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Analysis of Lubrication of Rolling Movable Tooth Reducer for Polar Deep Ice Drilling

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Abstract. Aiming at the wear of rolling movable teeth reducer used in polar deep ice drilling, this paper studies the wear law of tooth profile and analyses the sliding rate of rolling movable teeth transmission. The study of sliding rate can effectively provide the basis for the design of rolling movable teeth reducer, and improve the wear resistance, transmission efficiency and service life of reducer. In this paper, according to the extreme cold environment in the polar region, the type of lubricating oil is selected so as to maintain a certain oil film thickness on the tooth surface, which is an important method to improve the service life of the reducer.

1. Introduction

Rolling movable teeth reducer is used in polar deep ice drilling because of its compact construction, light weight, small volume, wide transmission ratio, high transmission efficiency, multi-teeth meshing, strong bearing capacity, long life, and high torque. During the working process of the reducer, the wear of the tooth surface has a great influence on the transmission accuracy, efficiency, fatigue life and dynamic characteristics of the reducer. The failure modes of rolling movable teeth reducer are tooth surface wear, pitting and cementing. Lubrication failure is the main cause of tooth surface cementation. Correct lubrication method can reduce the wear between the teeth, reduce friction power loss and prolong their service life. If lubrication is not proper, premature failure will occur. Through the analysis of sliding rate of movable teeth transmission and the selection of lubricating oil, the premature failure of reducer can be effectively avoided, the service life can be prolonged, and the bearing capacity, transmission efficiency and anti-gluing ability of gear can be improved. Therefore, lubrication research has practical guiding significance for improving the working condition of reducer for polar deep ice drilling.

2. Structure of the Rolling Movable Tooth Reducer

It was shown by Terada [1] that a rolling movable teeth transmission device is composed of three basic components: center wheel, eccentric wheels, and rollers. As shown in Figure 1, the reduction transmission device of an ice drill consists of an input shaft, an eccentric wheel, an eccentric disc, rollers, and a center wheel. The input shaft is equipped with an eccentric section, i.e., an eccentric wheel in the eccentricity of the input shaft segment. The inner surface of the eccentric disc is connected by an eccentric wheel and eccentric wheel bearing connection. The outer surface of the eccentric disc rolls as it is connected with the rollers. The rotating input shaft is connected to the intermediate bearing and tooth rack. The rollers are arranged inside the groove of the tooth rack, and the outer surfaces of the rollers are connected with the enveloping surface of the inner surface of the center wheel [2].



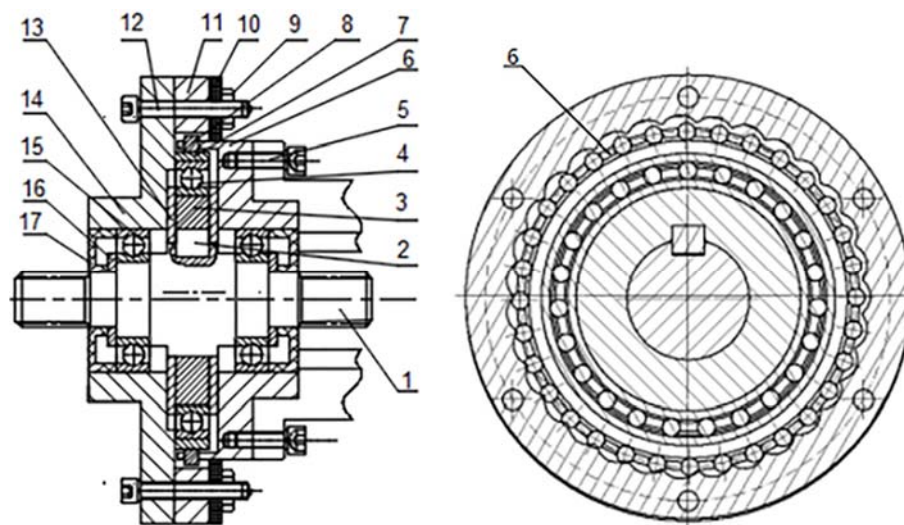


Figure 1. The structure of rolling movable teeth transmission.

(1) Input shaft; (2) key; (3) eccentric wheel; (4) eccentric bearing; (5) socket head cap screw; (6) tooth rack; (7) eccentric disc; (8) rolling movable teeth; (9) nut; (10) seal ring; (11) center wheel; (12) bolt; (13) shaft check ring; (14) left end cap; (15) bearing; (16) shaft check ring; (17) oil seal.

3. Sliding Rate of Rolling Movable Tooth Drive

In the process of transmitting motion and power, the relative sliding between two high pair elements of conjugate tooth profile is caused by the different tangent velocity of contact point. In the sliding contact process, the conjugate tooth profile produces friction and wear, which destroys the correct tooth profile, shortens the service life of the rolling movable tooth reducer, causes transmission instability, reduces transmission efficiency and increases noise. The transmission accuracy, efficiency, fatigue life and dynamic characteristics of the reducer will be affected by a series of adverse effects. The wear law of tooth profile depends to a certain extent on the change rate of relative sliding velocity of conjugate tooth profile at each contact point. The sliding rate U indicates the characteristics of harmful sliding action between two conjugate tooth profiles. Therefore, in order to improve the wear resistance, transmission efficiency and service life of the reducer, it is very important to study the sliding rate U in the design of the rolling movable teeth reducer.

The sliding rate U of conjugate teeth is defined as the ratio of the relative tangential velocity between two conjugate teeth at the contact point B of conjugate teeth to the tangential velocity at that point [3]. In general, the movement of movable teeth transmission is more complex, which can be rolled, slid or sliding and rolling. From the analysis of movable teeth motion state, it can be seen that in the rolling movable teeth transmission, it is assumed that the movable teeth and the eccentric wheel are rolling purely and there is no friction between the rolling movable teeth and the movable teeth rack. There are both relative rolling and sliding at the contact between the rolling movable teeth and the working tooth surface of the center wheel, and the wear between the center wheel and the movable teeth is the most serious. When the gear transmission is running under the condition of no lubrication or insufficient lubrication, the more serious the relative sliding between conjugate teeth is, the worse the wear of the teeth surface is. For better lubricated gear transmission, there is an appropriate relative sliding speed between conjugate teeth, which is beneficial to the formation of oil film. Therefore, in order to analyse the sliding rate of rolling movable teeth reducer, only qualitative analysis of the relative sliding situation between rolling movable teeth and center wheel, as well as improvement methods are needed. The sliding rate of rolling movable teeth U_G and sliding rate of center wheel U_K are as follows

$$\begin{cases} U_G = (v_{BK}^t - v_{BG}^t) / v_{BG}^t \\ U_K = (v_{BG}^t - v_{BK}^t) / i \cdot v_{BK}^t \end{cases} \quad (1)$$

Where v_{BG}^t is the tangential velocity of the rolling movable teeth at the contact point K, m/s; where v_{BK}^t is the tangential velocity of the center wheel at contact point K, m/s; $i = z_G / (z_G - z_K)$, i is the transmission ratio of the reducer; z_K is the number of teeth of the center wheel; z_G is the number of teeth of the rolling movable teeth.

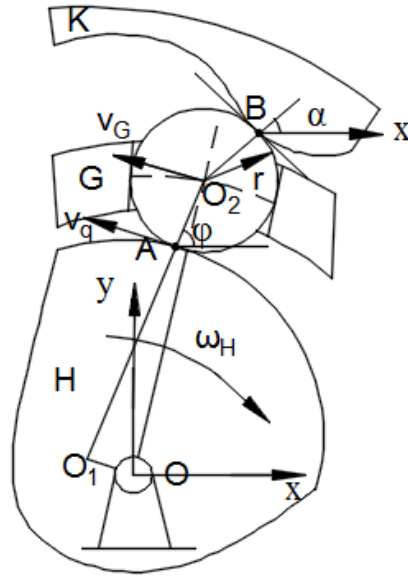


Figure 2. The relative motion sketch of the transmission component of the rolling movable teeth reducer.

The relative motion sketch of the transmission component of the rolling movable teeth reducer is shown in Fig. 2. The meshing pair of the transmission component of the rolling movable teeth reducer is composed of the rolling movable teeth, the eccentric wheel H, and the center wheel K and the tooth track G. The rolling movable teeth and the high pair of the center wheel form conjugate teeth, the rolling movable teeth and the high pair of the eccentric wheel form conditional motion pairs, and the rolling movable teeth and the movable teeth structure form rotational pairs. According to the hypothesis and mechanics [4, 5], the motion of the movable teeth in plane motion relative to the working surface of the center wheel can be decomposed into the translation of point B at the contact point between the movable teeth and the center wheel and the rotation of the movable teeth relative to point B of the center wheel. The V_G in Figure 2 is the speed of the tooth track, and the V_q is the speed of the movable teeth relative to the tooth track. If the component of V_G along the tangent line of the center wheel profile at point B is in the positive direction, then there is

$$\begin{cases} v_{BK}^t = V_G^t = 2l_{OO_2} \omega_H \cdot \cos(\varphi - \alpha) / z_G \\ v_{BG}^t = -V_q^t = -l_{OA} \omega_H \end{cases} \quad (2)$$

Where V_G^t is the tangential velocity of the tooth track along point B, m/s; V_q^t is the tangential velocity of the movable teeth relative to the tooth track along point B, m/s; ω_H is the angular velocity of the

eccentric wheel, rad/s; l_{OO_2} , l_{OA} , α , φ , from the following center wheel profile curve equation and geometric relationship. l_{OO_2} is the distance from the eccentric point O to the geometric center O_2 of the rolling movable teeth, mm; l_{OA} is the distance from the eccentric point O to the tangent point A of the movable teeth and the eccentric wheel, mm; α is the angle at which the tooth track rotates, rad; φ is the angle at which the eccentric wheel rotates, rad; Formula (2) is substituted into Formula (3).

$$\begin{cases} U_G = l_{OO_2} \cdot \cos(\varphi - \alpha) / i \cdot l_{OA} + 1 \\ U_K = -l_{OA} / l_{OO_2} \cdot \cos(\varphi - \alpha) - i \end{cases} \quad (3)$$

The rectangular coordinate system OXY are selected, as shown in Figure 3. The eccentric wheel is rotated clockwise around eccentricity point O, and the rollers are rotated counter-clockwise in the cage. In the rolling process of the rollers, the rollers are always tangent to the eccentric wheel. The distance between geometric centers O_1 and O_2 of the eccentric wheel and rolling, respectively, remains unchanged. The distance between geometric center O_2 of the rolling and eccentricity point O changes, and the center wheel, which is always in contact with the rolling, is fixed. Therefore, the shape of the center wheel profile is the outer envelope curve formed by the moving track of the rollers along O_1' . Therefore, for the center of the tooth profile curve equation, the curve equation is computed firstly out rolling trajectory [6].

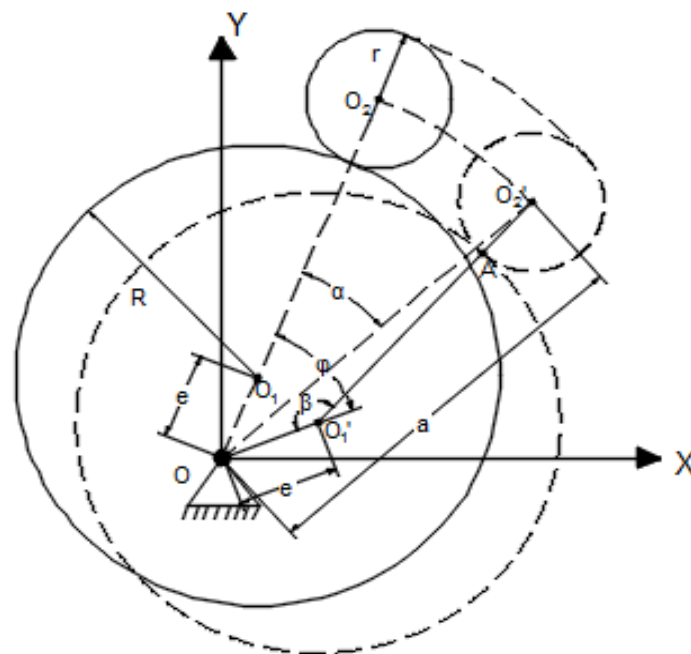


Figure 3. Motion diagram of the equivalent mechanism of movable tooth transmission.

Known formula

$$i = \varphi / \alpha = Z_G / (Z_G - Z_K) = Z_G \quad (4)$$

The geometric relationship in Figure 3 can be obtained as,

$$(R + r)^2 = e^2 + l_{OO_2}^2 - 2 \cdot e \cdot l_{OO_2} \cos(\varphi - \alpha) \quad (5)$$

$$l_{OA} = \sqrt{e^2 + R^2 - 2e \cdot R \cdot \cos \beta} \quad (6)$$

Where R is the radius of the eccentric wheel; r is the radius of the roller; e is eccentric distance;
 $\cos \beta = [(R+r)^2 + e^2 - l_{oo_2}^2] / 2e \cdot (R+r)$; $\varphi - \alpha = (z_G - 1)\alpha$;
 $l_{oo_2} = e \cos((z_G - 1)\alpha) + \sqrt{(R+r)^2 - e^2 \sin^2((z_G - 1)\alpha)}$. The design parameters of the rolling movable teeth reducer are shown in Table 1.

Table 1. The design parameters of the reducer.

Parameters and Dimensions	Numerical number
Drive ratio i	32
Number of rollers Z_G	32
Number of teeth in the center wheel Z_K	31
Radius of rolling movable teeth r / mm	2
Eccentric distance e / mm	0.95
Radius of eccentric wheel R / mm	42.05

According to the parameters in the Table 1, they are substituted into the above formulas. The sliding rate U_G of the center wheel and the sliding rate U_K of the rolling movable teeth are shown in Fig. 4 and Fig. 5. [7].

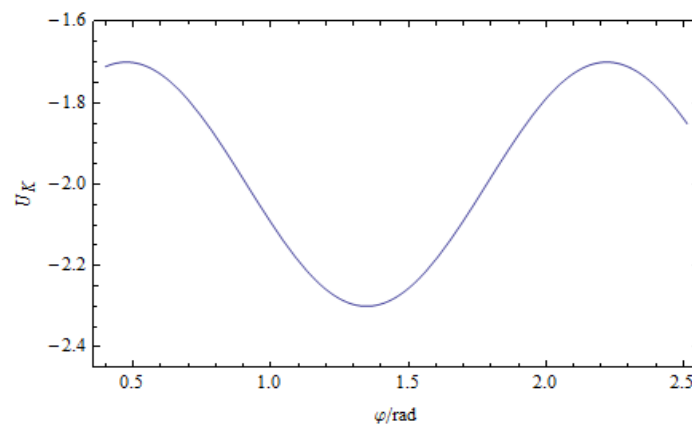


Figure 4. The sliding rate of the center wheel.

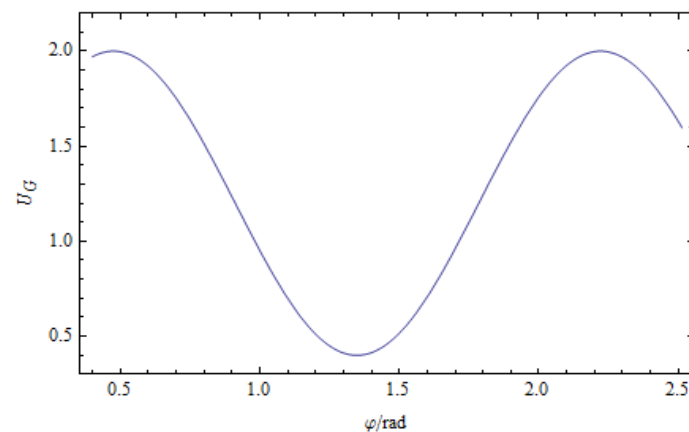


Figure 5. The sliding rate of the roller.

From the sliding rate curve, it can be seen that the sliding rate of the conjugate tooth surface of the center wheel and the rolling movable teeth is a function of the position of the meshing point, and the changing law of the two is the same. The sliding rate varies smoothly and fluctuates in a small range. When the rolling movable teeth mesh with the center wheel at the root and top of the teeth, the sliding rate of the movable teeth surface is the largest, so the wear of the rolling movable teeth surface is serious, and the tooth surface cementation is easy to occur. The conjugate tooth profile in practical application needs to be modified so that the top and root parts of the teeth do not engage in meshing. In this way, the actual maximum slip rate will be less than the theoretical maximum. The study of sliding rate in this paper lays a theoretical foundation for further study of tooth surface wear, transmission efficiency and elastohydrodynamic lubrication of rolling movable teeth transmission, and also provides a basis for the design of rolling movable teeth reducer in the future.

4. Lubrication Analysis

The average thickness of the Antarctic ice sheet is about 2000-2500m. The thickest part can reach 4000m. The surface layer is 15 m thick. The bottom 80 m is the deposited snow layer. The next is the permafrost layer, and the bottom is the subglacial bedrock. Temperature increases with depth in the ice. Deep ice layer is called warm ice layer. The temperature of warm ice layer is between - 2.8 and - 7.9 C [8]. Because of the staff work in the polar region having an extreme cold and anoxic environment, the labor intensity of the personnel and number of maintenances of equipment should be minimized in polar drilling. Therefore, lubrication operations such as refueling and oil exchange are difficult and there are safety risks. In addition, the oil consumption of reducer is relatively large, the replacement cost is high, the replacement of oil products and the cleaning of oil tank are also difficult, and the waste oil treatment is difficult and costly. As a reducer lubricant, it should generally have good extreme pressure and wear resistance, oxidation stability, anti-corrosion and rust resistance [9].

According to the structural characteristics and working environment of the reducer for polar deep ice drilling, there are some special requirements for lubricant besides the basic requirements mentioned above [10, 11, 12]. (1) For the good lubrication of the reducer at low temperature, the lubricant must have good low temperature performance. (2) Because of the small contact area of the rolling movable teeth in the process of meshing and meshing and at the inflection point of the central gear profile, the condition of forming oil film is poor and the thickness of oil film is small. Therefore, lubricants with relatively high viscosity and good lubrication performance should be selected. (3) When the reducer for polar deep ice drilling is lubricated by rollers rolling with lubricant, it is easy to cause lubricant loss, which is not conducive to the formation of dynamic pressure oil film. Therefore, the lubricant with good adhesion should be selected to ensure that enough oil film can be formed when movable teeth mesh. (4) Polar deep ice drilling is a very complex dynamic process. In order to buffer the larger load and impact load generated by drilling rig during drilling process, and to reduce the failure of fracture, pitting and fatigue wear, the lubricant with extreme pressure antiwear agent should be selected as far as possible to disperse load, improve oil film strength and reduce tooth surface wear. (5) In order to avoid impurities such as drilling fluid, lubricants with good water resistance, corrosion resistance and rust resistance should be selected. Therefore, lubricants should meet the requirements of good oxidation resistance, slow attenuation of various indicators, long oil exchange cycle and low cost of comprehensive management. In order to better solve the lubrication problem of the reducer for polar deep ice drilling, the extreme pressure and anti-wear semi-fluid lubricant with low temperature similar viscosity not less than 150 Pa · s and mass grade not less than CKC can be selected. The product has many advantages, such as good low temperature performance, anti-impact load, improving the leakage prevention of reducer, long oil exchange period, and adapting to field conditions.

5. Conclusion

In this paper, the sliding rate of rolling movable tooth reducer for polar deep ice drilling is analyzed. The analysis results show that the sliding rate of the movable teeth surface is the largest when the rolling movable teeth mesh with the center wheel at the root and top of the teeth, so the wear of the rolling

movable teeth surface is serious, and the tooth surface cementation is easy to occur. The conjugate tooth profile in practical application needs to be modified so that the top and root parts of the teeth do not engage in meshing. In this way, the actual maximum slip rate will be less than the theoretical maximum. The selection criteria of reducer lubricating oil are also described in this paper. Lubrication analysis of rolling movable teeth reducer for polar deep ice drilling is carried out. Combining friction and lubrication of movable teeth transmission with traditional design method can improve the design level and overall performance of movable teeth transmission.

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