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Structural Modelling And Stress Analysis Of Rack And Pinion Steering Mechanism

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Abstract:
Automobiles frequently utilize rack and pinion steering because of its ease of use, portability, and effectiveness in converting rotational motion into linear motion for accurate steering control. In order to assess a rack and pinion steering system's performance under various loading scenarios, this study focuses on structural modeling and stress analysis. Finite element analysis (FEA) is used to ascertain the stress distribution, deformation, and factor of safety once a 3D model of the mechanism is created using CAD software. To guarantee the system's dependability and longevity, the study takes into account boundary circumstances, material attributes, and realistic loading situations. The findings allow for optimization of design for improved performance and longevity by shedding light on potential failure sites and important stress areas. This research contributes to the advancement of steering system design by ensuring structural integrity and improving vehicle safety.
Index Term-*Rack and Pinion, Steering Mechanism, Structural Modeling, Stress Analysis Rack and Pinion.*

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I. INTRODUCTION

The most conventional steering arrangement is to turn the front wheels using a hand–operated steering wheel which is positioned in front of the driver, via the steering column, which may contain universal joints, to allow it to deviate somewhat from a straight line. Other arrangements are sometimes found on different types of vehicles, for example, a tiller or rear–wheel steering.

Many modern cars use rack and pinion steering mechanisms. The rack and pinion design has the advantages of a large degree of feedback and direct steering "feel". A rack and pinion is a pair of gears which convert rotational motion into linear motion. The circular pinion engages teeth on a flat bar – the rack. Rotational motion applied to the pinion will cause the rack to move to the side, up to the limit of its travel.Here the basic thing taken into consideration is the stresses acting on the tooth when the load is applied, which in turn is done by rotating the steering wheel. The comparison, of stresses induced on the tooth profiles of different types of gears (spur & helical gear), gives the better type of gear to be taken into consideration

II. Fundamentals Of Gears

A gear is a rotating machine part having cut teeth, which mesh with another toothed part in order to transmit torque. Two or more gears working in tandem are called a transmission and can produce a mechanical advantage through a gear ratio and thus may be considered a simple machine. Geared devices can change the speed, magnitude, and direction of a power source. The most common situation is for a gear to mesh with another gear; however a gear can also mesh a non-rotating toothed part, called a rack, thereby producing translation instead of rotation.

TYPES OF GEARS



Fig 1: Spur Gear Fig 2: Helical Gear Fig 3: Double Helical Gear

Fig 4: Bevel Gear **Fig 5: Worm Gear**

GEAR NOMENCLATURE

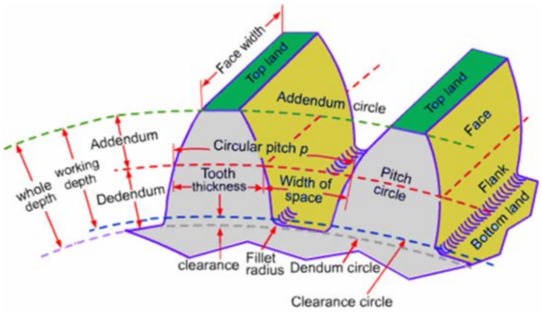


Fig 6: Worm Gear

III. INTRODUCTION TO RACK AND PINION:

A **rack and pinion** is a pair of gears which convert rotational motion into linear motion. The circular pinion engages teeth on a flat bar – the rack. Rotational motion applied to the pinion will cause the rack to move to the side, up to the limit of its travel

The rack and pinion arrangement is commonly found in the steering mechanism of cars or other wheeled, steered vehicles. This arrangement provides a lesser mechanical advantage than other mechanisms such as recirculating ball, but much less backlash and greater feedback, or steering "feel". The use of a variable rack (still using a normal pinion) was invented by Arthur E. Bishop, so as to improve vehicle response and steering "feel" especially at high speeds, and that has been fitted to many new vehicles, after he created a specialized version of a net-shape warm press forging process to manufacture the racks to their final form, thus eliminating any subsequent need to machine the gear teeth.

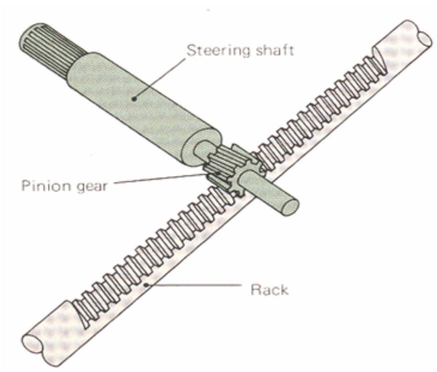
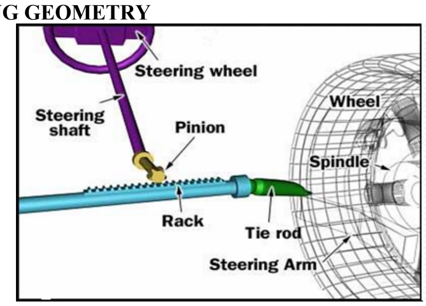


Fig 7: Rack and Pinion

PARTS OF STEERING GEOMETRY



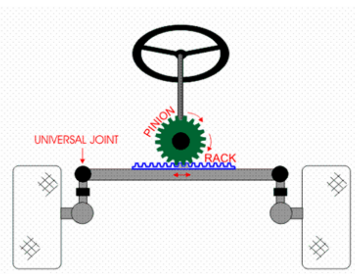


Fig 8: Rack and Pinion in steering **Fig 9: Rack and Pinion arrangement**

IV. Design Considerations

INTERFERENCE:

Most of the gears are manufactured by involute profile with 20° pressure angle. When two gears are in mesh at one instant there is a chance to mate involute portion with non-involute portion of mating gear. This phenomenon is known as INTERFERENCE and occurs when the number of teeth on the smaller of the two meshing gears is smaller than a required minimum. To avoid interference we can have undercutting, but this is not a suitable solution as undercutting leads to weakening of tooth at its base. In this situation Corrected gears are used

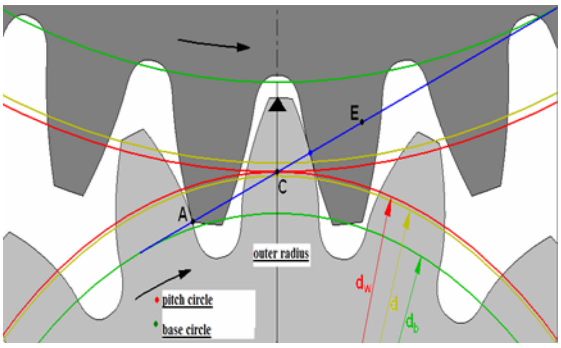


Fig 10: Interference in Gears

BACKLASH:

It is the difference between the tooth space and the tooth thickness as measured on the pitch circle. It exists because there is always some gap between the trailing face of the driving tooth and the leading face of the tooth behind it on the driven gear, and that gap must be closed before force can be transferred in the new direction. The term "backlash" can also be used to refer to the size of the gap, not just the phenomenon it causes; thus, one could speak of a pair of gears as having, for example, "0.1 mm of backlash." A pair of gears could be designed to have zero backlash, but this would presuppose perfection in manufacturing, uniform thermal expansion characteristics throughout the system, and no lubricant. Therefore, gear pairs are designed to have some backlash. It is usually provided by reducing the tooth thickness of each gear by half the desired gap distance. In the case of a large gear and a small pinion, however, the backlash is usually taken entirely off the gear and the pinion is given full sized teeth. Backlash can also be provided by moving the gears farther apart.

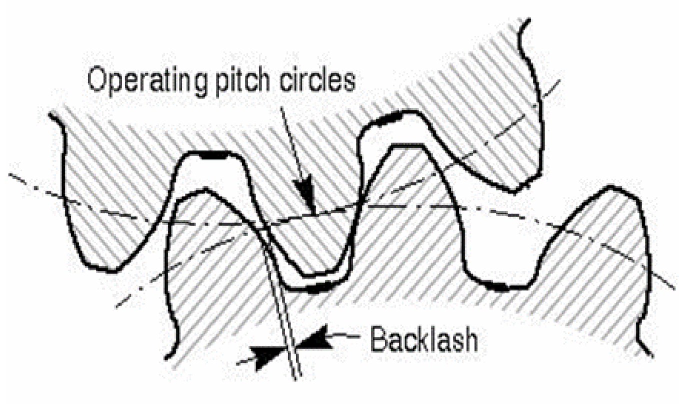


Fig 11: Backlash in Gears

INVOLUTE PROFILE OF TOOTH:

An involute of a circle is a plane curve generated by a point of tangent, which rolls on the circle without slipping or by a point on a string which is unwrapped from a reel. The involute gear profile is the most commonly used system for gearing today. In an involute gear, the profiles of the teeth are involutes of a circle.

In involute gear design contact between a pair of gear teeth occurs at a single instantaneous point. Rotation of the gears causes the location of this contact point to slide over the respective tooth surfaces. The path traced by this contact point is known as the Line of Action (also called Pressure Line or Line of Contact). A property of the involute tooth form is that if the gears are meshed properly, the line of action is straight and passes through the Pitch Point of the gears. When this is true, the gears obey the Fundamental Law of Gearing.

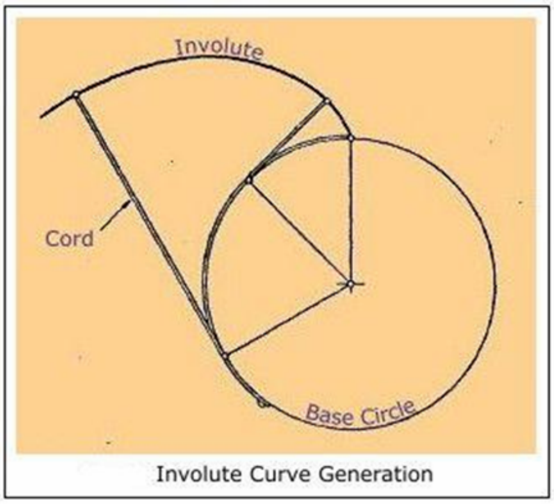


Fig 12: Involutes Curve Generation

Modeling Procedure

Computing the following data:-

- Module
- Addendum
- Dedendum
- Outer diameter
- Pitch diameter
- Base circle diameter
- Root diameter
- Circumference of base circle(CB)
- 1/20th of base circle radius(FCB)
- NCB= FCB/CB
- ACB= 360˚/NCB
- Gear tooth spacing

Helical Gear 3d Model

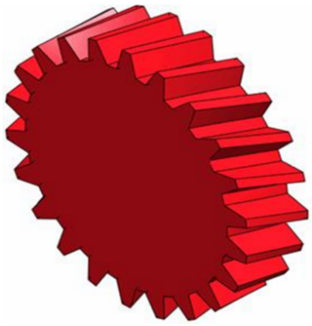
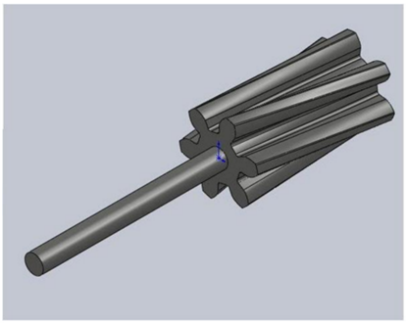



Fig 13:CAD Model of Helical Gears **Fig 14: CAD Model of Helical Gear Pinion**

Spur Gear 3d Model

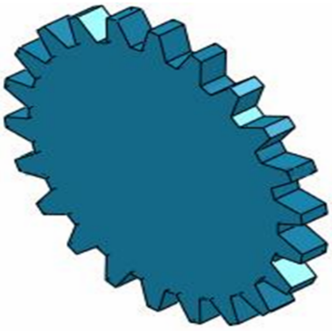
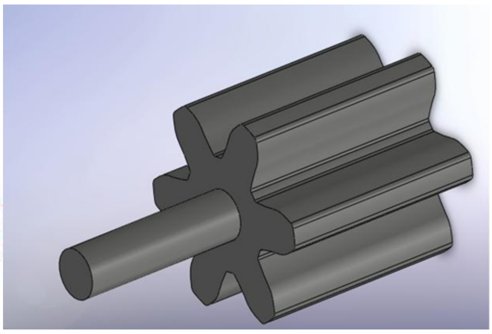



Fig 15: CAD Model of Spur Gears **Fig 16: CAD Model of Spur Gear Pinion**

The racks for the respective pinion gears are created and are shown in figures.

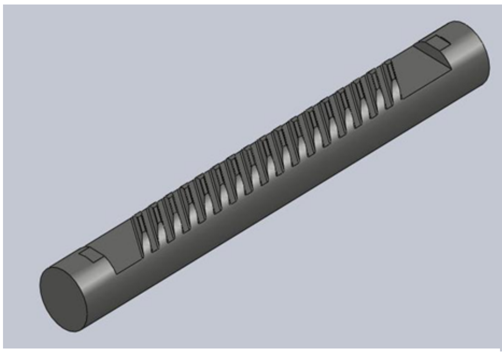


Fig 15: CAD Model of Rack For Helical Pinion

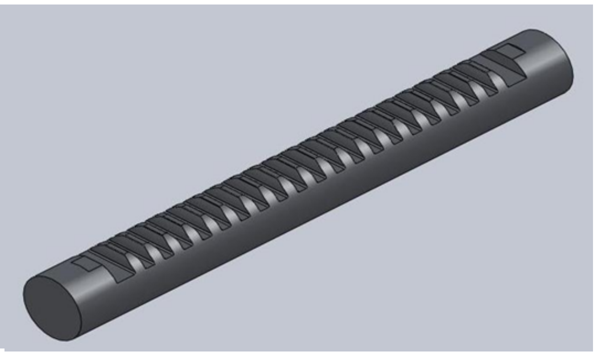


Fig 15: CAD Model of Rack For Spur Pinion

Modeling Calculations:
Measured Values From The Pinion of Maruti 800

- MODULE (m) = 2.25mm
- ADDENDUM (a) = 0.8*m
- DEDUNDUM (d.e) = 1*m
- OUTER DIAMETER (measured) = 17.1 mm
- P$ITCH DIAMETER ($p_d$) = O.D - 2*a

= 13.5mm

- BASE CIRCLE DIAMETER = D*cos(P.A)

= 17.1*cos (20)
= 12.685 mm

- ROOT DIAMETER = D-2*d_e

= 13.5-2*2.25
= 9 mm

- FACE WIDTH (b) = (1.5*3.14*m)/tan(α)

= 30.17 mm (helical)
= 18 mm

Load Calculations

- LOAD = (W*μ)/4

= (650*0.3)/4
= 48.74*9.81 (kg-m/s^2)
= 478 N
Where W = weight of car
μ = coefficient of friction

- TANGENTIAL LOAD ACTING ON THE TOOTH OF GEAR

(W_T) = (σ_W*b*t^2) / 6 * h = 3117N (helical)
= 1860N (spur)

V. Results And Discussion

The results are obtained using the simulation software ANSYS.
PINION ANALYSIS: Structural Analysis Of Pinion,
NODAL SOLUTION:
DISPLACEMENT

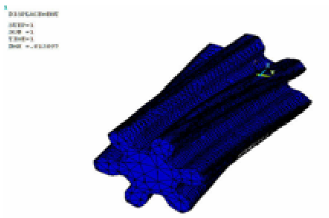


Fig : Nodal Solution For Helical Gears

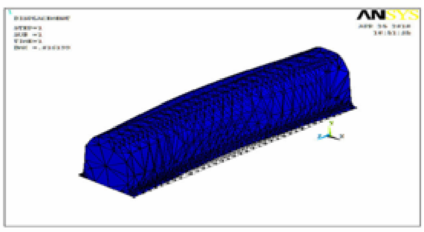


Fig : Nodal Solution For Spur Gear

Deflection Plots For Helical And Spur Gear

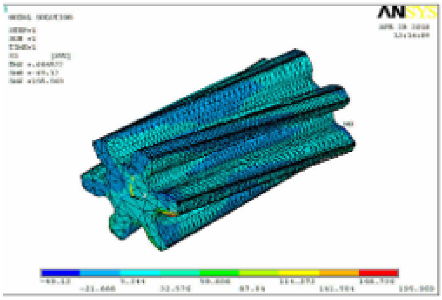


Fig : Displacement Results For Helical Gears

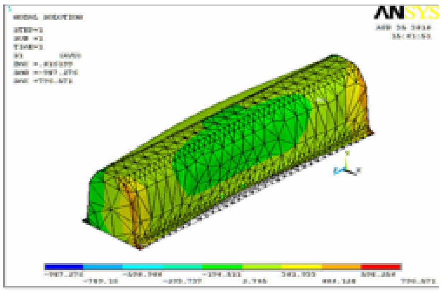


Fig : Displacement Results For Spur Gear

Stress Plots For Helical And Spur Gear

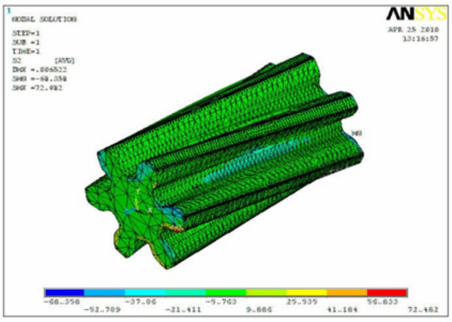


Fig : Stress Plot For Helical Gears

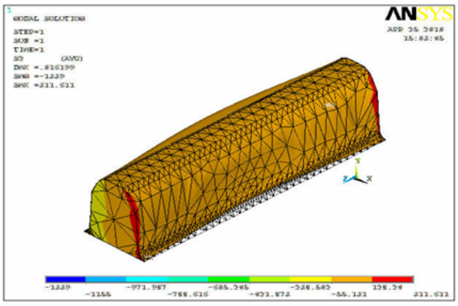


Fig : Stress Plots For Spur Gear

Principal Stress Plots For Helical And Spur Gear :

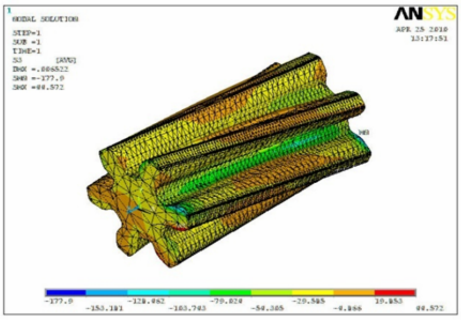


Fig : Principal Stress Plot For Helical Gears

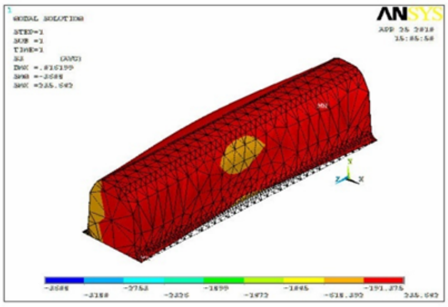
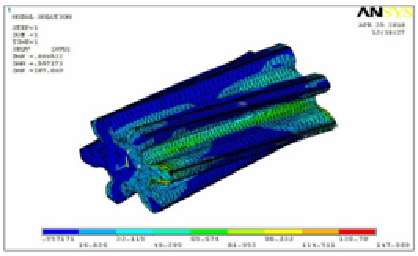
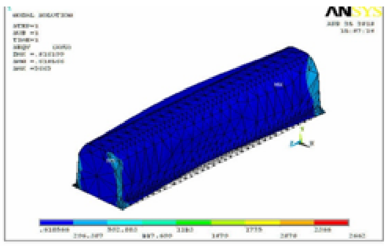


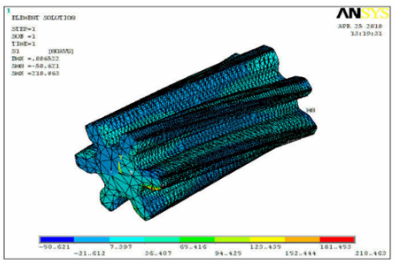
Fig : Principal Stress Plots For Spur Gear

Vonmisesstresses

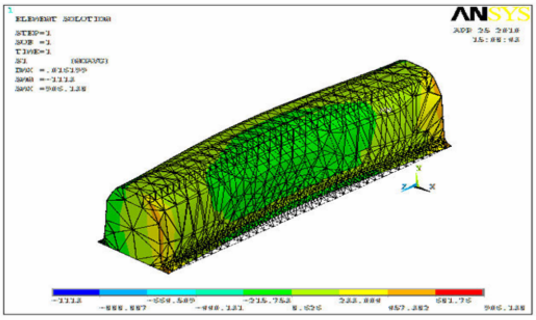




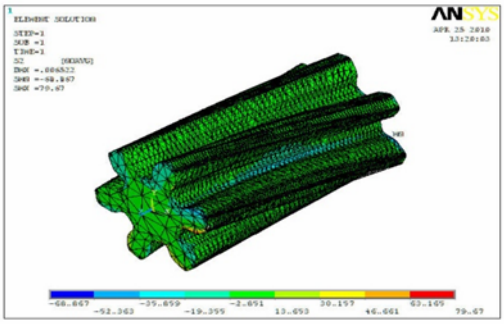
Element Solution:

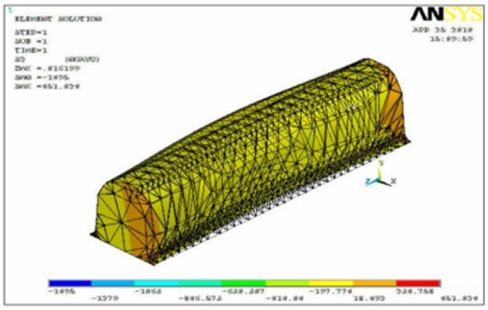


1st Principal Stress

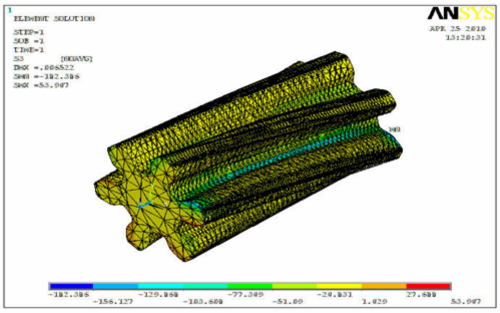


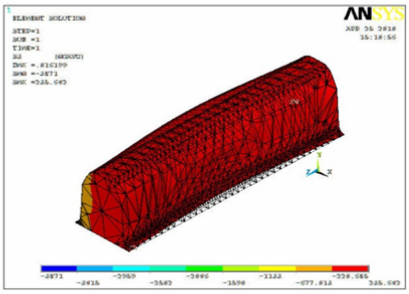
Second Principal Stress



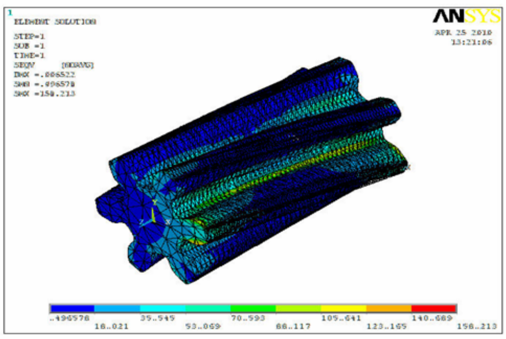


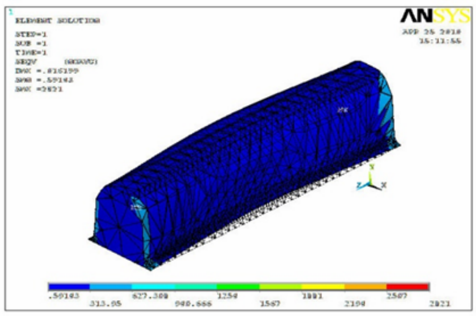
Third Principal Stress



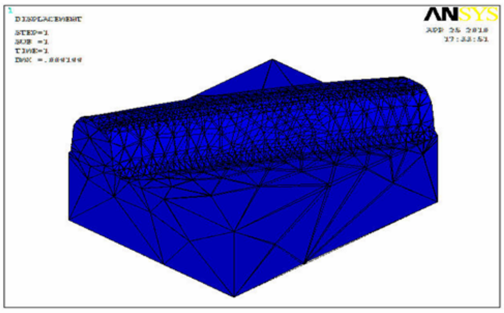
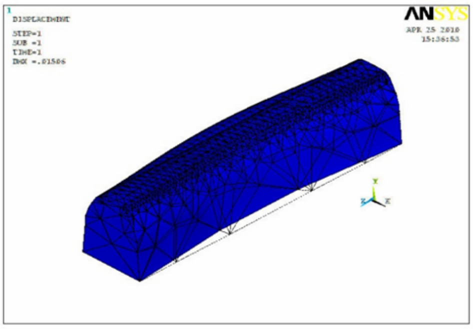


Von Mises Stresses

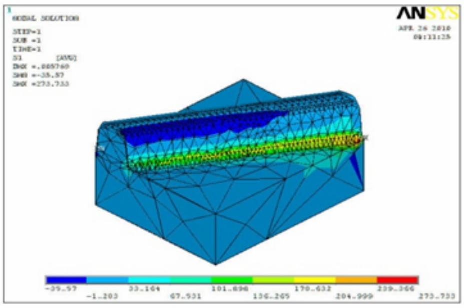




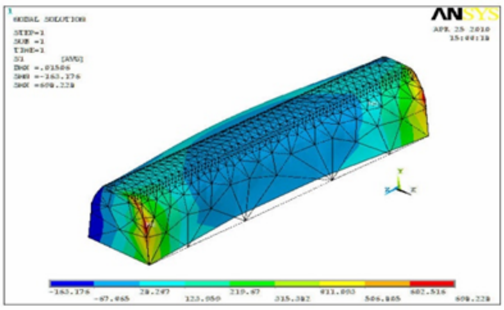
Rack Analysis:
Nodal Solution
Displacement

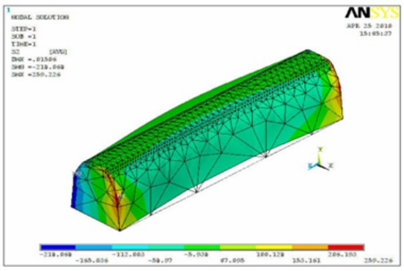
First Principal Stress



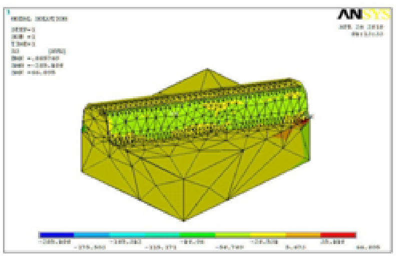
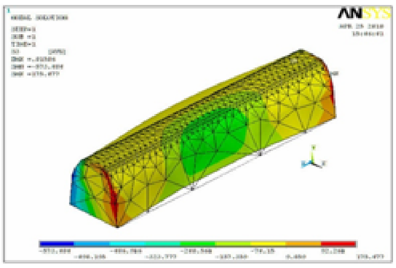
Spur



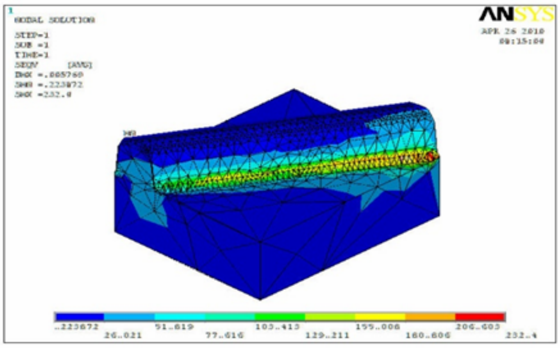
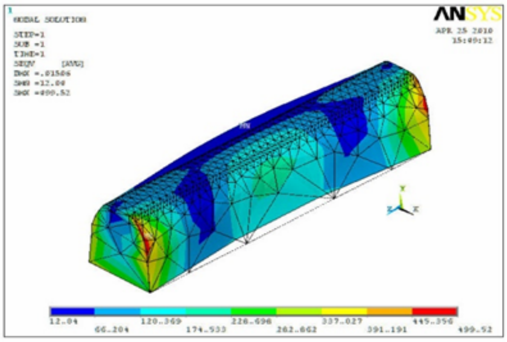
Second Principal Stress



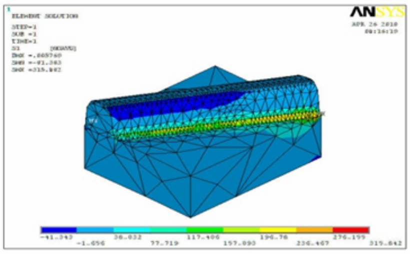
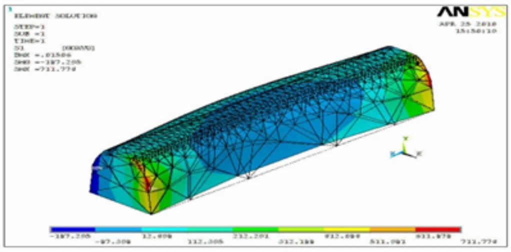
Third Principal Stress

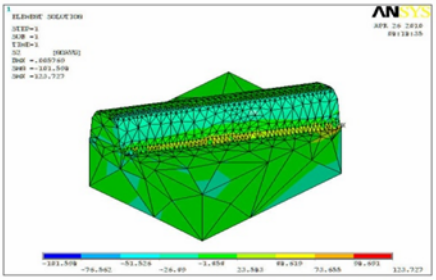
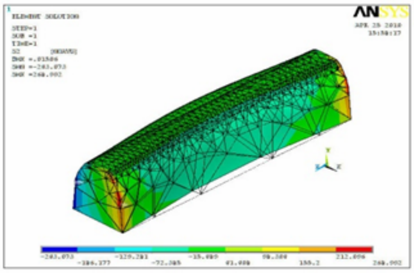
Von Mises Stresses

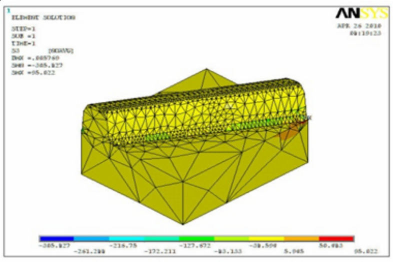
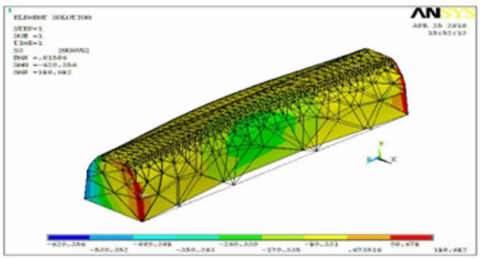
Element Solution
First Principal Stress

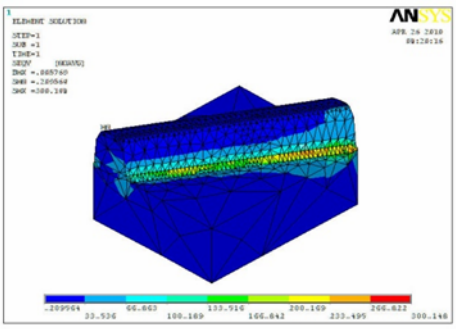
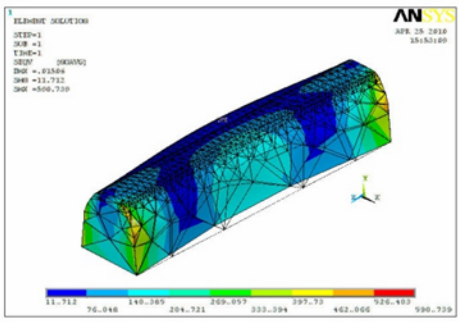
Second Principal Stress

Third Principal Stress

Vonmises Stress

Discussion:

Comparision Between Stresses Induced:

	HELICAL GEAR	**SPUR GEAR**
DISPLACEMENT	MAX DISPLACEMENT = 0.00652mm	MAX DISPLACEMENT =0.016199mm
NODAL SOLUTION	******************	***************
1.1ST PRINCIPLE STRESS	MAX STRESS = 195.969 N/mm² MIN STRESS = -49.12 N/mm²	MAX STRESS = 796.571 N/mm² MIN STRESS = -987.376 N/mm²
2.2ND PRINCIPLE STRESS	MAX STRESS = 72.483 N/mm² MIN STRESS = -98.258 N/mm²	MAX STRESS = 79.67 N/mm² MIN STRESS = -68.867 N/mm²
3.3RD PRINCIPLE STRESS	MAX STRESS = 44.572 N/mm² MIN STRESS = -177.9 N/mm²	MAX STRESS = 235.642 N/mm² MIN STRESS = -3608 N/mm²
4.VON MISES	MAX STRESS = 147.069 N/mm² MIN STRESS = 0.5571741 N/mm²	MAX STRESS = 2662.2 N/mm² MIN STRESS = 0.610566 N/mm²
ELEMENTAL SOLUTION	******************	***************
1.1ST PRINCIPLE STRESS	MAX STRESS = 210.463 N/mm² MIN STRESS = -50.621 N/mm²	MAX STRESS = 906.138 N/mm² MIN STRESS = -1113 N/mm²
2.2ND PRINCIPLE STRESS	MAX STRESS = 79.67 N/mm² MIN STRESS = -68.867 N/mm²	MAX STRESS = 451.0245 N/mm² MIN STRESS = -1495 N/mm²
3.3RD PRINCIPLE STRESS	MAX STRESS = 53.947 N/mm² MIN STRESS = -182.389 N/mm²	MAX STRESS = 235.642 N/mm² MIN STRESS = -3871 N/mm²
4.VON MISES	MAX STRESS = 158.213 N/mm² MIN STRESS = 0.49657 N/mm²	MAX STRESS = 2821 N/mm² MIN STRESS = 0.59143 N/mm²

Comparision Between Stresses Induced:

	RACK FOR HELICAL PINION	**RACK FOR SPUR PINION**
DISPLACEMENT	MAX DISPLACEMENT 0.005769mm	MAX DISPLACEMENT =0.01506mm
NODAL SOLUTION	******************	***************
1.1ST PRINCIPLE STRESS	MAX STRESS = 273.733 N/mm² MIN STRESS = -35.57 N/mm²	MAX STRESS = 698.228 N/mm² MIN STRESS = -163.17 N/mm²
2.2ND PRINCIPLE STRESS	MAX STRESS = 97.948 N/mm² MIN STRESS = -66.319 N/mm²	MAX STRESS = 259.226 N/mm² MIN STRESS = -218.068 N/mm²

3.3RD PRINCIPLE STRESS	MAX STRESS = 66.095 N/mm² MIN STRESS = -205.804 N/mm²	MAX STRESS = 175.447 N/mm² MIN STRESS = -573.404 N/mm²
4.VON MISES	MAX STRESS = 232.4 N/mm² MIN STRESS = 0.2238 N/mm²	MAX STRESS = 499.52 N/mm² MIN STRESS = 12.04 N/mm²
ELEMENTAL SOLUTION	******************	****************
1.1ST PRINCIPLE STRESS	MAX STRESS = 315.842 N/mm² MIN STRESS = -41.343 N/mm²	MAX STRESS = 711.774 N/mm² MIN STRESS = -187.295 N/mm²
2.2ND PRINCIPLE STRESS	MAX STRESS = 123.727 N/mm² MIN STRESS = -101.598 N/mm²	MAX STRESS = 268.992 N/mm² MIN STRESS = -243.073 N/mm²
3.3RD PRINCIPLE STRESS	MAX STRESS = 95.022 N/mm² MIN STRESS = -305.82 N/mm²	MAX STRESS = 180.682 N/mm² MIN STRESS = -629.682 N/mm²
4.VON MISES	MAX STRESS = 300.148 N/mm² MIN STRESS = 02095 N/mm²	MAX STRESS = 590.739 N/mm² MIN STRESS = 11.712 N/mm²

VI. CONCLUSION

- Finally it can be concluded that the results what we got show that helical gear has a displacement value of 0.00652mm, and maximum stress induced is 195.969N/mm² and minimum stress induced is -49.12N/mm² which is less compared to displacement and stresses induced in spur gear.
- The actual applied load on the gear is less than the calculated load.
- And the helical gear what we modeled and analyzed is safe for the practical application.
- Helical gear is better in withstanding loads and stresses when compared with spur gear.

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