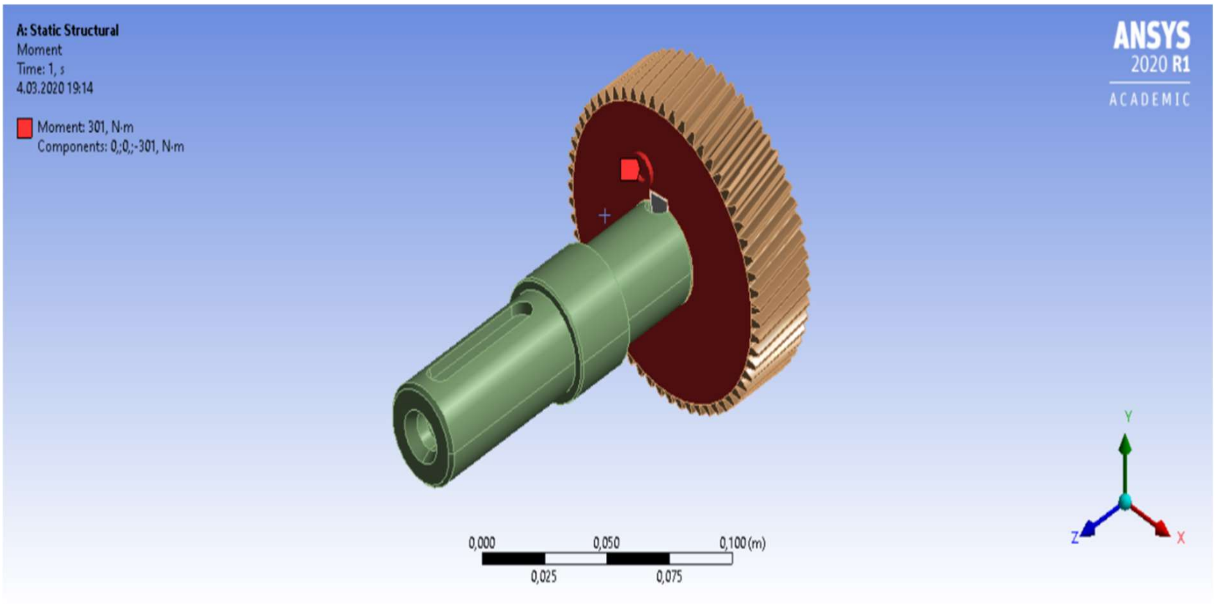

Figure 3.4.10 Gear Z6-Output shaft

3.4.1 Load Application Collectively

Here, we fixed the output shaft and bottom surfaces of the casing after giving the calculated moments to each shaft to see the static analysis of whole system together with the gearbox case.

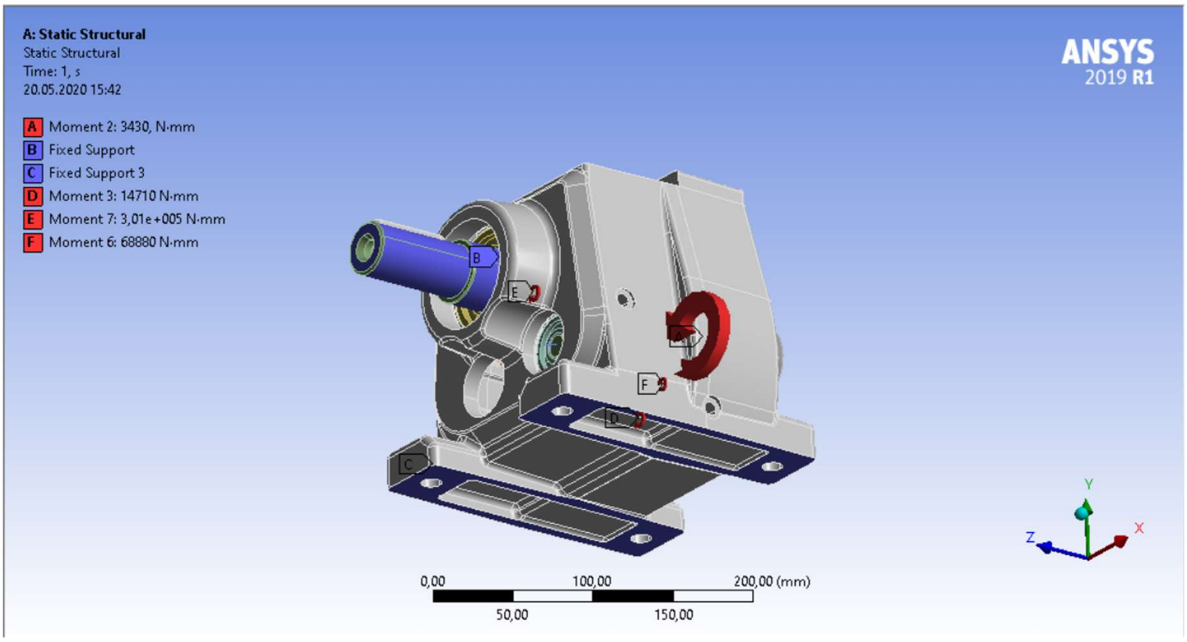

Figure 3.4.11 Fixed Supports

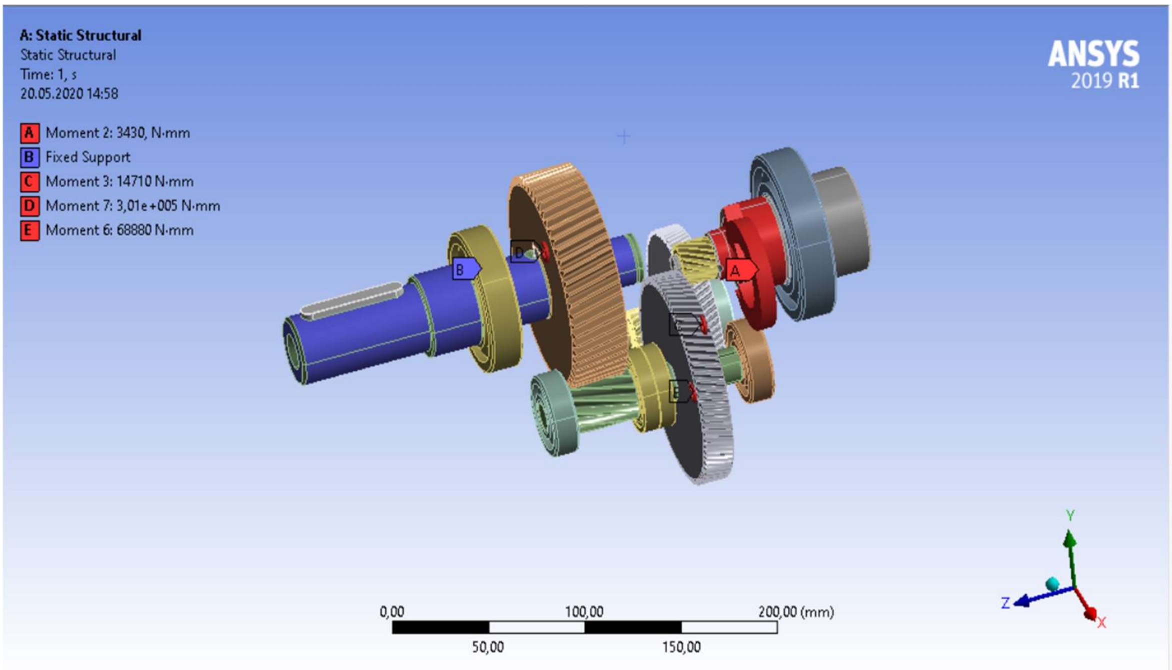

Figure 3.4.12 Moment Application to Shafts and Fixed Supports

4-RESULTS AND DISCUSSIONS

4.1 RESULTS

We have 6 teeth in contact with each other. We kept one of the gears constant in each part and gave the other the moment we calculated previously. We found 4 types of stress in the solutions. These; Total deformation, Equivalent (von-Mises) stress, Maximum principal stress, minimum principal stress.

The maximum principal stress is the most tensile (least compressive) and the minimum principal stress is the least tensile (most compressive).

4.1.1 RESULTS OF STATIC ANALYSIS

Respectively

Gear Z1

In order to find the deformation on gear Z1, we kept the gear Z1 fixed and gave the torque value that calculated in the torque section (14.71 N.m) to inside and outside surfaces of gear Z2. For solution part we selected Total Deformation, Maximum and Minimum Principal Stresses and Equivalent (von-Mises) stress.

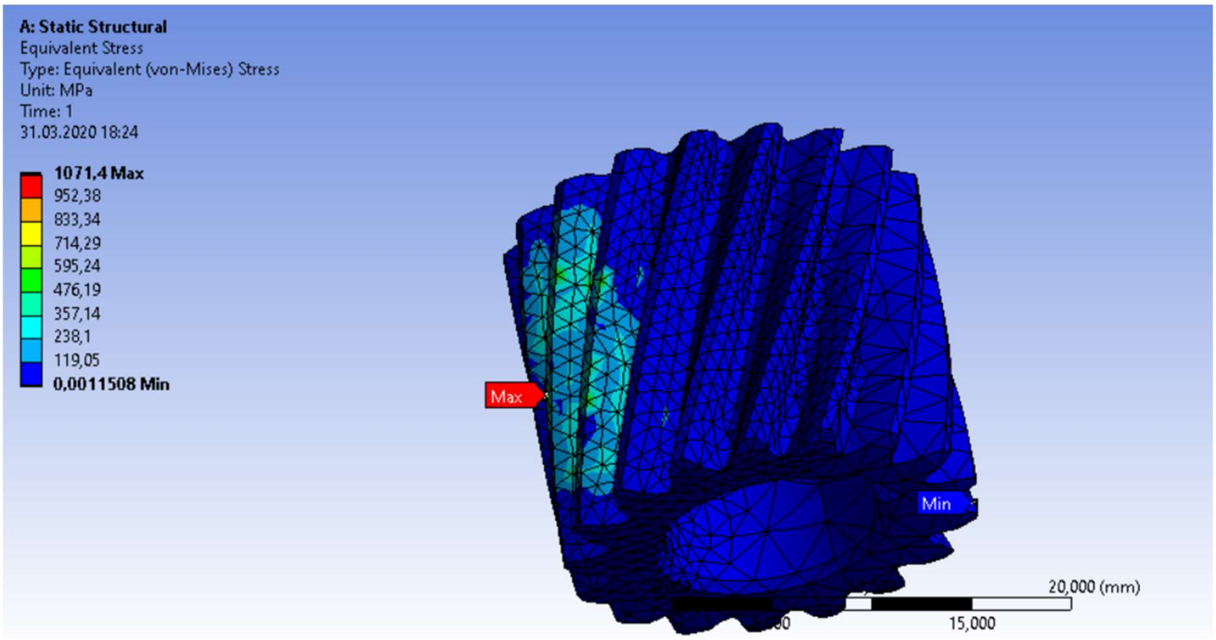

Figure 4.1.1 Static Analysis Results of Gear Z1

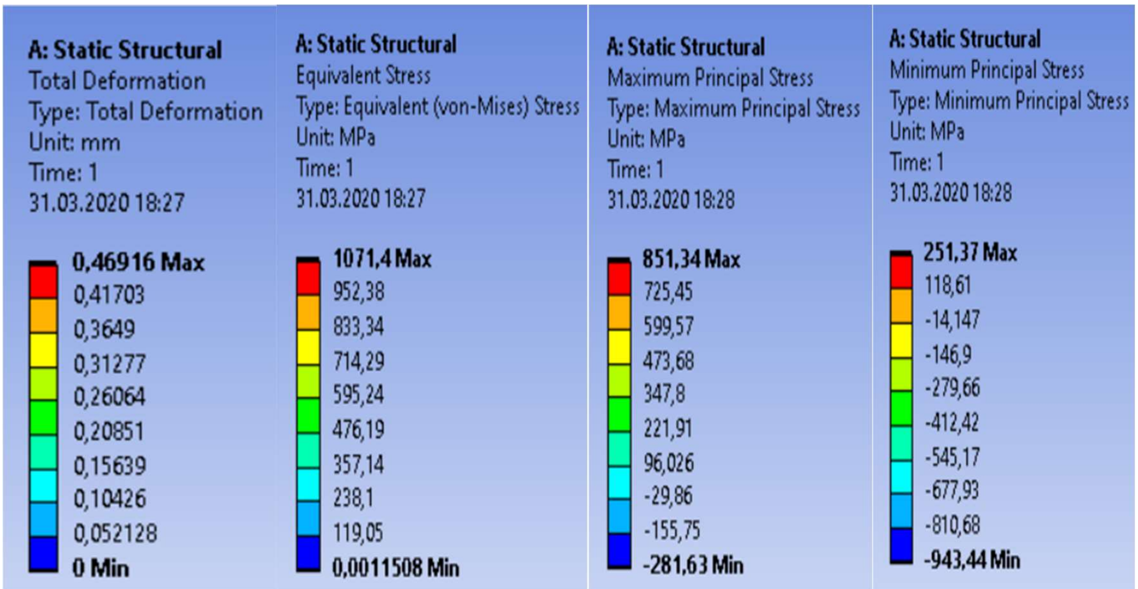

Figure 4.1.2 Stress solutions of Gear Z1

Gear Z2

For gear Z2, same way with the gear Z1 but this time we fixed gear Z2 and gave the moment (3.43 N.m) to the surfaces of gear Z1. Again for solution part we selected Total Deformation, Maximum and Minimum Principal Stresses and Equivalent (von-Mises) stress.

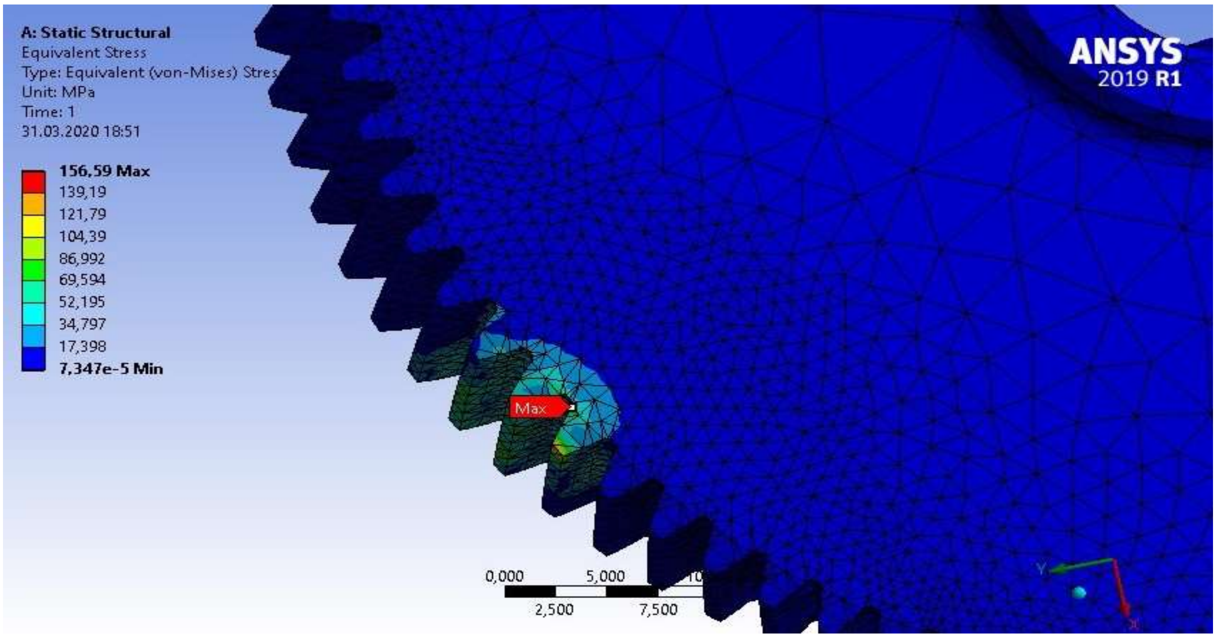

Figure 4.1.3 Static Analysis Results of Gear Z2

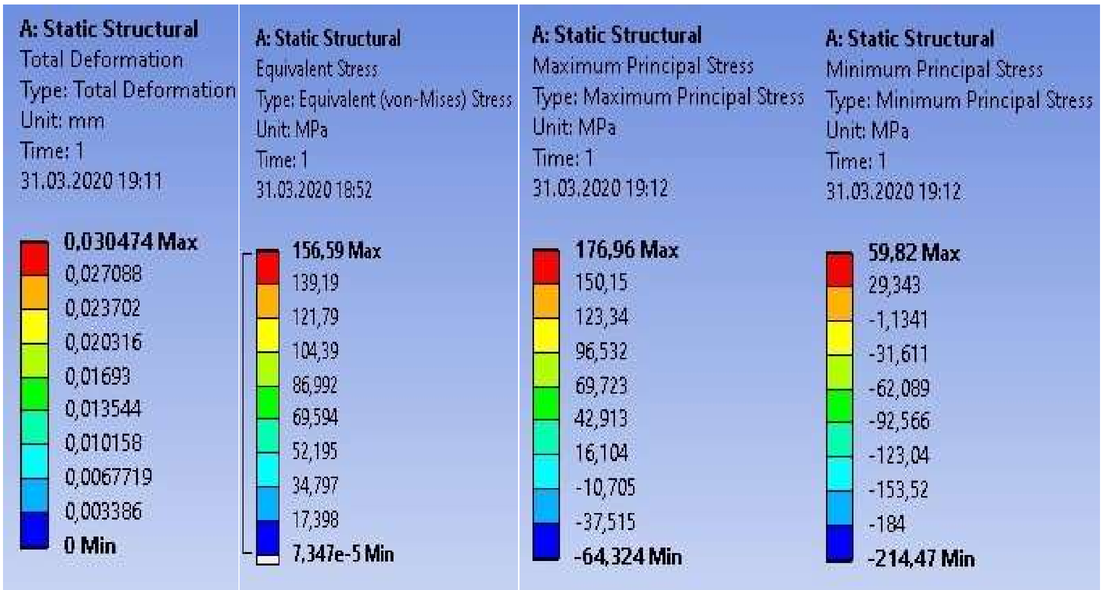

Figure 4.1.4 Stress solutions of Gear Z2

Gear Z3

To be able to find the deformation on gear Z3, we kept the gear Z3 fixed and gave the moment (68,88 N.m) to surfaces of gear Z4. For solution part we selected Total Deformation, Maximum and Minimum Principal Stresses and Equivalent (von-Mises) stress.

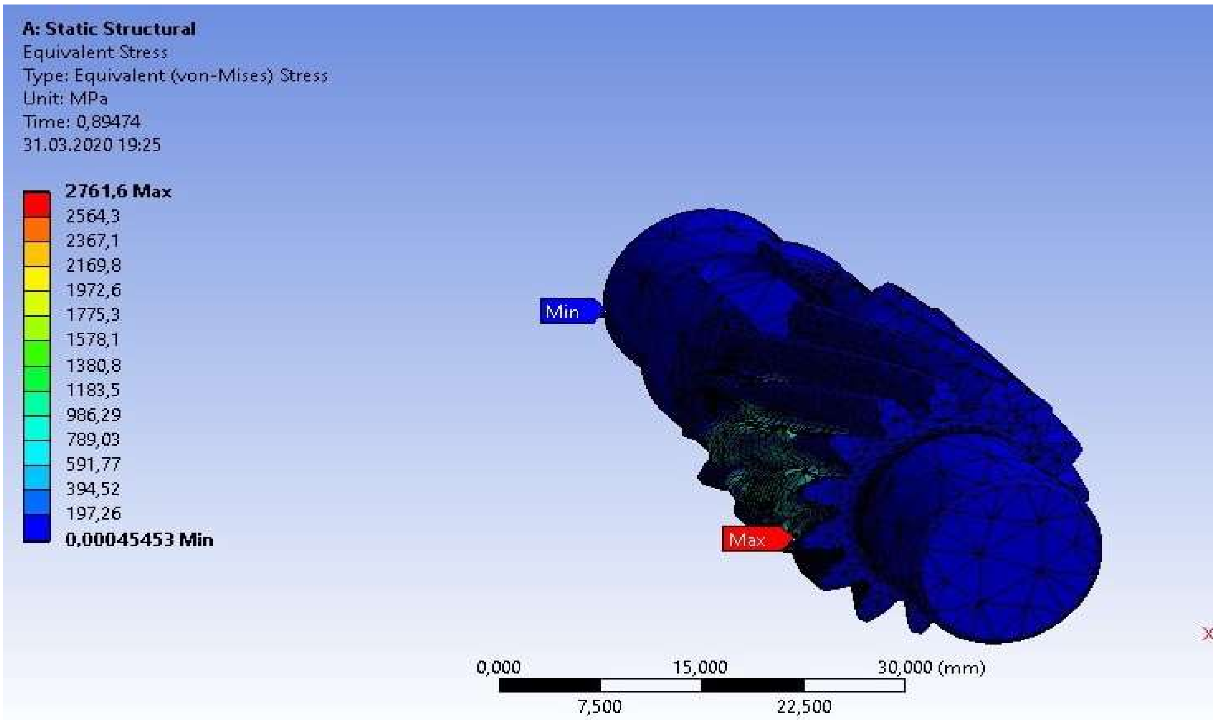

Figure 4.1.5 Static Analysis Results of Gear Z3

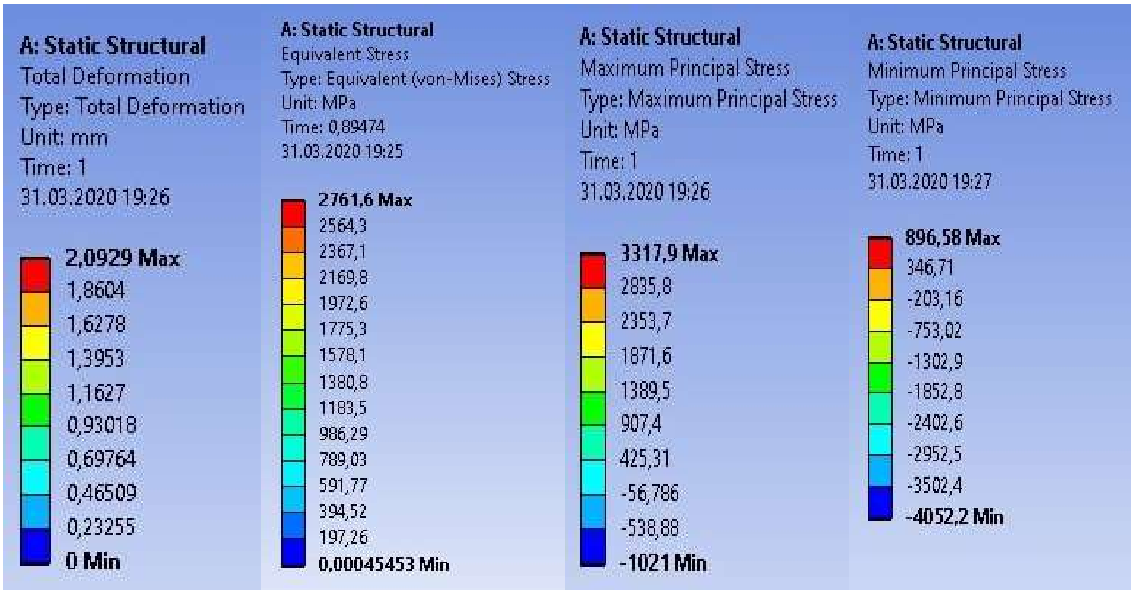

Figure 4.1.6 Stress solutions of Gear Z3

Gear Z4

In order to find the deformation in gear Z4, we kept the gear Z4 fixed and gave the torque value that calculated in the torque section (14,71 N.m) surfaces of gear Z3. For solution part we selected Total Deformation, Maximum and Minimum Principal Stresses and Equivalent (von-Mises) stress.

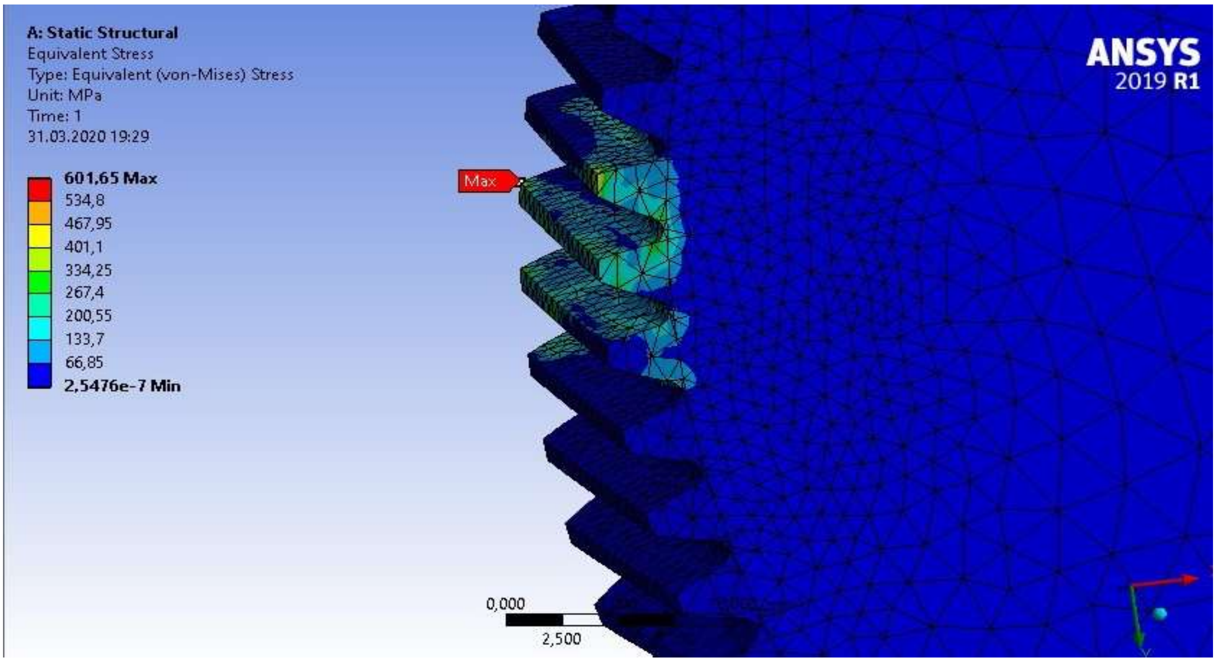

Figure 4.1.7 Static Analysis Results of Gear Z3

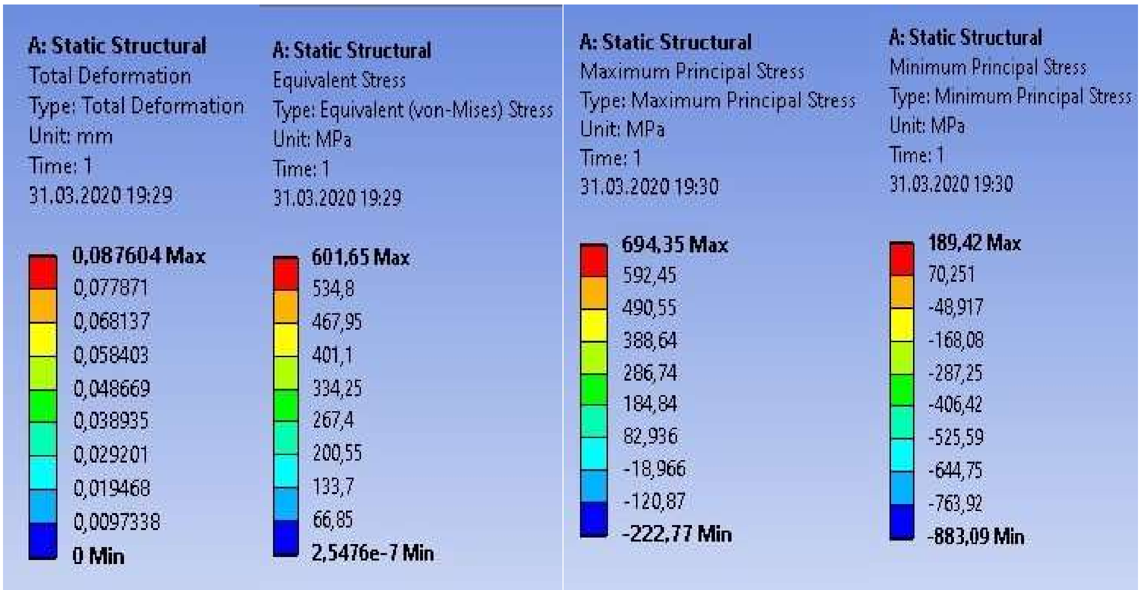

Figure 4.1.8 Stress solutions of Gear Z4

Gear Z5

In order to find the deformation in gear Z5, we kept the gear Z5 fixed and gave the torque value that calculated in the torque section (301 N.m) surfaces of gear Z6. For solution part we selected Total Deformation, Maximum and Minimum Principal Stresses and Equivalent (von-Mises) stress.

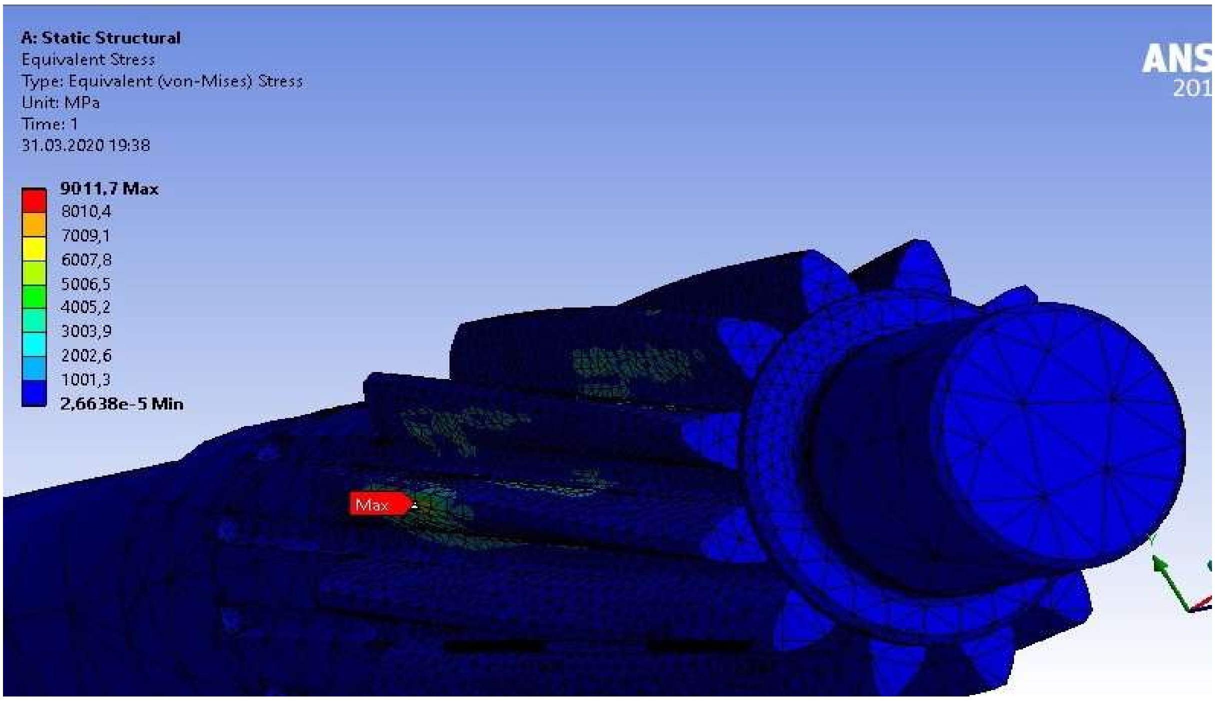

Figure 4.1.9 Static Analysis Results of Gear Z5

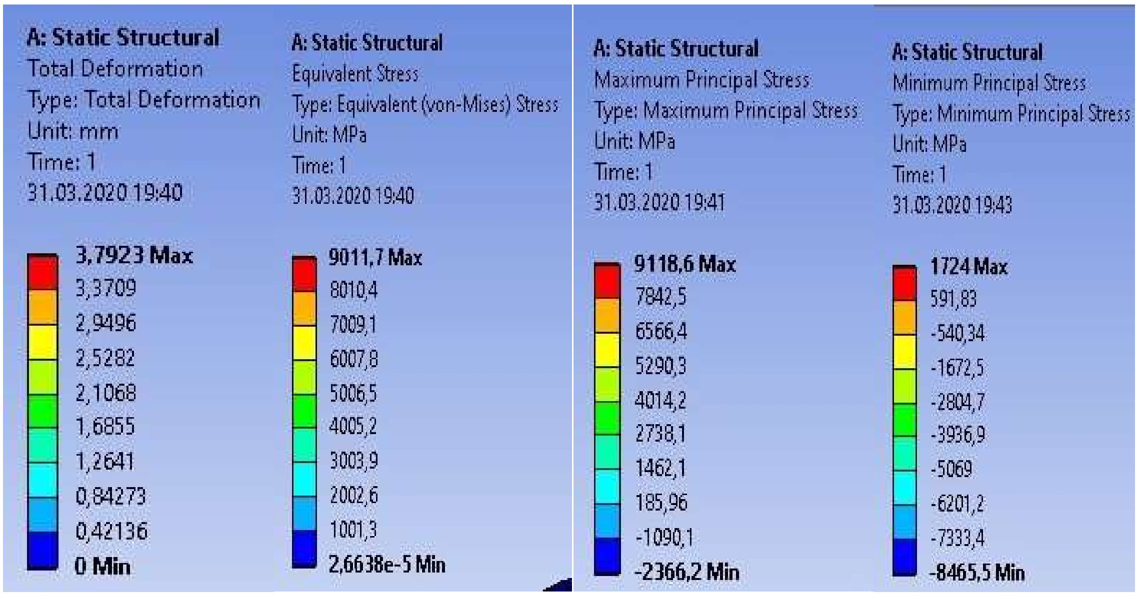

Figure 4.1.10 Stress solutions of Gear Z5

Gear Z6

In order to find the deformation in gear Z6, we kept the gear Z6 fixed and gave the torque value that calculated in the torque section (68,88 N.m) to surfaces of gear Z5. For solution part we selected Total Deformation, Maximum and Minimum Principal Stresses and Equivalent (von-Mises) stress.

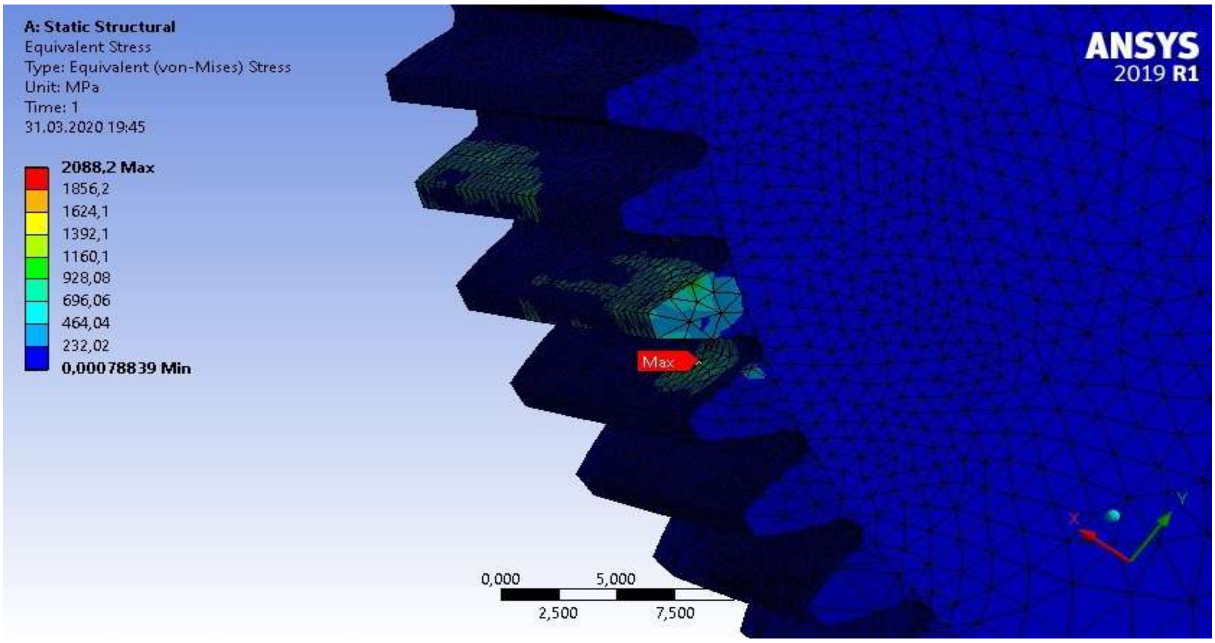

Figure 4.1.11 Static Analysis Results of Gear Z6

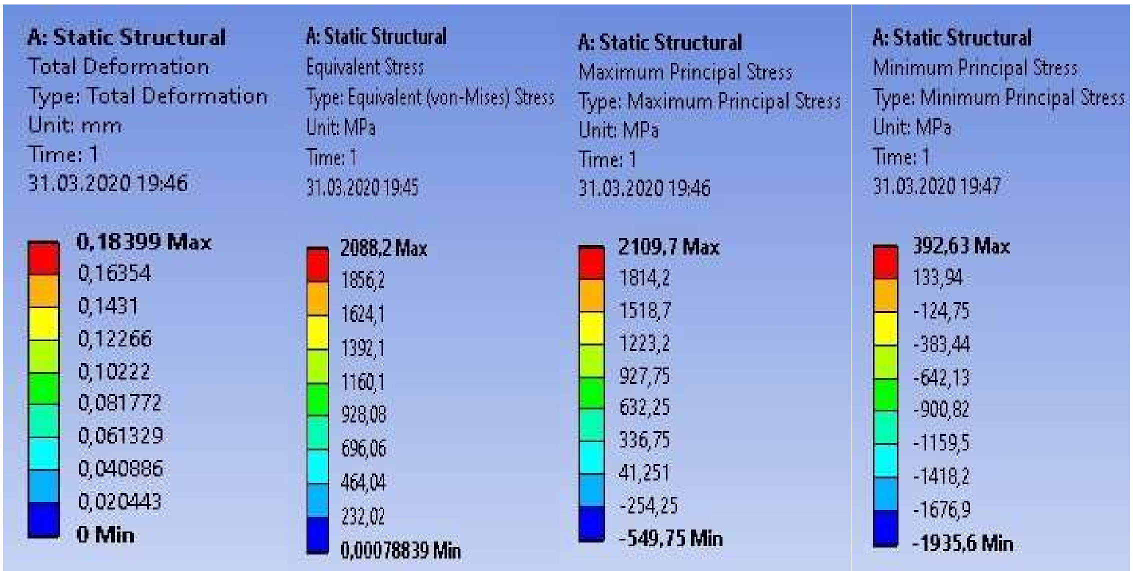

Figure 4.1.12 Stress solutions of Gear Z6

Collectively

In this part of our analysis, we keep the output shaft and the bottom surfaces of the box fixed and applied 4 moments for each shaft on our three-stage gearbox. The reason we do this is to see the static analysis of the gearbox collectively, after seeing the relationship of the gears between each other. For solution part we selected Total Deformation, Maximum and Minimum Principal Stresses and Equivalent (von-Mises) stress. Logarithmic scale was chosen for color change and chart bar. In order to see the Total Deformation, Equivalent, Maximum and Minimum Stresses better, we used an auto scale.

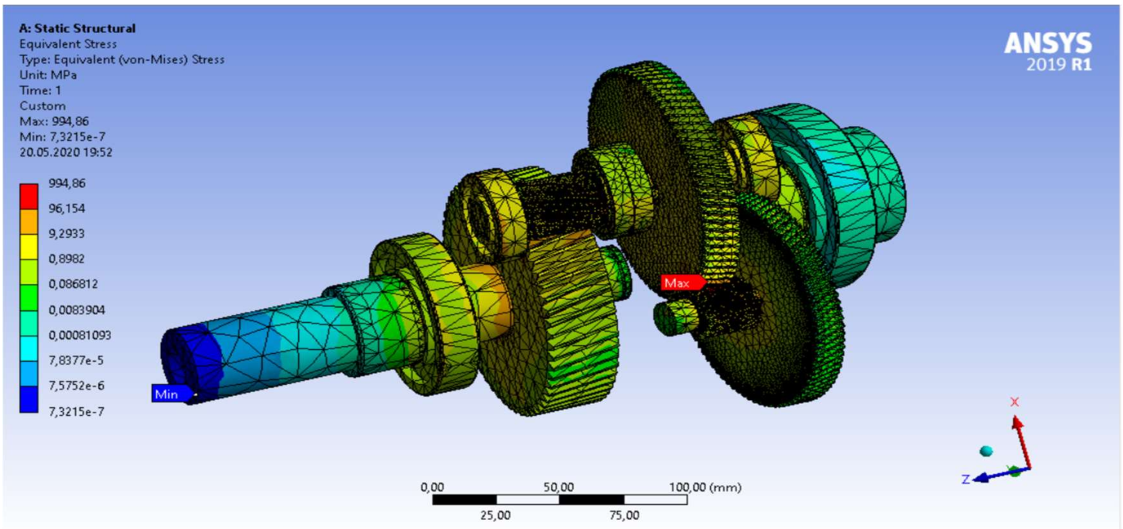

Figure 4.1.13 Equivalent(von-Mises) Stress Results of Gearbox

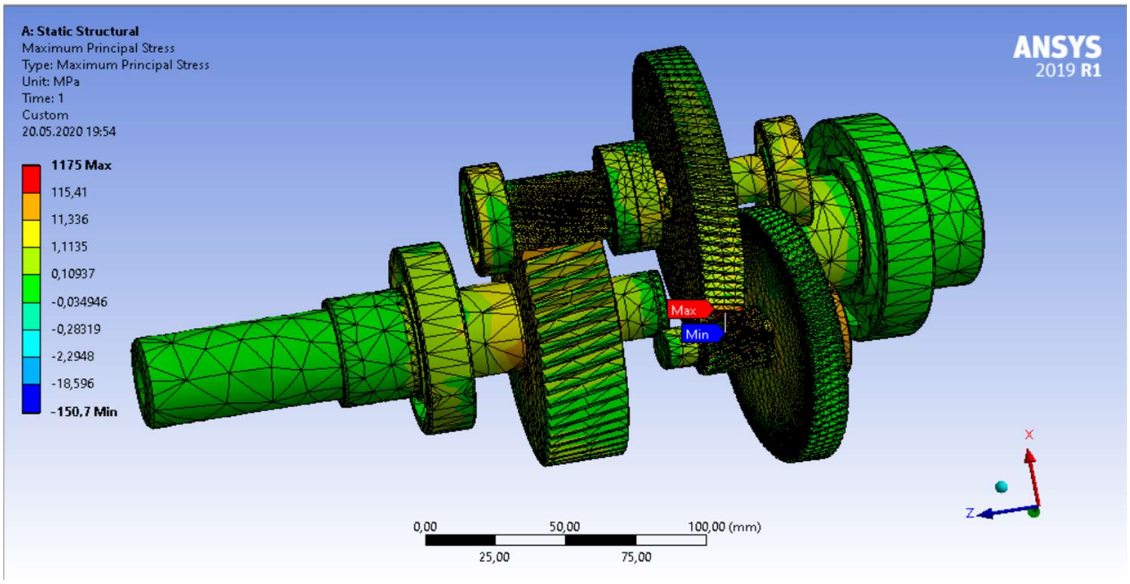

Figure 4.1.14 Maximum Principal Stress Results of Gearbox

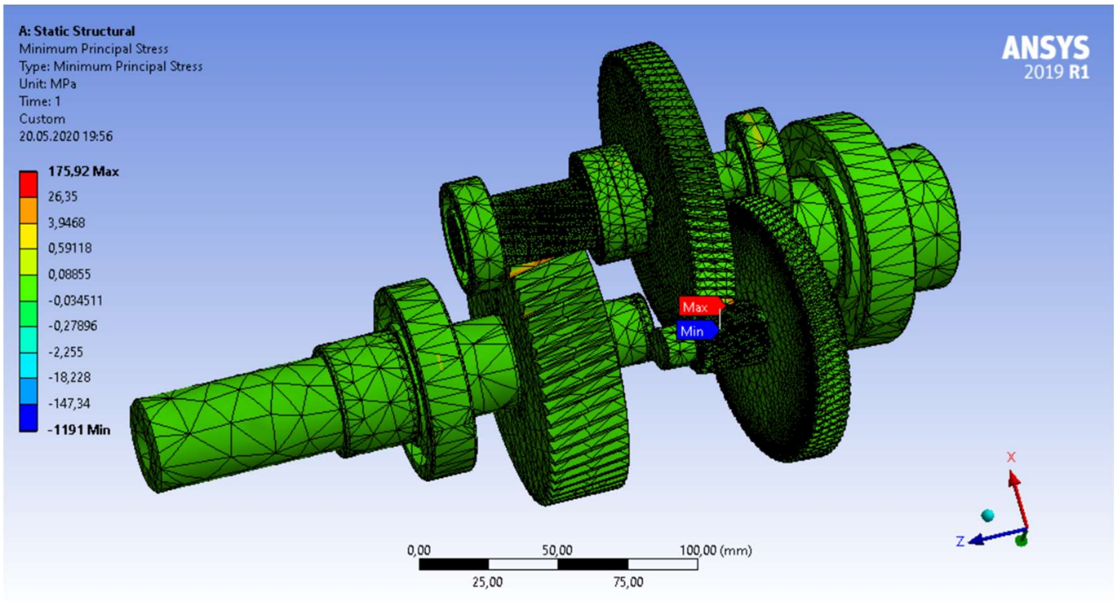

Figure 4.1.15 Minimum Principal Stress Results of Gearbox

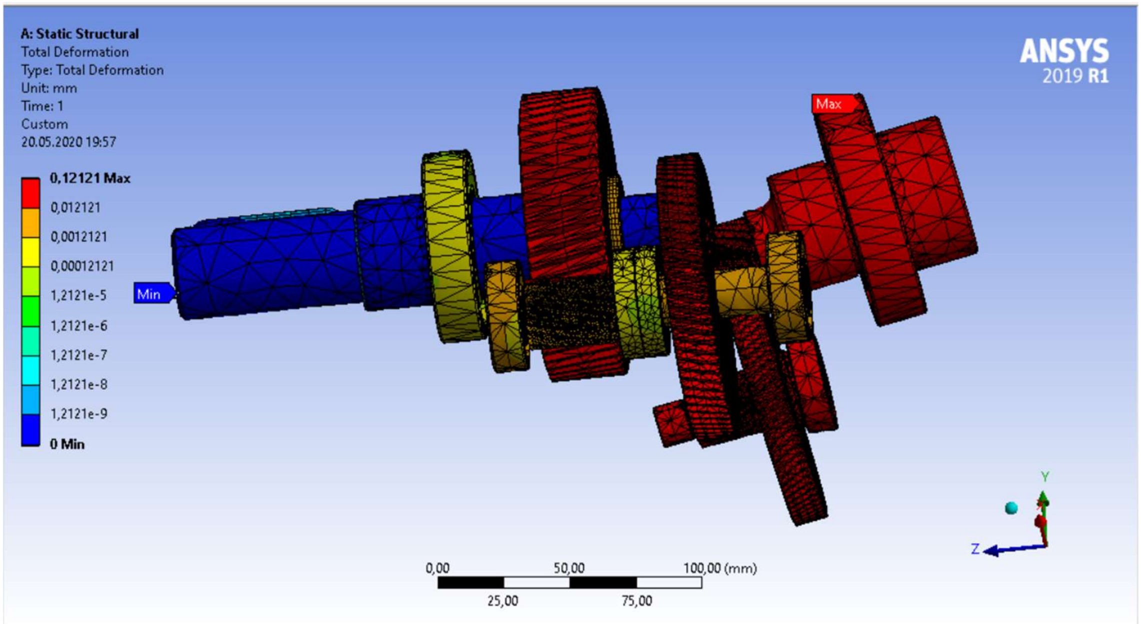

Figure 4.1.16 Total Deformation of Gearbox

Gearbox Casing

To be able to see the deformation and stress on Gearbox Casing, we fixed input shaft, output shaft and bottom of the casing.

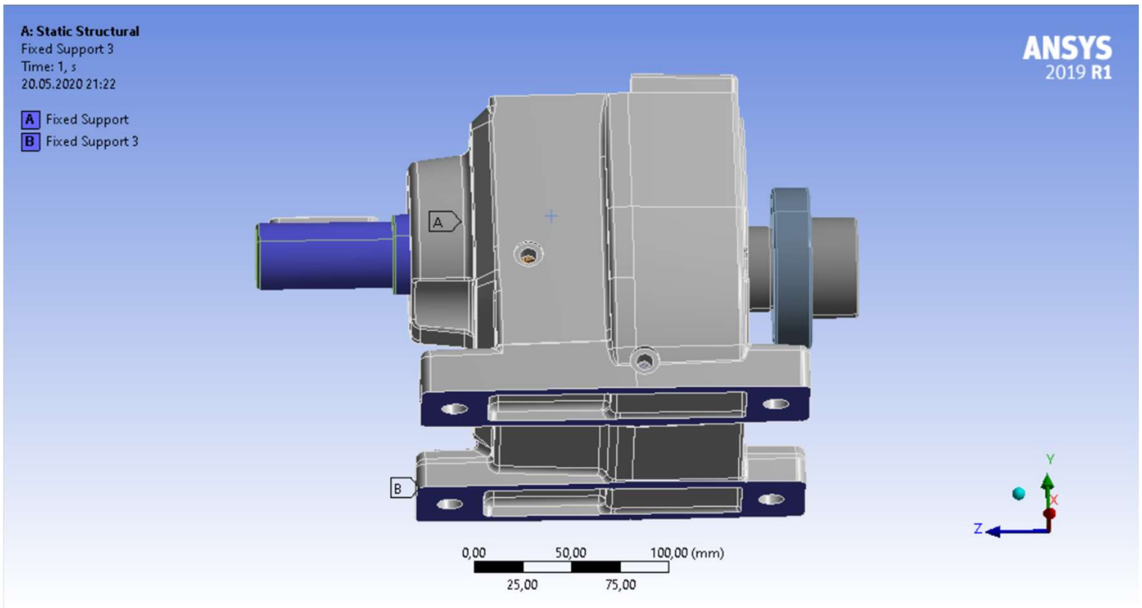

Figure 4.1.17 Fixed Supports at Casing, Input and Output Shafts

We chose the logarithmic scale for a better visual. Equivalent(von-Mises) stress and Total deformation results are below.

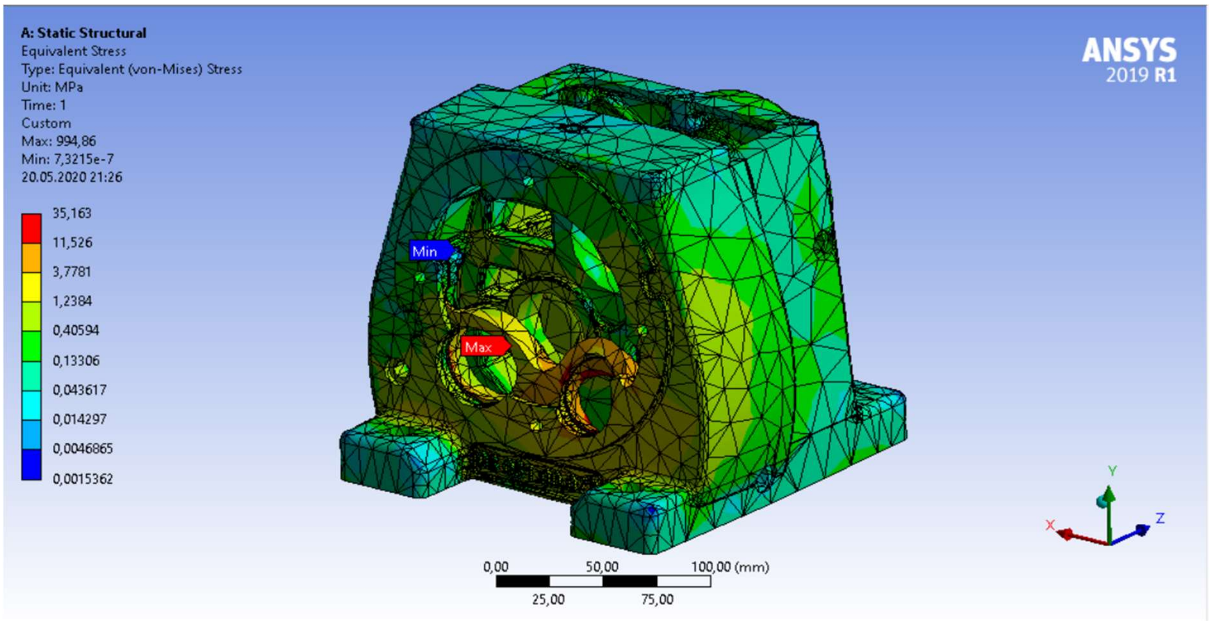

Figure 4.1.18 Equivalent(von-Mises) Stress Results of Gearbox Casing

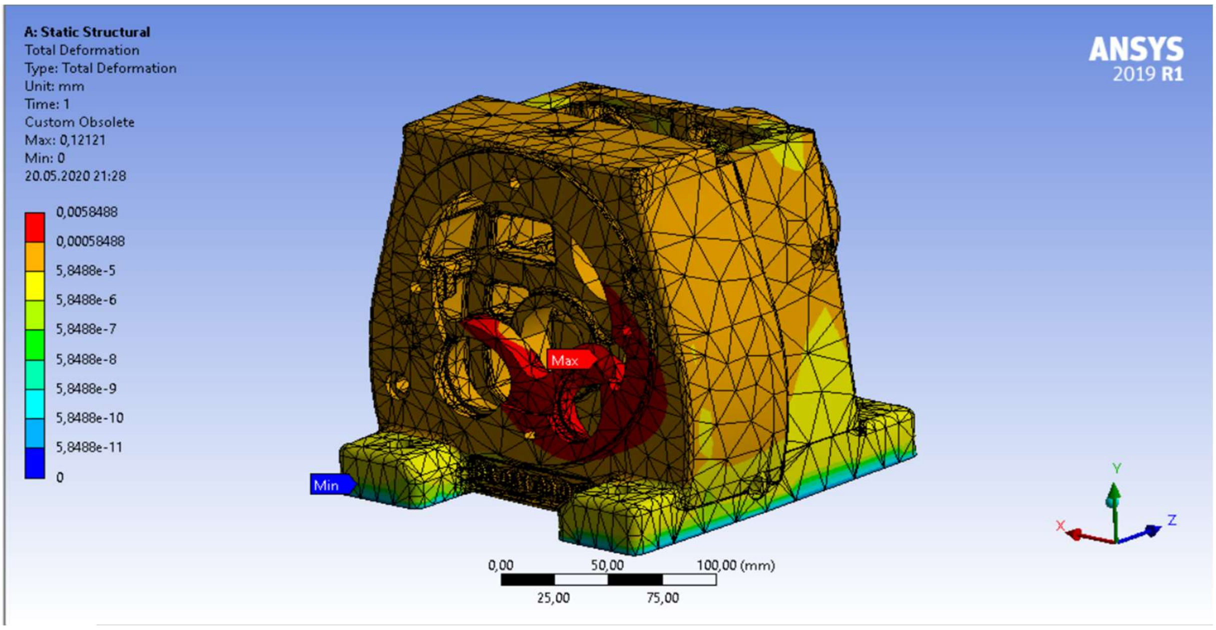

Figure 4.1.19 Total Deformation of Gearbox Casing

4.1.2 RESULTS OF DYNAMIC ANALYSIS

Mode Shape Analysis

The mode shape is a special vibration pattern that is carried out by a mechanical system at a certain frequency. Different mode shapes will be associated with different frequencies. The experimental technique of modal analysis explores these modal shapes and frequencies.

Maximum 6 modes were selected in this analysis. Total deformation is solved by selecting modes from 1 to 6. And the image of the maximum and minimum points of the frequencies formed in the gears are given below. And also the frequencies formed in the gears are given in a table. These solutions made 2 reviews as gears and gearbox case.

Gears

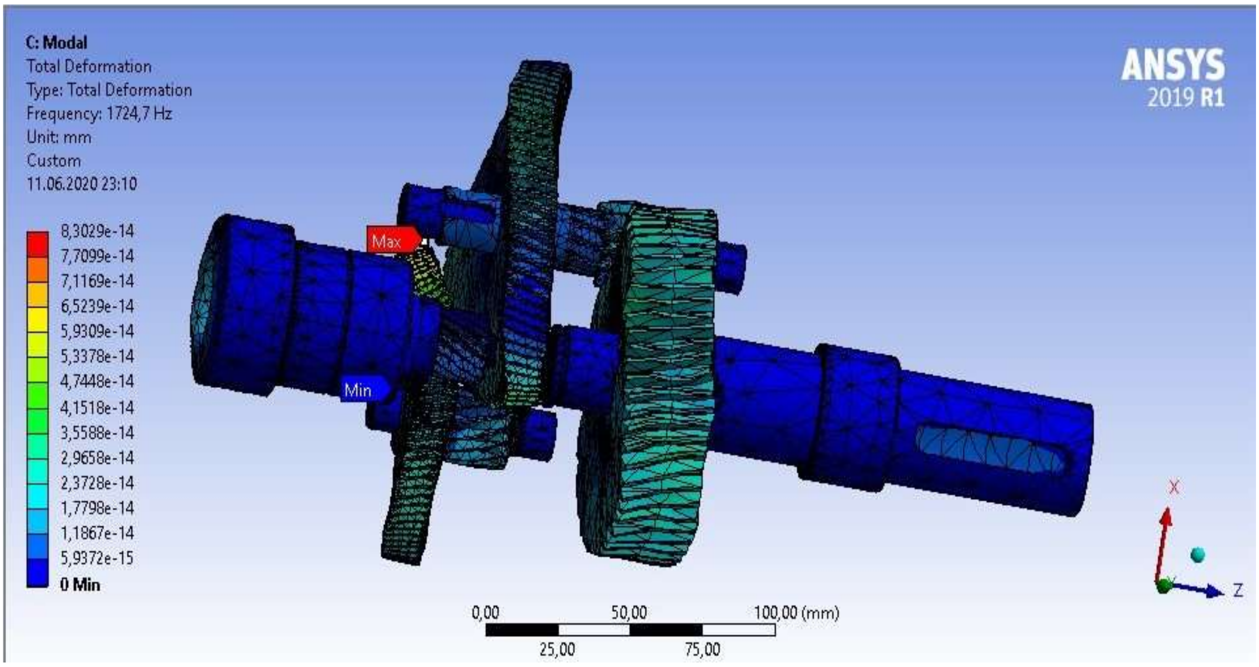

Figure 4.1.20 Results of Mode 1 Analysis of Gears

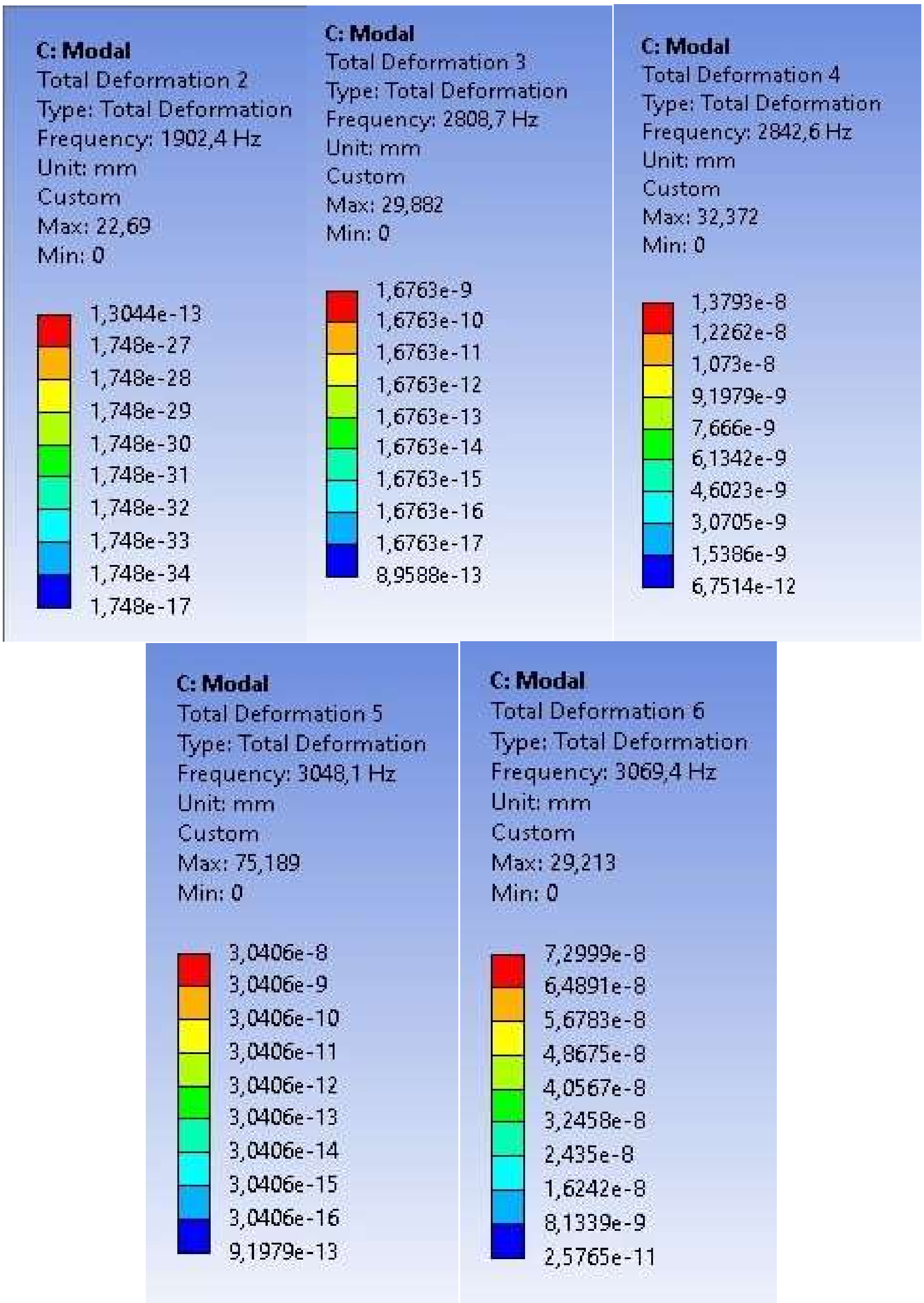

Fig 4.1.21 Results of Mode 2 to 6 Analysis of Gears

As seen in the results, different mode shapes were created using different modes and different frequencies were obtained. The frequency results obtained are given in the table below.

Table 4.1 Table of Frequency

Mode	Frequency [Hz]
1	1724.7
2	1902.4
3	2808.7
4	2842.6
5	3048.1
6	3069.4

Gearbox Casing

In this part, mode shape analysis was carried out in maximum 6 modes using case together with gears. The shape and the result in mode 1 are given in the figure below.

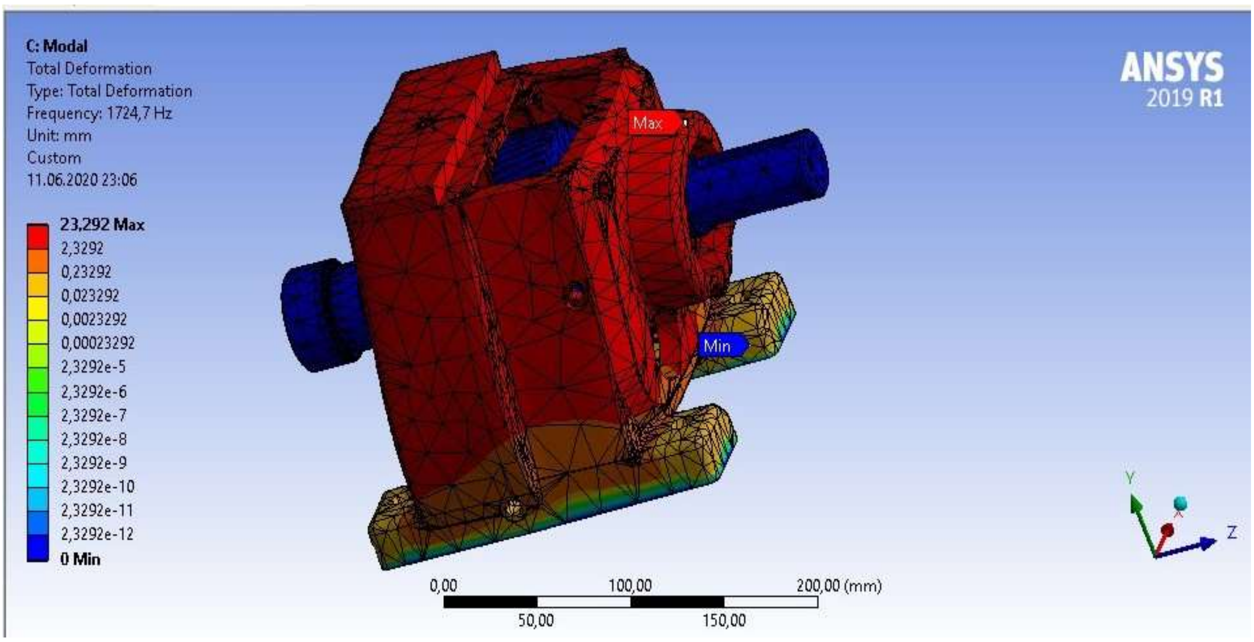

Figure 4.1.22 Results of Mode 1 Analysis of Gearbox Casing

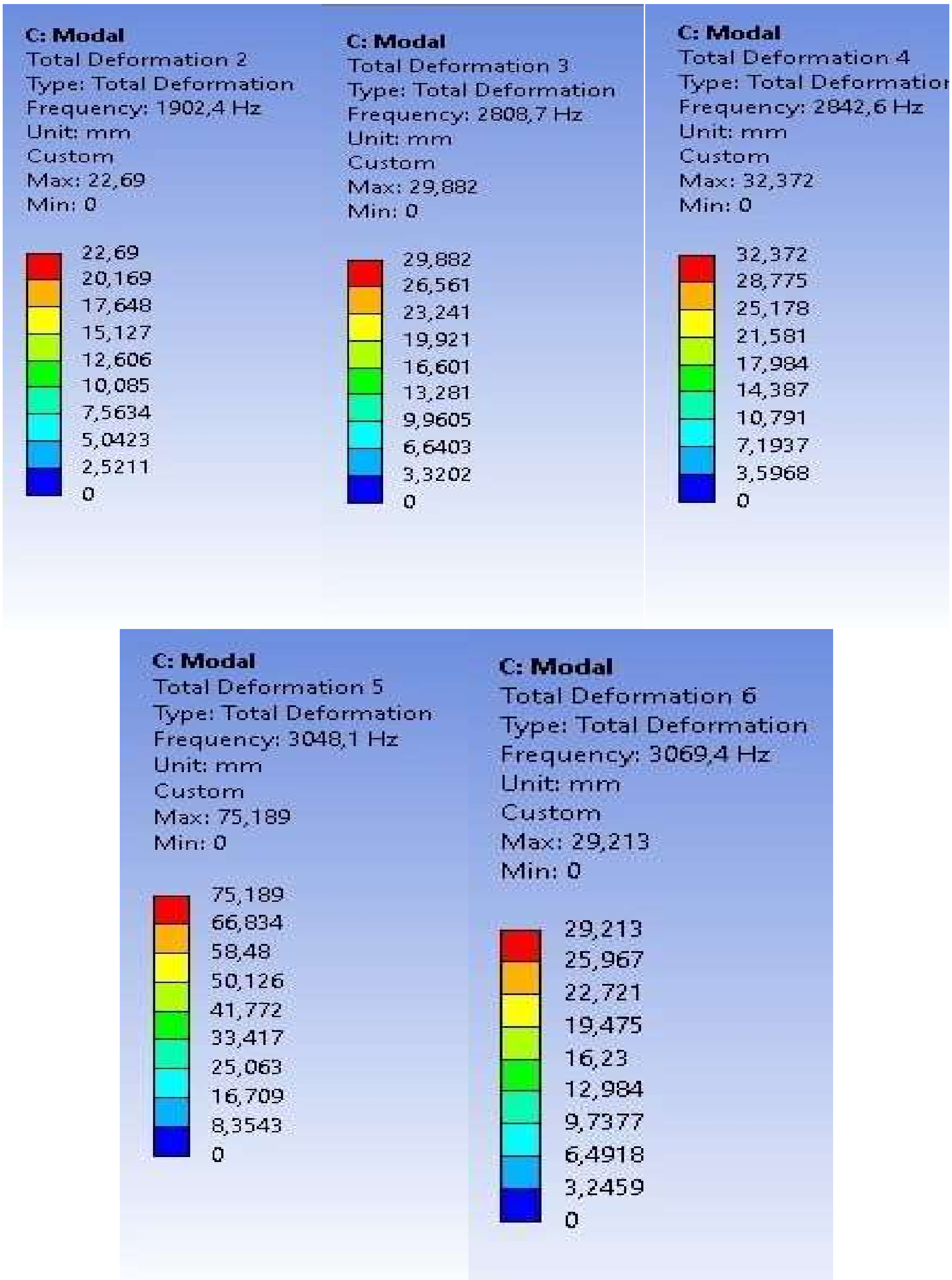

Fig 4.1.23 Results of Mode 2 to 6 Analysis of Gearbox Casing

The results of the solutions made from mode 1 to 6 are given above. Frequency values are same values for gears and casing and consider table 4.1 above.

Natural Frequency Analysis

Natural frequency analyses are used to determine the dynamic properties of a system and to identify its resonant frequencies.

In this analysis, gears and casing were analyzed in 6 modes. To know the behavior of gear train at resonance frequency, the natural frequencies of the gear train is compared with operating frequency. Operating speed of the vehicle is 10,86 to 1390 rpm.

$$Fn = \frac{1}{2\pi}\omega$$

$$\omega = \frac{2\pi N}{60}$$

$$Fn = \frac{1}{2\pi}\frac{2\pi * 1390}{60} = 23{,}16 \text{ Hz}$$

$$Fn = \frac{1}{2\pi}\frac{2\pi * 10{,}86}{60} = 0{,}181\text{Hz}$$

The range of operating frequency is in between 24 Hz to 0.1 Hz approximately. This range is very small when compared with lowest natural frequency of 1724.7 Hz. Hence FEM reveals that resonance will not take place on any gear train and the design is safe for both gears and casing. Min frequency is 1724.7 Hz and the maximum frequency is 3069.4.

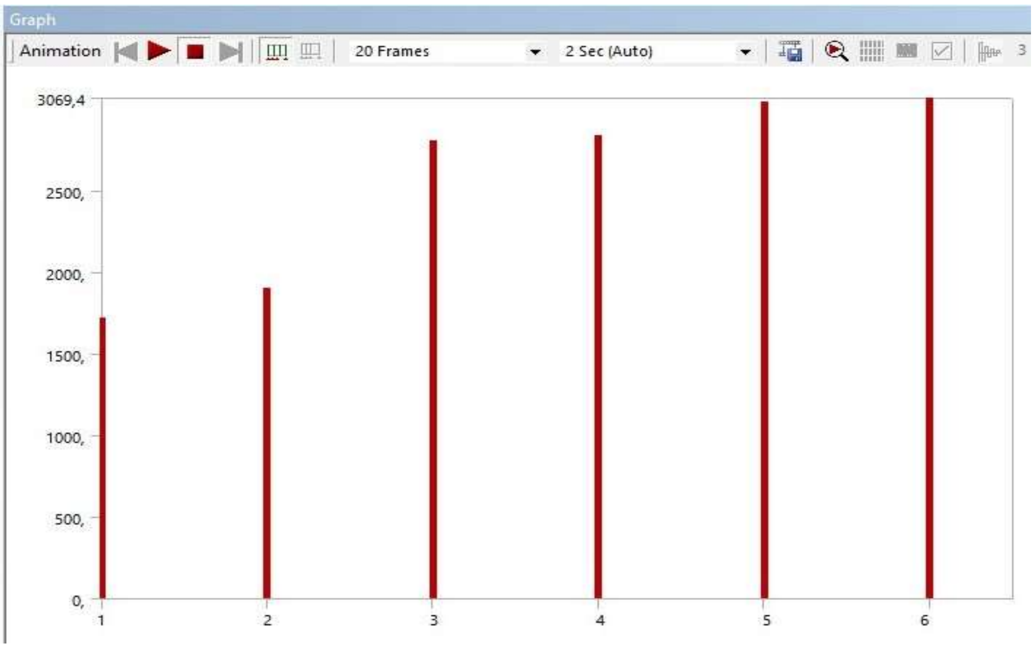

Fig 4.1.24 Frequency of Modes 1 to 6

4.2 DISCUSSIONS

4.2.1 DISCUSSIONS OF STATIC ANALYSIS

Respectively

In this section, the results we find in the results section will be evaluated and comments on why these values are so. Static stress analysis results of gears in a gear box with 6 teeth will be examined respectively.

Gear 1 Results Discussion

Gear 1 is made of AISI 8620 material and the yield strength value of this material is 833 MPa. When von mises stress is examined in the Z1 gear, it is observed that the maximum stress is at the extremities of the body and is 1071 MPa. It is observed that the average stress on the body is around 400MPa, except for the ends. Since the stress observed at the ends is larger than the yield stress, breaks will be observed at the ends. however, the reason for this value to be higher than the yield stress may be due to the drawing of the 3D model, and since there are gaps at the end points where the two gears touch each other, the measured value is too high.

Gear 2 Results Discussion

Gear 1 is made of AISI 5110(MnCr5) material and the yield strength value of this material is 484 MPa. Maximum stress is observed at the end points in this gear. And maximum stress is 156 MPa at these points so that there is no breakage in the gear. The drawings and material selection for this gear has been made correctly.

Gear 3 Results Discussion

Gear 3 is made of AISI 8620 material and the yield strength value of this material is 833 MPa. As stated in the results section for this gear, the average stress value at the contact points is around 600-700 MPa. But the same cannot be said for the endpoints . The maximum stress value formed at the end points reaches 2761 MPa and breaks occur at the end parts of this gear, as in the first gear.

Gear 4 Results Discussion

Gear 4 is made of AISI 5110(MnCr5) material and the yield strength value of this material is 484 MPa. For the 4 th gear, 601 MPa values are seen at the end points. This gear will also break at the contact points, but the average stress values observed at the contact points are around 250 MPa. Breaks may also be seen at the ends of this gear.

Gear 5 Results Discussion

Gear 5 is made of AISI 8620 material and the yield strength value of this material is 833 MPa. When we look at the results for gear 5, it was seen that the maximum stress was found to be 9000 MPa point by point. The average stress values in this gear are within the yield strength limit values but there will be breaks for the end points. There are several different reasons that affect maximum stress values, these are not very fine mesh values and drawing errors when drawing the contact points of the gears.

Gear 6 Results Discussion

Gear 6 is made of AISI 5110(MnCr5) material and the yield strength value of this material is 484 MPa. Maximum stress for gear 6 is 2088 MPa but this maximum stress is seen at the end points of the gear. The average value is between 400-500 as shown by the blue indicator and is within the yield stress limits. As can be seen in other gears, there will be breaks at the ends of the gear.

Collectively

In the collective analysis, we observed the highest stress value of 994.86 MPa and this value on the 3rd gear. In our respectively analysis, the maximum stress value on this gear was 2761 MPa. We think that the collectively analysis is a more accurate analysis, but nevertheless maximum stress is above yield strength which is 833 MPa and as a result breaks appear on the gear ends. The average value is between 200-300 MPa as shown by the orange indicator and is within the yield stress limits.

In the analysis we conducted respectively, we saw that the most stress was in the 5th gear, and in this analysis, although the maximum stress is at the ends of the 3rd gear, we see that the orange indicator is at the 5th gear as the average. In other words, in this analysis we conducted collectively, we observed that the most stress was in the 5th gear, as we did respectively.

Gearbox Casing

In this section, the stresses on the gearbox case will be discussed. As a Casing material Structural Steel is used in the analysis. As seen in the results, the maximum stress is in the bearing of the shaft carrying the 4th and 5th gears and 35,16 MPa. And on average, it is seen from the orange color, where the values are 20-30 MPa, and this is well below yield strength (460 MPa). As a result, there is no break in the gearbox case and it is safe.

4.2.2 DISCUSSIONS OF DYNAMICS ANALYSIS

Mode Shape Analysis

In this section, 6 modes were selected and modal analysis was performed and mode shapes were obtained. The most noticeable in the mode shapes with gears, shafts and casing, the total deformation of the casing at all natural frequency values is much higher than the gears. As the main reason for this, it can be considered that the materials of gearbox casing and gears are different and the area covered by gearbox casing is more than gears.

Natural Frequency Analysis

When the results of the natural frequency analysis are examined, it can be observed that it is much less than the operation frequencies calculated based on the input and output rpm values. The result of this analysis is that this gear system using the specified materials is safe against resonance.

5.CONCLUSION

In this study, analysis of the 3-stage gearbox was carried out in ansys using the finite element method. Casing and gears of the gearbox were analyzed both collectively and separately. After the necessary boundary conditions are given by assigning the gears and materials used for casing, the analysis settings steps are given in the above sections and then they are a source for other experiments. Meshes and contacts are carefully created to try to find a finer solution and meshes are reinforced at critical points for this. After all boundary conditions were created, total deformation, von misses, maximum principal stresses and minimum principal stresses options were used for static analysis results. The results were used both stresses between 2 gears and 6 gears working collectively with 3 stages. Although static breaks occurred at the end points caused by the drawings of the gearbox, it was observed that the material generally resisted the given speed and torque. In calculations section, it is calculated whether it is able to withstand the torques given for each gear by using 1.5 safety factor and theoretically all materials are resistant to these torques.

Besides static analysis, it was carried out in dynamic analysis. Modal analysis option of Ansys software was used in dynamic analysis. Thus, mode shape analysis and natural frequency analyzes were carried out. While creating mode shape analyzes, maximum 6 modes were used and frequency values in each mode were taken. In natural frequency analysis section, F_n values are calculated by calculating the input and output rpm values in the gearbox. the values we calculate are in a much smaller range than the frequency values we obtain in different modes in the program. So there will be no resonance for the gearbox design and it can be used safely.

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