REST Journal on Emerging trends in Modelling and Manufacturing
Vol: 6(3), 2020
REST Publisher
ISSN: 2455-4537

Website: www.restpublisher.com/journals/jemm

Modelling and Analysis of Rack and Pinion Using Different Material Combination

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Abstract

A rack and pinion comprises a pair of gears that converts rotational motion to linear motion and vice versa. This is commonly used in steering systems of small trucks and SUV's other than in lifting mechanisms, machine tools etc. The magnitude of contact stresses on the pinion and rack is the same. However, the number of stress cycles is lower for the rack. Additionally, the rack and pinion tend to have greater amounts of backlash and consequently a significant amount of friction. In this project, modelling of rack and pinion in SOLIDWORKS and contact analysis in ANSYS Workbench using different material combinations of rack and pinion is performed and the stress distribution is analysed.

Keywords: Rack and pinion; Contact Analysis; SOLIDWORKS; ANSYS

1.Introduction

A linear actuator is an actuator which creates motion in straight line and it is needed in many mechanisms for transmission of motion and force. Rack and pinion is one such linear actuator which has a circular gear called pinion and a linear gear (infinite radius) called rack. These gear are meshed together in such a way to convert the rotary motion of pinion to linear motion of rack and vice versa. Rack and pinion is used in many applications such as stair-lifts, rack railways, steering mechanism, etc., The rack and pinion used in steering mechanism, rack and pinion are made up of same material like carbon steel. This conventional rack and pinion made of same material is subjected contact stress and often gets worn out easily. By using different material for rack and pinion we could possibly stress and also reduce the weight of rack and pinion as compared to the conventional rack and pinion which is already existing. With an intention to optimize the existing rack and pinion model, it would be effective to take an existing model, collect all data and parameters related to it and make a virtual prototype of it. By creating a CAD model of rack and pinion in software like SOLIDWORKS, it can be analyzed and the drawbacks existing can be found out. Furthermore, a methodology can be set to get a possible solution to optimize it. After the finding the drawbacks in it we could try different combination of materials of rack and pinion instead of same carbon steel. After using different combination, the model can be subjected to stress analysis in software like ANSYS for failure testing and test passed combination can be compared to the existing carbon steel model so we can find whether we can reduce stress and weight of the rack and pinion model with this new material combination. The idea of trying out different material combination was extracted from worm and worm wheel gear pairs which uses different material for worm and worm wheel. By doing so they have increased performance parameters. Thereby, trying out different material combinations in rack and pinion, the existing drawbacks similar to that of worm and worm wheel can be eliminated.

- Designing a CAD model similar to existing rack and pinion model in SOLIDWORKS.
- Selection of appropriate material combinations
- Contact Stress Analysis for the selected material combinations in ANSYS
- Comparison of the results with existing model and arriving at a conclusion by doing so, it is expected to have the following results:
- Reduced contact stress
- Reduction in weight.

Notations

b	Face width
C_s	Service factor
C_v	Velocity factor
l	Total width of contact line
m	Module
M_t	Maximum Torque
N_p	Number of teeth on pinion
N_r	No of teeth on rack
P_t	Tangential load
R_p	Radius of pinion
S_b	Beam strength

Y	Lewis form factor
α	Pressure angle
μ	Poison ratio
σ_b	Endurance limit stress
σ_H	Hertz contact stress

2.Literature Survey

Aksh Patel and et al. have provided the methodology for design and manufacturing of manual rack and pinion steering system. This paper provides efficient way to design durable and relatively inexpensive steering system for All-terrain Vehicle by using manual manufacturing of rack and pinion mechanism. It mainly provides the design calculation for rack and pinion for ATV steering system with rack travel as primary parameter. Khan Noor Mohammad and et al. have designed an efficient steering system for an All-terrain Vehicle using rack and pinion mechanism. C Factor was chosen as the primary parameter to design the rack and pinion and also calculation of steering effort was materialized. S. Rajesh and et al. have studied the load carrying capacity of normal contact ratio spur gear by high strength less weight materials and compared to EN24 steel gear which is predominantly used for gears. They have accomplished finding the bending and contact load capacity through finite element analysis in 2-D mode and compared the result with conventional gears. Xiao Yanjun and et al. have established a 3-D modal of rack and pinion transmission system in UG software and analysed it using finite element analysis software, ANSYS. Modal analysis to get first six natural frequencies and transient structural analysis to get dynamic contact behaviour is accomplished. By studying the analysis, they verified the contact strength and beam strength of the gear and also found the maximum equivalent stress position. These helped them to provide a theoretical base for the optimization of the gear and rack. WANG Qingguo and et al. have conducted finite element analysis for finding out the main parameters influencing the carrying capacity of pinion in a jacking system. The main parameter that influences the carrying capacity of pinion was found according to the calculation method in GB standard. The discussion was made by establishing the relationships between the carrying capacity of pinion can and parameters such as tooth width, normal modulus, pressure angle, modification coefficient from which it was concluded that increase in tooth width and module shows positive trend in carry capacity as well. Marimuthu and et al. have determined the maximum contact and bending stress through altering the geometry of asymmetric spur gear and examined through load sharing ratio using a numerical method. A unique Ansys parametric design language code was developed to find the load sharing ratio, maximum fillet and contact stresses. The fillet stress was calculated in terms of non-dimensional stress. The influence of gear drive parameters such as drive and coast side pressure angles, top-land thickness coefficients, contact ratio, coefficient of asymmetry, gear ratio and teeth number on load carrying capacity has been studied extensively on non-dimensional fillet stress and maximum contact stress and compared with that of the conventionally designed gears. Prabhu Sekar and et al. presented an idea to remove the inequality in maximum fillet stresses developed between pinion and gear of a step up gear drive. This uniform fillet strength of the gear drive can be achieved by using non-standard pinion and gear with appropriate addendum modifications generated by non-standard basic racks. The influence of gear parameters such as gear ratio, pressure angle, addendum factor, pinion teeth number and addendum modifications on the maximum fillet stress on the non-standard pinion and gears of different tooth thickness was analysed through finite element method (FEM).

3.**Methodology**

Step 1: Primary parameter assumption based on literature survey.

Step 2: Design calculation and validation of rack and pinion for different material combinations.

Step 3: Modelling and assembling 3D CAD model of rack and pinion using SOLIDWORKS.

Step 4: Performing Dynamic Contact Analysis using ANSYS workbench for different material combinations.

Step 5: Comparing results and conclusion based on the same.

3.1. Design Parameters

The design calculations of the rack and pinion was carried out for the intended application in steering system of off-road vehicles. The parameters for the same were taken with concordance to literature survey executed. The primary design parameters assumed ass shown in Table 3.1.

Table 3.1 Primary Parameters

Parameters Assumed			
Parameters	**Values**	**Round Off**	**Units**
C FACTOR	110	110	mm
Pressure angle	20	20	°
M_t	13000	13000	N-mm

C Factor: The amount of linear distance travelled by the rack for one complete rotation of pinion is known as C-Factor. It determines the type of steering response provided by the rack and pinion gearbox.

Pressure Angle: 20° full depth involute gear profile was chosen over 14.5° full depth involute gear profile for the following advantages:

a) Reduces risk of undercutting and interference

b) Increased pressure angle provides broader gear tooth at root and provides better strength, load carrying capacity and length of contact.

The torque was assumed based on the nominal steering effort required for off-road vehicles. Based on the above assumed parameters, the geometric parameters of the rack and pinion were evaluated as shown in the Table 3.2 by the following formula.

$$\mathrm{C\text{-}Factor} = 2\,\pi\,\mathrm{R_p} \tag{4.1}$$

$$\mathrm{m_{max}} = \mathrm{R_p}\,\sin 2\alpha \tag{4.2}$$

$$\mathrm{N_p} = 2\frac{\mathrm{R_p}}{\mathrm{m}} \tag{4.3}$$

$$\mathrm{N_r} = \frac{\mathrm{C\text{-}Factor}}{\pi\,\mathrm{m}} \tag{4.4}$$

$$\mathrm{Y} = 0.484 - \frac{2.87}{\text{no. of teeth}} \tag{4.5}$$

$$\mathrm{b} = 10\mathrm{m} \tag{4.6}$$

Table 3.2 Geometric Parameters defining Rack and Pinon

Parameters	**Values**	**Round Off**	**Units**
R_p	17.50704374	18	mm
m_{max}	2.047935084	2	mm
Assuming m	1.5	1.5	mm
N_p	24	24	-
N_r	23.34272499	24	-
Y_r	0.3644	0.3644	-
Y_p	0.3644	0.3644	-
b	15	15	mm

3.2. Design Calculation

C-45 Steel, Aluminium and Bronze were selected for Rack and Pinion due to their availability, ductility and heat dissipation respectively. The material properties are shown in Table 3.3. These values are substituted to get the Lewis Beam strength of gear tooth by assuming gear tooth as a cantilever and Hertz contact stress as specified in Table 3.4.

$$\mathrm{P_t} = 2\frac{\mathrm{M_t}}{\mathrm{d_p}} \tag{4.7}$$

$$\mathrm{P_{eff}} = \frac{\mathrm{C_s}}{\mathrm{C_v}} P_t \tag{4.8}$$

$$\mathrm{S_b} = \mathrm{bm}\sigma_{\mathrm{b}}\mathrm{Y_p} \tag{4.9}$$

$$\sigma_{\mathrm{b}} = \frac{\mathrm{UTS}}{3} \tag{4.10}$$

$$\mathrm{FOS} = \frac{\mathrm{S_b}}{\mathrm{P_{eff}}} \tag{4.11}$$

$$\sigma_H = \sqrt{\frac{P\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}{\pi l\left(\frac{1-\mu_1^{\,2}}{E_1}+\frac{1-\mu_2^{\,2}}{E_2}\right)}} \tag{4.11}$$

Table 3.3 Physical properties of materials chosen

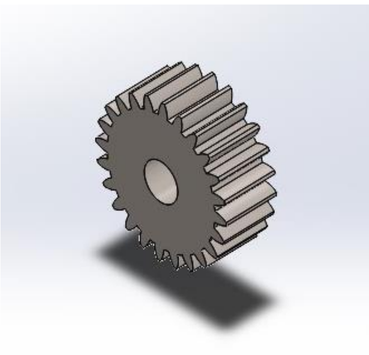
Material	**UTS (MPa)**	**Density (g/cm^3)**	**Young's Modulus (GPa)**	**μ**
STEEL (C45)	610	7.85	200	0.3
BRONZE	550	8.72	110	0.32
ALUMINIUM	310	2.77	68	0.32

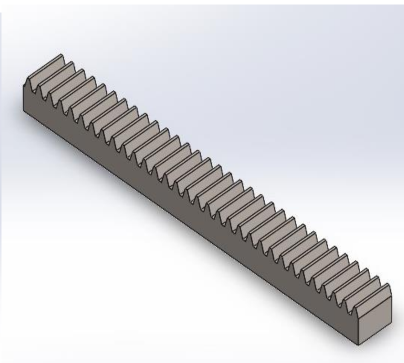
Table 3.4 Design calculation

Material	**P_t (N)**	**P_{eff} (N)**	**S_b (N)**	**FOS**	**σ_c (MPa)**
Steel (C45)	722.2222222	770.9773621	1667.13	2.162359211	180.963
Bronze	722.2222222	770.9773621	1503.15	1.949668141	154.68
Aluminium	722.2222222	770.9773621	847.23	1.098903861	130.97

3.3. Design Modelling

Having obtained the geometric parameters of rack and pinion, the CAD model was generated and assembled using SOLIDWORKS 2020 software as shown in Fig 3.1. The input parameters utilized to generate the model were Module = 1.5mm, Pitch Circle Diameter = 36mm, Face Width = 15 mm, number of teeth on rack = 24 and the number of teeth on pinion = 24. To get the rack and pinion assembled, rack and pinion mate was utilized from advance mechanical mate section. This enabled pinion to slide over rack simultaneously being in mesh with it as shown in Fig 3.2.





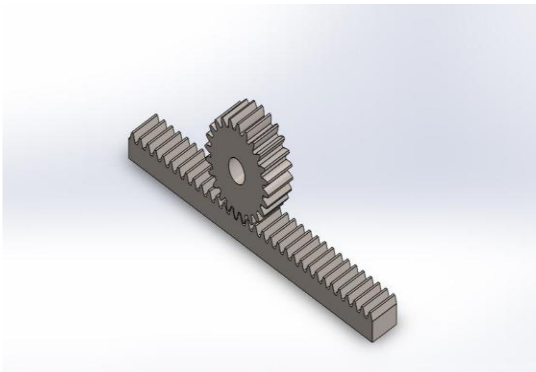


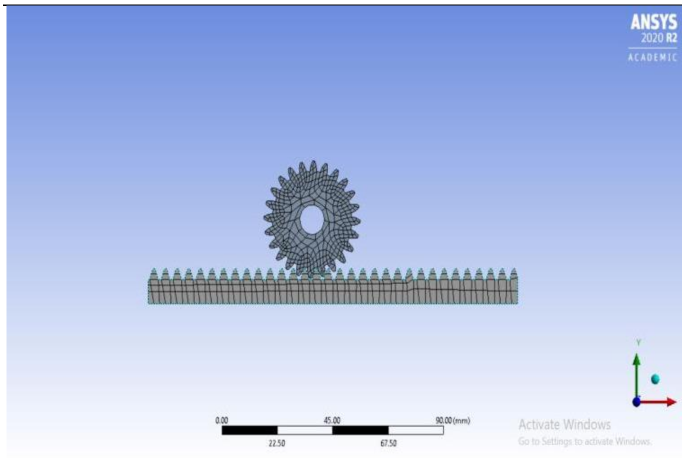
Fig.3.1. CAD model of Rack and Pinion; Fig. 3.2. Meshed assembly of Rack and Pinion

3.4. Design Analysis

Dynamic contact analysis is performed for the designed rack and pinion assembly using transient structural analysis module in ANSYS Workbench 2020 R2. The material combinations chosen were:

a) Steel Pinion - Steel Rack
b) Steel Pinion - Bronze Rack
c) Steel Pinion - Aluminium Rack

Predominantly, the contact stress of rack and pinion is calculated by traditional Hertz contact stress equation. However, the result of this standard formula is applicable only at pitch point and at the instant of gear meshing whereas the contact stress calculated by dynamic contact analysis is a range of values at different positions and instants of time which is more realistic and helps in predicting the physical behaviour without even building prototype. First, the CAD model is imported as. SLDASM file to ANSYS Workbench and the materials of our choice is imported to Engineering data. The contact is changed from Bonded to No Separation to have sliding in tangential direction but not the normal separation. The material assignment done by selecting the body and assigning the material. Optimal meshing can make the results more accurate and hence meshing is done with 0.272 transition ratio to get 11580 nodes and 1922 elements as displayed in Fig 3.3. The mesh size is taken such that not less than four elements can fill the tooth width of the gears. Dense meshing is done in contact region program controlled manner.



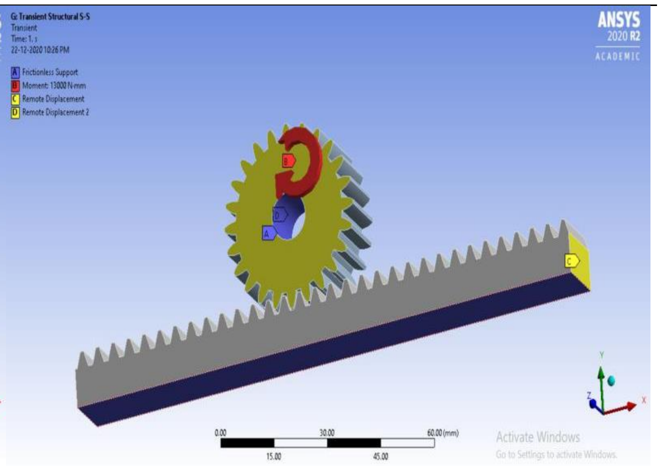


Fig. 3.3. Finite element model of Rack and Pinion; Fig. 3.4. Constrains on Rack and Pinion

In order to give the boundary conditions, constrains are imposed and the required degree of freedom is accomplished. For rack and pinion, the pinion must have only rotation about its axis and the rack must have only translation about the axis perpendicular to pinion's axis of rotation and along the length of rack. Therefore, remote displacement of rotation about Z axis and translation along X axis is given to pinion and rack respectively by making motion along and about other axis null. Furthermore, friction-less support is given between gears and casing. Moment of 13000 N-mm is applied about Z axis in a linearly increasing fashion from 0 N-mm at time = 0s to 13000 N-mm at time = 1s to accomplish dynamic load. The detailed schematic representation of constrains is shown in Fig 3.4. Using Mechanical ANSYS Parametric Design Language a unique code was developed to find the equivalent stress (Von Misses) developed in contact region to obtain results utilized to compare and conclude.

4.**Results and Discussions**

In order to find out the contact stress developed between the rack and pinion, transient structural analysis in ANSYS workbench is conducted. In steering application, the load on the pinion varies with respect to time since it is manually steered. Therefore, the value moment changes with respect to time. This is a typical characteristic of dynamic load which is reason for conducting transient structural analysis. In the ANSYS workbench, analysis is carried out for three different material combinations. In all these three models, pinion is assigned with C45 steel and only for rack, the materials assignment is altered. The three different materials used for rack are C45 steel, bronze and aluminium. The contact type has been changed from default bonded type to no separation for tangential sliding and normal restriction. The result from dynamic contact analysis had reported the maximum contact stress for steel-steel combination to be 162.6 MPa, for steel- bronze combination to be 152.8 MPa and for steel-aluminium combination to be 130.56 MPa. For comparison purpose, theoretically calculated the contact stress values for different material combinations using Hertz contact stress equation are provided along with the values from the FEA Analysis in the following table 4.1.

Table 4.1 Contact stress values for different material combinations (values in MPa)

Materials	**Hertz Contact Stress**	**FEA Contact Stress**
STEEL-STEEL	180.963	162.69
STEEL-BRONZE	154.68	152.83
STEEL-ALUMINIUM	130.97	130.56

The Fig 4.1 depicts the graphical representation of contact stress for different combination through the two methods. The different in values calculated by Hertz equation and FEA can be observed.

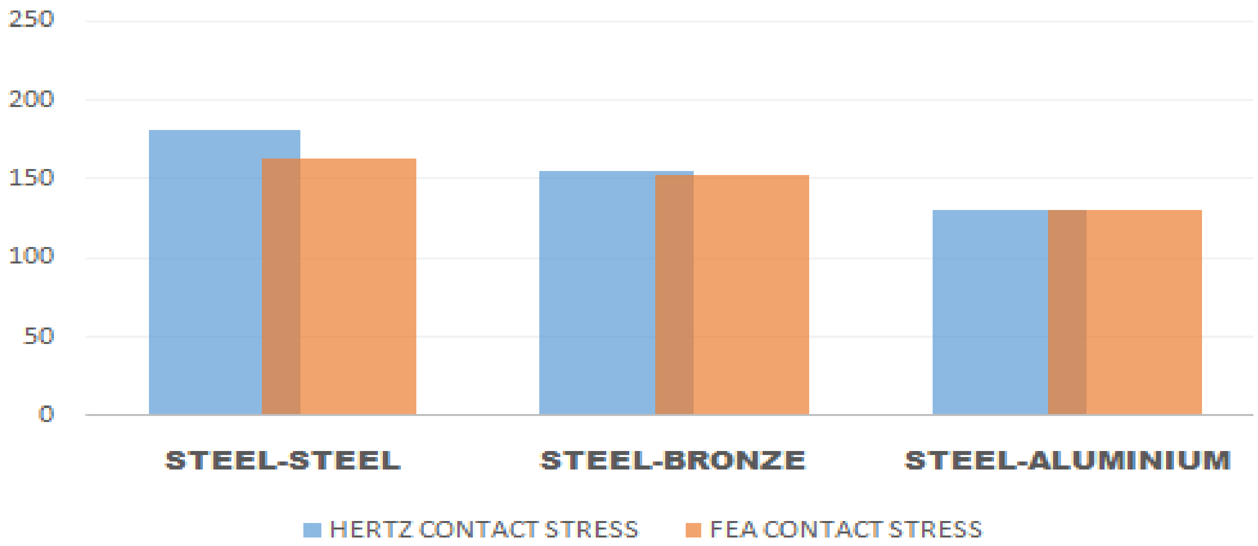
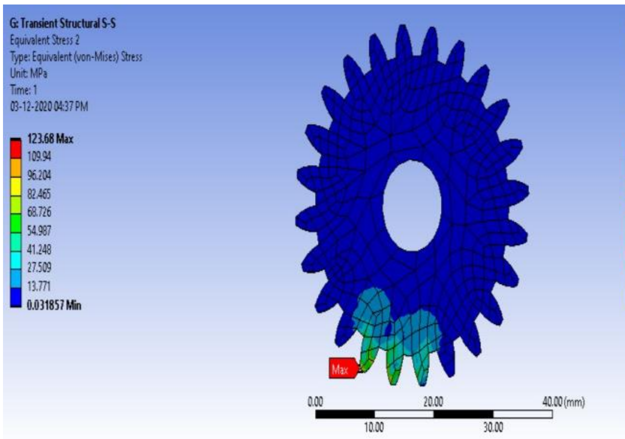


Fig. 4.1 Graph showing the comparison of contact stress values obtained from Hertz equation and FEA Analysis (Values in Y-axis represents contact stress in MPa).

The maximum stress on pinion for steel-steel combination is 123.68 MPa denoted by red colour and the tag max shown in the below Fig.4.2. Steel-Steel combination is the most widely used combination as of now in the Industry. Hence this combination was taken as reference to compare the results.



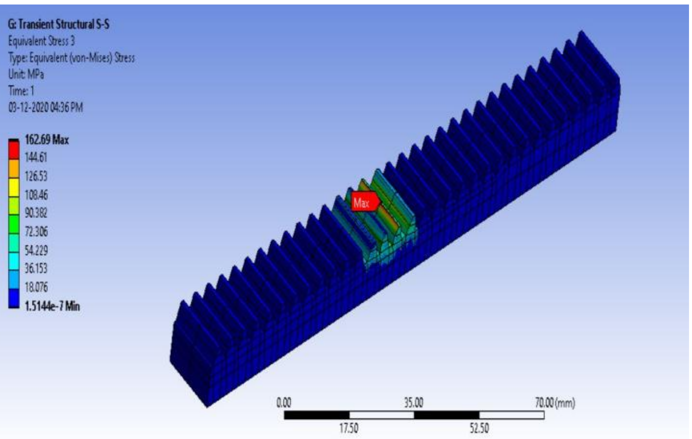
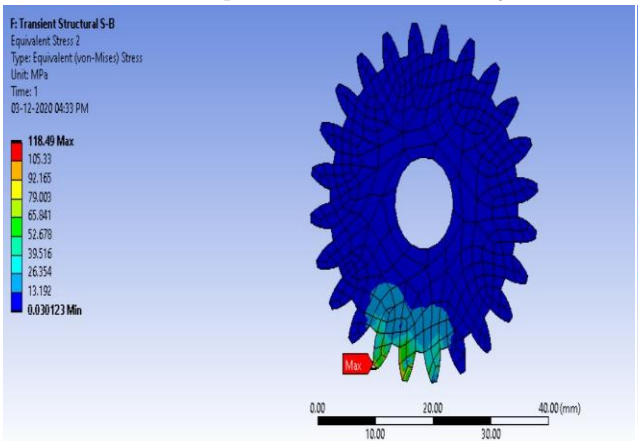


Fig. 4.2 Contact stress on pinion (Steel-Steel combination); Fig. 4.3 Contact stress on rack (Steel-Steel combination)

The maximum contact stress on rack for Steel-Steel combination is 162.69 MPa caused by the rotation of pinion over the rack and denoted by red coloured contour on the rack along with the tag max in the Fig.4.3. It is to be observed that the contact stress developed on the steel rack is higher than that of the steel pinion.



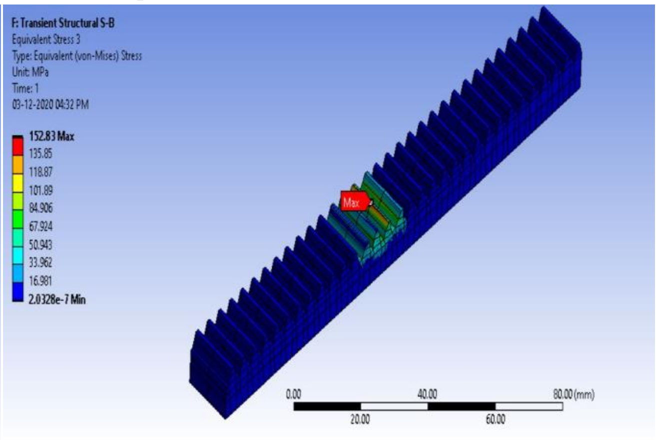


Fig. 4.4 Contact stress on pinion (Steel-Bronze combination); Fig. 4.5 Contact stress on rack (Steel-Bronze combination)

The maximum contact stress on pinion for Steel-Bronze combination is 118.49 MPa when the pinion is subjected to a moment of 13 N-m and it is denoted by red coloured contour on the pinion along with the tag max in Fig.4.4 which is lesser that of pinion for steel-steel combination. The maximum contact stress on rack for Steel-Bronze combination is 152.83 MPa caused by the rotation of pinion over the rack and it is denoted by red coloured contour on the rack along with the tag max in Fig.4.5 which is lesser than that of rack for steel-steel combination. It is to be observed that the contact stress developed on the bronze rack is higher than that of the steel pinion.

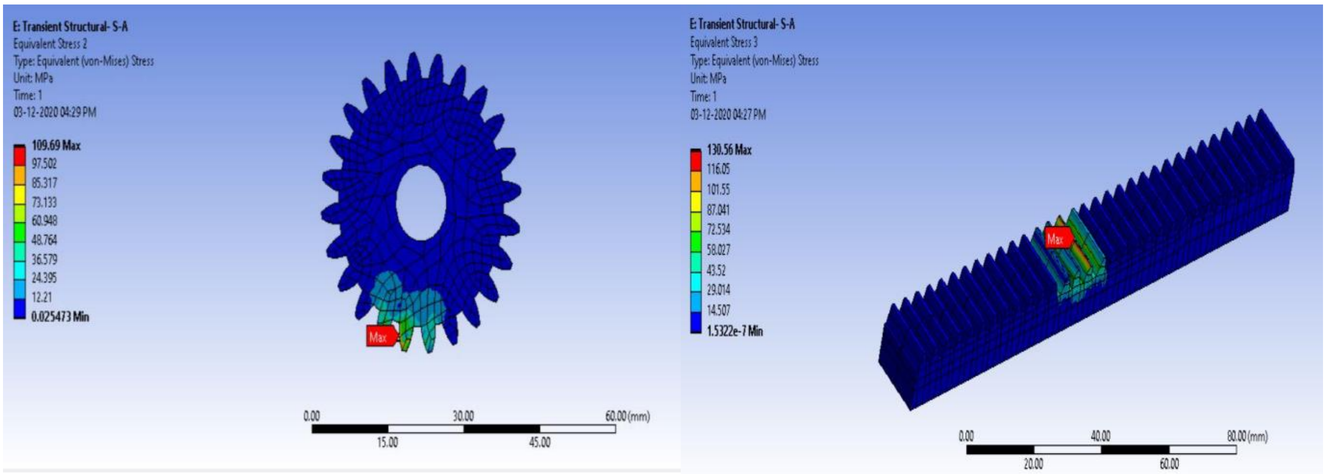


Fig. 4.6 Contact stress on pinion (Steel- Aluminium combination); Fig. 4.7 Contact stress on rack (Steel-Aluminium combination)

The maximum contact stress on pinion for Steel-Aluminium combination is 109.69 MPa when the pinion is subjected to a moment of 13 N-m and it is denoted by the red coloured contour along with the tag max in Fig.4.6 which is the lowest contact stress value for pinion among all the three models. The maximum contact stress on rack for Steel-Aluminium combination is 130.56 MPa caused by the rotation of the pinion over the rack and it is denoted by a red coloured contour along with the tag max in Fig.4.7. which is the lowest contact stress value for rack among all the three models.

5.**Conclusion**

The study of FEA results from ANSYS is validated analytically. The contact stresses determined for Steel-Aluminium and Steel-Bronze combinations are less compared to steel-steel combination by 19.45% and 6.06% respectively. The weight of Steel-Aluminium combination is found to be 64.71% lesser than that of Steel-Steel combination and the weight of Steel-Bronze combination is found to be 11.08% greater than that of Steel-Steel combination. It is also found that the stress developed on rack is more than that of pinion in all the three material combinations. For Steel –Steel combination stress on pinion is 123.68MPa and on rack is 162.69MPa, for Steel-Bronze combination stress on pinion is 118.49MPa and on rack is 152.83MPa and for Steel-Aluminium combination stress on pinion is 97.5MPa and on rack is 130.56MPa. The variation of contact stress values of rack and pinion based on Hertz equation and FEA lies between 0.3% - 10.09%. Hence the contact stress values resulted from FEA shows good agreement with values analytically calculated by Hertz equation, which indicates that the model is valid. Thus the Steel-Aluminium material combination reduces the contact stress and weight considerably.

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