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Removal of Backlash from Steering Systems

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Abstract

This design concept focuses on the elimination of backlash in steering systems by altering the design of a traditional rack and pinion. This can be accomplished by incorporating the features of the Roller Pinion System (RPS) technology; that has been extensively used to replace the rack and pinion mechanism in various industries. This improvised system is wear resistant and free of backlash owing to its superior constructional arrangement. A complete 3D-model of the system has been constructed using CATIA. SOLIDWORKS has then been used for simulating the motion and as well as for carrying out finite element analyses of the critical components.

1. Introduction

Most steering systems use rack and pinion mechanism to transfer steering torque to the wheels. This consists of a helical gear pinion that rotates over a rack, causing it to move in a to-and-fro linear motion.

Backlash is defined as the lost motion between mechanisms caused by gaps between mating parts. Due to the involute teeth profile, a certain amount of clearance (gap) is required between mating gears, which is the main cause for the occurrence of backlash. This hampers the functionality of the linkages; for instance, a steering ratio of 15:1 implies that on rotating the steering wheel by 30°, the car wheels will rotate by 2°. However because of backlash, the wheels will lose motion, and may only rotate for 1.2° (say).

Some systems try to avoid backlash by reducing the distance between centers of the rack and the pinion, which results in an increase in interference. This interference causes excessive wear and tear of the parts and can only delay the oncoming of backlash but not eliminate it completely.

The rack and pinion system is simple in construction but cannot provide as much mechanical advantage as the recirculating ball. However, it is advantageous over the latter due to lighter design and greater feedback. Thus the primary focus of this paper is to improve the functioning of the rack and pinion by slightly altering the current design. The approach of using a roller pinion instead of the helical gear was adopted so as to achieve complete elimination of backlash.

In this scheme, the commonly used helical gear pinion is replaced by a roller pinion; wherein the roller pinion is made of cylindrical rollers - fitted on to a hub using needle bearings - which can rotate about their individual axes. The rolling motion of the pinion adds to the design by reducing friction to a large extent. The rack is accordingly modified based on a cycloidal profile. The modified design makes the pinion fit on to the rack in such a way that at every instant two rollers position each other in opposing rolling directions, thus eliminating any unnecessary movements and hence backlash.

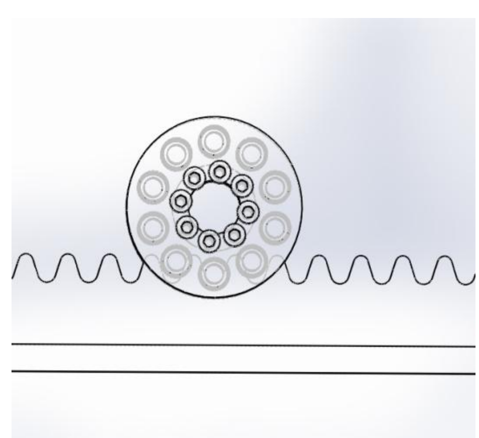

Figure1. Three rollers are always in contact, and two of these rollers are in opposition, preventing backlash.

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The Roller Pinion System (RPS) technology is an alternative to rack and pinion systems. It has completely replaced the traditional rack and pinion in various automation industries. However, its application within the automobile industry was found to be limited. The roller pinion can be incorporated into vehicular steering if the dimensions are optimized correctly.

2. Constructional Features

The roller pinion system consists of a driving gear or the roller pinion which has an unusual design and a uniquely constructed rack with perfectly meshing teeth. Following is a description of the design involved in making the rack and the roller pinion.

2.1 Roller Pinion

For the pinion, a standard model of RPS16 (Copyright of Nexen Group) was selected and constructed with a 3D geometry in CATIA. This model consists of ten equally spaced rollers fitted between the pinion body using needle bearings so as to provide free rotational movement about their individual axes. The dimensional aspects of the pinion are as follows:

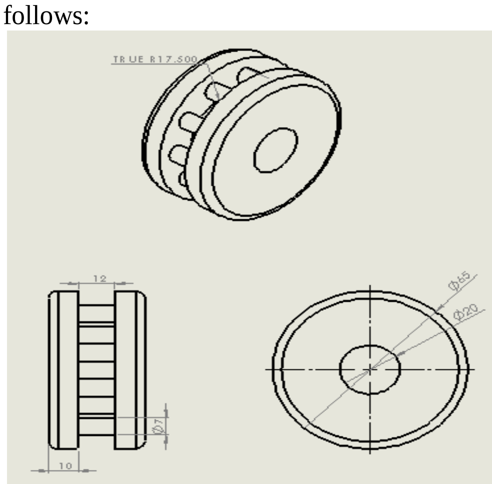

Figure 2. CAD drawing of roller pinion – FV, SV and Isometric.

Table 1.
Dimensions of Roller Pinion Parts

Element	Dimension (mm)
Diameter of Flange	67
Diameter of Pitch Circle	49
Diameter of Hub	32
Diameter of Bore	18
Diameter of Roller	7
Width of Roller	14

2.2 Modified Rack

The RPS tooth design is conceptually different from traditional gearing. It behaves like a cam and follower versus the typically sliding spur gear used in traditional rack and pinion gear sets.

A cycloidal curve is created when a point drawn on a circle rolls on a flat plane to another point without slipping. When multiple points are placed on the circle at regular intervals, the cycloidal curves are repeatedly created on the flat plane, and developed into a tooth-like profile.

A roller then is placed at each of these constructional points to act as pinion teeth and thereby modifies the tooth profile to create the rack teeth. The rollers meet the tooth with a tangent path and smoothly roll down the tooth face. This eliminates tooth slap, sliding friction, fatigue, noise, and low precision associated with traditional gearing [3]. The dimensional aspects of the rack are as follows:

Table 2.
Dimensions of the Modified Rack

Element	Dimension (mm)
Height of Rack Tooth	10.5
Width of Rack	12
Pitch of Rack	16
Length of Rack	As per requirement

3. Methodology

The first step was to select an appropriate roller pinion model based on load requirements. As mentioned under constructional features, the pinion used for the current model is RPS 16. Selection of the size of the roller pinion was done by considering factors like maximum torque involved and space available.

3.1 3D Modeling

The next step was to construct its 3D model using CATIA (V5). The 3D model has been created such that it adherently follows the actual design. The dimensions used, have already been mentioned in Table1. For practical purposes we created two models of the pinion, one is the working model that depicts only the basic features (fig. 3); the other is a part by part assembled model (fig. 4).

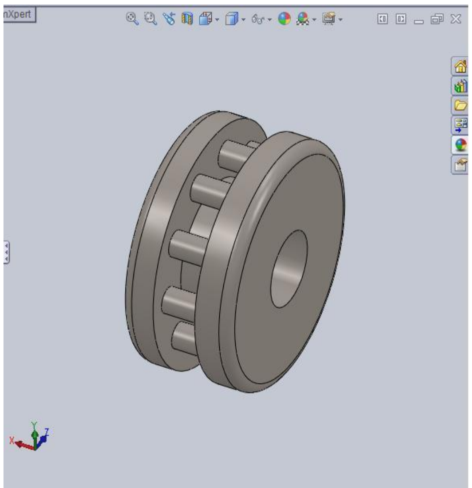

Figure 3. Working Model of Pinion

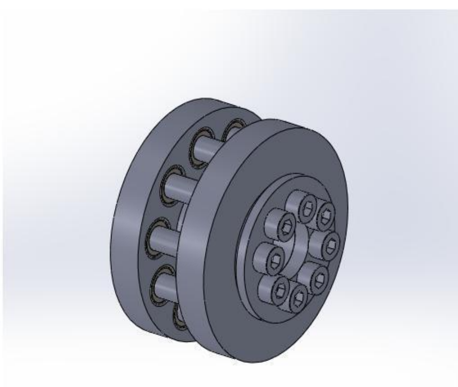

Figure 4. Fully assembled model of pinion.

Similarly, using the tooth profile generated before (as shown in the design abstract), a model of the rack was created as follows. The curve of the tooth is the same as the cycloidal profile generated previously.

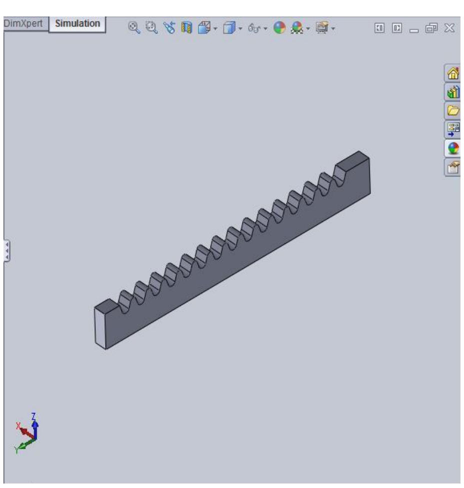

Figure 5. Rack with cycloidal tooth profile.

The above models are two individual parts of a single system. For this mechanism to be valid, the two parts should mesh perfectly with each other. Also, an additional feature will be the continuous contact of the rack with three rollers. Two of these rollers will be in opposing motion to each other, thus preventing backlash at every instant.

The above mechanism was found to have meshed perfectly and thus, the next step was to fit it at the end of a steering assembly; as shown below:

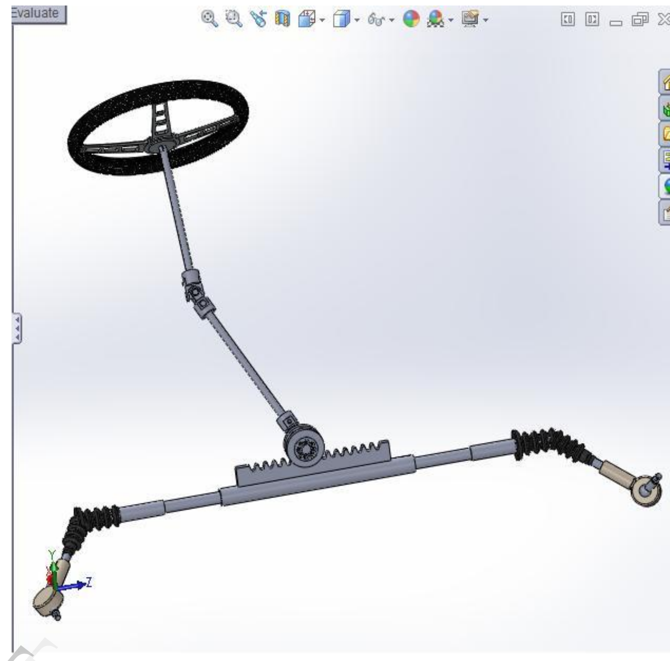

Figure 6. Entire steering assembly, with roller pinion as the driving gear instead of the typical helical gear pinion.

3.2 Loads

First and the foremost, the average torque applied at the steering wheel of the system was found out [2]. Using this torque as the starting point, the tangential load carried by the pinion was calculated.

Next, assuming the contact ratio as 2 for practical reasons (since, 3 will only be possible in the ideal case), the tangential load was divided by 2.

Now for the conduction of FEA on the pinion, this value of force was taken to be the most significant component. Ignoring shear forces that occur due to the rolling motion of the roller about their individual axes, we could assume that a particular roller will act as a fixed beam under a point load equal to half of the total tangential force.

Following are the values of torque and forces that were calculated:

Maximum torque applied at the steering by human
= 30 Nm
Tangential force transmitted by the pinion
= (Torque) ÷ **(**Pitch circle radius)
= 30×10^3 Nmm ÷ (49÷2) mm
= 1224.489796 N
Effective number of rollers in contact with rack
= 2

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Force acting on each roller
= (Total Force) ÷ 2
= 1224.489796 ÷ 2
= 612.244898
= 612 N (approx.)
Force acting on the rack tooth
= Force acting on each roller
= 612 N

3.3 Maximum Deflection

Using strength of materials, the maximum deflection acting on both pinion and rack can be calculated. These values are then matched against the FEA results later in the report.

3.3.1 Roller Pinion. For simplicity, the rolling motion of the roller has been ignored and considered as a fixed beam wherein, the ends of the roller are assumed to be fixed on to the pinion hub (when in reality it is free to rotate).

Thus the pinion is now basically a fixed beam with uniform loading.

The amount of load per unit length will be
= (Total Load) / (Length of roller)
= (612 N) / (14 mm)
= 43.7 N/mm
From SOM,
Maximum deflection for a fixed beam with UDL:
$y_{max} = (wL^4)/(384EI)$

Where,
w = UDL (N/mm)
L = Length of roller = 14mm
E = Young's Modulus = 200GPa
I = Area moment of inertia

$= (\pi/64)x(d)^4$
$= (\pi/64)x(7)^4 = 117.8588\ mm^4$

Thus,
$y_{max} = (43.7 \times 14^4) / (384 \times 200 \times 10^3 \times 117.8588)$
$= 1.85468579e^{-04} mm$
$y_{max} = 1.855\times10^{-7} m$

3.3.2 Rack. The rack tooth just like any other gear must be treated like a cantilever beam. In this case, it is observed that the most critical areas are near the crest (because of least cross sectional area). And hence, the cantilever must be considered to withstand a point load at the free end (when in reality, the whole of the flank is put under loading – this is considered for FEA in the next section).

Total load acting at the end of the cantilever beam,
P = 612 N
From SOM,
Maximum deflection for a cantilever beam with a point load on its free end:
$y_{max} = (PL^3)/(3EI)$
$= (612 \times 10.5^3) / (3 \times 200 \times 10^3 \times 512)$
$= 2.3062061e^{-03} mm$

$y_{max} = 2.3062\times10^{-06} m$
Where,
L = Length of rack = 16 mm
I = Area of moment of inertia
$= (1)/(12) x (b)(h)^3$
$= 1/12 x (12 x 8^3)$
$= 512\ mm^4$

here,

b=12mm

3.4 Finite Element Analyses

As the creation of the 3D geometries formed an important part of this project, the logical step was to analyze them as three dimensional components as well. Thus, SOLIDWORKS Simulation software was chosen to carry out the FEA analyses. The material for both components was chosen to be alloy steel.

3.4.1 Pinion. The working model was imported into SOLIDWORKS and constraints over the entire model were given. Since the deflection of the roller will be relative to the bore of the pinion, therefore, the entire bore, where the steering shaft is supposed to be press fitted, was supposed to be constrained. However, for more relevant results, the inner faces were taken to be fixed and not the bore.

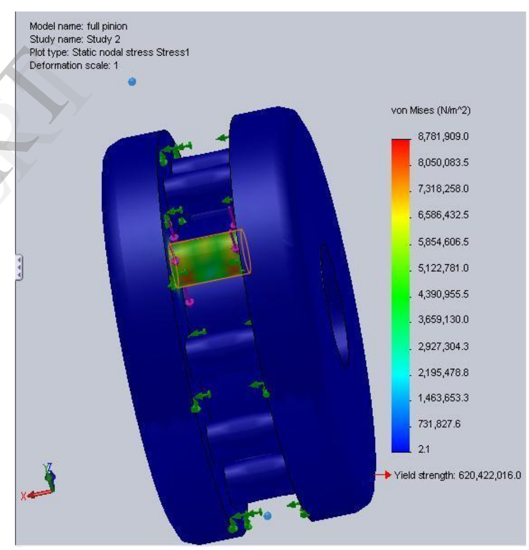

Figure 7. Von Mises Stress distribution for the pinion – Max. Stress = 8.781MPa

Factor of Safety, FOS = (620.42MPa)/(8.781MPa)
= 70.66, which is very safe.

3.4.2 Rack. The entire rack was not taken into consideration. Only one tooth was cut out. The root of the tooth was given a fixture (fixed) and a calculated pressure was applied to the entire flank.

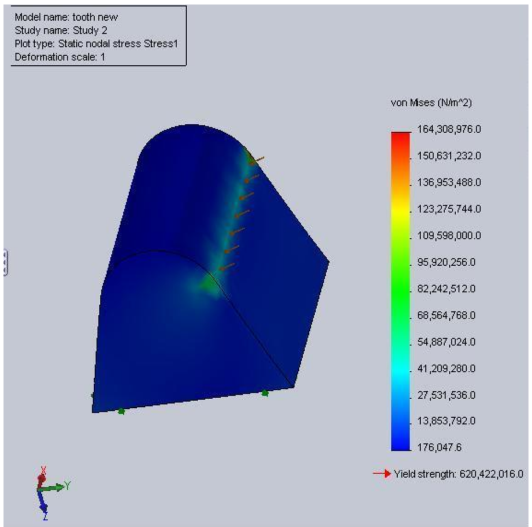

Figure 8. Von Mises Stress distribution for a single rack tooth – Max. Stress = 164.31 MPa

Factor of Safety, FOS = (620.42MPa)/(164.31MPa) = 3.78, which is reasonably safe.

4. Conclusion

The corresponding Von Mises Stress Distributions were found to be unusually low. This might have been because of the bigger size of the components, resulting in the presence of excessive material.

The size of the roller pinion that was assumed at the beginning of the selection process had a pitch circle diameter of around 50mm, which is more than twice that of a regular helical gear (≈20mm) used in traditional rack and pinion steering. The large size was maintained because of the discontinuous nature of the pinion. But as the FEA results now show that the stresses produced are very low; there is enough scope for optimization.

Optimization of the current component will involve reducing it in size such that it will still be able to withstand the same amount of load. This will be a highly beneficial procedure, as a smaller sized component means a more compact steering system for the vehicle.

In this project an attempt was made to improve an available mechanism by altering the design of one of the components. The new design was modelled and tested and was found to carry excessive weight to it. It now has to be subjected to optimization, so as to arrive at a final size that will still be able to withstand the load. Upon optimization, the design can be successfully incorporated into an automobile steering system.

The roller pinion version will make the car steering highly effective because it will completely eliminate losses in motion and as a result, improve the feedback mechanism. Also, because of the rolling motion involved, the maximum torque supplied at the steering wheel to turn the pinion over the rack will reduce.

5. Acknowledgments

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