Developing a Virtual Prototype of a Rack and Pinion Steering System

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Abstract: The operation of an automobile steering gear is intermittent with low angular velocities and frequent direction reversals. For a typical Rack and Pinion Steering (RPS) gear used in light vehicles, the span of movement of the rack is limited to approximately four rotations of the pinion. For this limited span of rack travel, the torque required to rotate the pinion is tested for satisfactory functioning of the steering gear. The torque variation due to engagement and disengagement of the pinion teeth, mesh friction variation, and spring force that keeps the gears in mesh, etc., are obtained from the virtual prototype developed in Automatic Dynamic Analysis of Mechanical Systems (ADAMS) software, where, the manufacturing errors are also incorporated. Comparisons of the results from the virtual prototype with those from laboratory tests validate the correctness of the proposed prototype in ADAMS. This prototype lends itself for the implementation of improvement concepts and performance testing.

Keywords: virtual prototype; Rack and Pinion Steering (RPS) gear; Free Pinion Torque (FPT).

Reference to this paper should be made as follows: Kamble, N. and Saha, S.K. (200x) 'Developing a Virtual Prototype of a Rack and Pinion Steering System', *Int. J. Vehicle Systems Modelling and Testing*, Vol. x, No. x, pp.xxx–xxx.

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1 Introduction

In this paper, a virtual prototype with complete CAD geometry of the Rack and Pinion Steering (RPS) gear used in small passenger cars, is proposed to reproduce the torque behaviour of the pinion. This prototype is developed in the ADAMS software. It is important to know the pinion torque characteristics as the vehicle handling comfort of the driver is governed by it. The aspect of handling comfort has motivated us to study the gear mesh phenomenon for the corresponding mesh friction excitation (Velex and Sainsot, 2002; Vaishya and Singh, 2001) for each rotation of the pinion. The rack and pinion gear is used for steering the vehicle. Its operating span is usually two pinion rotations in either direction. The performance tests for such steering gear are standardised to take into account for the cyclic and once per tooth variation in torque, and the effect of spring force keeping the gears in mesh (IS 13476, 1992).

The rack and pinion of a steering gear has different hands of helices. The crossed helical rack and pinion assembly has non-parallel and non-intersecting axes. For M1 type vehicles (ISO 3383, 1972), which constitutes of small and medium passenger cars, the RPS gear has overall steering ratio of 15–20:1 (Stoll and Reimpell, 2001). To achieve this gear ratio, the possible rack displacement in the vehicle is limited to about 140 mm. Hence, for this rack displacement the pinion diameter should be 18–20 mm. But, for M1 category vehicles, the pinion should be designed to carry a maximum of 30 Nm (92/62/EEC, 1992), while the vehicle is in motion. The demand for high torque carrying capability by the pinion teeth is met by increasing its tooth thickness, which is possible only if there is a reduction in the number of teeth. The tooth strength of the pinion subjected to this torque is increased by increasing the tooth thickness with corresponding decrease in number of teeth. The reduced number of teeth may lead to undercutting (Merrit, 1971; Buckingham, 1949). This undercutting is avoided by applying the profile shift to the pinion and the rack. The profile corrected pinion has long addendum and short dedendum. The virtual prototype of RPS gear made in ADAMS incorporates all these geometric details of its profile to reproduce the meshing phenomenon and its frictional behaviour. In order to have realistic teeth profiles, they are generated by calculating successive x and y coordinates of the involute profile of the tooth flank and trochoid at the root of the tooth (Celik, 1999). The line joining those points generates the tooth profile. In general, to simulate any gear kinematics, detailed gear geometry is required, where the kinematic relationship is represented by two discs of diameter equal to their pitch circle diameters. However, this representation is unable to produce the effects of backlash, friction, combination of rolling and sliding along the involute flanks. Hence, the CAD model with explicit parameters which influences the frictional torque are included in the prototype. Moreover, the realistic gear efficiencies can be calculated by considering the approach and recess action of the teeth mesh and the corresponding friction lever arm (Buckingham, 1949). This method provides a fairly accurate representation of the cyclic variation of the small magnitude from engagement to disengagement of one tooth.

Following aspects are included in the ADAMS virtual prototype of the RPS:

- three dimensional CAD models of the gears in mesh
- spring force acting on the rack and corresponding dual contact of mating teeth
- crossed helical gearing

- incorporation of the misalignments of the gear axes
- profile shifted gear.

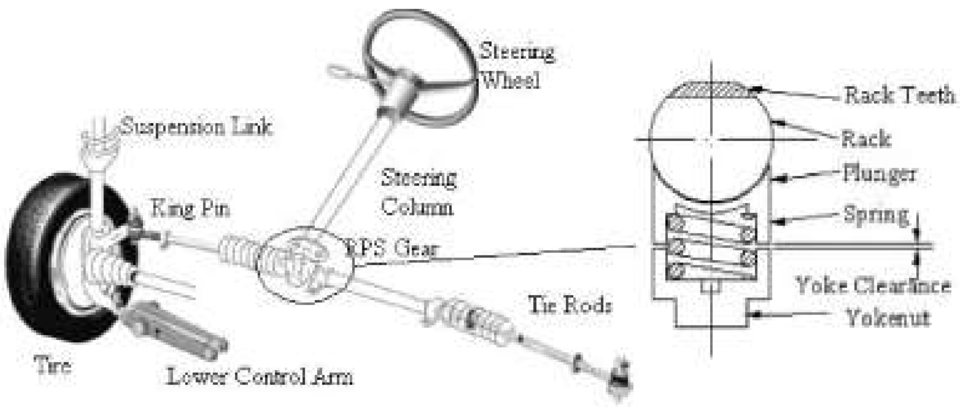
Literatures on steering systems generally focus on synthesis and analysis of linkage system. (Simionescu and Smith, 2000; Simionescu et. al., 2000) Even though there are vast literatures on gear designs and analysis (Vaishya and Singh, 2001; Zang and Fang, 1999; Michilin and Myunster, 2002), modelling of the steering gears in a steering system is rare, except by the authors in Kamble and Saha (2005a, 2005b) and few others (Filho, 2003; Baxter et. al., 2001). This paper presents the modelling of different components like rack, pinion, spring etc. in ADAMS environment as realistically as possible to reproduce the pinion torque characteristics as obtained in the laboratory tests. The contribution of this paper is modelling approach in ADAMS environment for realistic output. The paper is organised as follows: Section 2 presents description of a RPS system and its performance criteria, whereas Section 3 explains the basics of gear mechanics required in the generation of the gear prototype. Section 4 describes the advantages of using the complete gear model instead of using a gear joint tool in the software. Section 5 explains the modelling procedure with the results in Section 6. Finally, conclusions are given in Section 7.

2 Rack and Pinion Steering (RPS) gear

The manual Rack and Pinion Steering (RPS) system, as shown in Figure 1, consists of a crossed helical gear pair of rack and pinion. A crossed helical gear has poor contact properties and its use is limited to low load application (Michalec, 1966). This gearing arrangement resembles a worm and worm wheel, where the worm wheel is replaced by a rack with inclined teeth. The rack and pinion gear has considerable sliding due to

- differences in transverse pitches along the direction of rack translation
- all addendum pinion that has increased recess action.

Figure 1 A typical Rack and Pinion Steering (RPS) System (a) Rack and Pinion Steering System, and (b) Yokenut assembly



(a) (b)

The use of an RPS is limited to small and medium passenger cars with independent front wheel suspensions. In such arrangements, the rack is fixed to the car chassis through a tube constraining it. The steering rack is connected to the front road wheels through a ball and socket joint at each end. This system is compact and easy to manufacture. Also the anticipated wear can be compensated by meshing the gears under a spring preload. As a result, steering performance is retained over a long period of time in spite of the wear of the different components. In addition, the spring preload provides required stiffness to the steering systems to maintain good self-steering efficiency (IS 13476, 1992). The steering ratio of passenger cars is about 15–20:1 (Stoll and Reimpell, 2001). Approximately four turns of the steering wheel are required to translate the rack through about 140 mm, from left to right lock position. Four pinion revolutions turn the road wheels through a total of 60–80°. The distance travelled by the rack for one revolution of the pinion is termed as rack gain. The pinion diameter is usually small, equal to 18–20 mm, as dictated by the required rack gain. Moreover, as per the steering standards (92/62/EEC, 1992) the steering gear should not exceed a torque of about 30 Nm torque when installed on vehicle. Similarly, standalone steering gear should not exceed the torque of about 1.5 Nm. The nominal value of the torque on a standalone steering gear is raised to 1.5 Nm by meshing the rack and pinion under spring preload. This spring preload also serves the purpose of backlash elimination. The pinion needs to transmit the rated torque of about 30 Nm while steering the vehicle in motion, the pinion tooth strength is increased by increasing the tooth thickness. Consequently, the pinion has unusually low number of teeth, generally 5 or 6 teeth. Fewer number of teeth increases the tooth strength but the contact ratio becomes poor. In such situations, the helical gear helps to contact ratio by contributing its face contact ratio to the total contact ratio. On the other hand, flexible mounting of the rack and pinion with spring loaded rack takes care of any centre distance variations between the rack and pinion that may have resulted due to the backlash or gear errors. Also note that, while the circular cross section of the rack is suitable for the ease of manufacturing and assembly, it has a tendency to roll about its own axis when it is translating.

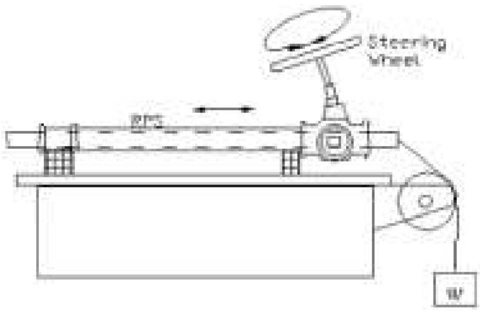
2.1 Performance of a steering gear

This subsection presents the different ways a steering system is assessed. It is done in two ways. One way is to test the steering system for the maximum steering effort while the vehicle negotiates a curve of minimum turning radius. Alternatively, the steering gear is tested for the torque requirement, efficiency and endurance, while it is in a standalone mode. The scope of this work is limited to the testing of a standalone steering gear for the torque requirement and efficiency. The torque required to rotate the pinion of a steering gear in stand alone mode as shown in Figure 2(a), with no load is termed as 'Free Pinion Torque' (FPT), where the term 'Free' differentiate it from the torque required by the steering gear while it is fitted on the vehicle. A typical FPT plot is shown in Figure 2(b). The efficiency test is conducted by attaching a weight at the end of the rack, as indicated in Figure 2(a). The forward or steering-in efficiency (IS 13476, 1992) is recorded when the pinion rotation raises the weight. On the contrary, resisting torque required by the pinion to avoid the weight from descending due to its own weight gives the reverse or self-centring efficiency of the steering gear (IS 13476, 1992). The forward and reverse efficiency is given by the following formulae.

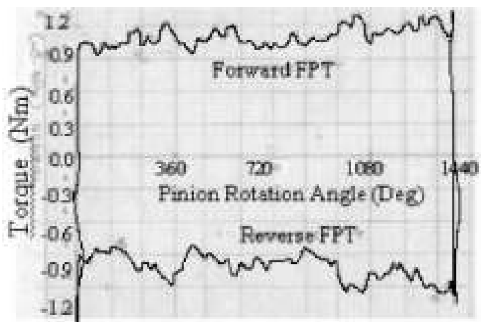
$$\eta_f = \frac{w \times s}{2 \times \pi \times \tau_f}; \text{ and } \eta_\tau = \frac{2 \times \pi \times \tau_\tau}{w \times s} \tag{1}$$

where η_f and η_r are the forward and reverse efficiencies, respectively, whereas w: dead weight and s = rack gain. The forward efficiency is important for the steering effort required for turning manoeuvre. The reverse efficiency is important for self-centring ability of the vehicle after negotiating a turn, as well as to reduce the sensitiveness of the steering system to road profile disturbances. The self-steering efficiency should be greater than 90% of the forward efficiency.

Figure 2 Performance of an RPS system (a) RPS test rig in standalone mode and (b) a typical plot of Free Pinion Torque



(a)



(b)

The mesh stiffness and the frictional behaviour of the crossed helical gear under varying spring force during motion have considerable influence on the torque characteristics of the steering gear. This aspect is successfully incorporated in the ADAMS' virtual prototype discussed ahead.

3 Pinion profile modification

To include the torque variation due to engagement and disengagement of the teeth and the effect of profile corrected pinion (Velex and Sainsot, 2002); the rack and pinion is modelled with actual involute profile instead of using ready-made gear joint in ADAMS which do not provide realistic results, as explained in Section 1. For an involute gear mesh, the conjugate gear tooth action (Buckingham, 1949) gives uniform angular velocity between the driver and the driven gears. Conversely, the path of contact generated by both the profiles is identical. The latter property states that, by knowing any one profile, the conjugate profile can be generated to achieve identical path of contact. A rack with straight flanks is generally used to generate gear profiles. Such a rack is called Basic Rack (ISO 53, 1974) as shown in Figure 3. Both the involute flank and the trochoid curves at the tooth fillet are dependent on the basic rack profile, which is the starting point of the profile generation. As explained earlier, the pinion has 5 or 6 teeth with major diameter of about 18–20 mm. With reduction in number of teeth, the gear has a tendency of undercutting. Due to undercutting, the useful portion of the involute already generated is destroyed by the cutting tool as the generation continues. The resulting tooth has reduced tooth width at the root called as undercut. This makes a part of the useful involute flank unavailable. The undercut can be avoided by displacing the cutter rack away from the blank. This condition is explained in Figure 4. The point ‘C1’, Figure 4, is the point where straight flank of the rack merges with tip radius. To avoid interference, a horizontal line passing through point ‘C1’ should lie above the point T, where the point T is the point of tangency of line of action with the base circle. In order to avoid the interference, the cutter rack can be withdrawn by an amount as shown in Figure 4. This shifting of cutter rack away from the gear blank is known as the ‘profile shift’ or ‘addendum correction’ (ISO 53, 1974). The profile shift increases the tooth width at critical section, but, reduces the crest width at the tip. This reduced crest width is prone to crumbling or chipping away of the tooth tip during operation. To overcome this problem, a part of the addendum is cut off, which is termed as ‘Topping up’. The increased addendum ‘a’ and reduced dedendum ‘b’ increase the angle of recess and reduce the angle of approach as shown in Figure 6. The approach and recess distances are L_a and L_r, respectively. For a tooth without profile shift, both approach and recess distances are nearly equal. With positive profile shift, Figure 5(b) the increase in L_r, increases the frictional torque, while the tooth advances from pitch point towards disengagement point. The pinion of the RPS gear under investigation is an all addendum gear, i.e., dedendum, $b = 0$. It increases L_r, which induces more sliding friction in the steering gear pair. The pinion of the RPS gear under investigation is an all addendum gear, i.e., dedendum, $b = 0$. It increases L_r, which induces more sliding friction in the steering gear pair.

Figure 3 A basic rack

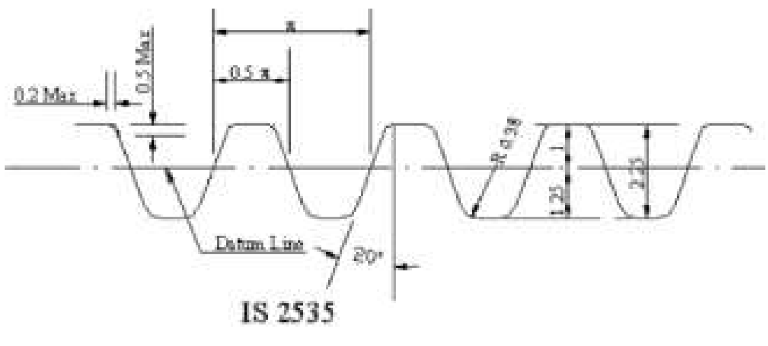


Figure 4 Schematic representation of undercut

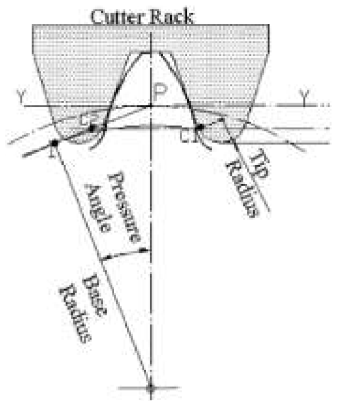
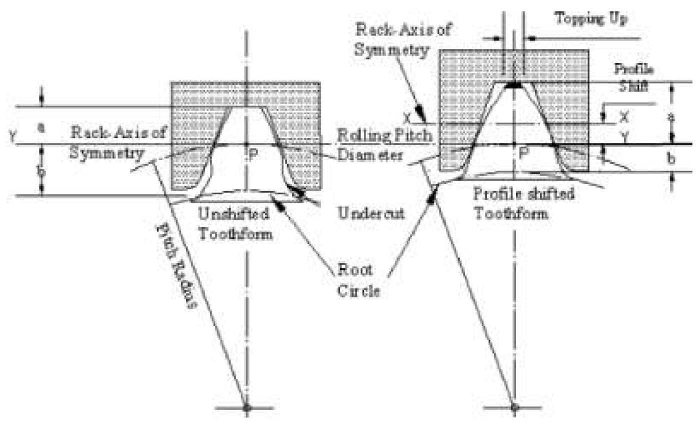
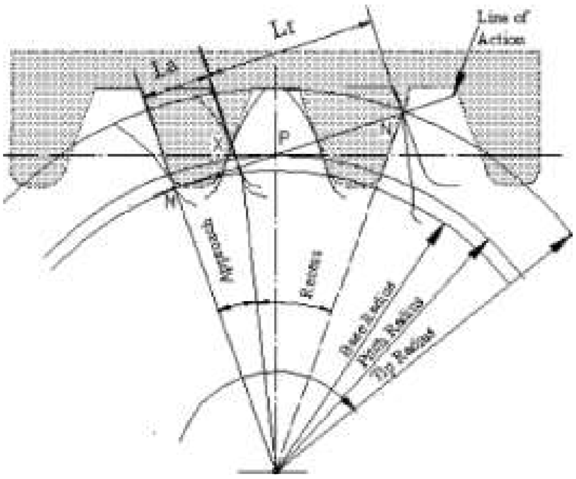


Figure 5 Schematic representation of the profile shift (a) tooth without profile shift and (b) profile shifted tooth



(a) (b)

Figure 6 Approach and recess action of a profile shifted pinion and rack pair



4 Virtual prototyping in ADAMS

ADAMS is a dynamic simulation software which has modelling as well as analysis capabilities (ADAMS-2005). Variety of mechanical systems can be modelled in ADAMS and simulated to check its performance. Such computer models are often referred as virtual prototypes. They are very useful to reduce the time from design to the actual product, as many critical analyses can be performed with the click of a button in minutes, whereas building actual prototypes will take at least days or months together before it can be even put to tests. To build a prototype, two-dimensional basic building blocks like points, lines, markers, arcs, and spline are the features available in ADAMS. Several types of solid geometries can be created with the available solid primitives like box, cylinder, frustum and cone. Commonly applied features like fillet, chamfers and surfaces of revolution are also available in the modelling tools. Further complex objects can be created by constructing planar section with the help of lines, arcs and spline, and extruded along an axis to model objects like spur gears, etc. Various solid objects can also be created using Boolean geometry, where one or more features are added or subtracted to and from an existing one, respectively. Various joint constraints and forces are provided in ADAMS to replicate the actual mechanical systems. On the other hand motion generator can be used to apply motion to the joints to determine the torque or force required for that desired motion. Impact or restitution based contact can be defined between two entities. The contact forces and frictional forces can be represented too by defining the friction coefficients and static and dynamic friction transition velocities.

4.1 Realistic model of the gear

In this subsection, the departure from the use of available features in ADAMS, e.g., gear joint, is emphasised, as they do not reproduce the meshing phenomenon of RPS gear realistically. For example, the involute profile and the contact phenomenon do not exist in the gear joint of ADAMS. Moreover, the crossed helical gear of the RPS system behaves as a worm with the number of teeth on the pinion equal to the number of starts on the worm. The motion transmitted is governed by the transverse pitch in the direction in which the rack moves. Note that the gears being spring loaded there is a tendency of centre distance variation, which is necessary to avoid backlash or binding of the gears at any point of mesh. The spring preload also introduces additional frictional resistance in the gear, so that road disturbances are not easily transmitted to the steering wheel. Each of these aspects is incorporated in the prototype to make it realistic.

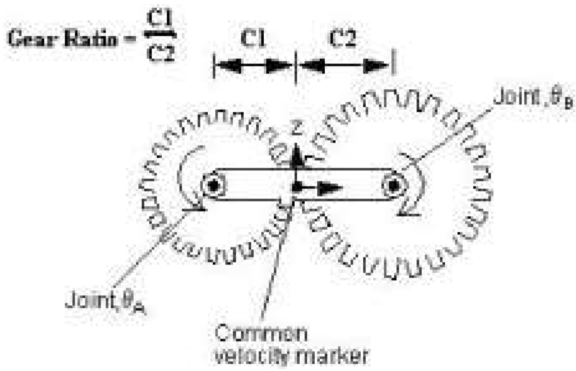
Gear joint in ADAMS resembles two rolling discs with diameters representing the pitch circle diameters. Each of these discs should have revolute or cylindrical joint. The gear joint needs three parameters, namely,

- first revolute joint
- second revolute joint
- common velocity marker.

The common velocity marker defines the direction of peripheral velocity of the gears. The gear pair then behaves like two spur gears meshed together with their axis at right angles to the common velocity marker. The gear ratio is purely defined by the distances

C1 and C2, i.e., the radii of the discs, as shown in Figure 7. The gear joint is able to satisfy only the kinematic constraint of relative angular velocity between the two mating discs or gears. There is no direct provision for backlash incorporation, etc. Helical gears can be indirectly modelled by inclining the common velocity marker in the direction of the helix angle, but for the crossed helical gears, the helices on both the gear are different. So the direction of the peripheral velocities of the mating gears are different, thus making the use of ready-made gear joint of ADAMS with common velocity marker is unsuitable for the present purpose of developing a realistic RPS gear model.

Figure 7 A gear joint in ADAMS

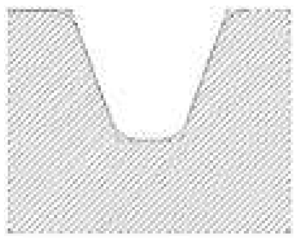


The ADAMS' toolbox, however, allows the definition of a physical contact between any two rigid bodies, and frictional properties can be defined at the point of contact. If the modelled entities conform to required involute profile then the contact pattern on the surfaces of the contacting bodies will resemble the contact pattern occurring in actual gears. This has led to the concept of modelling the three-dimensional (3D) model of the gear pair, and imposition of contact between them. Since the RPS gear resembles a worm gear, the true rolling diameter of the pinion depends on the transverse pitch of the pinion along the direction of motion of rack. Also, the torque required on the pinion largely depends upon the frictional and sliding losses at the gear mesh. So the condition of spring loaded gears needs to be taken into consideration while modelling the torque requirement. The sliding losses depend upon the kind of gearing, location of the point of contact with respect to the pitch point. This makes it necessary to consider the gear mechanics also while prototyping the RPS gear with the non-standard gear pair. With the gears modelled as 3D entities and with due considerations to the profile accuracies, the contact point follows the path of contact as in actual gear pair. The motion of the driven gear in this case is not due to any gear joint available in ADAMS but the teeth of the driver teeth pushes the driven gear teeth through the 'Contact' function of ADAMS. To reproduce the actual behaviour of the RPS gear, the 3D solid models of gear entities are made and the boundary conditions are applied to replicate the actual system.

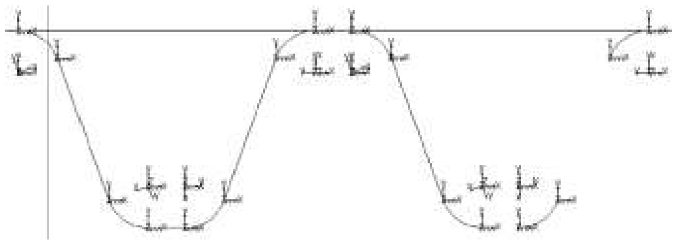
5 Modelling of RPS gear components

In RPS, the teeth of both rack and pinion have conjugate profiles which are generated by the same basic rack that is used as rack cutter. Thus, the first step of modelling is to make the basic rack. A two dimensional geometry of the basic rack as shown in Figure 8(a) is generated in AUTOCAD (AUTOCAD-2002). The data points are imported through a .dwg file from AUTOCAD to ADAMS and then connected by lines to make flanks and arcs of tip-root radii. All the line and arc segments are linked together by a continuous multi-line, Figure 8(b), to make a closed 2D rack cutter geometry with required number of teeth. This 2D rack is then extruded to cover the face-width of the pinion, as shown in Figure 8(c).

Figure 8 Generation of a rack cutter (Basic rack) (a) one tooth span of basic rack in AUTOCAD (b) rack profile in ADAMS with data points imported from AUTOCAD and (c) extrusion of the planar rack section



(a)



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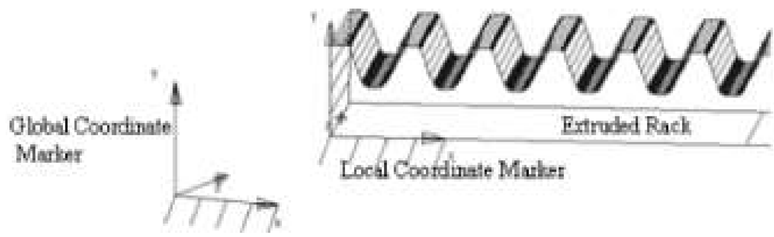


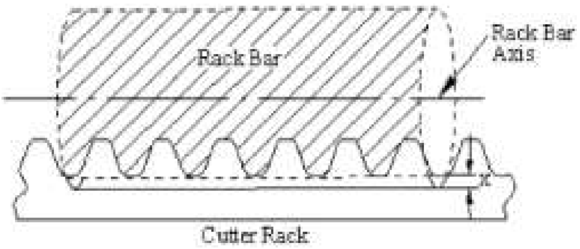
(c)

5.1 Rack

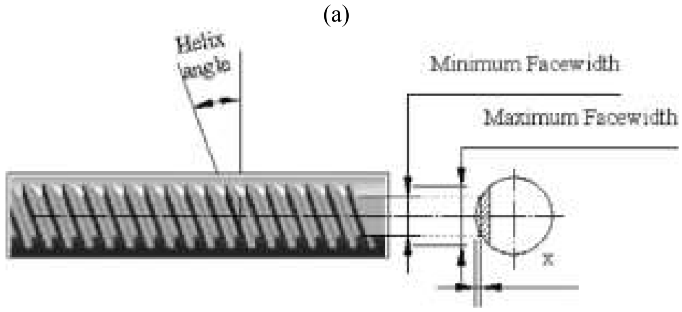
The rack of the steering gear is a circular bar with teeth cut across a part of its length. The steps followed during the generation of rack are as follows:

- A cylindrical bar of diameter equal to the rack diameter is drawn in ADAMS using ‘Cylinder’ option in ADAMS. The rack cutter is plunged into the blank by amount ‘x’, as shown in Figure 9(a), to achieve a flat top land for the teeth cut on circular bar.
- Two markers defined in the workspace of the model to represent the global coordinates and the local coordinates of the rack, as in Figure 8(c). In the prototype, the global and local coordinate markers represent the normal and transverse pitches of the gears respectively. The cutter rack is oriented in relation to the blank according to the helix angle of the rack.
- With the use of the option ‘Cut out solid with another’ from the Boolean geometry of the ADAMS toolbox, the cutter rack geometry is subtracted from the rack bar. The resulting body is the rack of the RPS gear (Figure 9(b)). The face-width of the rack teeth varies from top to the bottom of the teeth due to the circular cross section of the rack bar. It has minimum face-width at the top of the teeth, which gradually rises to maximum at the bottom of the teeth.

Figure 9 Modelling of rack (a) the rack bar and the basic rack cutter and (b) rack bar with helical teeth



(a)



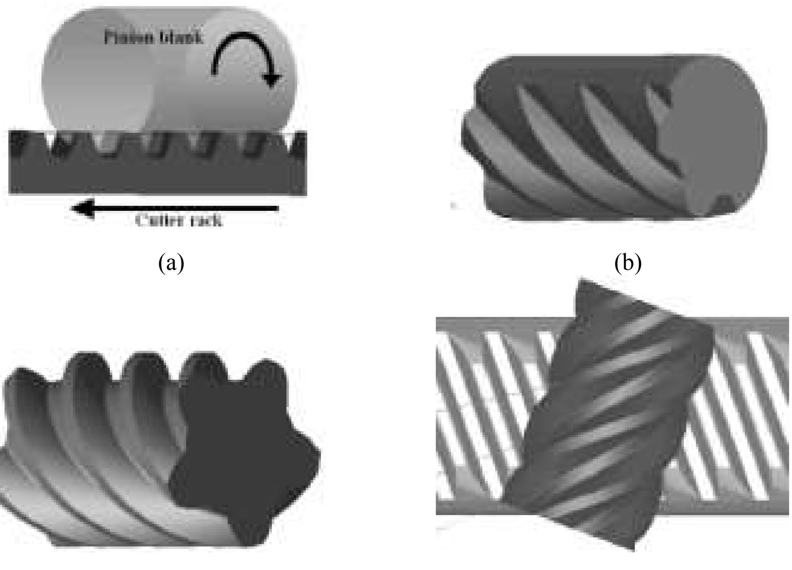
(b)

5.2 Pinion

The generation of pinion in ADAMS is done in a manner similar to the gear shaping process (Merrit, 1971). The gear shaping and hobbing are similar in their principle of involute generation. The hobbing is a continuous generation process where the hob is continuously fed into the workpiece, whereas, the gear shaping is an intermittent process where the cutter rack makes series of cut on blank. For each cutting instance during shaping, the rack and blank are moved through a small incremental displacement, as if they are in mesh. At the end of rack-stroke, it is withdrawn from the blank and brought to the original position. The principle of gear shaping is used while modelling the pinion in ADMAS, i.e.,

- A cylinder of nominal diameter is made first. The pitch diameter of the pinion blank and the line of symmetry of the rack are separated by the amount of profile shift. The cylinder and the cutter rack is oriented to the required helix angle of the pinion.
- Similar to the local coordinate marker for rack, a local coordinate marker representing transverse section of the pinion is placed at the pinion centre. Both the local coordinate markers for rack and pinion helps to provide relative motion between the cutter rack and the pinion blank.
- For each incremental cutting instance, both the cutter rack and pinion are given a relative displacement. The motion of the rack occurs along the local X coordinate of the rack, with an amount equal to the circumferential distance travelled by the pinion, as indicated in Figure 10(a).
- For each cutting instance, the cutter rack geometry is subtracted from the pinion blank. For each incremental revolution of the pinion by 3.6 degrees, the rack is translated along the normal pitch direction, as if the both are in mesh. The cutting operation is repeated until one complete rotation of the pinion. The pinion blank after 90° and 360° of cutting motion is shown in Figure 10(b) and (c), respectively, whereas its mesh with the rack is shown in Figure 10(d).

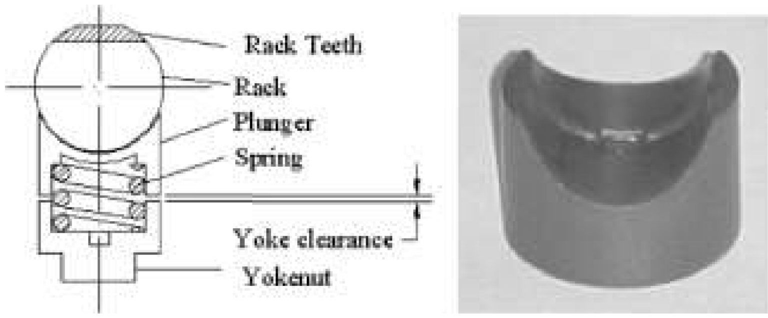
Figure 10 Generation of pinion using gear shaping method (a) pinion ready to cut; (b) pinion after 90° of cutting motion; (c) pinion after 360° of cutting motion and (d) pinion meshed with rack



5.3 Yokenut assembly

After modelling the rack and pinion, the yokenut assembly of the RPS that keeps the rack in proper mesh with the pinion is modelled. Since the steering motions include frequent direction reversals as per the driving needs, any backlash is undesirable. Besides, the rack teeth portion is subjected to uneven wear. Hence, a yokenut is fixed to the vehicle frame while the plunger rests against the rack with a spring between the yokenut and plunger as shown in Figure 11(a). The spring is compressed by tightening the yokenut. The yoke clearance can be adjusted by changing the position of yokenut. The rack bar is pushed against the pinion due to the plunger.

Figure 11 Yokenut assembly (a) different components and (b) a plunger



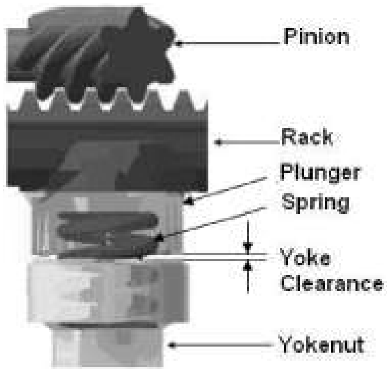
(a) (b)

In order to introduce rack-plunger frictional interface, a plunger as shown in Figure 11, is generated in ADAMS, followed by the application of the surface contact between the plunger and the rack bar. The steps followed during modelling the yokenut assembly are as follows:

- A plunger is modelled with cylindrical resting surface for rack, as seen in Figure 11(b). To represent rack-plunger frictional interface, a 'contact' is defined between the two bodies. The coefficient of friction for the interface is taken from the actual test data for the metallic rack bar and non-metallic plunger pair.
- Another cylinder representing the yokenut is made with axis of both the plunger and the yokenut aligned. Both the plunger and yokenut are separated by a distance equal to the yoke clearance, as depicted in Figure 12.
- The distance between the yokenut and plunger thus represents the yoke clearance. While simulating the gear motion, the impact of the plunger on yokenut due to centre distance variation needs to be modelled. So another contact is placed between the plunger and the yokenut.
- The yokenut is fixed to the ground representing the car chassis, with the help of 'Lock' joint and a translational joint is applied between the plunger and ground, so that the plunger is free to move axially with respect to the yokenut.
- A spring is placed between the plunger and the yokenut. The attachment points for the placement of the spring are the 'centre of mass markers' for the respective bodies. The damping coefficient is set to zero, as there is no damping arrangement.

- In the spring placement dialogue box, the spring preload value is set to the rated spring preload applied to the assembly in practice. This preload can be varied by 'Spring-Modify' dialogue box to simulate the effect of the plunger load variation on the RPS assembly.
- During simulation, the impact between the plunger and yokenut occurs due to the movements of the plunger within the yoke clearance against the spring preload. The completed yokenut assembly is shown in the Figure 12.

Figure 12 Yokenut assembly in ADAMS



5.4 Constraints

The rack in RPS gear is not completely constrained to move axially. The rack has a cylindrical joint with a bush at one end, where the bush itself is flexible. The rack is supported by a spring loaded plunger at the other end. This arrangement allows slight misalignments of rack axis due to swivelling about the flexible bush. The joint friction values from the experimental studies are incorporated in the model.

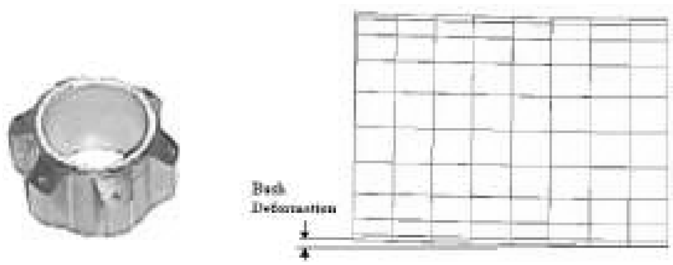
5.4.1 Translation joint of rack

In the real system, the rack slides within a bush which is flexible. The rack and bush are assembled under certain preload to avoid slackness. The bush material being flexible also allows small movements of the rack corresponding to the variation in backlash. Thus, it also accommodates centre distance variation. The bush is either metallic with outer lining of hard rubber or a polymer bush as shown in Figure 13(a). While modelling, the bush is made up of a flexible link between the ground and rack. It is composed of 25 discrete elements. Hence, the bush acts as a cylindrical joint between the rack and the ground (fixed link), and allows it to translate as well as swivel. The steps followed for the modelling of bush, as shown in Figure 13(a), are listed next:

- To reproduce the flexibility of the bush, the bush is modelled with the following modelling option of ADAMS: Build-Flexible Bodies-Discrete Flexible Link. This link (bush) is made up of 25 hollow cylindrical segments which have flexible connections between each pair of segments.

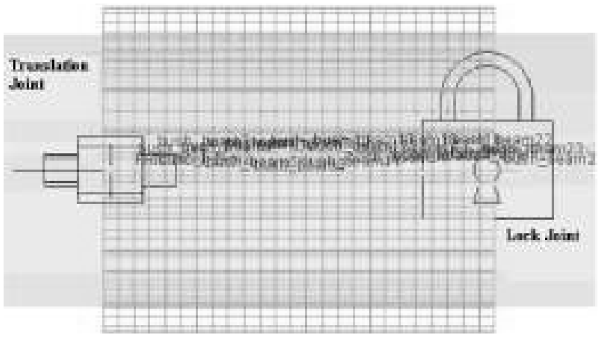
- The first and last link segment has attachment points with other bodies. Thus the flexible link has two attachment points. One of the attachment points, away from the pinion, is locked with the ground. The attachment point nearer to the pinion is linked to the rack with a cylindrical joint, as illustrated in the Figure 14.
- During motion, the centre distance between the pinion and rack varies. This movement of rack is accommodated by deflection of bush as shown in Figure 13(b).
- The rack is thus flexibly mounted and is free to translate, rotate as well as swing about the fixed end of the bush to accommodate backlash at the teeth mesh.
- Certain amount of drag force is required to push the rack through the bush, as the bush is fixed with preload to avoid slackness during the operation or due to wear. This drag force is introduced in the model by incorporating a known value of friction preload at the cylindrical joint between the rack bar and the bush.

Figure 13 A bush and its model (a) a real bush and (b) bush deflection in ADAMS model



(a) (b)

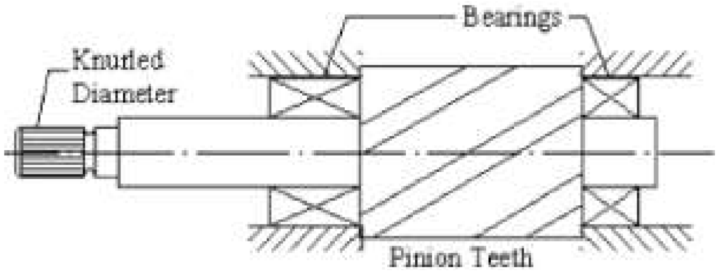
Figure 14 Bush with fixed and translation joints



5.4.2 Revolute joint of pinion

In practice, the revolute joint between the pinion and the ground is achieved by assembling the pinion within the pinion housing with bearings, as indicated in Figure 15(a). The pinion has fixed axis of revolution and is assembled in the pinion housing with the help of needle bearings. The pinion when mounted on the pinion housing with the bearings, it needs certain amount of torque to overcome the bearing friction. The experimental values of the bearing friction are introduced in the revolute joint as the revolute joint friction.

Figure 15 Mounting of the pinion



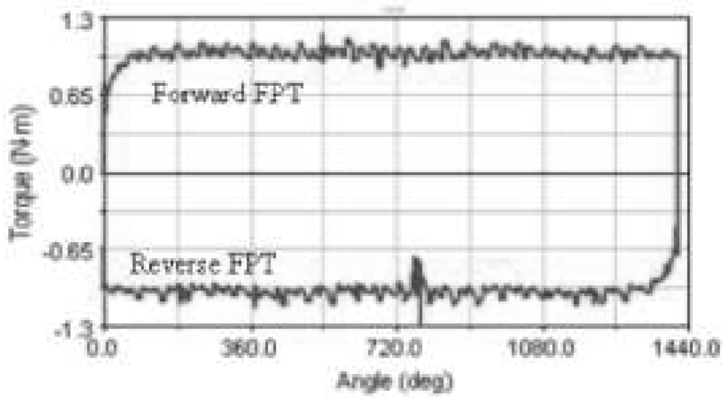
6 Simulation results

For the rack and pinion gear pair, the corresponding two joints are revolute and translational joints. The prototype can be simulated by applying motion either to the revolute or translational joints, and checking the torque or rack push force required, respectively.

6.1 Free Pinion Torque (FPT)

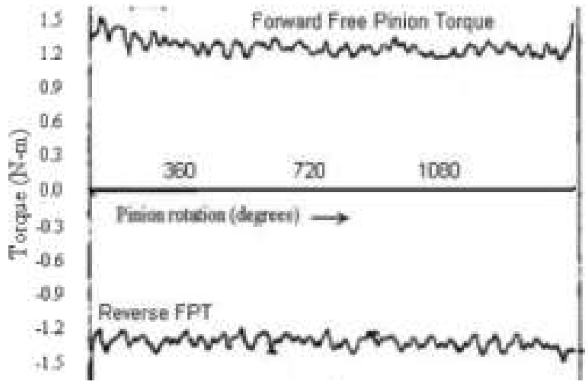
The Free Pinion Torque (FPT) serves as a performance standard for the satisfactory functioning of the RPS gear. The FPT is the torque required to rotate the pinion on a stand alone assembly under no load condition. Once a motion is applied to the pinion revolute joint and the model is simulated, the post processor in ADAMS is used to check the various motion and force components for the model. The torque required for the motion applied to the revolute joint gives the FPT. A typical FPT obtained using ADAMS and the experimental set-up, as shown in Figure 2(a), are shown in Figure 16.

Figure 16 FPT plots for ADAMS and experimental set-up (a) ADAMS FPT plot and (b) experimental FPT plot



(a)

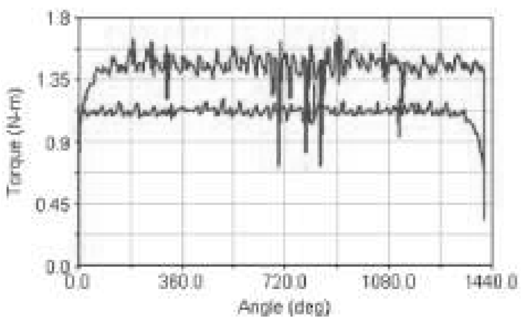
Figure 16 FPT plots for ADAMS and experimental set-up (a) ADAMS FPT plot and (b) experimental FPT plot (continued)



(b)

In practice, the efficiency test is also carried out for the RPS by attaching a weight to the end of the rack. By reversing the direction of rotation of pinion, both the forward and reverse torques and corresponding efficiencies are obtained. The forward and reverse torque in this case differs by an amount equal to the frictional resistance within the gearbox. During the forward torque test, the load falling under its own weight helps to rotate the pinion. The friction within the gearbox opposes the downward motion of the load. On the other hand, during the reverse torque test, the load is raised against the gravity and frictional resistance. The reverse efficiency of the RPS should be greater than 90% of the forward efficiency. The forward efficiency determines the effort required to turn the vehicle and the reverse efficiency is responsible for self-centring ability of the vehicle (IS 13476, 1992). This can be simulated in the virtual prototype by applying a force along the axis of rack equal to the weight to be attached during the actual test. The FPT values for various plunger loads can also be tested for any change in the spring preload by changing the spring properties of the model. A typical plot for the load of 10 kg is shown in Figure 17, which gives the reverse efficiency, η_r, calculated using equation (1) as about 90% of the forward efficiency, η_f, also given in equation (1). Frictional coefficients between the rack-plunger and rack-pinion interfaces can also be changed to know the frictional losses, etc.

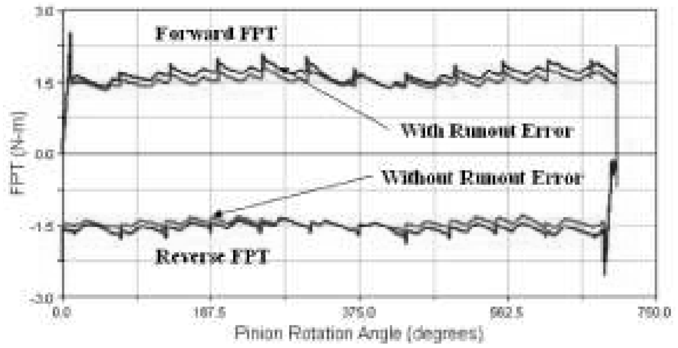
Figure 17 Efficiency measurement using ADAMS prototype



6.2 Gear errors

The prototype lends itself for the incorporation of some of the manufacturing and assembly errors in the RPS gear. As an example, runout error is modelled by displacing the pinion axis of rotation. Also, the installation angle can be deviated from the correct value to check its effect. A FPT plot for 'no runout error' and runout error of 40 μm is shown in Figure 18.

Figure 18 Runout error in ADAMS model



7 Conclusions

The paper presents the modelling of a realistic RPS gear using ADAMS software. The model lends itself for easy manipulation of the parameters, and functions in a way similar to the physical system. Hence, the model is referred as the virtual prototype of the RPS gear, which can be used to incorporate any proposed modifications in variety of ways. The FPT plots closely resemble the test results. Hence, the RPS virtual prototype is satisfactory. The contribution of this paper lies in the novel way of modelling the gear teeth in ADAMS, along with the incorporation of errors which are generally difficult to incorporate using the existing tools in ADAMS.

Acknowledgement

The research work reported in this paper has been supported by 'Sona Koyo Steering Systems Ltd.', Gurgaon, India. The authors sincerely acknowledge their support.

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