Study on Swing Oscillating Gear Transmission with Hypocycloid Shockwave

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ABSTRACT

Transmission principle is the base of design theory. This research presents a new type of oscillating gear transmission—the swing oscillating gear transmission with hypocycloid shockwave. This type of transmission device enables transmission with one teeth difference. The transmission is simple and easy to be optimized. This research also explains transmission principle of swing oscillating gear transmission with hypocycloid shockwave and the calculation of transmission ratio. Tooth profile equation of shock diffuser and center gear is deduced based on speed change and enveloping principle. A design condition that no tooth profile interference exists in central gear lays a foundation for designing this type of transmission device. Computing examples are provided accordingly.

KEYWORDS: cam shockwave; hypocycloid; swing oscillating gear transmission; tooth profile equation; curvature radius.

INTRODUCTION

Oscillating gear transmission with small tooth difference is an active branch in the area of mechanical transmission research. In the transmission system, oscillating gear-group is employed to transmit rotary motion and motive power. It is highly efficient with compact structure, large transmission ratio and high load-carrying capacity. This transmission system has become very popular in the field of planetary gear transmission [1,2]. Two-tooth difference oscillating gear transmission has more promising future than one-tooth difference oscillating gear transmission because of its axisymmetric shockwave generator, small axial dimension and compact structure. Moreover, it has become an attractive topic in transmission research field around the world as it can adopt single-row oscillating teeth and be completely self-balanced [3]. Oscillating gear transmission with small tooth number difference has various structures, such as oscillating gear transmission with cycloid cam[4], oscillating gear transmission with spatial cam[5], rolling oscillating gear transmission with cam shockwave[6,7,8], oscillating gear transmission with two-tooth difference push-rod[9] and swing oscillating gear transmission with two-tooth difference cam shockwave.

Swing oscillating gear transmission is an important type of oscillating gear transmission. A distinct feature of the transmission is that swing-oscillating teeth revolve around evenly-distributed dowel pins in oscillating-tooth carrier. This feature improves transmission efficiency by effectively reducing slippage of oscillating teeth in oscillating slot [9] and sliding wear[3,11,12]. Oscillating gear transmission with two-tooth difference is an advanced version of that with one-tooth difference. It has improved its performance but not the scale of transmission ratio. Oscillating gear transmission with multi-tooth difference is worthy of more attention for its large transmission ratio scale.

Research in the study of reference [15] studied the cosine curve of tooth profile. For pure swing oscillating gear transmission of arbitrary tooth difference in reference [16], equidistant line of cosine curve was the profile of both shockwave gear and solid gear. Isokinetic conjugate transmission of arbitrary tooth difference could be operated in this type of transmission. This design was simple to be optimized. Rolling bearing components were adopted in oscillating tooth to achieve pure rolling contact transmission. Reference [17] was about the matching curve of dual-cosine tooth transmission. Reference [18] focused on the three-shockwave roller tooth transmission, which was actually a special case of cosine shockwave.

Research mentioned in references [16, 17, 18] all paid attention to oscillating planetary transmission based on the rolling structure of cosine shockwave. Reference [19] was a study on push-rod oscillating transmission with high oval tooth profile of arbitrary tooth difference. Tooth profile of high-order polynomial curve of arbitrary push-rod oscillating tooth was examined in reference [20]. Profile of oscillating tooth has great influence on performance of oscillating gear transmission. It is important to conduct research on new type of oscillating gear transmission about tooth profile of inner gear and performance of oscillating planetary transmission. This research focuses on swing oscillating gear planetary transmission with hypocycloid shockwave. Transmission principle and calculation of transmission ratio of swing oscillating gear transmission with hypocycloid shockwave are briefly introduced. Tooth profile equation of shock diffuser and center gear is deduced based on speed change and enveloping principle. A design condition with no tooth profile interference in central gear lays a foundation for designing this type of transmission devices. Moving parts in the new transmission mechanism such as input shaft are self-balanced. On the base, speed variator with arbitrary tooth difference can be realized. This design is specifically suitable for mechanical transmission device with heavy load due to high removability and great transmission torque.

STRUCTURE OF SWING OSCILLATING PLANETARY TRANSMISSION WITH HYPOCYCLOID SHOCKWAVE

Shockwave gear, inner gear and oscillating tooth group share a rotation center. Each one of them can be fixed while other two are acting as input link and output link separately to obtain different speed variation. The principle of swing oscillating gear transmission with hypocycloid cam shockwave can be explained as: shock cam rotates at a uniform speed and generates radial forces under the effect of input drive torque; it promotes oscillating tooth to swing around stud center at the same time; meanwhile, the forces, generated by engagement of swing oscillating gear and working tooth profile of inner gear, drive gear rack into uniform circular motion at a given ratio; the transmission from motion to driving force is realized after the torque is transmitted to output shaft of gear rack; shock cam is driven into constant rotary motion when the input shaft is in uniform rotation. Restricted by shock gear, inner gear and oscillating-tooth carrier, the oscillating gear is in rotary motion at constant speed ratio; rotary motion of oscillating tooth is periodically driven by shock gear; shock gear oscillating gear and inner gear maintain a continuous relationship with each other at a fixed transmission ratio.

The structure of swing oscillating gear transmission of hypocycloid shockwave is shown in Figure 1. It mainly consists of shock diffuser H, oscillating gear G (oscillating gear and gear rack) and central inner gear K. A remarkable feature of swing oscillating gear transmission is that swing oscillating tooth rotates around evenly-distributed studs on gear rack. The transmission efficiency of this device can be significantly enhanced by effectively reducing the slippage of tooth in oscillating-tooth carrier and sliding wear. Shock cam is connected with input shaft through flat keys, with equidistant curve of hypocycloid curve as its profile. Oscillating gear rack consists of output tray and floating tray, which are fixed together by circumferentially-arranged oscillating-tooth pins.

As the swinging locus of oscillating tooth, profile of central inner gear is driven by shock cam, around oscillating gear rack pin. Meanwhile, shock cam and oscillating gear rack draw envelope line of uniform circular motion at a given ratio. Tooth number of central inner tooth and oscillating gear is defined as arbitrary tooth difference. Three-section composite structure is adopted in swing oscillating gear as shown in Figure 2. It consists of oscillating tooth matrixes (labeled as 3 in Figure 3), two needle bearings (labeled as 1) and pinhole (labeled as 2) of eccentric point. The middle section of oscillating tooth is a part of tooth matrix. Its excircle surface is meshed with shock cam. The two needle bearings, as truncated needle bearings without inner ring, are installed on both sides of the middle segment of tooth matrix; its outer surface is meshed with central inner gear. Tooth matrix is eccentrically mounted on gear rack pin roll. Moreover, a shallow annular groove is placed in the center of inner gear to avoid contact friction between excircle surface of middle tooth and inner gear. Meanwhile, the width of shock cam should be smaller than that of middle section of oscillating tooth, thus avoiding interference of shock cam and outer surface of the two needle bearings.

There are several major features of this transmission device: (1) Profile of shock cam is axisymmetric hypocycloid curve. Shockwave motion on oscillating tooth can be realized only by rotating around its geometric center. Self-balance under force of shock cam and central inner gear can be ensured as well. Excitation force and overturning moment will not exist in this transmission device, whether using single or multiple-piece shock cam. Vibration excitation can be avoided from principle. (2) Double needle bearing was planted on matrix, appropriately reducing the friction between oscillating gear, central inner gear and shock cam. Swing tooth is linearly connected with shock cam, enhancing entrainment rate between oscillating gear and cam. The transmission efficiency is effectively improved with simple structure of gear rack, making a more compact structure. (3) The cam doesn’t need to be eccentrically mounted on shaft due to the application of symmetrical structure. Therefore, the angular position of each row of chains can keep consistent when using the multiple-row transmission chains. The chains are composed of shock cam, oscillating gear, gear rack and inner gear. It is convenient for transmission device assembling. Moreover, different from traditional oscillating tooth of one-tooth difference or two-phase cam shockwave of two-tooth difference, the wave number of the cam can be arbitrary tooth difference. Research with broader selection range of transmission ratio is more valuable.

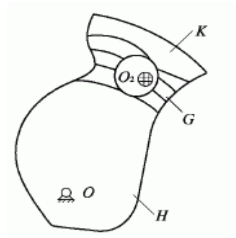

Figure 1: Principle of swing oscillating gear transmission actuated by cam with cosine curve

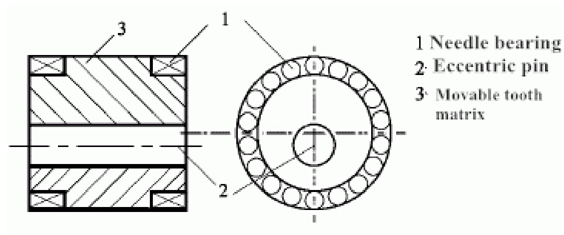

Figure 2: Structure sketch of the swing oscillating tooth

CALCULATION OF TRANSMISSION RATIO

Within the structure of swing oscillating gear transmission with hypocycloid shockwave, if the shockwave gear is fixed, the structure can be seen as a conversion mechanism without planetary gear. A precondition is set that an additional angular velocity ω_H is given. Its magnitude is equal to that of shock gear but in opposite direction to the shockwave oscillating tooth system. Then calculation can be made through the formula of transmission ratio of gear train with fixed axis:

$$i_{GK}^{H} = \frac{\omega_G^H}{\omega_K^H} = \frac{\omega_G - \omega_H}{\omega_K - \omega_H} = \frac{Z_K}{Z_G} \tag{1}$$

From Formula (1), we can obtain:

$$\omega_G = \omega_K i_{GK}^{H} + \omega_H (1 - i_{GK}^{H}) \tag{2}$$

H represents shock gear; G oscillating gear; K inner gear; transmission ratio; the superscript letters indicate corresponding fixing member, while subscript letters the state of corresponding members of left letters with respect to that of the right letters. For example, i_{GK}^{H} is the ratio between relative angular velocities of G and K to that of shock cam H. Superscript letters of ω is corresponding fixing member, while subscript letters is the corresponding members; Z_H the wave number of shock gear; Z_G the tooth number of oscillating gear; Z_K the wave number of inner gear. Transmissions ratios in different installation forms are shown in Table 1.

Table 1: Transmission ratios of different types of installation

Transmission scheme	Transmission ratio	Steeling direction	Tooth number of oscillating gear	Deceleration or acceleration
Fixed inner gear $\omega_K = 0$	$i_{HG}^{K} = \frac{Z_G}{Z_G - Z_K}$	$Z_G > Z_K$ Same direction	$Z_G = Z_K + Z_H$	Deceleration
		$Z_G < Z_K$ Opposite direction	$Z_G = Z_K - Z_H$	Deceleration
	$i_{GH}^{K} = \frac{Z_G - Z_K}{Z_G}$	$Z_G > Z_K$ Same direction	$Z_G = Z_K + Z_H$	Acceleration
		$Z_G < Z_K$ Opposite direction	$Z_G = Z_K - Z_H$	Acceleration
Fixed oscillating gear $\omega_G = 0$	$i_{HK}^{G} = \frac{Z_K}{Z_K - Z_G}$	$Z_G > Z_K$ Opposite direction	$Z_G = Z_K + Z_H$	Deceleration
		$Z_G < Z_K$ Same direction	$Z_G = Z_K - Z_H$	Deceleration
	$i_{KH}^{G} = \frac{Z_K - Z_G}{Z_K}$	$Z_G > Z_K$ Opposite direction	$Z_G = Z_K + Z_H$	Acceleration
		$Z_G < Z_K$ Same direction	$Z_G = Z_K - Z_H$	Acceleration
Fixed shock gear $\omega_H = 0$	$i_{GK}^{H} = \frac{Z_K}{Z_G}$	$Z_G > Z_K$ Same direction	$Z_G = Z_K + Z_H$	Acceleration
		$Z_G < Z_K$ Same direction	$Z_G = Z_K - Z_H$	Deceleration
	$i_{KG}^{H} = \frac{Z_G}{Z_K}$	$Z_G > Z_K$ Same direction	$Z_G = Z_K + Z_H$	Deceleration
		$Z_G < Z_K$ Same direction	$Z_G = Z_K - Z_H$	Acceleration

TOOTH PROFILE EQUATION

In swing oscillating gear transmission with hypocycloid shockwave, the profile of shock cam is equidistant curve of cosine shock curve. Standard cylindrical curve is adopted as profile of swing gear. Profile of central inner gear is swinging locus of gear moving around shock rack pin driven by shock cam. Meanwhile, shock cam and oscillating gear rack draw envelope line of uniform circular motion at a given ratio.

Tooth profile equation of shock gear

Hypocycloid profile equation of shockwave gear [21,22] is:

$$\begin{cases} x_1 = (R - r_0)\cos\theta + nr_0 \cos[(\frac{R}{r_0} - 1)\theta] \\ y_1 = (R - r_0)\sin\theta - nr_0 \sin[(\frac{R}{r_0} - 1)\theta] \end{cases} \tag{3}$$

,

where R is the radius of fixed circle; r_0 the radius of floating circle; $\theta \in [0, 2\pi]$; Z_H shock number of shockwave gear; $R = Z_H r_0$; $n \in [0, 0.27]$.

Actual tooth profile of shock gear is the inner equidistant curve of theoretical profile with offset r_1. Therefore, actual tooth profile equation can be deduced from mechanical principles [23].

$$\begin{cases} x_2 = x_1 - \dfrac{r_1 \frac{dy_1}{d\theta}}{\sqrt{(\frac{dx_1}{d\theta})^2 + (\frac{dy_1}{d\theta})^2}} \\ y_2 = y_1 + \dfrac{r_1 \frac{dx_1}{d\theta}}{\sqrt{(\frac{dx_1}{d\theta})^2 + (\frac{dy_1}{d\theta})^2}} \end{cases} \tag{4}$$

where $\frac{dx_1}{d\theta} = -\sin\theta(R - r_0) - nr_0 \sin[R / r_0 - 1)\theta](R / r_0 - 1)$,

and $\frac{dy_1}{d\theta} = \cos\theta(R - r_0) - nr_0 \cos[R / r_0 - 1)\theta](R / r_0 - 1)$.

Tooth profile equation of inner gear

In Figure 3, XOY is set as the fixed coordinate system connected with annular gear while the origin coordinate O is the geometric center. $X'OY'$ and X_1OY_1 are defined as conjoined coordinate systems for shock cam and oscillating-tooth carrier separately. They share the same origin coordinate with the fixed coordinate system X_2OY_2. In the initial configuration of transmission, coordinate systems XOY, $X'OY'$, X_1OY_1 and X_2OY_2 coincide with each other. Pinhole center O_1 is on axis OX_1, while the gear center O_2 is on axis OX_2. According to transmission principle of swing oscillating tooth with shock cam, the motion locus of oscillating gear center O_2 around coordinate system $X'OY'$ of shock cam is the theoretical profile of shock cam. The inner equidistant curve, with oscillating gear radius r_1 as the offset, is called working profile of shock cam. The motion locus of oscillating gear center O_2 around XOY is the theoretical tooth profile of central inner gear (motion locus of gear center). The working tooth profile of central inner gear is the equidistant curve of theoretical tooth profile with offset radius r_2.

Figure 3 shows that the pinhole center of oscillating gear rack O_1 is on the standard circle with the center of coordinate origin O and radius R_0; the offset of gear is d (distance between O_1 and O_2); the angle between lines connecting two centers (OO_1 and OO_2) with O is β. Preconditions are set as follows: firstly, coordinate system $X'OY'$ turns around angle α ($\angle XOX'$) in fixed coordinate system XOY at any time; then X_1OY_1 turns around angle θ ($\angle XOX_1$); after that, gear center O_2 is driven around angle $\alpha-\theta$ ($\angle XOX_2$) relative to coordinate system XOY. Moreover, α and θ meet the given condition of input-output ratio: $i = i_{HG}^{K} = \dfrac{\alpha}{\theta}$ (i_{HG}^{K} is the transmission ratio). It is can be seen in Figure 3 that the angle between OO_2 (the line from coordinate origin O and oscillating gear center O_2) and coordinate axis OX' is $(\alpha+\beta-\theta)$, while the angle between coordinate system XOY and OX is $(\theta-\beta)$. Thus, the length of OO_2 can be calculated according to Formula (3) as follows:

$$\left|OO_2\right| = \rho = \sqrt{x_{30}^2 + y_{30}^2} \tag{5}$$

In the equation,

$$\begin{cases} x_{30} = (R - r_0)\cos[(i-1)\theta] + nr_0\cos[(\dfrac{R}{r_0}-1)(i-1)\theta] \\ y_{30} = (R - r_0)\sin[(i-1)\theta] - nr_0\sin[(\dfrac{R}{r_0}-1)(i-1)\theta] \end{cases}$$

Then the theoretic tooth profile curve of inner gear in coordinate XOY is:

$$\begin{cases} x_3 = \rho\cos(\theta-\beta) \\ y_3 = \rho\sin(\theta-\beta) \end{cases} \tag{6}$$

In the triangle OO_1O_2, $\left|OO_2\right| = \rho$, $\left|O_1O_2\right| = d$, and $\left|OO_1\right| = R$. According to the cosine law of triangle,

$$d^2 = R^2 + \rho^2 - 2R\rho\cos\beta \tag{7}$$

When θ is known, β and ρ can be calculated from Formulas (5) and (7) using Function fsolve.m of MATLAB platform. Meanwhile, $\dfrac{dx_3}{d\theta}$ and $\dfrac{dy_3}{d\theta}$ in Formula (6) can be deduced through numerical differentiation.

Actual tooth profile of inner gear is the equidistant curve of theoretical tooth profile with offset r_2. Therefore, based on mechanical principle, the actual tooth profile equation of inner gear is as follows [23].

$$\begin{cases} x_4 = x_3 + \dfrac{r_2 \dfrac{dy_3}{d\theta}}{\sqrt{(\dfrac{dx_3}{d\theta})^2 + (\dfrac{dy_3}{d\theta})^2}} \\ y_4 = y_3 - \dfrac{r_2 \dfrac{dx_3}{d\theta}}{\sqrt{(\dfrac{dx_3}{d\theta})^2 + (\dfrac{dy_3}{d\theta})^2}} \end{cases} \tag{9}$$

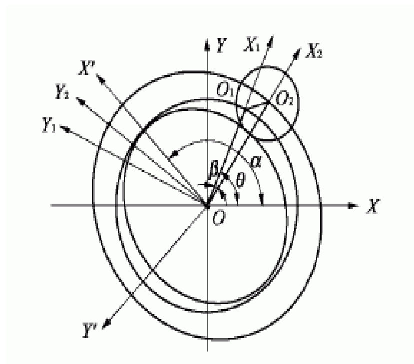

Figure 3: Generation principle sketch of tooth profile of inner gear

CURVATURE RADIUS OF INNER GEAR TOOTH PROFILE

Curvature of certain point of central gear tooth profile shows the curvature degree of tooth profile curve around this point. It describes the geometric feature of tooth profile curve, which is an important parameter in analyzing the loading capacity and lubrication condition of oscillating gear transmission. Curvature of central gear profile can be deduced by differential geometry through the equation below:

$$Kr = \frac{\dot{x}_3 \ddot{y}_3 - \ddot{x}_3 \dot{y}_3}{(\dot{x}_3^2 + \dot{y}_3^2)^{\frac{3}{2}}}$$

Therefore, curvature radius of central gear profile is

$$\rho = \frac{1}{Kr} = \frac{(\dot{x}_3^2 + \dot{y}_3^2)^{\frac{3}{2}}}{\dot{x}_3 \ddot{y}_3 - \ddot{x}_3 \dot{y}_3} \tag{8}$$

Actual curvature radius of central gear is $\rho_w = |\rho| \pm r_2$. Curvature of concave section around tooth roots of central gear is positive (+), while that of convex section is negative (-).

COMPUTING INSTANCES

Hypocycloid shockwave parameters are given as: $Z_H = 3$; R=120mm; r0=40mm; n=0.1; radius of oscillating-tooth carrier R0=80; tooth offset d=10mm; inner isometric radius of shockwave gear r01=10mm; outer isometric radius of inner gear r02=10mm; reduction ratio i=3; wave number of shockwave gear zh=3; tooth number of inner gear zk=6; Geneva numerate of planetary carrier zg=9; inner gear is fixed.

Theoretic and actual tooth profiles of shockwave gear are calculated by MATLAB programming, which are shown in Figure 4. In Figure 4, the inner curve refers to actual tooth profile while the outer curve is theoretic tooth profile. Theoretic and actual tooth profiles of inner gear are presented in Figure 5. In the figure, the outer curve represents actual tooth profile when the inner curve is theoretic tooth profile.

Curvature radius of theoretic tooth profile of inner gear of convex section near tooth top is 29.1967mm, while that of actual tooth profile is 19.1967mm. The $\rho - \theta$ curve of theoretic profile of oscillating gear is shown in Figure 6.

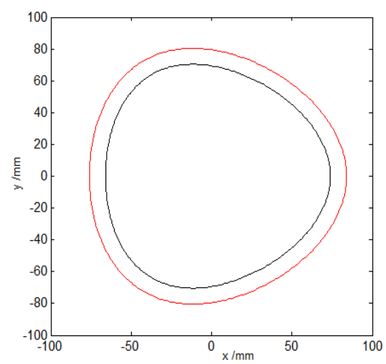

Figure 4: Tooth profile curve of shock cam

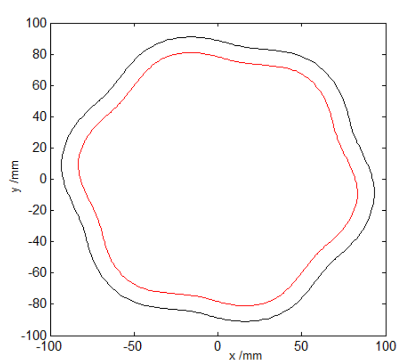

Figure 5: Tooth profile curve of inner gear

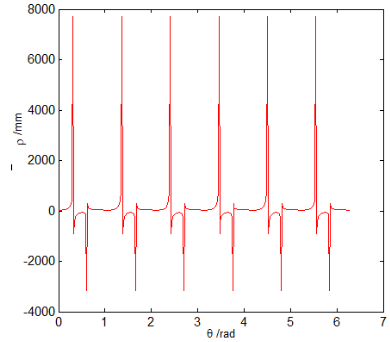

Figure 6: Theoretic tooth profile curve $\rho - \theta$ of oscillating gear

CONCLUSIONS

In swing oscillating shockwave transmission with arbitrary tooth difference, standard cylindrical curve is adopted as the equidistant line of cosine curve in shockwave gear and oscillating tooth profile. Profile of central tooth gear is swinging locus of tooth moving around shock rack pin under the drive of shock cam. Meanwhile, shock cam and oscillating gear rack draw envelope line of uniform circular motion at a given ratio. This type of transmission enables isokinetic conjugation transmission with arbitrary tooth differences. The design of this transmission is simple and easy to be optimized. Transmission principle of swing oscillating gear transmission with hypocycloid shockwave and the calculation of transmission ratio are also explained in this research. Tooth profile of shock diffuser and center gear is deduced based on speed change and enveloping principle. We obtain a design condition that there should be no tooth profile interference in center gear, laying the foundation for designing this type of transmission devices. There are also computing examples related to the design. A solid model of oscillating tooth with swing shockwave can be constructed by programming calculation with Matlab program and Solidworks parametric modelling. The application of motion simulation of Solidworks enables transmission to perform well without interference. Through this process, the deduction above is proved to be theoretically correct. Formulas used in the deduction are also suitable for swing oscillating gear transmission with arbitrary hypocycloid shockwave.

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REFERENCES

1 Qu, Jifang (1993) "*Theory of Oscillating Gear Transmission*". China Machine Press, Beijing, 1-10.

2 Wang Shuying and Xu Liju (2006) "Research on Analysis of Force and Contact Strength for Translation Oscillating Gear Drives". Journal of Sichuan University: Engineering Science Edition, 35(6):82-85.

3 Liang Shangming, Zhang Jie, Zhou Rongliang, (2012) "Virtual Assembly and Kinematics Simulation of Two-tooth Difference Swing Oscillating Gear Transmission". Coal Mine Machinery, 33(1):40-42

4 Chen Bingkui and Li Zhaoyang (2003) "Study on the Tooth Profile of Cycloid Cam Sliding Tooth Gearing". Journal of Mechanical Transmission, 27(3):14-17.

5 Chen Bingkui, Mou Huan, Li Zhaoyang (2004) "New Spatial Cam Sliding Tooth Planetary Gearing". Journal of Mechanical Engineering, 40(1):29-32.

6 Li Jianfeng, He Aiying, Dong Xinrui, (2011) "Geometrical Design of the Oscillating Rolling Gear Transmission with Cam Actuating". Journal of Mechanical Engineering, 47(1) : 24-30 .

7 Li Jianfeng and Dong Xinrui (2008) "Method of Meshing Force Analysis for the Oscillating Rolling Gear Transmission with Cam Actuating". Journal of Mechanical Engineering, 44(5):39-44.

8 Li Jianfeng and Dong Xinrui (2008) "Inner Gear Profile Modificationm Research on Cam Profile Rolling Oscillating Gear Transmission". Journal of Beijing University of Technology, 34(10):1009-1014.

9 Li Jianfeng, Chen Xing, He Aiying, (2012) "Tooth Profile Analysis and Geometrical Design of the Two-tooth Difference Pushrod Oscillating Gear Transmission". Journal of Beijing University of Technology, 38(7):967-972.

10 Qiao Hui, Li Jianfeng, Chen Xin, (2012) "The Equations of Tooth Profile and Geometry Analysis of Two-Tooth Difference Swing Oscillating Gear Transmission". Machinery Design & Manufacture, (11):40-42..

11 Li Jianfeng, Dong Xinrui and Zhou Liyan (2008) "Tooth Profile and Meshing Force Analysis of the Swing Oscillating Gear Transmission Actuated by Cam". Journal of Beijing University of Technology, 34(1):7-13.

12 Dong Xinrui, Li Jianfeng, Sui Yan, (2007) "Structure and Profile Analysis of Cam Disk Profile Swing Oscillating Gear Transmission". Journal of Mechanical Transmission, 31(3):1-6.

13 Liang Shangming, Zhou Rongliang and Zhang Jie (2011) "Study on the Strength of Components of Two-tooth Difference Swing Oscillating Gear Transmission". Journal of Sichuan University (Engineering Science Edition), 43(2):207-210.

14 Liang Shangming,Huang Bo,Zhang Jie, Yao Jin (2012) "Study on Meshing Efficiency of the Swing Oscillating Gear Transmission Actuated by Cam". Journal of Sichuan University (Engineering Science Edition), 44(1):192-195.

15 Shanming Luo, Yue Wu and Jian Wang (2008) "The Generation Principle and Mathematical Models of a Novel Cosine Gear Drive". Mechanism and Machine Theory, 43:1543-1556.

16 Huang Jingzhi, Cheng Shigan and Chen Xianxiang (2008) "Oscillating Gear Transmission of Pure Rolling with Arbitrary Tooth-difference". Machine Design and Research, 24(4):44-46.

17 Huang Jinzhi, Chen Xianxiang and Cheng Shigan (2008) "Study on the Meshing Curve of Dual-cosine Oscillating Gear Transmission". Journal of Mechanical Transmission, 32(4):6-7.

18 Huang Nan,Tao Dongcai,Li Ming, (2013) “Research of Oscillating Rolling Gear Planetary Transmission Theory with Three Actuating Wave”. Journal of Mechanical Transmission, 37(3):5-7,23.

19 Youxin Luo (2013) “Study on the Principle of Higher Order Elliptic Rolling Oscillating Gear Planetary Transmission”. Wulfenia Journal, 20(9):169-181

20 XuanYali, Liu Pengpeng, An Zijun and Qu Jifang (2014) “Research on High-order Polynomial Curve Tooth Profile of Oscillating Gear with Arbitrary Tooth Difference Push-rod”. Journal of Mechanical Transmission, 38(4):1-4

21 Liu Xiling and Liu Xianghong (1998) “Planar Motion of Hypocycloid”. Journal of Changsha Railway University, (3):75-79.

22 Sun Ke and Bu Rui (2014) “The Application of Hypocycloid in Keyless Connection” . Mechanical Engineer, (2):14-15.

23 Sun Hen, Chen Zuomo and Ge Wenjie (2006) “*Theory of Machines and Mechanisms*”. Higher Education Press, Beijing.