ANALYSIS OF RACK AND PINION OF DAVIS STEERING MECHANISM

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Abstract

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.

In this paper 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.Modelling is done using Solid works software and Analysis using Ansys.

Key Words : *Rack and Pinion, Steering Mechanism*

1. 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.

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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.

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.

Fig 1 : Spur gears

Spur gears or straight-cut gears are the simplest type of gear. They consist of a cylinder or disk, with the teeth projecting radially, and each tooth is straight and aligned parallel to the axis of rotation. These gears can be meshed together correctly only if they are fitted to parallel axles

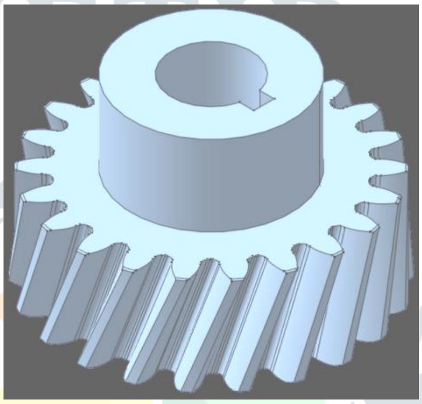


Fig 2: Helical gears

Helical gears offer a refinement over spur gears. The leading edges of the teeth are not parallel to the axis of rotation, but are set at an angle. Since the gear is curved, this angling causes the tooth shape to be a segment of a helix. Helical gears can be meshed in a parallel or crossed orientations. The former refers to when the shafts are parallel to each other; this is the most common orientation. In the latter, the shafts are non-parallel.

2.Gear Nomenclature

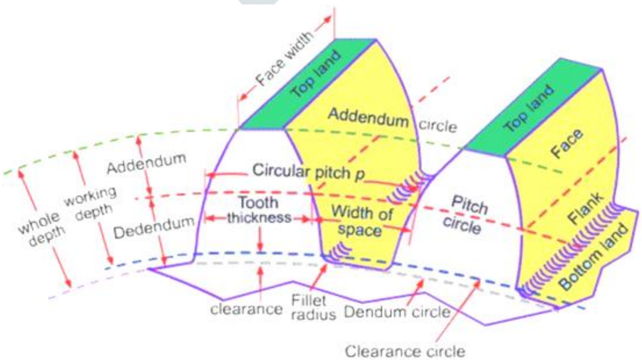


Fig3. :Gear Nomenclature

3. DESIGN CONSIDERATIONS:

3.1 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.

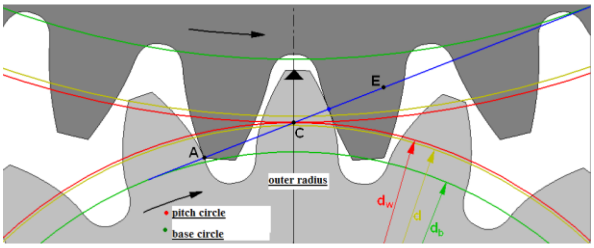


Fig 4 : Interference in Gear

4.MODELING PROCEDURE:

4.1 Involutes curve

- 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

After computation of the above values, the modeling procedure of the involute is carried on as follows in Solidworks (CAD Software).

- Drawing concentric circles of the Pitch Diameter (D), Base Circle diameter (DB), Outside Diameter (DO), and Root Diameter (DR).

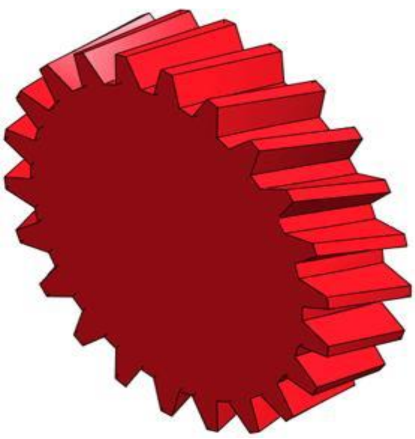


Fig 5: Helical Gear

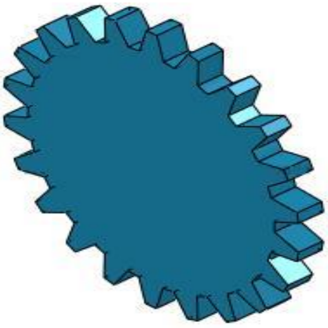


Fig 6 : Spur Gear

Model created in Solidworks using the module of maruti 800 pinion

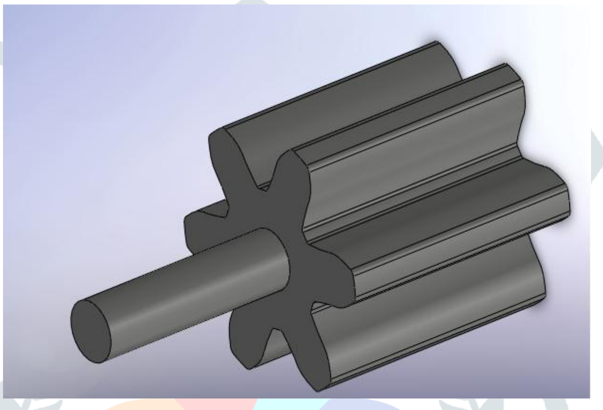


Fig 7: Model created in Solidworks

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

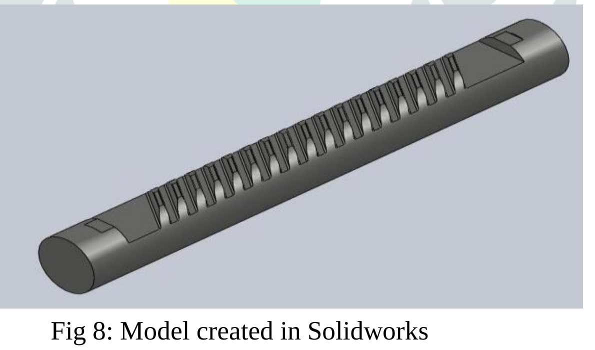


Fig 8: Model created in Solidworks

RACK FOR SPUR PINION

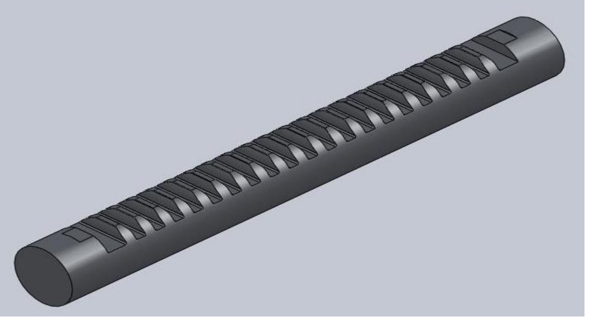


Fig 9: Model created in Solidworks

5.Analysis

5.1 Pinion Analysis:

5.1.1Nodal Solution:Displacement

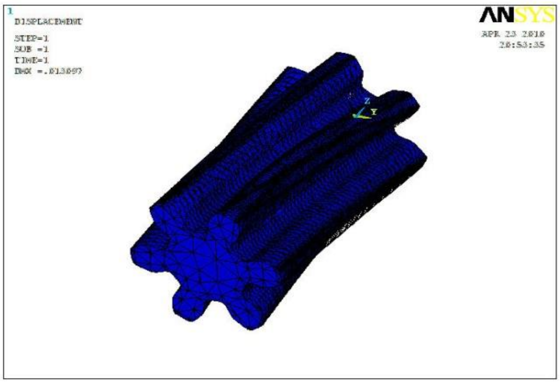


Fig :Nodal Solution of helical gear

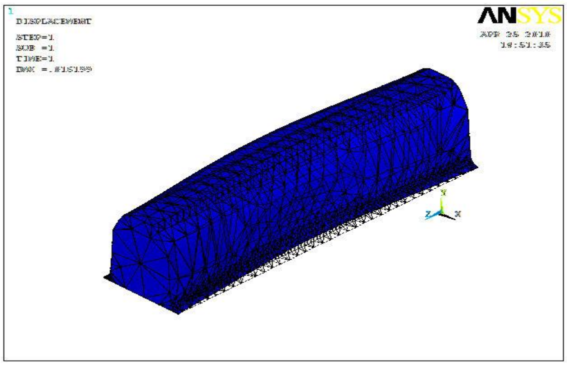


Fig :Nodal Solution of spur gear

FIRST PRINCIPAL STRESS

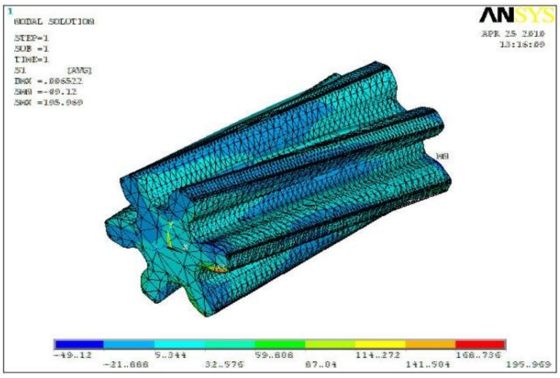


Fig : Deformation Plot of helical gear

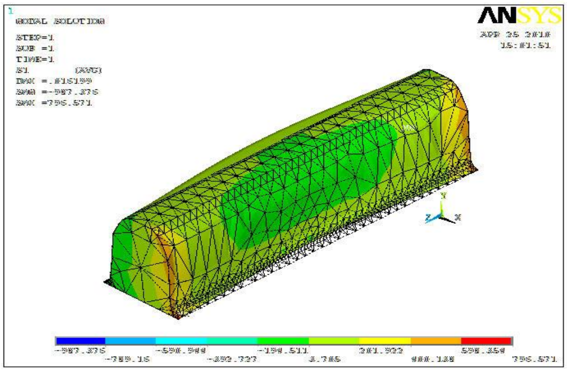


Fig : Deformation Plot of spur gear

VON MISES STRESSES

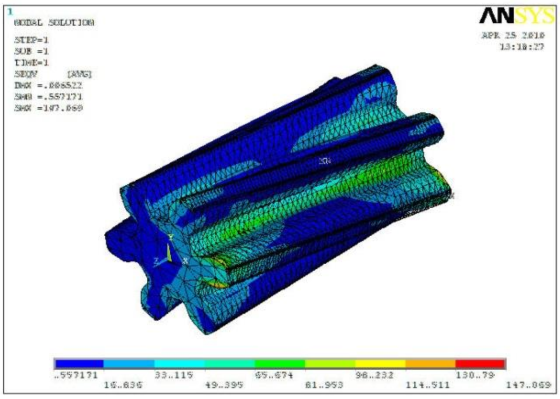


Fig Stress Plot of Helical Gear

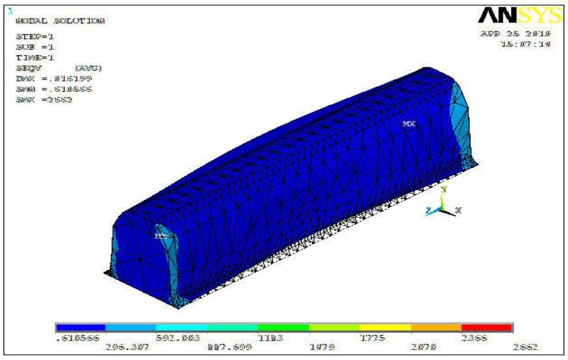
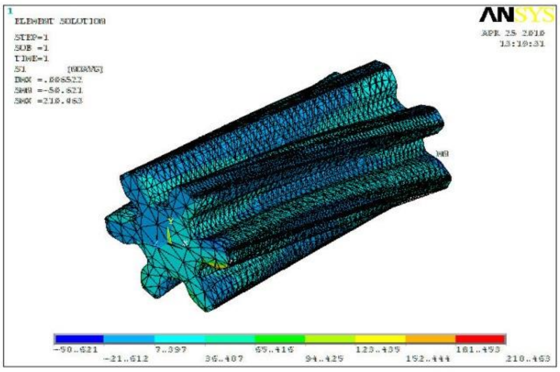


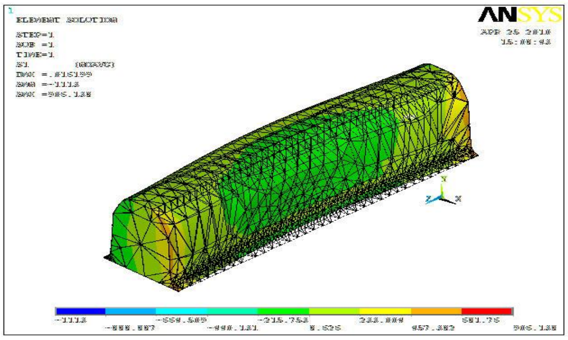
Fig Stress Plot of Spur Gear

ELEMENT SOLUTION:

1ST PRINCIPAL STRESS



HELICAL



SPUR

6.CALCULATIONS:

6.1 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

6.2 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

$(W_T) = (\sigma_W*b*t^2) / 6 * h$ = 3117N (helical)

= 1860N (spur)

6.3 COMPARISON BETWEEN STRESSES INDUCED

	HELICAL GEAR	**SPUR GEAR**
DISPLACEMENT	MAX DISPLACEMENT = 0.00652mm	MAX DISPLACEMENT =0.016199mm
1ST PRINCIPLE STRESS	MAX STRESS = 195.969 N/mm^2 MIN STRESS = -49.12 N/mm^2	MAX STRESS = 796.571 N/mm^2 MIN STRESS = -987.376 N/mm^2
VON MISES	MAX STRESS = 147.069 N/mm^2 MIN STRESS = 0.5571741 N/mm^2	MAX STRESS = 2662.2 N/mm^2 MIN STRESS = 0.610566 N/mm^2
ELEMENTAL SOLUTION	******************	***************

1ST PRINCIPLE STRESS	MAX STRESS = 210.463 N/mm^2 MIN STRESS = -50.621 N/mm^2	MAX STRESS = 906.138 N/mm^2 MIN STRESS = -1113 N/mm^2
VON MISES	MAX STRESS = 158.213 N/mm^2 MIN STRESS = 0.49657 N/mm^2	MAX STRESS = 2821 N/mm^2 MIN STRESS = 0.59143 N/mm^2

7.CONCLUSION :

- The maximum stress induced in the helical gear are 195.969 N/mm^2
- The maximum stress induced in the spur gear are 796.571 N/mm^2
- The Von Mises stress value of helical gear is less than the spur gear

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