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Finite element stress analysis of three-stage gearbox

Üç kademeli bir dişli kutusunun sonlu elemanlar ile gerilme analizi

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Abstract

The stress analysis of the three-stage gearbox is carried out by the finite element analysis method using the Ansys commercial program. The triple reduction helical gearbox is manufactured from AISI 5115 (16MnCr5) and AISI 8620 steels. Static structural and rigid dynamic analyzes are performed in this work. The three-stage helical gearbox is a three-stage gearbox transmitting 0.5 kW at 1390 rpm with a reduction ratio of 127.99:1. After setting the boundary conditions for helical gears, the analysis is performed. Rotational velocity is given to rotating gears for static structural analysis. Analyzing these types of gearboxes to be produced enables the product to be produced more consciously with optimum design parameters with its right safety coefficients.

After the static analysis, the dynamic analysis is also 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 considering the input and output rpm values in the gearbox. These calculated values are compared with the normal operating condition to see if there will be any resonance in the gearbox design during normal operating conditions.

Keywords: Three-stage gearbox, Finite element analysis, Computer simulation, Static analysis, Modal analysis

Özet

Bu çalışmada üç kademeli bir dişli kutusunun sonlu elemanlar analizi kullanılarak gerilme analizi gerçekleştirilmiştir. Üç kademeli dişli kutusu AISI 5115 (16MnCr5) ve AISI 8620 çelikleri kullanılarak üretilmiştir. Bu çalışmada yapısal statik ve katı dinamik analizler gerçekleştirilmiştir. Göz önüne alınan üç kademeli helisel dişli kutusu 0.5 kW gücünü 1390 dev/dak da 127.99:1 dönüştürme oranı ile iletmektedir. Helisel dişliler için sınır şartları belirlendikten sonra analizler gerçekleştirilmiştir. Statik analizler için dönme hızları girilmiştir. Üretilecek olan dişli kutuları için bu tür analizlerin yapılması ürünleri optimum tasarım parametreleri ve doğru emniyet katsayıları göz önüne alınarak daha doğru bir şekilde üretilmelerine imkan tanımaktadır.

Statik analizden sonra, ANSYS programının modal analiz seçeneği kullanılarak dinamik analizler gerçekleştirilmiştir. Serbest titreşim analizleri ile birlikte doğal frekans analizleri de gerçekleştirilmiş olup, giriş ve çıkış devir sayıları göz önüne alınarak dişli kutusunun normal çalışma frekansları belirlenmiştir. Dişli kutusunda normal çalışma koşularında bir hasarın oluşup oluşmayacağını belirlemek üzere hesaplanan bu değerler ile dişli kutusunun normal çalışma koşullarındaki değerleri kıyaslanmıştır.

Anahtar kelimeler: Üç kademeli dişli kutusu, Sonlu elemanlar analizi, Bilgisayar simülasyonu, Statik analiz, Modal analiz

1 Introduction

This article aims to analyze the three-stage gearbox and find strain and stress values at each component under normal operating conditions. The typical components of the gearbox are gears, shafts, bearings, pins, screws and case. Three-stage means, the power and speed are reduced in a three-stage gear system so one can get higher torque at each stage, thus gear reduction has the opposite effect on torque.

The analysis of these stresses is important 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 materials. This is mainly due to the well-known fact that a chain is as strong as its weakest ring. In addition to the static analysis, dynamic analysis is also important in terms of observing how much stress and strains this gear system undergoes at what frequencies. Furthermore, by using modal analysis, one can observe whether there is resonance at certain speed ranges and at what speeds the resonance occurs in mode shape analysis.

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

2 Literature survey

There are several works done on the finite element stress analysis of gearbox [1-9]. Bathe [2] described that the Finite Element Analysis (FEA) is an important part of engineering analysis and design. Finite element analysis is a practical application for the analysis of all the physical phenomena. Designers use it to minimize the number of physical

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prototypes so it can be easier to solve and to improve the design in minimum time by optimizing the components.

Yang and his co-workers [3] studied the dynamic characteristics of the brake drum with the finite element in using ANSYS software. After the analysis, the natural frequencies and vibration shapes are also computed.

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

Vijaykumar and his co-workers [5], by using ANSYS finite element analysis software, determined the vibration analysis of the transmission case using a harmonic frequency response for the case to prevent resonance. It is observed that to prevent resonance, the frequency ratio should be set to 0.25 from the first modal natural frequency.

Ramamurti [6] made a comparison of stress results obtained from the 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 and Muruganantham [7] stated in their works 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 operating frequency calculated using the input and output rpm values of the gear system.

Yesilyurt and his co-workers [8] measured the reduction of stiffness of gear teeth with the help of modal analysis. Besides, with the help of modal analysis, they determined that the gear tooth can be detected as damage and wear damage. To prove these analyzes in a physical environment, an experiment is also conducted to obtain the Frequency Response Function (FRF) of the gear tooth. The steel-tipped impact hammer is used as a stimulator and the accelerometer acts as a response detector.

Weis and his co-workers [9] tried to calculate the first 20 natural frequencies and corresponding mode shapes in the modal analysis of gearbox housing with Ansys workbench.

3 CAD design of three-stage gearbox

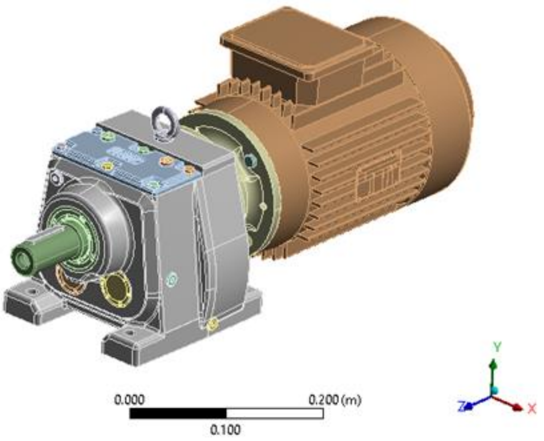

Figure 1. Whole design of three-stage gearbox

The drawing of the three-stage gearbox was created within the scope of this project in a computer environment. The design has "6 gears", "3 shafts", "6 keys", and a rotor engine, as shown in Figure 1 and Figure 2. For static analysis, the stresses at each gear, shaft, key, and gearbox case are calculated.

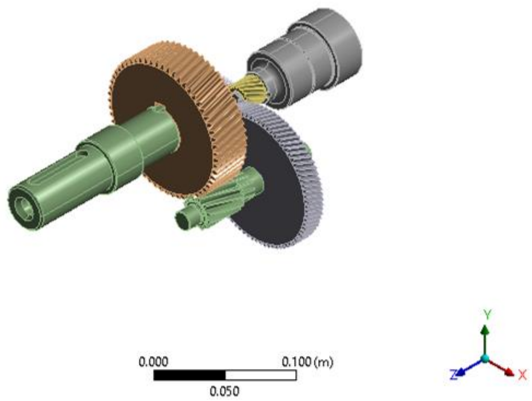

Figure 2. Gearing system in the whole design

4 Analytical calculations in gearbox

In this section, the analytical calculations of a three-stage gearbox design are explained. First, the speed, power, and torque transfer calculations of the 6 helical gears given in Table 1 are analytically calculated as following.

Table 1. Speed of gears

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

The CAD drawings of this gearbox are taken from industry, given in Figure 3.

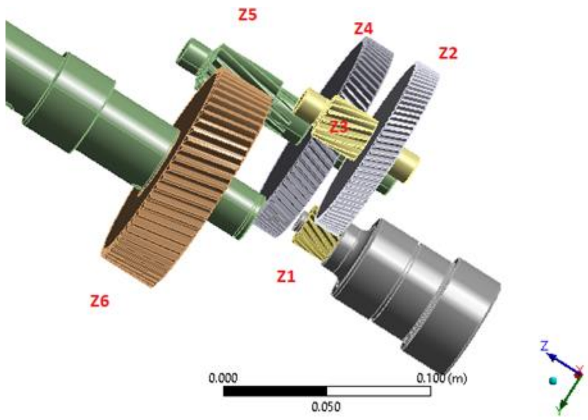

Figure 3. Gearbox CAD drawing

Table 2. Properties of the gears

Gear	Number of Teeth	Normal Module (m)	Pressure Angle (°)	Helix Angle	Helix	Status
Z1	17	1	20	20	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

4.1 Speed calculations

The input starts with 1390 rpm and the output is reduced to 10.86 rpm by applying the following formula at each stage.

N_i : Number of teeth in gear (i)
W_i : Rotational speed in gear (i)

$$W_2 = \frac{W_1}{N_2} * N_1 \quad (1)$$

4.2 Gear efficiency

For the value of pressure angle $\alpha_0=20°$ (shown in Table 2) the efficiency is approximately $\mu_1 = \mu_2 = \mu_3 = 0.89$ taken from references [1].

4.3 Power calculations

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:
$P_{input} - P_{output} = 0.5 - 0.352 = 0.148\ kW$

4.4 Torque calculations

$$\tau_Z = \frac{P}{W} \quad (2)$$

P : Power
W : Angular velocity

$$I_{12} = \frac{N_2}{N_1} \quad (3)$$

$$\tau_{Z2} = I_{12} \times \tau_{Z1} \times \mu_1 \quad (4)$$

I_{12}: Gear ratio
μ_1: Gear efficiency

Table 3. Torque of Gears

Gears	Torques of Gears (N.m)
τ_{Z1}	3.43
τ_{Z2}	14.71
τ_{Z3}	14.71
τ_{Z4}	68.88
τ_{Z5}	68.88
τ_{Z6}	301

4.5 Stress control in stages

The contact stress at the pitch point of gear is calculated using American Gear Manufacturer's Association (AGMA) stress equation [10]. The examined bending stress analysis in gear:

$$\sigma_{max} = W_t \times \frac{1}{b \times m} \times \frac{1}{Y} \quad (5)$$

W_t is the tangential transmitted load (N)
b is the face width of the narrower member, in (mm)
m is the metric module, in (m)
Y is the geometry factor for bending stress (which includes root fillet stress-concentration factor (K_f)
Forming Factor Y is obtained from Figure 4.

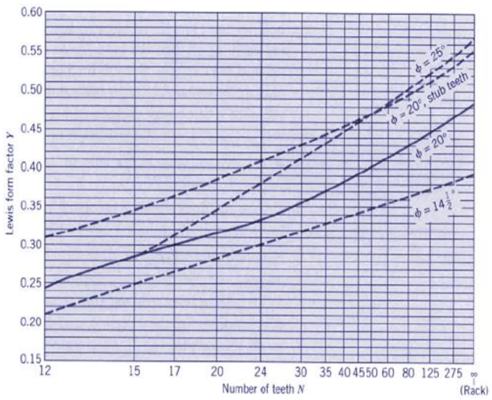

Figure 4. Forming factor Y [10]

Based on the AGMA equation, the maximum stresses in the gears are calculated as in Table 4.

Table 4. Maximum calculated stresses in the gears

Gears	Maximum Stresses σ_{max} (MPa)
$\sigma_{max\ z1}$	67.255
$\sigma_{max\ z2}$	40.72
$\sigma_{max\ z3}$	185.96
$\sigma_{max\ z4}$	103.81
$\sigma_{max\ z5}$	428.41
$\sigma_{max\ z6}$	226.64

The mechanical properties of the gear material are presented in Table 5 [11-12].

Table 5. Properties of gear materials

Properties	*AISI 5110 (16MnCr5)* [11]	*AISI 8620* [12]
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}

The safety factor for all gears is assumed as $S=1.5$. From the comparison of Table 4 and Table 5, the calculated maximum stresses are lower than allowable stresses so one can see that all gears are durable from an analytical static stress analysis perspective.

5 Finite element analysis

5.1 Materials

The ANSYS program begins by defining materials [13]. First, the materials to be used must be written in the engineering data section. 3 materials are used in this work. 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 a pitch diameter bigger than 50 mm (Gear 2, 4 and 6), respectively. Structural steel is used for keys, which is the default option of the program. After entering the engineering data tab, add material and change the data of the structural steel option and write the properties of the desired material according to the material in the design.

5.2 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 the key and shaft must be bonded because these parts will move together.

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

"No separation" option is selected since gears cannot rotate in the same direction.

5.3 Meshing

Mesh generation is one of the most important parameters 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. The important thing here is, considering the computing resources, how better split the model into smaller pieces using the selected element.

In 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 is increased by increasing the number of meshes and nodes in the sensible areas where higher stress concentrations are expected.

Sphere-shaped body sizing is considered in the region where the highest stresses are expected, Figure 5. In these regions, remarkable finer mesh densities are used for more accurate results. Outside of the spherical region where lower stresses are expected, coarse meshes are used.

To create a body sizing mesh, a new coordinate is created from the coordinate system and its spherical shape is created according to that coordinate system. Considering this shape, the spherical body radius is 10 mm smaller, but it may take more time to solve the process according to the performance of the computer.

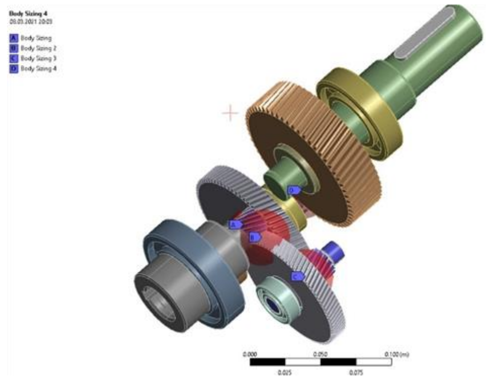

Figure 5. Influence of spheres used for mesh refinement

These operations are repeated in all areas between gear-gear and gear-wedge locations. It is worth pointing out that finer mesh densities may give more accurate results, but they may exceed the capacity of the compute or may take longer CPU time.

5.4 Loading

The FEM analysis of the whole gearbox is performed at this stage. To achieve this, the end of the output shaft and bottom surfaces of the casing is fixed, and the input moments are introduced from the input shaft, as shown in Figure 6 and Figure 7. By doing this, the static analysis of the whole system together with the gearbox case is obtained.

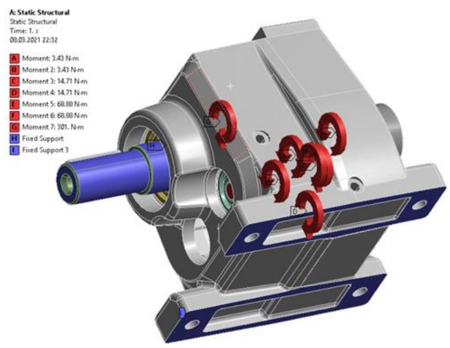

Figure 6. Fixed supports

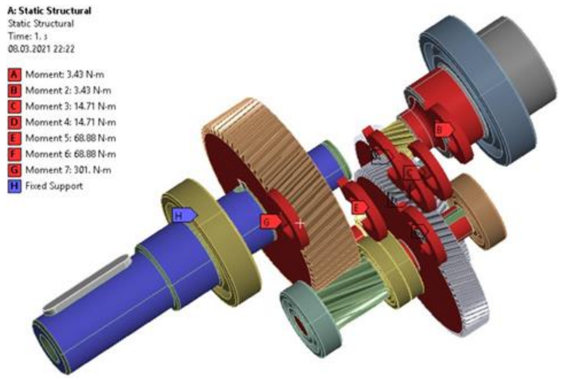

Figure 7. Moment application to shafts and fixed supports

6 Analysis results

There are 6 gears in contact with each other. Two parameters of the analysis results are presented here. These are; total deformation and equivalent (von Mises) stress.

6.1 Results of static analysis

In this part of our analysis, the end of the output shaft and the bottom surfaces of the gearbox case are fixed, and a torsional moment is applied from the input shaft of the gearbox. The driving force behind this analysis is to calculate the static stresses and deformations of the gearbox, namely total deformation and maximum and minimum principal stresses, and equivalent (von Mises) stress. A logarithmic scale is chosen for color change and results chart bar. Those calculated parameters for gears and shafts are depicted in Figure 8 and Figure 9.

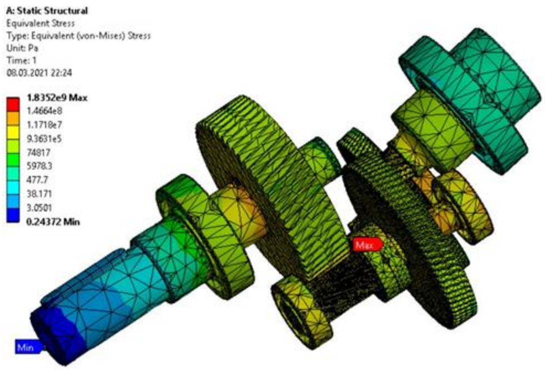

Figure 8. Equivalent (von Mises) stress results of gearbox

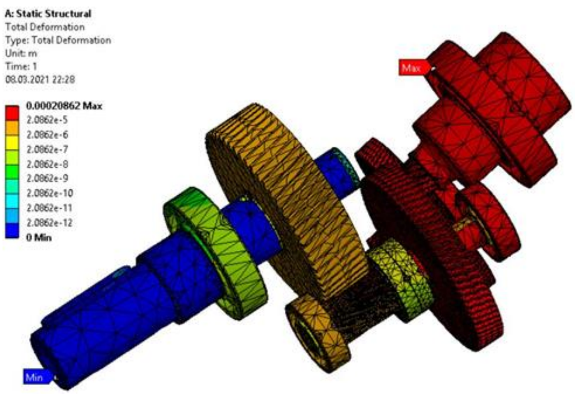

Figure 9. Total deformation of gearbox

To be able to see the deformation and stress on the gearbox casing, torsion is applied from the input shaft, and the output shaft and bottom of the casing are fixed, as presented in Figure 10.

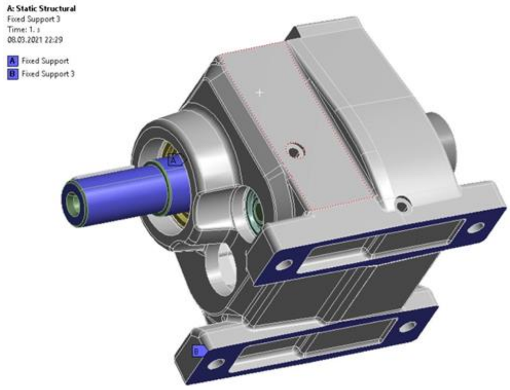

Figure 10. Fixed supports at casing, input, and output shafts

The logarithmic scale is selected for presenting the calculated parameters in a better visual format. Equivalent (von Mises) stress and total deformation results are presented in Figure 11 and Figure 12, respectively.

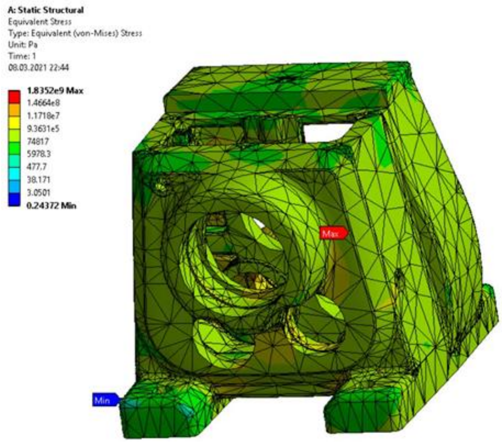

Figure 11. Equivalent (von Mises) stress results of the gearbox casing

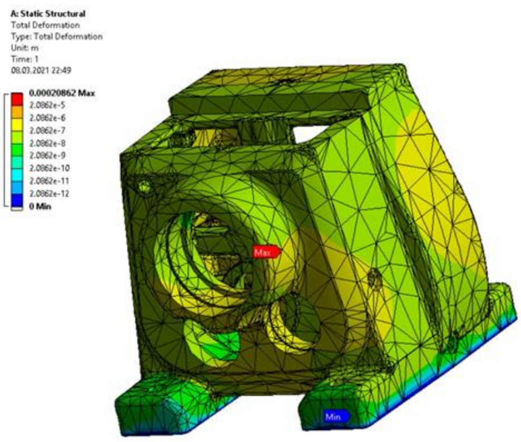

Figure 12. Total deformation of the gearbox casing

6.2 *Results of dynamic analysis*

6.2.1 *Mode shape analysis*

The mode shape is a special vibration pattern that is carried out by a mechanical system at certain frequencies. Depending on the stiffness of the engineering structures, different mode shapes is associated with different frequencies. The experimental technique of modal analysis explores these modal shapes and frequencies, but it is rather time-consuming and costly.

In the mode shape analysis of this three-stage gearbox, maximum 6 modes are selected. Total deformation is solved by selecting modes from 1 to 6. The image of the maximum and minimum points of the frequencies formed in the gears are depicted separately for gears and gearbox case in Figure 13.

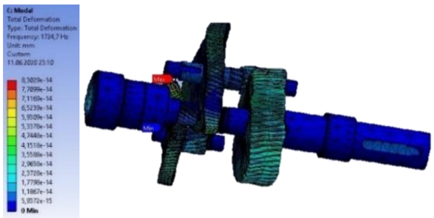

Figure 13. Results of mode 1 analysis of gears

6.2.1.1 *Gears*

The analysis results have revealed different mode shapes at different frequencies. The frequency results obtained are depicted in Table 6.

Table 6. Frequency of modes

Mode	*Frequency (Hz)*
1	1724.7
2	1902.4
3	2808.7
4	2842.6
5	3048.1
6	3069.4

6.2.1.2 *Gearbox casing*

In this part of the analysis, the mode shape analysis is carried out in maximum 6 modes using a case together with gears. The shape and the result in mode 1 are given in Figure 14.

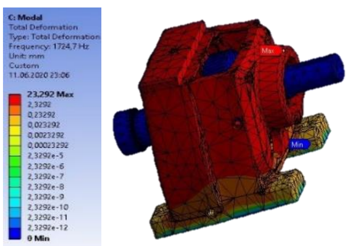

Figure 14. Results of mode 1 analysis of gearbox casing

6.2.2 *Natural frequency analysis*

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

In this analysis, gears and casing are analyzed in 6 modes. To evaluate the behavior of gear at the resonance frequency, the natural frequencies of the gear are compared with operating frequency. The gearbox manufacturer has foreseen the operating speed of the gearbox for the range of 10,86 to 1390 rpm.

$$Fn = \frac{1}{2\pi}\omega \qquad and \qquad \omega = \frac{2\pi N}{60} \tag{5}$$

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

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

Thus, the operating frequency range of the gearbox is in between 24 Hz to 0.1 Hz approximately. This range is very small compared with the lowest natural frequency of 1724.7 Hz. Hence FEM results reveal that resonance will not take place on the gearbox and the design is safe for both gears and casing. Min frequency is 1724.7 Hz, and the maximum frequency is 3069.4.

7 Discussions

7.1 *Discussion of static analysis*

In this section, the results presented here are discussed. The stress occurring on each gear under static loading is questioned compared with the yield strength of the gear material.

In the analysis, we observed the maximum von Misses stress value of 994.86 MPa on the 3rd gear. It is foreseen that this excessive von Misses stress is above the yield strength, which is 833 MPa, and as a result, it appears that a failure is inevitable on the gear ends. These higher stresses are attributed to the geometric incompatibility in the gearbox system, mainly the distance between the related shaft axes. Another source of these high stresses could be the misalignments of the shafts [14]. These misalignments may result from elastic deformation, manufacturing, and/or assembly errors.

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 has a value of 35,16 MPa which is well below the yield strength of the casing material, (460 MPa). As a result, it could be concluded that there is room for reducing the weight of the gearbox casing.

7.2 *Discussions of dynamic analysis*

7.2.1 *Mode shape analysis*

In this section, 6 modes are selected, the modal analysis is performed, and mode shapes are obtained. The most noticeable result is to observe that, regarding the mode shapes of gears, shafts, and casing, the total deformation of the casing at all the natural frequency values is much higher than the gears. It might mainly since the material and the stiffness of the gearbox casing is different from the shafts and gears.

7.2.2 Natural frequency analysis

When the results of the natural frequency analysis are examined, it is much less than the calculated operation frequencies of the system. Thus, the results of this analysis reveal that this gear system with the given design parameters is safe against resonance.

8 Conclusions

In this study, analysis of the three-stage gearbox is carried out in Ansys using the finite element method. Static and dynamic analyses of gears, shafts, and casing of the gearbox are performed. After the necessary boundary conditions are set, the parts' materials are assigned. After carefully creating the meshes setting the boundary conditions, for obtaining more accurate solutions, finer meshes are crated at critical locations. After obtaining the solution, total deformation, von Misses, maximum principal stresses, and minimum principal stresses options are paid special attention in static analysis results. In the calculations section, it is examined whether the gearbox can withstand the targeted torques with a 1.5 safety factor.

Besides the static analysis, the dynamic analysis is carried out in the system. The modal analysis option of Ansys software is used in dynamic analysis. Thus, the mode shape analysis and the natural frequency analyzes are carried out. While creating mode shape analyzes, maximum of 6 modes are used and frequency values in each mode are examined. In the natural frequency analysis section, F_n values are calculated by considering the input and output rpm values in the gearbox. The targeted operating values are in a much smaller range than the frequency values obtain in different modes in the program. Thus, there will be no resonance for the gearbox design, and it can be used safely.

In this study, the static and dynamic analyzes of the three-stage gearbox are targeted and the results of the analysis are presented and discussed. To increase the accuracy of the analysis and obtain more precise results, the analyses could be made on powerful computers. Furthermore, the gears and wedges in contact with each other have been contacted using the no separation option, more powerful computing resources may give better results if the frictional option is considered in the analysis. Another issue is that in this study spheres are created to tighten the mesh processes, but in the regions where higher stresses are expected, the body sizing method could be considered for more accurate and sensible results.

Declaration of interest

The author declares that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Similarity rate (iThenticate): %12

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