Effects of Gear Manufacturing Errors on Rack and Pinion Steering Meshing.

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Abstract

In the present paper the effects of rack manufacturing errors on the meshing of crossed helical rack and pinion are analyzed. The study is performed using multibody simulations that include the effects of the forces acting at the gear mesh to determine the influence of rack geometrical errors on operating center distance; results are in good agreement with a simplified analytical formulation.

1. Introduction

The steering system is a group of parts that transmit the movement of the steering wheel to the front, and sometimes the rear, wheels. The purpose of the steering system is to allow the driver to guide the vehicle and provide a feedback through the steering wheel of road surface conditions. Several types of steering systems are used on modern cars, based on different types of gearing arrangements in the steering gearbox. The most common gear mechanisms used in automobile power steering are rack and pinion and recirculating ball (worm and roller, worm and nut, warm and sector) type arrangement. Many innovations in steering system are available today to improve the steering confort level and give the best possible directional stability to the cars, some of the improvements include Active Kinematics Control (AKC®) and steer-by-wire.

1.1 Rack and Pinion Steering system

The crossed helical rack and pinion drive is a common component in automotive steering system technologies. This system converts the rotational motion of the steering wheel, connected with a pinion gear, into the linear motion of the rack; this motion applies steering torque to the front wheels via tie rods and a short lever arm (steering arm), as shown in Figure 1.

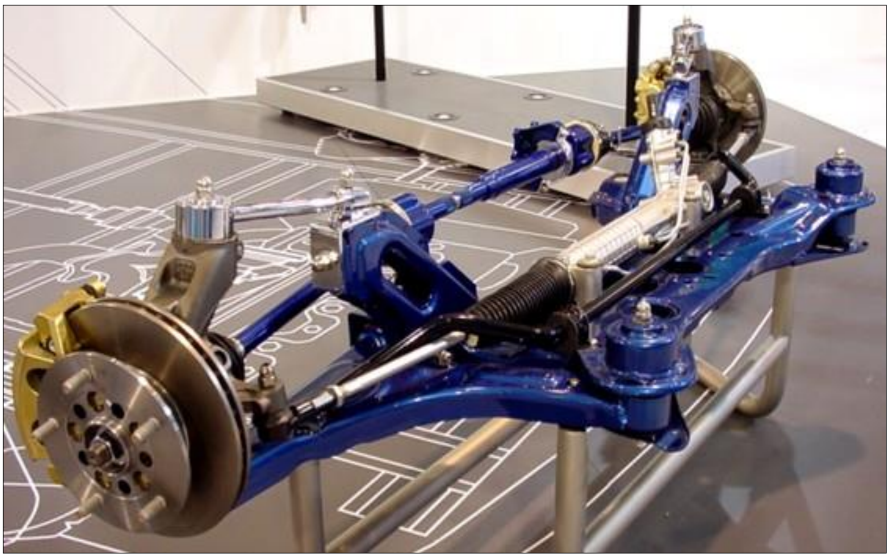


Figure 1. Rack and Pinion steering system

A typical rack and pinion system is shown in Figure 2: the pinion gear teeth are in mesh with the rack bar teeth; the movement of the rack bar relative to the housing is supported and guided by the yoke sliding bearing and a bushing.

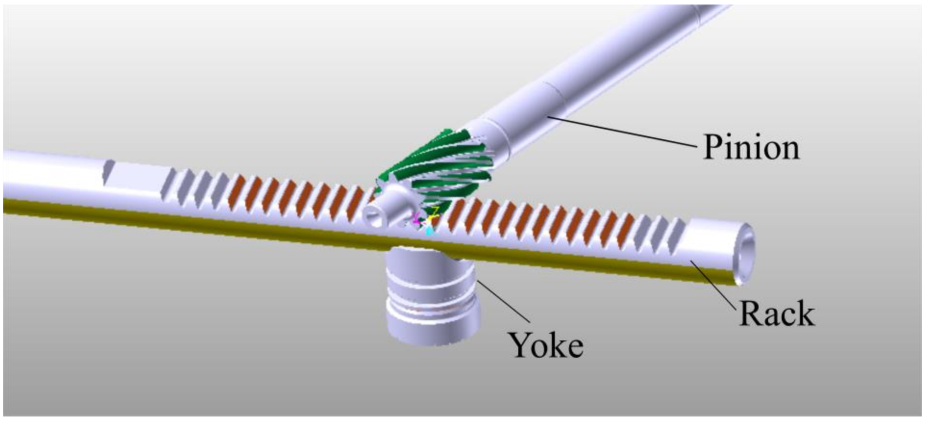


Figure 2. Rack and Pinion assembly

The yoke bearing preloads the pinion against the rack by a spring to eliminate backlash, compensate for initial geometrical imperfections, and recover wear over usage
Helical gear mesh misaligns have been studied by Houser[1] and Reiter[2,3]; no prior work in literature is focused to steering rack and pinion mesh misalignments. To overcome this void in literature, kinematic effects of rack manufacturing errors on the meshing of crossed helical rack and pinion with double flank contact are studied in this paper. The study is performed by means of a multibody model, developed in Recurdyn, that includes the effects of the forces acting at the gear mesh to determine the influence of rack geometrical errors on center distance variation; results are in good agreement with a simplified analytical formulation.

2. Manufacturing gear errors

Gear characteristic dimensions are inspected by means of a computer controlled coordinate measuring machine (CMM); measurements are defined by mechanical probes attached to the moving axis of the machine. Several types of gear deviations are provided in ANSI/AGMA ISO 1328-1/2[4,5], or equivalently DIN 3962 Part 1/2/3.
In the following, the geometrical errors of rack and pinion gears are shortly listed and the contribution of each error on operating center distance is analytically extimated.

2.1 Rack manufacturing errors

2.1.1 Over roller error
The rack is measured by means of a spherical probe placed successively in the tooth spaces. In measuring a rack, the pin is ideally tangent with the tooth flank at the pitch line; a variation of the over pin measure is thus related to a pitch line shift.
In the case of a helical rack, normal module m_n, and normal pressure angle α_n, the ideal pin diameter is given by:

$$d_p' = \frac{\pi m_n - s'}{\cos(\alpha_n)} \quad (1)$$

where s' is the rack tooth thickness (at pitch line). The ideal over pin measurement is given by:

$$M = K - \frac{\pi m_n - s'}{2\tan(\alpha_n)} + \frac{d_p}{2}\left(1 + \frac{1}{\sin(\alpha_n)}\right) \quad (2)$$

where d_p is the rounded value of calculated pin diameter; relevant dimensions are shown in Figure 3.

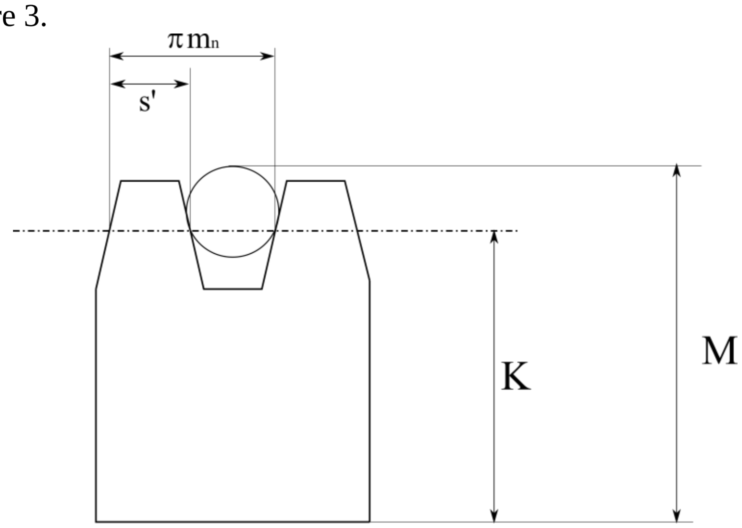


Figure 3. Over Pin measurement for a rack using a pin or a ball

An over roller error thus changes the center distance (CD) of:

$$\Delta CD_{RackOverRoller} = M_{measured} - M_{nom} \quad (4)$$

Overoller measurements performed with rollers of different diameter (e.g. $D_{roller,2}$ and $D_{roller,1}$), differ of a constant quantity, ΔM, calculated as follows:

$$\Delta M = M_2 - M_1 = \frac{D_{roller2} - D_{roller1}}{2}\left(1 + \frac{1}{\sin(\alpha_n)}\right) \qquad (5)$$

2.1.2 Pitch Error

The axial pitch is measured inserting sequentially the pin in successive tooth spaces and calculating the axial distance between the two adjacent measured values; a deviation from the nominal pitch value is thus related to a variation of tooth width.

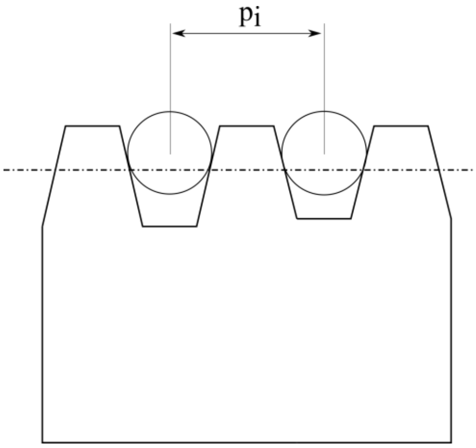


Figure 4. Pitch measurement for a rack using a pin or a ball

A variation of pitch Δp_i, affects the operating center distance of:

$$\Delta CD_{PitchError} = \frac{|\Delta p_i|}{2\tan(\alpha_n)} \qquad (4)$$

2.1.3 Lead Angle Error

In Figure 5 a section normal to rack axis is shown; the nominal lead angle is denoted by β_{rack}, the rack facewidth by *k*. A lead angle deviation $\Delta\beta$, affects the operating center distance by:

$$\Delta CD_{LeadError} = Y = \frac{k\tan(\Delta\beta)}{2\tan(\alpha_n)} \qquad (5)$$

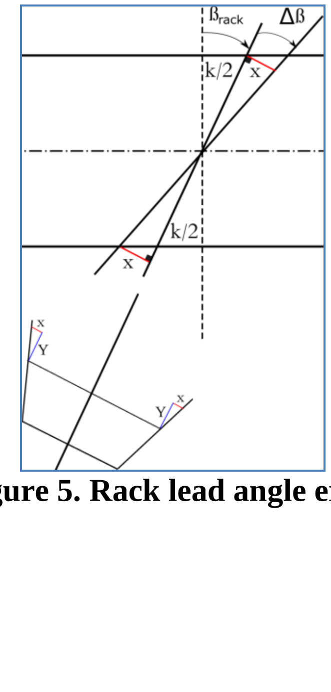


Figure 5. Rack lead angle error

2.2 Pinion manufacturing errors

2.2.1 Radial Measurement Error

The individual radial measurement, r_i, is defined as the radial distance from the gear axis to the centre or other defined location of a probe (ball or cylinder), which is placed successively in each tooth space. During each check, the probe contacts both the right and left flanks at approximately midtooth-depth. Since the individual radial measurement r_i is by definition a radial distance, it directly influences the operating center distance measure:

$$\Delta CD_{RadialError} = r_{i,measured} - r_{i,nom} \quad \text{..(4)}$$

2.2.2 Pitch deviation error

The individual single pitch deviation $f_{pi,}$ is defined as the algebraic difference between the actual pitch and the corresponding theoretical pitch in the transverse plane on the measurement circle of the gear. A variation of pitch Δf_{pi}, affects the operating center distance of:

$$\Delta CD_{PitchError} = \frac{|\Delta f_{pi}|}{2\tan(\alpha_n)} \quad \text{..(4)}$$

3. Effects of rack manufacturing errors on meshing stability

The effects of rack geometrical errors on meshing stability are investigated by means of a multibody model. The contact forces acting at the gear mesh are calculated to determine the influence of each error on center distance; results are compared with the analytical formulation presented in Sect. 2.

3.1 Multibody model

The rack and pinion model used for simulation of rack geometrical errors is shown in Figure 6.

3.1.1 Kinematic constraints

In real rack and pinion steering gear (RPS) the rack is not completely constrained to move axially, it is supported by a cylindrical joint with a flexible bush at one end and by a spring preloaded at the other end; this arrangement compensates slight misalignments of rack axis. In practice, the pinion is assembled within the housing with needle bearings, thus having fixed axis of revolution.

Since the present study is focused on understanding the impact of rack errors on meshing quality, the proposed kinematic model is slightly different from the real RPS.

The rack is supported by an ideal cylindrical joint, to which is imposed a translational motion with constant velocity; the joint is assumed to be frictionless. The pinion is constrained with a revolute joint to a massless dummy body, in turn constrained with a prismatic joint to ground (Figure 6).

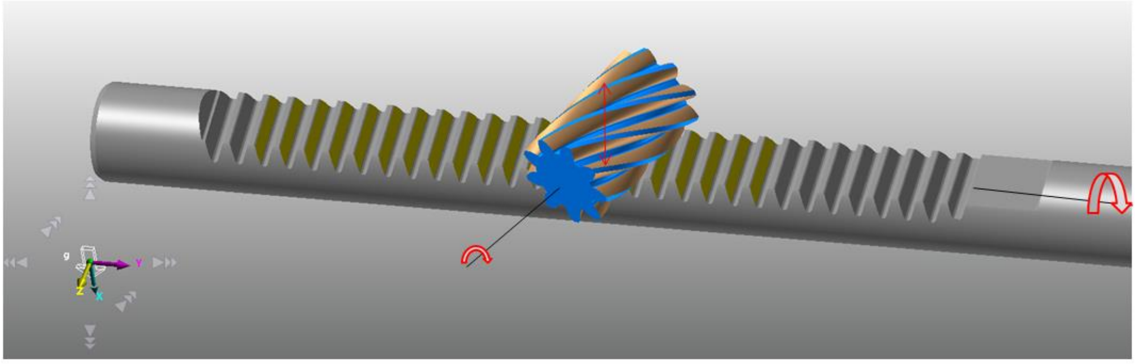


Figure 6. Multibody kinematic model

3.1.2 Compliant Contact Force Model

A robust contact algorithm for a compliant contact force model between bodies of complex geometry is implemented in the software Recurdyn [6,7,8]. A surface-to-surface contact problem can be replaced by multiple sphere-to-surface contact problem: the boundary of the pinion teeth is thus represented by a set of spheres, and the flanks of rack are approximated by triangular patches, as shown in Figure 7. The normal contact force is calculated as a function of the penetration, δ, of the pinion tooth flank nodes into the rack flank patch as follows:

$$f_n = k\delta^a + c\frac{\dot{\delta}}{|\dot{\delta}|}|\dot{\delta}|^b\,\delta^c \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

where k and c are the spring and damping coefficients, which are determined by an experimental method, and $\dot{\delta}$ is the time differentiation of δ. The exponents *a* and *b*, generate a non-linear contact force and the exponent *c*, yields and indentation damping effect. When the penetration is very small, the contact force may be negative due to a negative damping force; this situation can be overcome by using the indentation exponent greater than one.
The contact friction force is obtained by:

$$f_f = \mu|f_n| \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

where μ is the friction coefficient, and its sign and magnitude can be determined from the relative velocity of the pair on contact position.

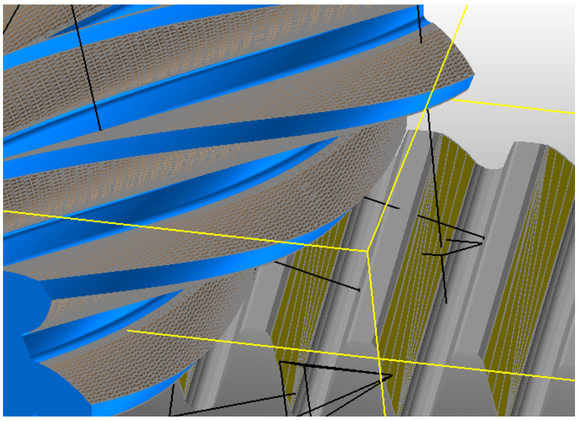


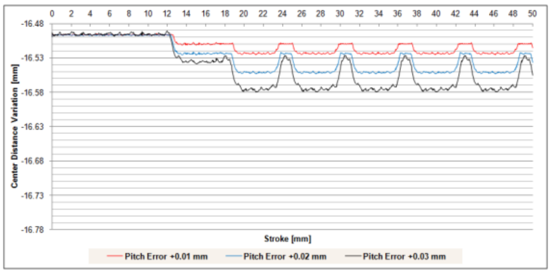
Figure 7. Discretization of rack and pinion surfaces

3.2 Parallel misalignment (Center distance change)

In the following, the contribution of rack manufacturing errors (Sect. 2.1), on the operating center distance is numerically determined; results are compared with the analytical formulas, showing a good agreement. The simulation is performed meshing the pinion with a parametric rack, composed of two parts: the former with nominal parameters, the latter with the desired errors. Results are shown in Tables 1,2,3 respectively for pitch, lead angle and overoller error.

Table 1. Rack Pitch Error and Center Distance Variation

Δp_i [mm]	**Analytical ΔCD [mm]**	**Multibody ΔCD [mm]**
+0.01	0.014	0.017
+0.02	0.028	0.040
+0.03	0.041	0.059



Δp_i [mm]	**Analytical ΔCD [mm]**	**Multibody ΔCD [mm]**
-0.01	0.014	0.015
-0.02	0.028	0.034
-0.03	0.041	0.056

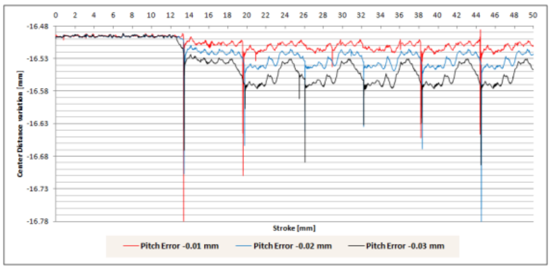
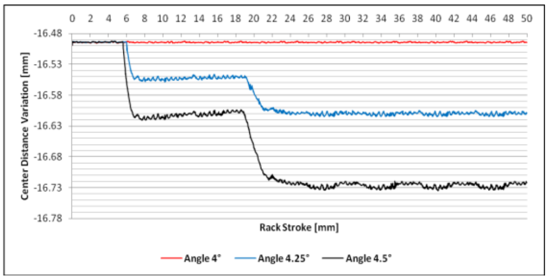


Table 2. Rack Lead Angle Error and Center Distance Variation

$\Delta\beta$ [mm]	**Analytical ΔCD [mm]**	**Multibody ΔCD [mm]**
0	0	0
+ 0.25°	0.119	0.115
+ 0.5°	0.237	0.231



$\Delta\beta$ [mm]	**Analytical ΔCD [mm]**	**Multibody ΔCD [mm]**
0	0	0
-0.25°	0.119	0.106
-0.5°	0.237	0.219

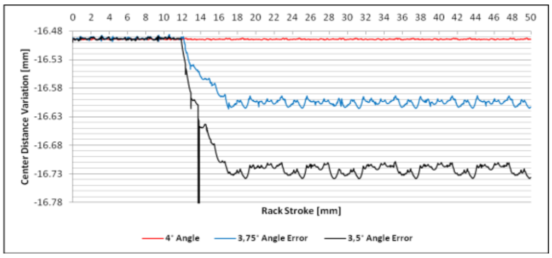
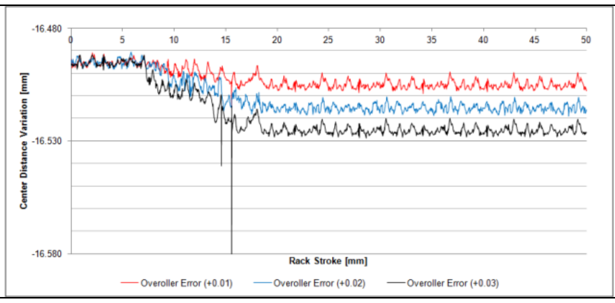
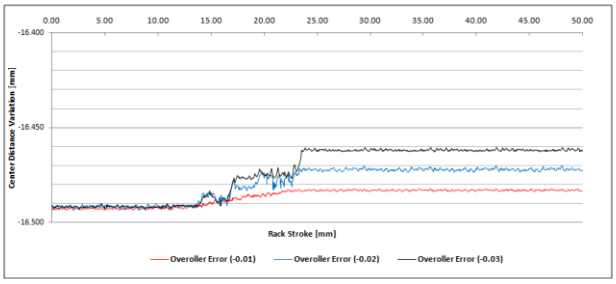


Table 3. Rack Over Roller Error and Center Distance Variation

ΔM [mm]	Analytical ΔCD [mm]	Multibody ΔCD [mm]
+ 0.01	+0.010	+0.010
+ 0.02	+0.020	+0.020
+0.03	+0.030	+0.030



ΔM [mm]	Analytical ΔCD [mm]	Multibody ΔCD [mm]
- 0.01	-0.010	-0.010
- 0.02	-0.020	-0.020
-0.03	-0.030	-0.030



4. Conclusions

In the present paper the effects of rack manufacturing errors on the meshing of crossed helical rack and pinion has been investigated. The contribution of each error to the center distance variation has been determined by analytical formulation and numerically validated by multibody simulation of meshing.

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