Investigation of Lateral Force Effect In Steering gear

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Abstract—Road vehicles are controlled by the driver almost entirely via the steering system. During cornering steering wheel translates the rotary motion into a linear motion. The linear motion is transmitted to the wheel carrier via the tie rod with its ball joints. These are transferred via the steering column to the steering gear. This research work is deals with the investigation of the effect of lateral force on helical rack and pinion steering gear. This lateral force is generated by the tyre during cornering. The magnitude of this lateral force is dependent upon the vertical load on tyre. The steering gear is used in manual steering system for light weight road vehicle. After the design, the steering gear is modeled using the Catia V5 software and further static stress analysis is carried out using PTC Creo Simulate 3.0 M010 software. To study the impact of lateral force the stresses and deformations obtained from static stress analysis are compared for different cases of lateral force, keeping material and geometry of steering gear same. It is seen that as the lateral force is increases, the stress and deformation in gear also increases.

Keywords—Steering gear; Static stress analysis; Lateral force; Von Mises stress; Deformation

I. INTRODUCTION

In an automobile, a steering wheel manages the directional movement of a vehicle. The components of the steering system are the steering wheel, steering column, steering gear and tie rod. Steering system requires a steering gear to manage the directional changes of the automobile during cornering.[1] In this study manual steering system for light weight vehicle is used. The steering gear being analyzed here is a helical rack and pinion steering gear. During cornering the lateral forces of the tyre are produced by a lateral deformation of the rubber enclosed in the tread between the road surface and the ply. The magnitudes of lateral force are dependent upon the slip angle and the vertical loading of tyre. The impact of this lateral force on rack and pinion steering gear is studied through the results obtain from the static stress analysis of gear for bending stress and displacement.

Although there are different types of steering gear available e.g. worm and worm wheel, worm and sector, recirculating ball, rack and pinion etc., rack-and-pinion steering gear proved to be inexpensive and due to the direct transmission of the steering forces to the tie rods they have a high degree of rigidity, which is advantageous in particular for the steering precision.[1] Also, there are some external forces due to bad road surface are acting on rack and pinion during cornering. But our research work only deals with the impact of lateral force generated by tyre on rack and pinion. The lateral forces of the tire are produced by a lateral deformation of the rubber enclosed in the tread between the road surface and the ply. Sandip Kumar et al. worked on the design and stress analysis of Rack using cae tool. Steering Rack is designed to sustain bending loads during vehicle running. The loads come from tire side and produce the bending loads on Steering Rack[3]. Babita Vishwakarma and Upendra kumar joshi performs experiment on helical gear to observe the effect of force on gear geometry in its finite element analysis of helical gear using three-dimensional cad model research paper, the stress generated in the tooth of gear using the lewis formula and agma formula and ansys. They got values of stress and deformation of gear geometry[5]. A.Y Gidado et al. presented design, modeling and analysis of helical gear according bending strength using agma and ansys. They estimates the bending stress, three dimensional solid models for different face width are generated by Pro/Engineer and the numerical solution is done by ANSYS 11.0[7].
II. STEERING UNIT FOR ANALYSIS PURPOSE

This is the car steering unit that we are using as a reference system for our steering gear analysis. In this steering gear both pinion and the rack teeth are helical gears. The rack and pinion system produces a ratio between the degrees of steering wheel movement and the degrees of wheel movement. The ratio is called steering ratio and is typically calculated by turning the steering wheel one time and checking the number of degrees that the wheel assembly moves. In this reference system the steering ratio is 18:1. Another parameter is the gear tooth profile, a 20° full depth involute profile system was selected for the our study because of the advantage of reduction in the risk of undercutting and interference. Also, due to increase in pressure angle, the tooth becomes slightly broader at the root. This makes the tooth stronger and increases its load carrying capacity. It provides better length of contact.

![Figure 1. Steering unit](image)

3.1. Force analysis

In a manual rack and pinion steering the parameters are the wheel angle and the wheel torque, which is initiated at the steering wheel and transformed by the dovetailing components, pinion and rack, into rack force and rack shift. Beyond these mechanical variables, no further energy is supplied to the manual rack and pinion steering to move the rack.[1] The rack displacement forces of mechanical gears are usually a little higher than those of rack-supported power-assisted gears. The reason is that all the rack forces of mechanical gears are directly transferred over the pinion-rack link. [1]

1) Tangential force ($F_t$): It acts in normal direction on rack tooth with magnitude approximately equal to torque applied at steering wheel. The tangential component is a useful component and it is responsible for transmitting the torque or power. 2) Radial Force ($F_r$): The radial force tends to separate the two gears. It acts along the radial line through the pitch point and is directed towards the center. 3) Axial Force ($F_a$): It acts in axial direction. The axial force acting on two meshing helical gears are equal in magnitudes and opposite in directions.
III. LATERAL FORCE

The motion of a vehicle is governed by the forces generated between the tire and the road. Lateral tire force is the force necessary to hold a vehicle through a turn. It is caused by the slip angle, which is defined as the angle between the direction of the wheel circumference and the direction of the movement of the wheel. The tread bars are increasingly deformed from the entry point of the contact patch to its exit point by the constant relative lateral movement between the road surface and the ply, as long as the traction between tread bars and road surface remains sufficient. On a dry surface the traction is largely upheld up to an acceleration of about 3–4 m/s².[1]

At higher acceleration, the slip angle increases and the deformation increases accordingly, the tread bars are torn off in the rear area of the contact patch at first and start to slip on the road surface. While the lateral force continues to increase in line with the slip angle in the linear operating range of the tire, it decreases digressively at an additional increase of the slip angle until a more or less pronounced maximum of force is built up. If the slip angle is very large, the tire starts to slide very close to the entry point of the contact patch. In that case any additional increase of the slip angle does not lead to an increase of the lateral force.
The total lateral force generated by the tyre acts on the various parts of the vehicle, e.g. front axle, tie rod, suspension system, etc. The major part of the tyre lateral force is taken by the front axle up to 80% to 90%. [2] In mechanical steering system lateral force acts on the rack which is transferred from tyre to tie rod and tie rod to rack. Here it is observed that, the line of action of the lateral force on the rack is such that it assists the linear movement of rack when steering wheel is rotate clockwise for changing the direction of the vehicle. E.g. According to general design of rack and pinion steering system if the vehicle is taking the right turn, then rack needs to be move linearly towards left direction. At the same time lateral force from the tyre is also acts in the left direction. i.e. it forces the rack to move in left direction. The magnitude of lateral force of tyre is depend upon tyre properties e.g. material, load, size, application etc. It is available in the manufacture’s catalogue. In our case we are using 55 R 13 78 S ‘82 series’ steel radial tyre, measured on a dry drum at pT = 1.8 bar. As shown in fig.4.4. It’s loading capacity is around 360 kg. In our case we assumed the mass of vehicle 800 kg, therefore the total mass on each of the front tyre is 200 kg ( i.e. 1962 N). Therefore the lateral force generated by the each tyre is 2000 N. Which means that during cornering total Lateral force generated by front tires is 4000 N. As we said above 80-90% of this lateral force is taken by front axle, therefore the remaining lateral force is taken by tie rod and other part of vehicle. We assumed that 20% of this lateral force is transfer by tie rod to the rack i.e. 800N. So the total lateral force (F_l) acting on the rack = 800 N.[2]
IV. DESIGN CALCULATIONS

It is necessary to design rack and pinion system in such a way that, it must provide required aligning torque while cornering without failure. The system was designed against bending as well as pitting failure. The detailed procedure of designing is as follow:

4.1. GEAR MATERIALS

The materials used in the steering system target precise operation and light weight components. Although precision and weight are the top priorities, cost, manufacturability, and reliability were also considered. Here we are using Alloy steel 40Ni3Cr65Mo55 (Hardened and Tempered) for pinion having Minimum Ultimate Tensile strength 1000 N/mm² and BHN 285-340. Similarly, Alloy steel40Ni2Cr1Mo28 (Hardened and tempered) is used for rack gear having Minimum Ultimate Tensile strength 800 N/mm² and BHN 230-275 [5]. The factor of safety \( N_f \) is taken as 2 as the both the material are hardened and tempered. For single helical gears, the helix angle normally ranges from 15° to 30°. The values of helix angle are not standardized. For our study, we assumed the helix angle to be 25°. In the design of a gear, it is necessary to express the face width in terms of module. In practice, the face width is taken such that, \( 9m_n < b < 15m_n \). In our case we assume the face width \( (b) \) as 12mₙ.

4.2. BEAM STRENGTH (\( F_B \))

Beam strength of gear tooth is the maximum tangential load that gear tooth can take without tooth damage. The min. number of teeth on pinion \( (z_p) \) is taken as 18.

The bending Endurance strength of pinion = \( \sigma_{bp} = \frac{5t_f}{3} = \frac{1000}{3} = 333.34 \text{ N/mm}^2 \).

The bending Endurance strength of rack = \( \sigma_{br} = \frac{5t_f}{3} = \frac{800}{3} = 266.67 \text{ N/mm}^2 \)

4.3. WEAKER OF PINION AND GEAR

The product \( (\sigma_b \times Y') \) decides the weaker member between the gear and pinion. In order to calculate lewis form factor \( (Y') \) we need to find out virtual number of teeth \( (z_p') \).

Lewis form factor for pinion,

\[ Y'_p = 0.484 - \frac{2.87}{z_p'} = 0.484 - \frac{2.87}{24.17939} = 0.365303879. \]

As rack has the infinite pitch circle diameter \( (d_p) \), therefore the number of teeth \( (z_r') \) on rack are unable to calculate. Hence,

\[ Y'_r = 0.484 - \frac{2.87}{\infty} = 0.484 \]

As we can see \( (\sigma_{bp} \times Y'_p) < (\sigma_{br} \times Y'_r) \), pinion is weaker than rack in bending. Hence it is necessary to design pinion for bending.

\[ F_{bp} = \sigma_{bp} \times \frac{m_n \times Y'_p}{2} = 333.34 \times 12 \times m_n \times \frac{0.3653}{2} \]

\[ F_{bp} = 1461.2155m_n^2 \]

4.4. EFFECTIVE FORCE (\( F_{eff} \))

In this design the torque (\( T \)) applied at pinion is assumed to be max. 6 Nm. Also, an average man can turn a simple steering wheel with an average force of 300N with an average 15 rpm (N) [1].

Therefore the power (\( P \)) to be transmitted is,

\[ P = \frac{2\pi NT}{60} = \frac{2\pi \times 15 \times 6}{60} = 9.4247 \]

2) Pitch line velocity\( (v) = \frac{\pi d_p \times \sigma}{60 \times 1000} = 0.015598637m_n \)
Now, the effective load on the gear tooth is the total maximum tangential force acting on the gear tooth. The theoretical tangential force acting on the gear tooth due to the power transmitted is given by,

\[ F_t = \frac{P}{V} = \frac{9.4247}{0.015598637m_n} = \frac{604.2001}{m_n} \]

In addition to this force \( F_t \), the lateral force \( F_l \) will also act in the same direction. The magnitude of lateral force is 800N as we explain in before. Therefore the total theoretical tangential force,

\[ F_{t,\text{total}} = F_t + F_l = \frac{604.2001}{m_n} + 800 \]

3) Effective force:

\[ F_{\text{eff}} = \frac{K_u \times K_m \times F_{t,\text{total}}}{K_v} \]

Where, \( K_u = 1.15 \), \( K_m = 2.2 \), \( K_v = \frac{5.6}{5.6 + \sqrt{v}} \)

\[ F_{\text{eff}} = \frac{5376 + \frac{4060.2245}{m_n} + 119.8987\sqrt{m_n} + \frac{90.5535}{\sqrt{m_n}}}{5.6} \]

4.5. ESTIMATION OF MODULE

In order to avoid the bending failure, \( F_b = N_f \times F_{\text{eff}} \).

Solving this equation by trial and error, we get, \( m_n = 1.465 \). The standard value of module \( m_n \) is taken as 2mm.

4.6. DIMENSIONS OF PINION

\[ m_n = 2 \text{ mm} \]
\[ z_p = 18 \]
\[ b = 12 \times 2 = 24 \]
\[ d_p = \frac{z_p \times m_n}{\cos \Psi} = \frac{18 \times 2}{\cos 25^\circ} = 39.7216, \text{ mm} \]
\[ h_a = 1m_n = 2 \text{ mm} \]
\[ h_f = 1.25m_n = 2.5 \text{ mm} \]

Calculation for other parameters,

1) Pitch line velocity \( (v) = 0.015598637m_n = 0.015598637 \times 2 = 0.031197275 \text{ m/s} \)
2) The minimum face width

\[ b \geq \frac{1.15 \pi m_n}{\sin \Psi} = \frac{1.15 \pi \times 2}{\sin 25^\circ} = 17.0973 \text{ mm} \]

3) Effective force

\[ F_{\text{eff}} = \frac{5376 + \frac{4060.2245}{m_n} + 119.8987\sqrt{m_n} + \frac{90.5535}{\sqrt{m_n}}}{5.6} = 1364.2332 \text{ N} \]

4) Safety Against Bending strength

\[ F_{bp} = 1461.2155m_n^2 = 1461.2155 \times 2^2 = 5844.862 \text{ N} \]

\[ N_{Fb} = \frac{F_b}{F_{\text{eff}}} = \frac{5844.862}{1364.2332} = 4.28 > 2 \text{ ................. Eq(1)} \]

3) Safety Against Wear failure:

According to Buckingham`s equation for wear strength of the helical gear tooth.
Hence, the factor of safety available against wear failure, 
\[
F_w = \frac{39.7216 \times 12 \times 2 \times 2 \times 1.21}{\cos^2 25^\circ} = 2808.6777 \text{ N}
\]
From Equation (1) and (2), the available factor of safety is higher than the required factor of safety, therefore our design is safe.

4.7. DIMENSIONS OF RACK
For the modeling of rack and pinion the dimension of the rack is taken as same to that of pinion. In other word rack modeling is developed according to the geometry of pinion. In our design of rack and pinion steering gear, the number of turns of steering wheel from lock to lock is taken as 2.5. Therefore the total length of the rack is, 
\[
(2 \times \pi \times r_p \times 2.5) = 311.9727 \text{ mm}
\]

4.8. FORCE COMPONENTS
1) Tangential Force \( F_r \) = \( F_{eff} \) = 1364.2332 N
2) Axial Force \( F_a \) = \( F_{eff} \times \tan \Psi \) = 1364.2332 \times \tan 25^\circ = 636.1523 N
3) Radial Force \( F_r \) = \( F_{eff} \times \frac{\tan \theta_r}{\cos \Psi} \) = \( \frac{1364.2332 \times \tan 20^\circ}{\cos 25^\circ} \) = 547.8715 N

V. STATIC STRESS ANALYSIS
5.1. PARAMETRIC MODELING OF HELICAL GEAR
During the gear design, the main parameters that would describe the designed gear such as module, pitch circle diameter, face width and number of teeth. CATIA V5 uses these parameters, in combination with its features to generate the geometry of the helical pinion and helical rack and all essential information to create the model.

![3D parametric modeling of helical rack and pinion](image)

Figure 7. : 3D parametric modeling of helical rack and pinion helical rack

5.2. BOUNDARY CONDITIONS
For meshing, element type is taken as Tetra and element size is given as 10mm. Poisson’s ratio is given as 0.3. Material properties for pinion and rack are specified. For both material densities is given as 7.85 gm/cm³. For pinion tensile yield stress = 630 N/mm² and young’s modullus = 210 Gpa. For rack tensile yield stress = 504 N/mm² and young’s modullus =190 Gpa.

5.3 RESULTS & DISCUSSIONS
The helical pinion was analyzed for the applied tangential, axial and radial force. The Figures shows the stress distribution plot along the tooth. Under static Condition Maximum von mises stress is 93.53Mpa which is under the permissible bending stress of 333.34 Mpa
Similarly, The helical rack was analyzed for the applied tangential, axial and radial forces. The Figures shows the stress distribution plot along the tooth. Under static Condition Maximum von mises stress is 109.9 Mpa which is under the permissible bending stress of 266.67 Mpa.

![Figure 8](image1.png)  
**Figure 8.** Von mises stress distribution in pinion and rack

The total maximum deformation of helical pinion is 0.00255 mm and The total maximum deformation of helical rack is 0.00486mm.

![Figure 9](image2.png)  
**Figure 9.** Deformation under static load in pinion and rack

As we can see from above results the maximum von mises stresses for both pinion and rack are under permissible bending stress and the deformation of gear tooth due the stresses is also small, therefore our design for steering gear is safe for applied lateral force of 800 N.

**VI. IMPACT OF LATERAL FORCE**

In order to study the impact lateral force on both pinion and rack, we calculated the lateral forces in increasing order for respective increasing tyre vertical load from the fig no. 6. Then for each lateral force steering gear is analyzed for static stress analysis, keeping the material and geometry of steering gear same. The graph for the magnitude of lateral force for respective vertical loading on tyre is plotted below. As the maximum loading capacity for the given tyre is 360kg, therefore the maximum lateral force acting on the steering gear is 1280. As we have already done with analysis for 800N, we started our analysis from lateral force of 960 N.

**Table 1.** Forces for different lateral loads

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Lateral Force (N)</th>
<th>Tangential Force ($F_t$)</th>
<th>Axial Force ($F_a$)</th>
<th>Radial Force ($F_r$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>800</td>
<td>1364.2332</td>
<td>636.0946</td>
<td>547.821</td>
</tr>
<tr>
<td>2</td>
<td>960</td>
<td>1562.2889</td>
<td>728.5072</td>
<td>627.4100</td>
</tr>
<tr>
<td>3</td>
<td>1060</td>
<td>1686.0738</td>
<td>786.229</td>
<td>677.1217</td>
</tr>
<tr>
<td>4</td>
<td>1160</td>
<td>1809.8587</td>
<td>843.9509</td>
<td>726.8333</td>
</tr>
<tr>
<td>5</td>
<td>1280</td>
<td>1958.4005</td>
<td>913.2171</td>
<td>786.4872</td>
</tr>
</tbody>
</table>

The corresponding values of lateral force, tangential, radial and axial force for given increase in vehicle weight are tabulated in above table.
VII. RESULTS AND DISCUSSION

The above analysis is performed on rack and pinion steering gear designed for light weight vehicle of mass 800Kg for which the lateral force of 800N is acting on steering gear. By keeping the material and geometry of steering gear, the results of analysis performed in remaining four cases are plotted in following figures. Figure 10 shows that as the lateral force is increases, the Tangential, Axial and Radial forces are also increases. The slope of Tangential force is slightly steeper than other two forces for the given increased in lateral force. It shows that load on gear tooth is getting more concentrated in tangential directions.

From following both graph it is clearly proved that as the lateral force increases the max. von mises stresses and max. deformation is also increases. Also in each case the max. deformation and max. von mises stress acting on rack is higher than pinion. This is because the ultimate tensile strength of rack material is lower than that for pinion material.

VIII. CONCLUSIONS

The results obtain from above analysis shows the impact of lateral force in the design of steering gear.
1. The maximum lateral force by the tyre is 1280 N as its maximum loading capacity is 960kg. If more load applied is more than this capacity then tyre will burst.
2. As the lateral force is increased, the force components radial force, axial force and especially tangential component \( f_{eff} \) acting on gear tooth is also increased which results into increase in the value of stress acting on gear tooth.

3. Also increased lateral force causes increased deformation of gear tooth.

4. In order to sustain for high lateral force the material of high strength need to be selected for gear.

5. The max. von mises generated in case 5 on rack and pinion are under permissible bending stress because the material selected is already of high tensile strength, therefore it will not failed for the max lateral force generated by this tyre.

6. Another way to sustain for this lateral force we can increased the module of the gear, but because of space limitation this method is difficult to apply.

REFERENCES