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RESEARCH ARTICLE OPEN ACCESS

Design Optimization and Analysis of A Gear Box Used In Agricultural Machinery

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ABSTRACT:
The agricultural machineries are used world-wide to facilitate the various agricultural work. In this work, we have considered the gear box of agricultural machine consisting of an arrangement of transmission shaft, spur and bevel gears. Presently, this gear box is facing the problem of heating up during the operations. To overcome this undesired problem, the limitations in the process of designing, are identified and analyzed with the help of the software tool used in the machine design applications. The designing of CAD model, structural and thermal analysis of the existing and modified gear box was carried out by CREO and ANSYS software. The main focus of the work was to reduce the stresses and temperature of the existing gear box. Firstly, the designs of existing spur gear pair are modified, by making the changes in their geometric dimensions and performed structural analysis. By considering the structural analysis results, the designed optimal gear box was selected and taken for further analysis. The designed gear box has shown the considerable reduction of the shear stress and von-mises stress as compared to the existing gear box. Secondly, performed the thermal analysis throughout the designed gear box. And on the comparison of the results of thermal analysis and temperature distribution of the designed gear box with the existing gear box, it was found that, there was reduction in temperature of designed gear box. The validation of the design optimization process was carried out by the experimental method. From the results, it was found that, the contact stress of the designed gear box was also reduced.
***Keywords*:** ANSYS, Bevel gear, CREO, Contact stress, Spur gear.

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I. INTRODUCTION

Due to the crisis of aging population and the sharp decrease of agricultural labour force, India attaches more and more importance to the development of agricultural machinery. However, there are many kinds of agricultural machinery, and the production batch is small. As a key part of agricultural machinery, gear is complex and difficult to model; there is a lot of repeated design work, which seriously affects the research and development cycle of agricultural machinery. The key to the sustainable development of agricultural machinery is to introduce modern design methods such as knowledge engineering, 3D reconstruction modelling and virtual simulation to improve the intelligence and automation of agricultural machinery product design in India.

Knowledge engineering is the basic theory of virtual design. In recent years, many organizations at home and abroad have carried out in-depth research on knowledge engineering, and achieved a series of important results. Agricultural University of India has carried out a lot of work on intelligent agricultural machinery and equipment, and established key model knowledge base such as crawler tractor.

In the process of gearbox virtual design and simulation, there are many gear models and low knowledge reuse rate, which affect the research and development cycle of the whole machine. In order to improve the level of intelligent design of agricultural machinery, this research takes the design of gearbox and spur gear assembly as an example, applies the methods of knowledge engineering and parametric design to the virtual design of gears, develops the virtual design system of gearbox gears based on knowledge, and verifies the feasibility and effectiveness of this method through in study.

II. RELATED WORK

Pradeep Kumar Singh et. al. (2014), studied the Hertz theory of contact stress calculations and used Lewis formula for calculating the bending stress in a pair of gear which plays significant role in the design of gears. Theoretically, the results obtained by Lewis formula and hertz equation are compared with the results of finite element analysis of spur gear. And from the results it was found that, if the contact stress minimization is the primary concern and large power is to be transmitted then spur gears with higher model be preferred.

Krishanu Gupta et. al. (2015), studied the equivalent stresses and strains of the symmetric involute spur gears through finite element meshing simulation for finding out the gear pair with least stress when the pressure angles are increased. A comprehensive study on the variation of static stresses with four different pressure angles that might be developed in spur gear teeth, using a commonly used finite element-based software package.

Ahmed Saeed Mohamed et. al. (2018), investigation was conducted in both time and frequency domains. A finite element analysis was performed to determine the variation in stiffness with respect to the angular position for different combinations of crack lengths. A simplified nonlinear lumped parameter model of a one-stage gear box with six degrees of freedom was then developed to simulate the vibration response of faulty external spur gears. Four different multiple-crack scenarios were proposed and studied.

B. Sivakumar et. al. (2018), analysis was carried out by optimizing the material such as carbon fibre high modulus. The results such as total deformation, equivalent elastic strain and equivalent stress for each material are determined. Compared the optimized materials and the conventional material, carbon fibre high modulus material has the low values of total deformation, stress and strain. The results found that, the carbon fibre high modulus material was suitable for the spur gear manufacturing.

Anirudh Tallam et. al. (2020), studied contact stress of spur gear by application of static analysis. Using Finite element method, a numerical study has been carried out to examine the effects of the material on contact stress behavioural characteristics of a gear. The gears used for the analysis are spur gears and the contact stress analysis is performed between pair of mating gear teeth. The benefit of comparing is to calculate the bending stress analysis using Lewis equation and contact stress with principle of Hertz equation. The results found are satisfactory in comparison with the Hertz equation.

III. PROPOSED METHODOLOGY

3.1 Problem Definition

a) We have studied that, when the torque is transmitted in the gear box through the transmission shaft the temperature increases due to the presence of friction between the mating teethes of the gear.

b) The lubricant is used to reduce the friction and hence the temperature of the gear box.

c) Checking through FEA, the temperature of the existing gear box and also of optimized or designed gear box by making the modifications in the surface area, diameter or teeth of the spur gear pair.

d) If the temperature of the designed gear box decreases, then we will compare the results with the existing one.

e) For performing analysis, the Power required is considered and the value of Torque for a given RPM is calculated. And carrying out the design calculations for the validation of the optimization process.

3.2 Objectives

a) To study existing design and identify the scope in design modifications in the gear box.

b) To study the geometric and operating parameters of the existing gear box and perform analysis.

c) To modify or design the gear box model by changing the geometric parameters of the spur gear pair and study its operating parameters.

d) To perform the structural and thermal analysis and undertake the comparative study of the designed gear box with the existing gear box.

e) To carried out the design calculations of the existing and designed gear box.

3.3 Select Appropriate Data

To perform the above processes of analysis the following steps are undertaken.

Mechanical Properties of the existing gear box materials are taken and the design calculations are done. The CAD Models of existing and modified gear box are created in the CREO Software. These mechanical properties are also used while performing structural and thermal analysis using ANSYS software.

A) Spur Gear & Pinion Shaft

The existing gear material given for our optimization process is 20MnCr5. The table 1 shows the mechanical properties of the material 20MnCr5.

Table 1: Mechanical Properties of 20MnCr5

Sr. No.	Mechanical Properties	Units	Value
1	Ultimate Tensile Strength	MPa	520
2	Young's Modulus	GPa	210
3	Shear Modulus	GPa	50
4	Poisson's Ratio	-	0.3
6	Density	Kg/m3	7800
7	Thermal Conductivity	W/m2 K	45.6

B) Bevel Gear

The existing gear material given for our optimization process is EN353. The table 2 shows the mechanical properties of the material EN353.

Table 2: Mechanical Properties of EN353

Sr. No.	Mechanical Properties	Units	Value
1	Ultimate Tensile Strength	MPa	1030
2	Young's Modulus	GPa	190
3	Shear Modulus	GPa	50
4	Poisson's Ratio	-	0.3
6	Density	Kg/m3	8000
7	Thermal Conductivity	W/m2 K	11.3

C) Gear box Casing

The existing gear box casing material given for our optimization process is SG Iron 450/15. The table 3 shows the mechanical properties of the material SG Iron 450/15.

Table 3: Mechanical Properties of SG Iron 450/15

Sr. No.	Mechanical Properties	Units	Value
2	Young's Modulus	GPa	204
3	Shear Modulus	GPa	79
4	Poisson's Ratio	-	0.29
6	Density	Kg/m3	7300
7	Thermal Conductivity	W/m2 K	36

IV. DESIGN CALCULATIONS

4.1 Design of Spur gear

Speed of pinion Np = 540 rpm

Power $p1$ = 41.0135 w

$velocity\ ratio\ = \frac{T_w}{T_p}$ = 1.058

$Pressure\ angle\ \emptyset = 20^0$

Max safe stress = 296 MPa i.e. 20 MnCr5 = 520 MPa

Shear stress = 94 MPa i.e. shear modulus = 50 GPa

Starting torque is 25 % higher than running torque, Spur gear should be design for power

$p1\ =\ 1.25 \times\ 41.035\ \times 10^3\ =\ 51266\ W$

m = modulus

for the Gear and pinion is

$m_{gear}\ = \frac{D}{T} = 6.66$

$m_{pinion}\ = \frac{D}{T} =\ 6.73$

velocity of the pitch line

$v = \frac{\pi \times D_p \times N_p}{60} =\ 3.235\ m/sec$ (1)

Assume steady load condition and 8-10 hours of service per day the Service Factor (Cs) is Cs =1

Design tangential tooth load

$W_T\ = \frac{P_1}{V} \times Cs\ =\ 15847.29$ N (2)

ordinary cut gear and operating at velocity up to 12.5 m/s

$velocity\ factor\ Cv\ = \frac{3}{3+V} =\ 0.4811$ (3)

we know that 20^0 involutes tooth from factor for pinion

$Y_p\ =\ 0.124\ - \frac{0.684}{\mathrm{T}} =\ 0.084$ (4)

b = force width for both pinion and gear

$W_T\ =\ Tangential\ tooth\ load\ design$

$W_T\ =\ W_P \times\ b\ \times\ \pi \times\ m\ \times\ Y_P$ (5)

$b\ =\ 17.168\ mm$

$force\ width\ =\ 6\ \times\ b =\ 103.00\ mm$

for other proportion for pinion and gear having 20^0 involute teeth

1] $Addendum\ =\ 1m\ =\ 6\ mm$

2] $Dedendum\ =\ 1.25m\ =\ 1.25\ \times\ 6\ =\ 7.5\ mm$

$working\ depth\ =\ 2m\ (m\ = module) =\ 12\ mm$

$minimum\ total\ depth\ =\ 2.25m\ =\ 2.25\ \times\ 6\ =\ 13.5\ mm$

$Tooth\ thickness\ =\ 1.5708m\ =\ 1.5708\ \times\ 6\ =\ 9.4248\ mm$

$minimum\ clearance\ =\ 0.25m\ =\ 0.25\ \times\ 6\ =\ 1.5\ mm$

Design of pinion shaft

Normal load acting between tooth surfaces

$W_n\ = \frac{W_T}{\cos \emptyset} =\ 16864.33\ N$ (6)

Weight of pinion

$W_p\ =\ 0.00118\ \times Tp \times b \times m^2\ =\ 74.38\ N$ (7)

Resultant load acting on pinion

$W_p\ = \sqrt{W_N^2 + W_p^2 + 2\ W_n \times W_p \times \cos \emptyset}$ (8)

$=\ 16934.24\ N$

Assuming that pinion is overhung on shaft taking 100 mm

Bending moment on shaft due to resultant load

$M_1\ =\ Wp\ \times\ 100\ \ =\ 1693424\ Nmm$ (9)

Twisting moment on shaft

$T\ =\ W_T\ \times\ \frac{D_p}{2} =\ 907257.35\ Nmm$ (10)

Equivalent twisting moment (Te)

$T_e\ =\ \sqrt{M^2 +\ T^2}\ =\ 1421145.52\ Nmm$ (11)

Equivalent twisting moment T_e

$T_e\ = \frac{\pi}{16} \times\ \tau\ \times\ d_p^3$ (12)

$d_p\ =\ 25.15\ mm$

$Diameter\ of\ pinion\ hub\ =\ 1.8\ \times\ dp\ =\ 1.8\ \times\ 25.15\ =\ 45.27\ mm$

$Lenght\ of\ Hub\ =\ 1.25\ \times dp\ =\ 1.25\ \times\ 25.15\ =\ 31.43\ mm$

Design of Gear Shaft

$W_N\ =\ 16864.33$

weight of gear

$W_G\ =\ 0.00118\ \times T_G \times b \times m^2 = 78.75$ (13)

Resultant load acting on gear

$W_R = \sqrt{W_N^2 + W_G^2 + 2\ W_n \times W_G \times \cos \emptyset}$ (14)

$=\ 16938.35\ N$

Taking over hung as 100 mm

$M\ =\ W_R \times\ 100\ =\ 1693835\ Nmm$

Twisting moment on shaft

$T = W_T \times \frac{D_G}{2} = 950820\, Nmm$ (15)

Equivalent twisting moment (Te)

$T_e = \sqrt{M^2 + T^2}$ = 1942456.09 Nmm (16)

Equivalent twisting moment T_e

$T_e = \frac{\pi}{16} \times \tau \times D_G^3$ (17)

$D_G = 25.24\, mm$

4.2 Design of Bevel gear

$\theta = 90^0$

$Power = 29840\, W = 40\, HP$

$T_P = 14$

$T_G = 22$

Ultimate stress = 1030

$\sigma_{tensile\ stress} = 551\, MPa$

$\sigma_{shear} = 0.5 \times 1030\, MPa = 515\, MPa$

$N_p = 571.76\, rpm$

$N_G = 363.84\, rpm$

$\emptyset = 22.30$

Material = EN353

Module = 6.25

$M_p = 6.25$

$\emptyset_p = 22.30$

$M_G = 2.75$

$\emptyset_G = 22.30$

shafts are at right angles,

Pitch angle for pinion

$\emptyset_{p1} = \tan^{-1}\left(\frac{1}{VR}\right)$ (18)

$\emptyset_{p1} = \tan^{-1}\left(\frac{T_P}{T_G}\right) = 32.47^0$

pitch angle for gear

$\emptyset_{p2} = 90^0 - 32.47^0 = 57.53^0$

Formative number of teeth for pinion

$T_{EP} = T_P \times \sec \emptyset_{p1} = 15.09$ (19)

Formative number of teeth for gear

$T_{EG} = T_G \times \sec \emptyset_{p2} = 23.72$ (20)

Tooth form factor for pinion

$Y_p' = 0.124 - \frac{(0.684)}{T_{EP}} = 0.0786$ (21)

form factor for gear

$Y_G' = 0.124 - \frac{(0.684)}{T_{EG}} = 0.095$ (22)

$\sigma_{tensile\ stress} \times \mathrm{y_p'} = 551 \times 0.0786 = 43$ (i)

$\sigma_{tensile\ stress} \times \mathrm{y_G'} = 515 \times 0.095 = 48.925$ (ii)

Product of (i) is less than (ii)

Gear is weaker

Thus, design is said to be based upon gear

Torque on gear

$T = \frac{P \times 60}{2\, \pi\, N_G} = 783.57\, N/m$ (23)

Tangential load on gear

$W_T = \frac{T}{\frac{D_G}{2}} = 25.9 \times 10^3\, N$ (24)

Pitch line velocity

$V = \frac{\pi\, D_G\, N_G}{60}$ (25)

$V = \frac{\pi\, m\, T_G\, N_G}{60} = 1.15\ \mathrm{m/s}$

Taking velocity factor

$C_v = \frac{6}{6 + V} = 0.845$

Length of pitch line elements

$L = \frac{D_G}{2 \sin \emptyset_{p2}}$ (26)

$L = \frac{m\, T_G}{2 \sin \emptyset_{p2}} = 80.75$ mm

Assuming face width (b) As 1/3rd of length of pitch line element L

$b = \frac{L}{3} = 26.91$ mm

Tangential load on gear

$W_T = \sigma_T \times C_v \times b \times \pi \times m \times Y_G' \times (\frac{L-b}{L})$ (27)

$m = 1.039$

$b = 10.67m = 11.08\, mm$

Gear diameter

Pitch diameter of pinion

$D_p = m\, T_p$ = 87.5 mm

pitch diameter of Gear

$D_G = m\, T_G = 60.5\, mm$

Check for dynamic load

Pitch line velocity

$V = 1.32m = 1.37\, m/sec$

Tangential tooth load on gear

$W_T = \frac{6820}{m} = 6564\, N$

V. CAD MODEL

The data from above design calculations are used for designing the models in CREO software. The CAD models of Spur gears Fig. 1 & 2, Bevel gears Fig. 3 & 4 and Gear box assembly with casing Fig. 5 are shown. The internal arrangement of gear box is shown hiding the casing from gear box assembly.

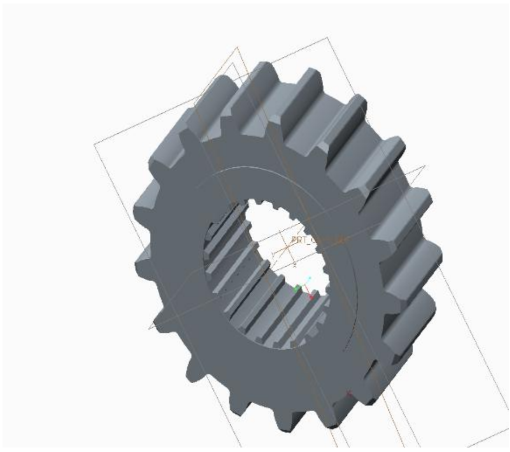

Figure 1: CAD model of spur gear z=17

Figure 2: CAD model of spur gear z = 18

Figure 3: CAD model of bevel gear z-22

Figure 4: CAD model of bevel gear z-14

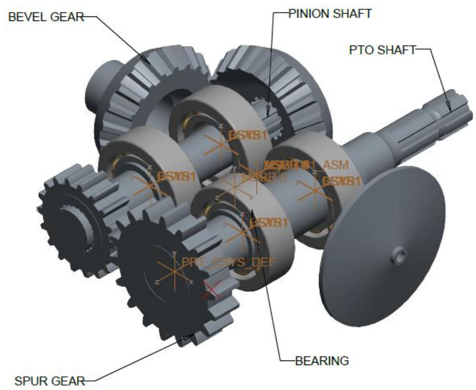

Figure 5: internal arrangement of gear box

According to the problem statement mentioned, we have mainly focused on the temperature reduction inside the gear box assembly by changing the dimensional parameters of the spur gear pair. Thus, we have increased the surface area of the designed CAD model see Fig. 7 and performed the structural analysis of this optimized gear pairs. From the results of the structural analysis see Fig. 10 & Fig. 11, the gear box with optimized spur gear pair has shown the less values of maximum von-mises stresses and shear stresses than the existing gear box. So, the optimized gear box obtained has been considered for thermal analysis.

Surface area of the gears has been increased by changing the radius of curvature of the gears. To validate the changes made in the radius of curvature, the contact stress of the gear pair is calculated and compared.

CAD Model of the Spur Gear Before changes made in the Radius of Curvature and the Curve Length.

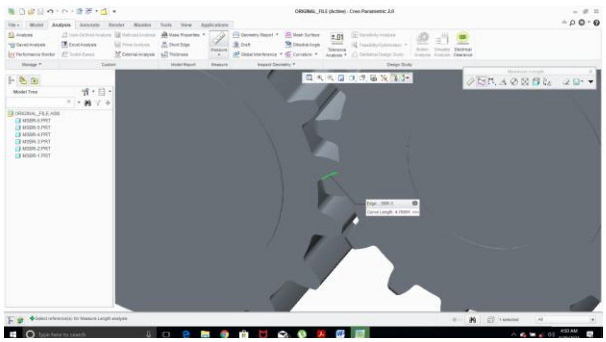

Figure 6: mentioning length l = 4.788

The contract stress is given by the formula.

$$\sigma_H = \sqrt{\frac{F_n \times \left(\frac{1}{P_1}+\frac{1}{P_2}\right)}{\pi \times \left[\frac{1-\mu_1^2}{E_1}+\frac{1-\mu_2^2}{E_2}\right] \times L}} \tag{28}$$

E1 = Elastic module of pinion
E2 = Elastic module of Gear
Fn = Normal Force acting on gear pair
P1 and P2 = Curvature radius of pinion and gear
L = total carrying length

$P_1 = 2.174\,mm \quad ; P_2 = 4.075\,mm$
$F_n = 15847.29\,N; E_1 = E_2 = 2 * 105\,N/mm^2$
$L = 4.788$

$$\boldsymbol{\sigma_H = \quad = 9.076 \times 10^3\ N/mm^2} \qquad (29)$$

CAD Model of the Spur Gear pair after changes made in the Radius of Curvature and the Curve Length. Length was reduced and radius of curvature was increased which leads to the increase in the surface area.

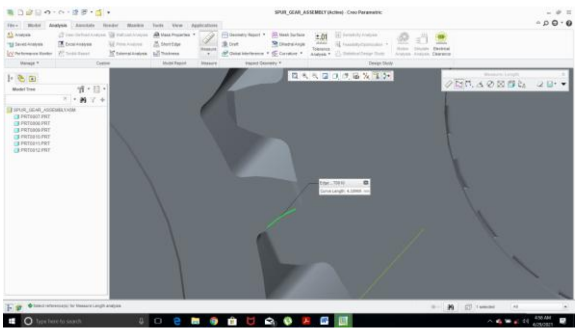

Figure 7: change made in length l = 4.329

$\rho_1 = 3.613$
$\rho_2 = 4.434$
$L = 4.329$

$$\boldsymbol{\sigma_H = 8.022 \times 10^3\ N/mm^2} \qquad (30)$$

From the calculations, the contract stresses before modification (29) and after modification (30) are compared. Now, we can say that, the stress is minimum when the radius of curvature is increased, which in turn results in increase of surface area. So, the CAD model Fig. 7 has been taken for the structural and thermal analysis performed in ANSYS.

VI. RESULTS AND DISCUSSIONS

After performing the structural and thermal analysis over the optimized gear box following results were recorded through ANSYS software. The structural analysis was performed by using the boundary conditions of the gear box. The rotational twisting moment on the transmission shaft was in all six degrees of freedom, only rotational degree was allowed while the other five were locked.

While during the Thermal Analysis, the boundary conditions, i.e. heat transfer co-efficient and the thermal conductivity of the material of components was used.

Finally, the results of the structural and thermal analysis were recorded and compared with the results of the analysis before modifications.

Structural Analysis Results

In Structural Analysis of the gear box, we have focused mainly on the results of the Shear and Von-mises stresses as all over the assembly, only rotational twisting moment over the shaft is considered throughout the complete analysis.

A) Results of the Existing Gear box Assembly

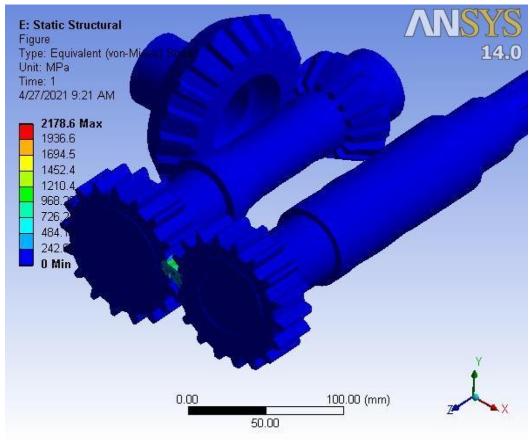

Figure 8: von mises stress

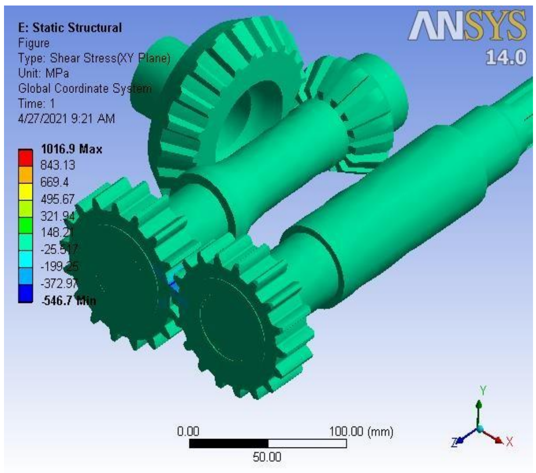

Figure 9: shear stress

B) Results of the gear box assembly when the Spur Gear pair was optimized and the bevel gear pair dimensions were not changed

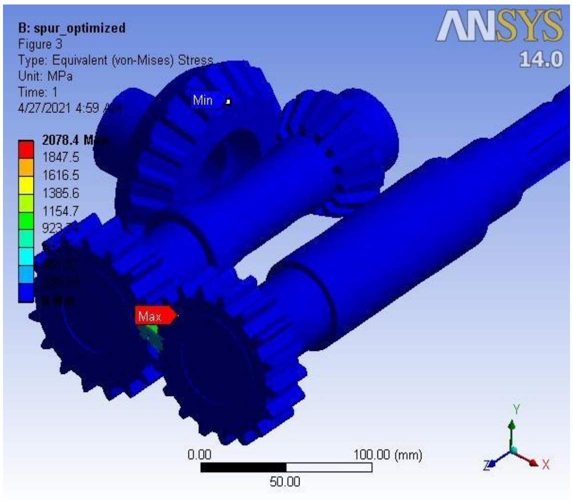

Figure10: von mises stress

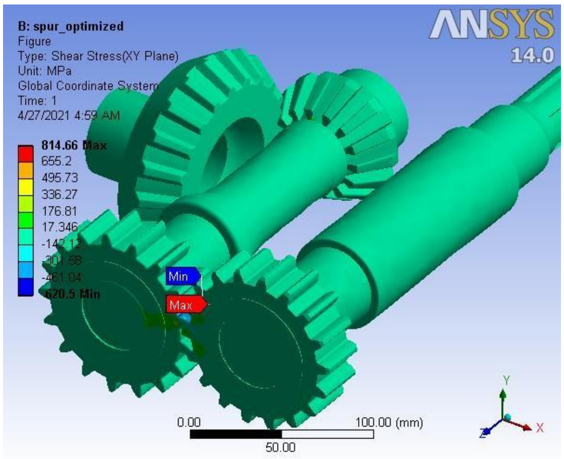

Figure11: shear stress

Thermal Analysis Results

During the thermal analysis of the gear box, we have focused mainly on the temperature reduction of the gear box. For this purpose, the changes were made in the designs i.e. dimensions of the spur gear pair. So, the results of the gear box with optimized spur gear pair were compared with the gear box before modifications. And it is found that, the maximum values of von-mises and shear stresses after modifications of the spur gear pair are less.

Temperature distribution of gear box Assembly Results**:**

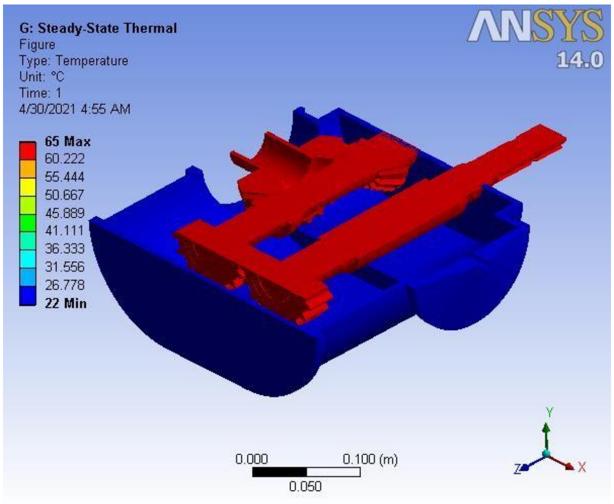

Figure 12: before modification

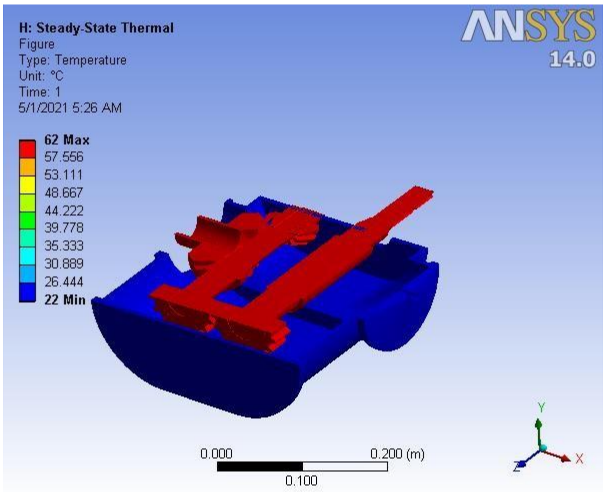

Figure 13: after modification

Table 4: Structural Analysis Results

Sr. No.	Existing Gear box Assembly		Optimized Gear box Assembly	
	Von Mises Stress (MPa)	Shear Stress (MPa)	Von Mises Stress (MPa)	Shear Stress (MPa)
1	242	-372	230	-461
2	484	-199	461	-301
3	726	-25	692	-142
4	968	148	923	17
5	1210	321	1154	176
6	1452	495	1385	336
7	1694	669	1616	495
8	1936	843	1847	655
9	2178	1016	2078	814

Table 5: Thermal Analysis Results

Sr. No.	Temperature Distribution Before Modification, (^{0}C)	Temperature Distribution After Modification, (^{0}C)
1	22	22
2	26	26
3	31	30
4	36	35
5	41	39
6	45	44
7	50	48
8	55	53
9	60	57
10	65	62

VII. CONCLUSION

From the results of structural analysis, we can conclude that, when the surface area of the spur gear pair is increased by changing the radius of curvature, the shear stresses as well as von-mises stresses are decreased. The maximum values of von-mises stresses and shear stresses before modifications of the gear box are 2178 MPa and 1016 MPa respectively. While the maximum values of von-mises stresses and shear stresses after modifications are 2078 MPa and 814 MPa respectively. The boundary conditions considered, while performing the structural analysis were rotational twisting movement over the shaft, torque for the motor power of 41 KWatts and speed of 540 rpm. Thermal analysis of the existing gearbox was performed by considering the boundary conditions of heat transfer coefficient, whose value is 11.53 W/m²·°C and the ambient temperature of the gear box casing to be 22^{0}C. From the results of thermal analysis of the optimized design, we can conclude that when the modified spur gear arrangement is taken, it leads to change in the value of heat transfer coefficient to 11.63 W/m²·°C. This certain change in the heat transfer coefficient leads to decrease the temperature inside the designed gear box and was found to be 62^{0}C after modification.

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