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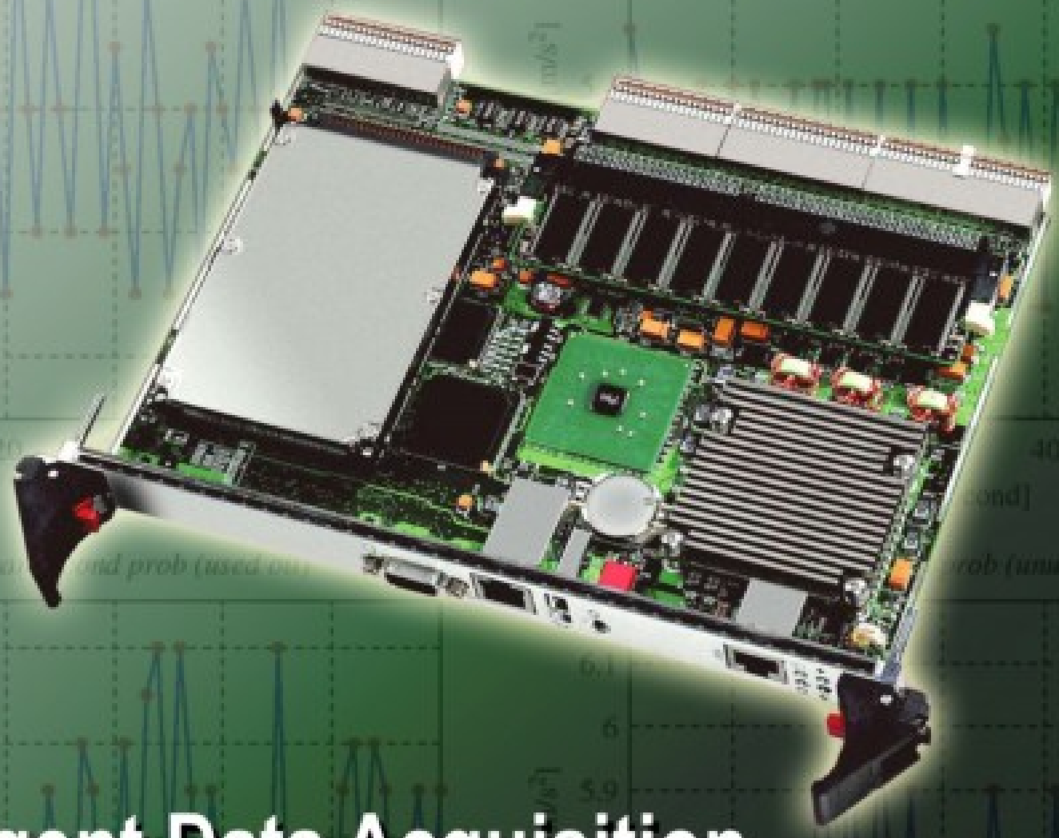

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Analysis on Sliding Velocity of Cycloid Gear Pair during Meshing

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Abstract: The sliding loss is caused by the relative sliding velocity vectors at contacting points. During meshing of cycloid gear pair, all teeth are in contacting, and half of the teeth in meshing with the cycloid gear. The sliding velocity vectors at all contacting points in cycloid gear pair are described, and the sliding velocity's variation of cycloid tooth pair during meshing is further developed. Compared with the general cycloid gear pair, sliding velocity vectors at contacting points of cycloid gear pair with the pin gear's tooth in different structures such as with rollers or half-embedded are developed. The structure of pins with rollers can reduce the sliding loss, and the sliding loss is directly influenced by the proportion of the pin to the roller. But half-embedded tooth structure wouldn't increase sliding loss with larger loads transferred. *Copyright © 2012 IFSA.*

Keywords: Sliding velocity, Cycloid gear drive, Rollers, Half-embedded.

1. Introduction

Cycloid drive has become more and more popular gear reducer, due to their large transmission ratio, compact size, large load capacity, and high efficiency. And in recent years, because of their advantages, especially for half of the teeth meshed simultaneously in theory and the outstanding averaging effect of errors, more and more new cycloid gear drives are designed for precision transmission or hard capacity in cases such as robot, aerospace and mining industry area [1-2].

The researchers have been done a lot of research about cycloid drives, including structure innovation design, dynamic analysis, meshing efficiency analysis and so on. Neagoe et al. proposed a new variant of a cycloid planetary transmission with modified structure, which is able to accomplish high kinematical ratio and high efficiency, and useful to fit mechatronic systems of Renewable Energy System [3]. Gorla et al. proposed a new cycloid gear reducer, which has an external ring gear and engaged with the planet wheel by means of cylindrical rollers, and also made a theoretical and experimental investigation on this cycloid speed reducer [4]. Chen et al. proposed a new conic cycloid gear reducer, which has a cycloid gear pair with conic-arc-inner gear as the pin gear and a conic cycloid gear, and can realize zero clearance by adjusting the relative position of the conic cycloid gear pair [5]. Shan et al. set up an analytic mathematic model of a double-crank ring-plate-typed pin-cycloid gear planetary drive [6]. Mirko Blagojević et al. developed analysis on the dynamic behavior of cycloid drive [7]. W. J. Xu et al. studied the power flow of cycloid gear train, and developed the control methodology for cycloid gear used in precision transmission [8]. And some researchers established the equation of cycloid gear tooth profile, analyzed the meshing characteristics in detail according to gear geometry, and also made geometric design of cycloid gear [9-11].

The meshing components of a typical cycloid drive has a planet cycloid gear, the profile of which is the inner offset of an epitrochoid, meshing with cylindrical pins mounted on the gear case. All teeth of cycloid gear pair are in contacting and half of the teeth in meshing with the cycloid gear during meshing. In some other cycloid gear drives, pins are designed with rollers for decreasing sliding loss, and some are designed as half-embedded or integral structure for high capacity. In all these cycloid gear pairs, sliding loss of the tooth surface at contacting points will occur from beginning to the end unavoidably, and the sliding loss ratio is directly influenced by the sliding velocity vector. With this reason, sliding velocity of cycloid gear pair during meshing is focused in this paper.

2. Sliding Velocity of Cycloid Gear Pair

Sliding loss of cycloid gear pair is caused by relative slip at contact points of the pin gear and the cycloid gear during meshing. The sliding of the tooth profile surface during meshing is represented in Fig. 1. In case that all cylindrical pins housing to the case along the distribution circle form the pin gear O_b, the pitch circle of the cycloid gear O_g will be pure rolling along the pitch circle of the pin gear O_b, and the eccentric distance of the cycloid gear and the pin gear is e. At this moment, the pitch point P is the absolute instantaneous center between the cycloid gear and the pin gear, all the teeth are in contacting, and the i^{th} tooth pair is contacting at point M_i. Because of the pin gear is fixed, so the velocities at contacting points of cycloid gear are the relative velocities. Form Fig. 1, the sliding velocity vectors at contacting points is different. The direction of velocity V_i is perpendicular to the line of the instantaneous center point P connecting to the contacting point M_i, so the velocity component in direction n-n is zero, which makes the cycloid gear contact with the pin gear throughout the meshing period. The sliding velocity V_i at contacting point M_i in t-t direction is equal to the velocity at point M_i of cycloid gear, and the value is directly proportional to the distance from point P to point M_i, which makes sliding occurred between the meshing tooth pair.

3. Sliding Velocity of Cycloid Tooth Pair during Meshing

For cycloid gear pair, sliding velocity vectors at contact point of one tooth pair is different with other tooth pair's. When sliding of one same tooth pair is focused, sliding velocity vectors is found to be varying with meshing position during meshing, shown in Fig. 2.

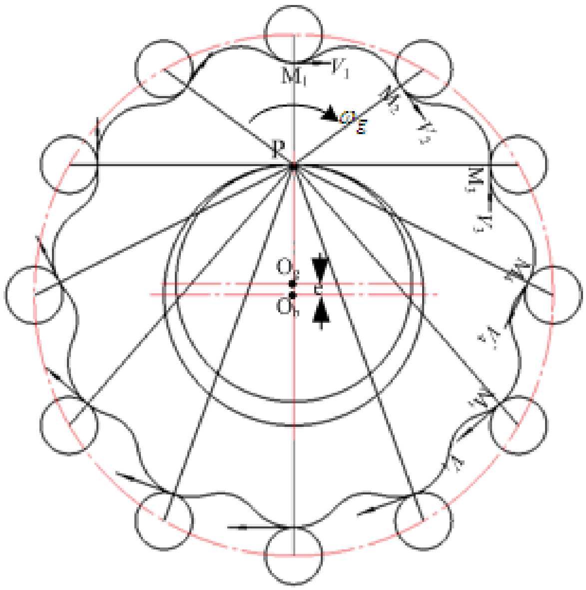
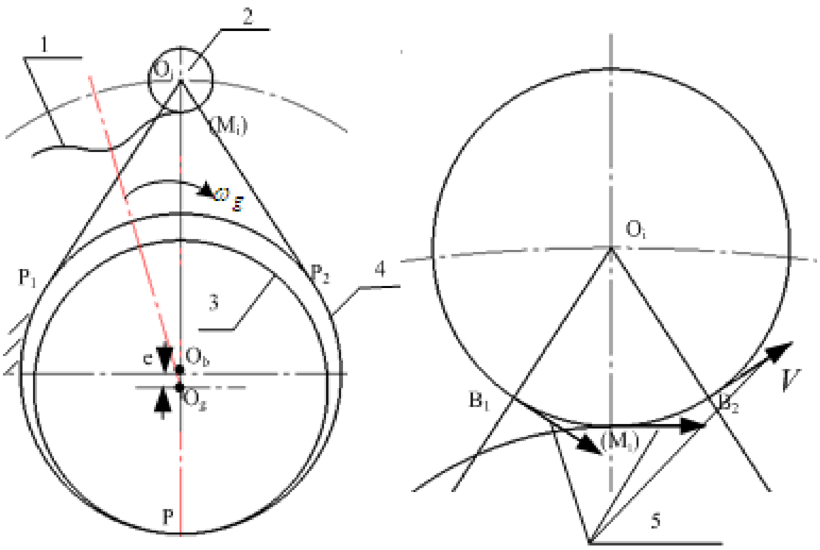


Fig. 1. The sliding of tooth surface during meshing in cycloid gear pair.



1- the cycloid gear; 2- the i th tooth of the pin gear; 3- the pitch circle of cycloid gear; 4- the pitch circle of pin gear; 5-velocity vectors at different contacting points.

Fig. 2. The velocity vector at contacting point of one cycloid tooth pair varying with meshing position changed during meshing.

For the i tooth of the pin gear O_b meshing with the cycloid gear O_g, the sliding velocity vector is always tangent to the circle of the i tooth O_i of the pin gear, and the value is determined by

$$V_i(\theta_{bi}) = \left(\overline{PO_i} - \overline{O_iM_i}\right)\omega_g = \left(\sqrt{R_z^{\,2}\sin^2(\theta_{bi}) + (R_z\cos(\theta_{bi}) - r_b)^2} - r_z\right)\omega_g \tag{1}$$

$$(0 + \frac{2i\pi}{z_b} \le \theta_{bi} \le 2\pi + \frac{2i\pi}{z_b}),$$

where ω_g is the angular velocity of the cycloid gear; Rz is the tooth distributing circle's radius of the pin gear; r_b is the pitch circle radius of cycloid gear; r_z is the radius of the pin tooth; θ_{bi} is the angle between $\overline{PO_i}$ and $\overline{O_b O_i}$, i is the tooth number of pin gear ($i \le z_b$).

4. Sliding Velocity of Cycloid Gear Pair with the Pin Tooth in Different Structures

4.1. Tooth Structures of the Pin Gear

In general cycloid drive, for transferring load, pin points usually can be designed as cylinders. But in some cycloid drive, cylindrical pins shown in Fig. 3 are further designed with rollers to decrease friction, some shown in Fig. 4 are designed with half-embedded cylindrical pins for a higher load-carrying capacity, and others are designed with an integral structure for higher strength and reducing installation difficulty, shown in Fig. 5.

The tooth of the pin gear designed with roller or half-embedded can rolling during meshing, the sliding velocity at contacting point may change, which may further affect the sliding loss of cycloid gear pair. With this reason, the variation of the sliding velocity with the tooth of the pin gear in different tooth structures (pins with rollers and half-embedded pins) is developed in detail in this paper.

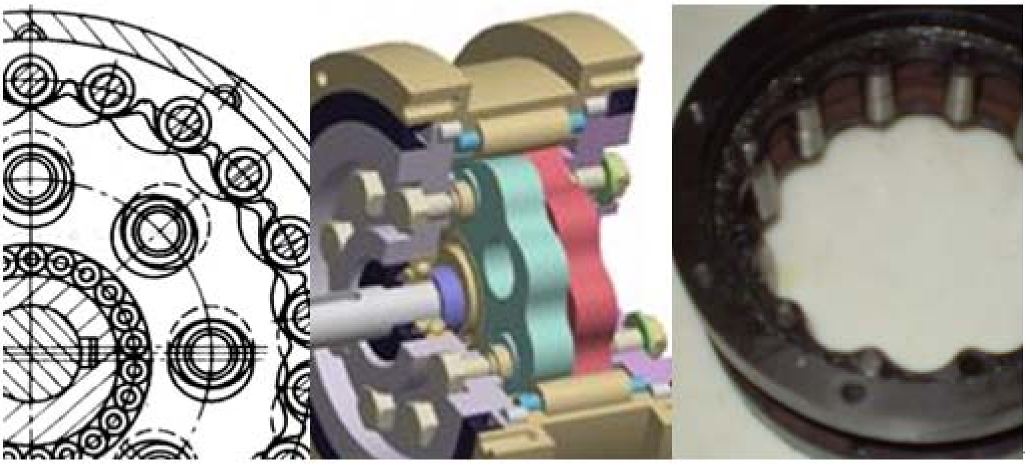


Fig. 3. The teeth of the pin gear designed as cylindrical pins with rollers.

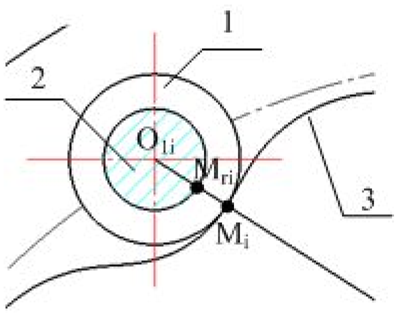
Fig. 4. The teeth of the pin gear designed as cylinders in half-embedded.

Fig. 5. The teeth of the pin gear designed as an integral structure.

4.2. The Sliding Velocity Vector at Contacting Points in Different Tooth Structure

For the i th cycloid tooth pair, if the strucutre of the pin gear's tooth is desigend as cylindrical pins with rollers, the contacting condition in detail is shown in Fig. 6. Fig. 6 shows that the roller is not only contacting with the profile surface of cycloid gear, but also is contacting with the pin. And the contacting condition of cycloid tooth pair with the pin gear's tooth in half-embedded structure, shown in Fig. 7. From this figure, we can find that the pin is in contact with both the profile surface of the cycloid gear and the cylindrical groove surface of the pin gear. In fact, many factors such as manufaturing errors or assembly errors of these parts will affect the sliding between meshing tooth pair. In order to facilitate research, the following assumptions are made.

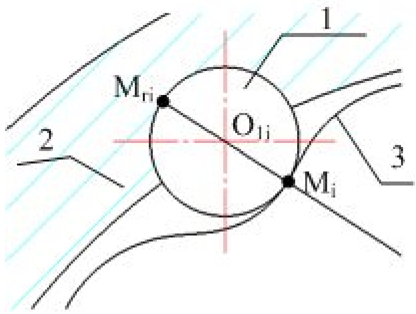
1. No sliding between pins and rollers during meshing;
2. No sliding between pins and semi-cylindrical groove surfaces on the pin gear in half-embededd structure;
3. Both rolling and sliding occur between meshing tooth profile surfaces.



1-the roller of the i th tooth of the pin gear;
2- the pin of the i th tooth of the pin gear; 3- the cycloid gear.

Fig. 6. The teeth of the pin gear designed with rollers.

Fig. 6 shows the i th tooth of the pin gear meshed with the cycloid gear, the roller of the i th tooth is in contact with the cycloid gear at point M_i, and is in contact with the pin of the tooth at point M_{ri}. In this case the pin mounted on the pin gear and the pin gear fixed, pure rolling at contacting point M_{ri} considered, the following equation is obtained.



1- the i th tooth of the pin gear; 2- the pin gear; 3- the cycloid gear.

Fig. 7. The teeth of the pin gear designed with half-embedded structure.

$$v_{1M_{ri}} = \omega_p r_p + v_{O_{1i}} = v_{2M_{ri}} = 0\,, \tag{2}$$

where $v_{1M_{ri}}$ is the velocity of roller 1 at point M_{ri}; $v_{2M_{ri}}$ is the velocity of pin 2 at point M_{ri}; ω_p is the angular velocity of roller 1 rotating around the center O_{1i}; $v_{O_{1i}}$ is the velocity of the pin at the center O_{1i}; r_p is the pin's radius of the i th tooth of the pin gear.

Both rolling and sliding at contacting point M_i considered, the value of the relative sliding velocity $v_{gM_ibM_i}$ at point M_i during meshing can be determined by

$$\left|v_{gM_ibM_i}\right| = \left|\omega_p r_w - \omega_g \overline{PM}_i\right| + v_{O_{1i}} \tag{3}$$

where r_w is the roller's radius of the i th tooth of the pin gear.

Because of the firiction force as the only driving force of the roller, the relative velocity always should be the mimimum, so the following equation is obtained.

$$\omega_g \overline{PM_i} = \omega_p r_w \tag{4}$$

Combining and rearranging equations (2), (3) and (4), the relative velocity $v_{gM_ibM_i}$ at contacting point M_i can be determined by

$$v_{gM_ibM_i} = -\omega_p r_p = -\omega_g \overline{PM}_i \frac{r_p}{r_w} = -v_{gM_i} \frac{r_p}{r_w} \tag{5}$$

From the equation, when the pin's radius of the i th tooth is close to zero, the relative velocity $v_{gM_ibM_i}$ at the contacting point M_i is close to zero, pure rolling between meshing tooth profile will occur. On the contrary, when the radius of the pin is near to the roller's, the relative velocity $v_{gM_ibM_i}$ is close to the velocity v_{gM_i} at point M_i of the cycloid gear, which is same the case with no rollers, only sliding and no rolling will occur. In other words, that is, sliding reaches the maximum.

For the tooth structure designed as the pin with roller, the sliding loss decreases with reducing the radius of the pin. But the pin is transferring loads, so the radius of the pin can't be too small. The pin's radius can be determined to as small as possible under strength condition satisfied.

Similar to the structure of pins with rollers, the tooth of the pin gear designed as half-embedded shown in Fig. 7, pure rolling at point M_{ri} assumed, both rolling and sliding occur at point M_i between meshing tooth profile. The following equation can be obtained.

$$v_{1M_{ri}} = \omega_p r_z + v_{O1_i} = v_{2M_{ri}} = 0 \tag{6}$$

Here, for the relative velocity be the mimimum, equation (4) will be

$$\omega_g \overline{PM}_i = \omega_p r_z \tag{7}$$

Substituting equations (6) and (7) into equation (3), and rearranging, the relative velocity $v_{gM_ibM_i}$ at contacting point M_i is determined by

$$v_{gM_ibM_i} = v_{O_{1i}} = -\omega_g \overline{PM}_i = -v_{gM_i} \tag{8}$$

And from above equation, half embedded pins designed for a higher load-carrying capacity will not reduce or increase the sliding loss, but sliding will be in the opposite direction.

5. Conclusions

Sliding velocity vectors at contacting points of cycloid gear pair and sliding velocity's variation of cycloid tooth pair during meshing are separately developed in this paper. Several common structures of the pin gear's tooth and the sliding velocity vectors at contacting points in these structures are further studied.

1) The sliding velocity at contacting points of cycloid gear pair is proportional to the distance from the instantaneous center point to the contacting point.

2) The sliding velocity vector is always tangent to the circle of the pin's tooth, and the value varies with the instantaneous center point moving along the pitch circle of the pin gear.

3) The sliding of the tooth surface during meshing will be decreased in the structure with the pin gear's tooth designed as pins with rollers. And the sliding decreases with reducing the pin's radius. Half-embedded pins designed for a higher load-carrying capacity will not decrease or increase the sliding loss. The direction of sliding velocity vectors of above two tooth structures will be in the opposite direction.

Acknowledgements

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