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Virtual Prototype of Rack and Pinion Steering Gear Meshing

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ABSTRACT

Rack and pinion steering gear being compact and light package with kinematically stiffer characteristics is commonly employed on passenger vehicle cars. Satisfactory performance of the steering system is determined by an acceptance test, which checks the composite error in the gear. The acceptance test checks the assembled gearbox for its torque characteristics instead of checking the individual components. The torque required by the pinion to rotate is the 'Free Pinion Torque' (FPT). FPT on Assembly varies within a range of 0.4-0.6 N-m, even if the tolerances on individual components (such as PCD run out of pinion, rack bend) are maintained within close tolerances. A virtual prototype of Rack and pinion steering gear is made in ADAMS [Automatic Dynamic Analysis of Mechanical Systems). This model will help to identify critical parameters and their effects on the assembly.

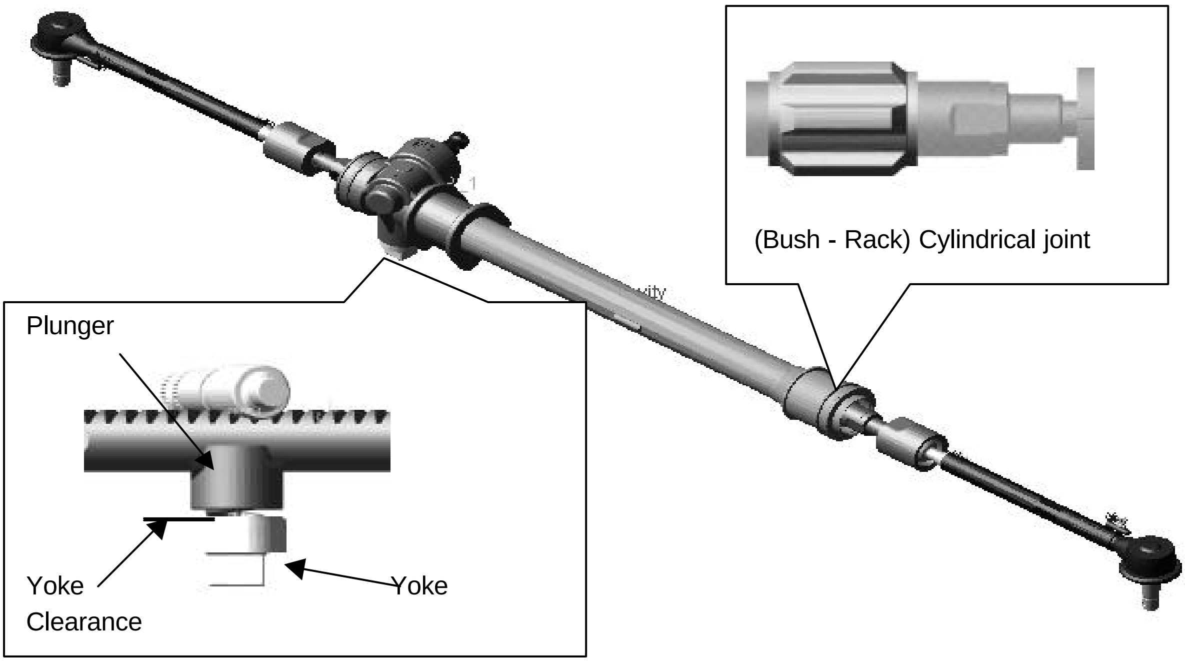

Figure 1. Rack and Pinion Steering Gear (ADAMS model)

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INTRODUCTION

To improve driver vehicle system performance, it is necessary that steering system convey correct information to the driver, giving good ‘feel’ of the road. For this the transmission error within the gearbox needs to be minimum. Simple and compact construction of Rack and Pinion Steering Gear as shown in Figure 1 favors its use over other types of steering systems like worm and gear type. The steering gear has steering ratio of 15-20 to 1 on passenger cars. Ratio lesser than this will require more effort to be put on steering wheel while turning. On the other hand, high steering ratio will need the steering wheel to be turned through a large number of rotations, which will make it inconvenient for the driver to negotiate sharp corner. Steering linkages and lengths of tie rods combinely determine the steering ratio. Specific case of Maruti 800 shows that the rack gain should be 34.1504mm per revolution of the pinion. This has led to use of non-standard pinion in the gearbox with 6 numbers of teeth with 20^o pressure angle. Steering gear has crossed helical gearing arrangement. Impact of this kind of gearing on steering performance is discussed later. Sources of error in transmission can be traced back to few of the following reason:

1) Friction: Gearbox assembly has three friction interfaces namely:

a. Rack and bush: Bush is made of flexible material. The frictional force cannot be simply product of friction coefficient and normal load at the interface. The reason is that the bush grips the rack surface circumferentially just like a rubber sleeve. This friction sufficient to hold the weight of rack when no lubricant is applied.

b. Rack and plunger: Friction at this interface depends upon the normal force applied by the spring preload. Any variation in the normal load is due to compression of the spring.

c. Rack and Pinion: Friction at the rack and pinion interface is of critical importance. It depends upon the geometry of the gears. Rack and pinion form a ‘Crossed type helical gearing arrangement’, which has poor precision rating, skewed shafting, point contact, high sliding. Here the point contact limits the capacity of the gearbox (less than 10 hp).

2) Deflection of bush: Bush deflection due to normal tooth load on rack changes the direction of motion of rack along the cylindrical joint.

3) Nature of contact: Pinion is mounted on bearings with a fixed axis of rotation. But rack is not completely constrained to move in axial direction only. It may deflect in vertical direction also. Thus the point of contact may not move along a fixed helicoids path. Point of contact may change as the distance between the axes of the gears is changed due to vertical displacement of the rack. This gives more frictional loss.

Figure 2 shows the ADAMS model of ‘Manual Rack and Pinion Steering’. Possible deviations from ideal motion expected in rack and pinion gear are also marked on the figure.

VIRTUAL PROTOTYPE OF THE GEARBOX:

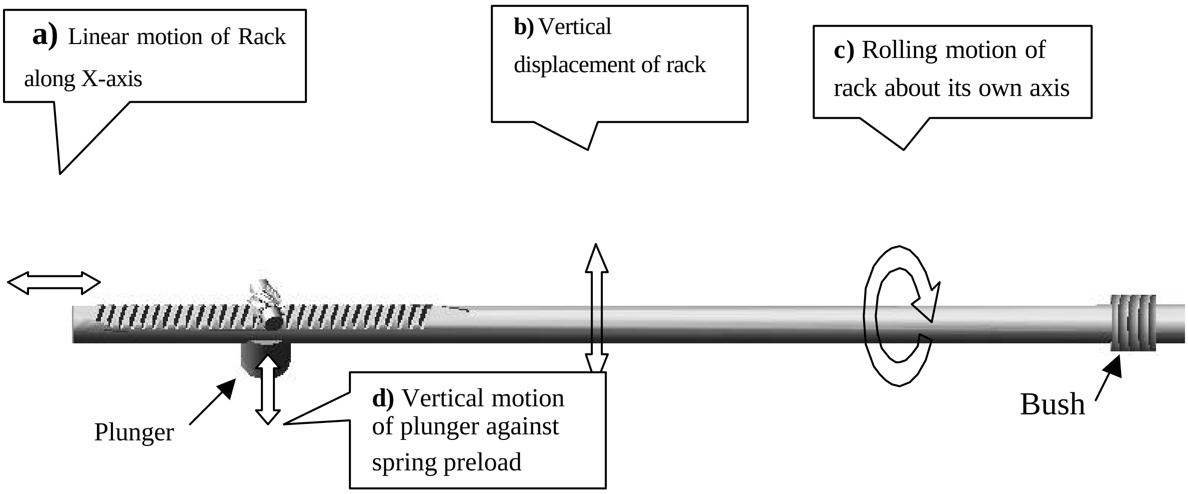


Figure 2. ADAMS model of Rack and Pinion Steering Gear with possible motions

Pinion: This is a non-standard pinion with 6 numbers of teeth and 20^o pressure angle. The profile needs to be modified to avoid undercutting. Profile modification changes the disposition of the addendum and dedendum with respect to the pitch line. Total depth of 3.213mm is divided in to 2.938mm addendum and only 0.275 dedendum. This pinion is generated by Boolean operations and generation by rolling. This pinion do not have smooth flank surface due to series of Boolean cut operation. So the friction between the rack and bush needs to be incorporated in form of some function at the interface of two entities.

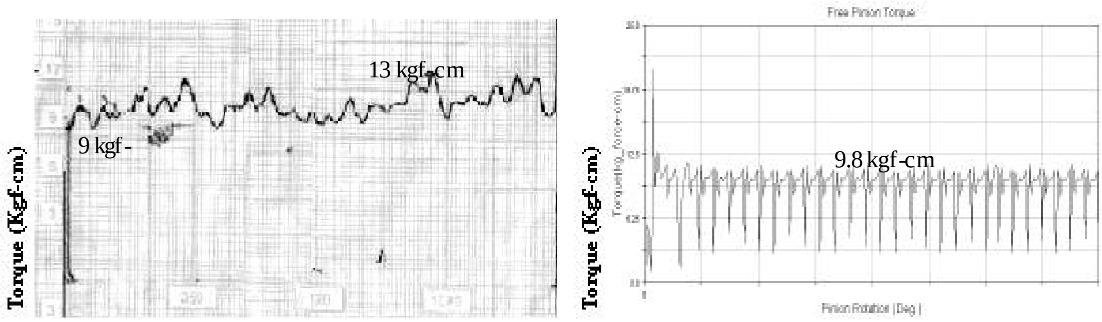
Rack: Rack with teeth inclined at 15^o is modeled. The material properties and weight of the rack is specified.

Bush: Flexibility of bush being important to predict the rack motion, it is made up of 25 discrete elements as a flexible body. Material properties are adjusted accordingly. The bush has a cylindrical joint with rack. Rack is not constrained to translate in horizontal direction. It moves in slightly downward direction when the bush deflects.

Plunger: Plunger has a prismatic joint with yoke nut, but kept apart by spring preload of 30 kgf. This spring preload helps to maintain contact between the rack and pinion continuously and also provides damping. Vertical movement of rack produces equal retaining force due to compression of the spring.

STEERING GEARBOX

Contact between steering rack and pinion is free of play and even internal damping is maintained with the help of yoke preload. Backlash may be due to departure in tooth thickness because of tolerance and allowances, tooth profile error, PCD run out, etc. The backlash is eliminated due to preload on the rack, thus maintains a constant dual flank contact. Baxter et al [1] has calculated the mesh friction and mechanical efficiency of the Rack and Pinion steering design considering a dual contact. In his model frictional torque rises with contact depth. Here the total contact force is the summation of frictional forces and normal forces at the point of contact on both the flanks. It is assumed that rack is not allowed to roll around its axis.

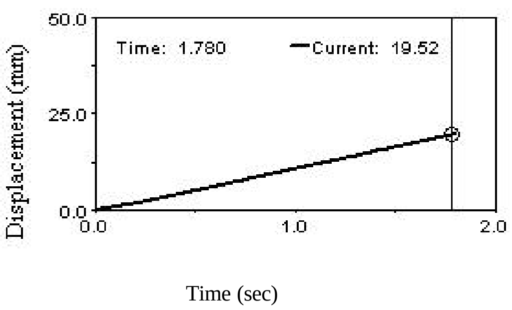


Pinion Rotation **Pinion Rotation**

(a) Actual FPT plot **(b) FPT plot generated in ADAMS**

Figure 3. Free Pinion Torque (FPT) plot for Rack and Pinion Steering Gear

On the contrary Bishop [2] has shown that rolling moment on the rack localizes the loads on the ends of the teeth. And to overcome this problem, the rack was forged in to 'Y' section to eliminate rolling of the rack. The coefficient of friction between the rack and pinion is considered to be constant. But at slow speeds (below 1500mm/min pitch line velocity) friction of approach appears to be double that of recess [3]. A momentary jump in the friction load when the contact passes through pitch point is observed. So some average values of two coefficients of friction need to be used for approach and recess. At the pitch point the sliding reduces to zero. Pure rolling occurs at pitch point corresponding to higher contact stresses and greater loss [4]. This gives non-uniform nature of frictional torque at the interface. Figure 2 shows the frictional torque on the pinion as seen in the ADAMS model. Contact ratio also plays a major role in the torsional stiffness of the gearing. Contact ratio is a measure of percentage of time two meshes share the load. Mesh stiffness variation, the change in stiffness of meshing teeth as the number of teeth in contact changes, causes instability and vibrations in the gear system. This mesh stiffness variation directly affects tooth deflections and transmission error. In case of helical gear this becomes little more complicated as total contact of a helical gear mesh is some combination of the profile and face contact ratio, where face contact ratio is the ratio of face width to axial pitch [5]. The sliding is maximum at engagement and disengagement. Sliding reduces to zero at pitch point where pure rolling occurs. The variation in friction along the line of contact and change in mesh stiffness due to contact ratio can thus generate friction excitations in errorless gears also [6][7]. Profile modification applied to the gear can substantially affect the noise and vibration in the gear [8]. The rack and bush form a cylindrical joint at one end of the gear case housing (Figure 1). This bush being flexible deforms easily and allows the rack to hang freely as a cantilever beam. This hanging motion is controlled at the other end by plunger (Figure 1). The bush is modeled as flexible body to reproduce the complex behavior of the joint. The rack swivels about the bush with each successive tooth engagement and disengagement, due to normal load on the teeth. Figure 4 and 5 shows the linear and vertical displacements of rack respectively.



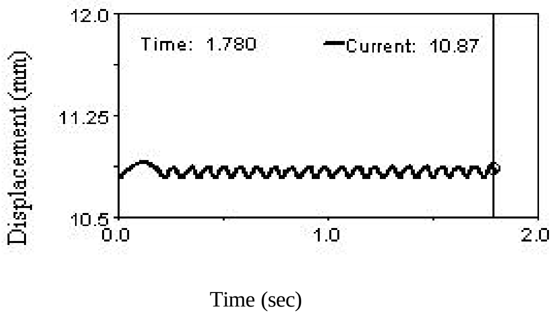


Figure 4. Linear displacement of rack

Figure 5. Vertical displacement of Rack

The yoke clearance of about 70 microns, as indicated in Figure 1, puts a limit to the vertical travel of the plunger. Thus any vertical movement of the rack is going to generate equal restoring force. This force variation is the cause of variation in normal load on the rack and pinion mesh. Friction varies with normal load at the mesh. [9] Subsequently the normal load changes are the causes of the variation in the frictional torque on the pinion. If the errors accumulate in the assembly, then the rack may be displaced to such an extent that it consumes all the yoke clearance. Where the plunger touches the yoke nut, generating very high forces for any further downward movement of rack. Vertical displacement of the plunger is shown in Figure 6.

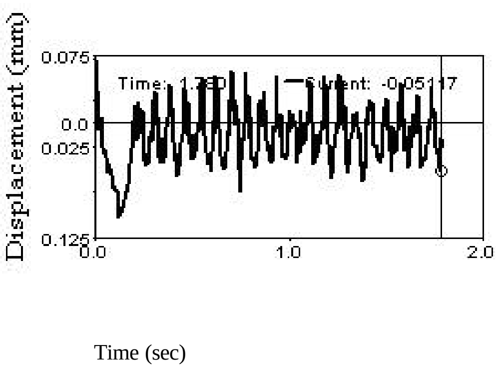


Figure 6. Plunger displacement due to yoke clearance

CONCLUSION

Free Pinion Torque is considered as representative measure of the performance of the steering system. This torque characteristic is the cumulative effect of various tolerances and errors in gearing. Contribution of each parameter to the total torque is important. Knowledge of these parameter will help us to release some of the unnecessary tight tolerances or apply stringent tolerances wherever required. But identifying those critical parameters physically is tedious. The virtual prototype can be easily used to verify effect of the change of various parameters. As an example, if the rack rolling is to be eliminated then it can be done by just changing the cylindrical joint to prismatic one, which allows only translation and no rotation, keeping all other constraint same. But practically it will require hours of input to change the rack cross section to verify the same effect.

First, the confidence level is being built by validating the model with results from actual laboratory tests. Next, any conceptual modification will be tested virtually with much lesser cost and effort.

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