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Original Research Article

Generation and analysis of gear drive with tubular tooth surfaces having double contact points

Dong Liang [a,b,*], Bingkui Chen [a], Chengli Hua [c], Ruijin Liao [b]

[a] State Key Laboratory of Mechanical Transmission, Chongqing University, No. 174 Shazheng Street, 400030 Chongqing, China

[b] College of Electrical Engineering, Chongqing University, No. 174 Shazheng Street, 400030 Chongqing, China

[c] City College of Science and Technology, Chongqing University, No. 368 Guangcai Street, 402167 Yongchuan Chongqing, China

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ABSTRACT

The gear drive with tubular tooth surfaces having double contact points is proposed. The generation of gear pair is investigated based on two mismatched rack cutters. The basic tooth profiles of rack cutters for generating the pinion and the gear are designed, respectively. Mathematical models of tooth surfaces are derived according to the geometric relationships. Solid models of gear pair are also established. Stress analysis of tubular tooth surfaces having double contact points is carried out using the finite element method. The results show that the proposed gear drive has the better transmission characteristics. Gear prototype is finished with CNC machining technology. The further studies on the dynamic characteristics and experimental test of this gear drive will be carried out.

1. Introduction

It is well known that gears are widely used in industry for power transmissions. Many scholars have developed various concepts, design and analysis approaches toward the high-performance gear drive. Chen et al. [1] presented a complete method to design tooth profile and its backlash assessment in harmonic drive to enable more teeth to participate in meshing. The parametric equations were expressed for all segments of the double-circular-arc common-tangent tooth profile (DCTP). Fuentes et al. [2] proposed two different versions of geometry of circular-arc curvilinear shaped teeth gears based on generation by face-milling cutters. The computerized processes of virtual generation of both members of the gear set were described and algorithms for simulation of meshing, tooth contact analysis, and finite element analysis were applied. Bahk et al. [3] investigated the impact of tooth profile modification on spur planetary gear vibration. An analytical model was proposed to capture the excitation from tooth profile modifications at the sun–planet and ring–planet meshes. Litvin et al. [4] studied a version of a worm gear drive with improved bearing contact, reduced level of transmission errors and lessened sensitivity to errors of

* Corresponding author at: The State Key Laboratory of Mechanical Transmission, Chongqing University, No. 174 Shazheng Street, 400030 Chongqing, China.

E-mail address: liangdong213@cqu.edu.cn (D. Liang).

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alignment. Unlike the existing approach, the hob and worm tooth surfaces are mismatched (in addition to the oversize of the hob). Bair et al. [5] provided an oval gear drive, which is used in an oil pump with larger pumping volume using a rack cutter with circular-arc teeth. The generated oval gears have convex–concave tooth profile contact. The mathematical models of oval gears with circular-arc teeth were derived based on the theory of gearing and gear generation mechanism. Luo et al. [6] proposed a new styled point-line meshing gear drive characterized by both easy manufacturing and divisibility of involute gears and high strength for contacting between convex and concave tooth profiles of a circular-arc gear. Wang et al. [7] presented a new type of spherical gear drive, which takes micro-segment involute profile as tooth profile. This new type of spherical gear drive has the potential to improve the load capacity and the performance of spherical gear transmission. Chen et al. [8] put forward a new involute-helix gear drive, which is point contact with convex and concave circular-arc tooth profiles. The basic principle characterized by the advantages of involute and circular-arc gears was provided.

Conjugate-curve gear transmission, a new type of spatial gear drive based on the theory of conjugate curves had been studied by authors [9–13]. During our researches, through a given contact curve in a plane or space, we can obtain its conjugated curve along the designated contact direction. And the engagement tooth surfaces are generated in terms of this developed conjugate-curve pair.

As a type of tooth surfaces of conjugate-curve gear drive, theoretical and experimental investigations on tooth surfaces with single contact point were provided [14]. Generally speaking, the double contact points have larger contact area and better mesh characteristics compared with the single contact point. A novel conjugate-curve gear drive with tubular tooth surfaces having double contact points is proposed. The circular-arc and parabola curves are designed as the basic tooth profiles of rack cutters, respectively, for generating the pinion and gear. Mathematical model of gear pair is established and a computerized simulation of the meshing process is conducted. Mechanics properties of gear pair are discussed, and the contact and bending stresses are analyzed. The gears are processed with milling method to achieve manufacturing precision utilizing the CNC technology.

2. Tooth profiles of rack cutters

Two rack cutters with mismatched surfaces are used separately for generating the pinion and the gear.

2.1. *Circular arc tooth profile of rack cutter for generating the pinion*

The normal tooth profile Σ_a of rack cutter for generating the pinion is shown in Fig. 1. The edge I is an arc with radius ρ_{ea} and it is used to form the root filet of the pinion. Edge II generating the working region of the pinion is a circular arc with radius ρ_a. The addendum and dedendum of normal tooth profile of rack cutter are h_{a1} and h_{f1}, respectively. h_1 represents the tooth

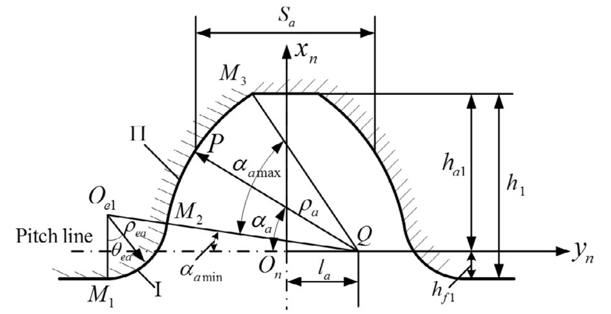

Fig. 1 – Normal tooth profile of rack cutter for generating the pinion.

depth and S_a is space width. Coordinate system $S_n(O_n - x_n, y_n, z_n)$ fixed to the rack cutter is defined and origin O_n located in the center.

Vector $\mathbf{r}_{an}^{(I)}$ of edge I in S_n is

$$\mathbf{r}_{an}^{(I)} = \begin{bmatrix} -\rho_{ea}\cos\theta_{ea} + (\rho_a + \rho_{ea})\sin\alpha_{amin} \\ \pm[\rho_{ea}\sin\theta_{ea} - (\rho_a + \rho_{ea})\cos\alpha_{amin} + l_a] \\ 0 \\ 1 \end{bmatrix} \tag{1}$$

Vector $\mathbf{r}_{an}^{(II)}$ of working edge II in S_n is

$$\mathbf{r}_{an}^{(II)} = \begin{bmatrix} \rho_a\sin\alpha_a \\ \mp(\rho_a\cos\alpha_a - l_a) \\ 0 \\ 1 \end{bmatrix} \tag{2}$$

where θ_{ea} is the location parameter of point on edge I. l_a is the distance from point Q to axis x_n. α_{amin} and α_{amax} are separately the minimum and maximum values of parameter angle α_a. The upper "+" represents the left-side normal section, while the lower "−" indicates the right-side normal section.

The transverse section of rack cutter can be determined utilizing the coordinate transformation as follows.

$$\mathbf{r}_{ap}^{(i)} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos\beta & \sin\beta & u_a\sin\beta \\ 0 & -\sin\beta & \cos\beta & u_a\cos\beta \\ 0 & 0 & 0 & 1 \end{bmatrix} \mathbf{r}_{an}^{(i)} (i = I, II) \tag{3}$$

where u_a is the distance between the two coordinate systems origins. β is the helix angle.

2.2. *Parabola tooth profile of rack cutter for generating the gear*

Fig. 2 shows the normal tooth profile Σ_f of rack cutter for generating the gear. It includes three parts: circular arc edge I, parabolic curve edge II, and chamfer edge III. h_2 is the tooth depth of normal tooth profile of rack cutter. h_{a2} and h_{f2} are the addendum and dedendum of normal tooth profile of rack cutter, respectively. Suppose that P_1 and P_2 are the contact points of tooth profile, the pressure angles of two points are α_{f1} and α_{f2}, respectively. l_f represents the distance from the center point Q to the symmetrical line $K - K$, and $l_f = l_a - j/2$.