Enhanced Load-Sharing Technology of Toroidal Drive

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Original Article

Keywords: Toroidal drive, Load sharing , Floating oil film , Load-sharing structure, Dynamic analysis, Load-sharing effect

Posted Date: May 28th, 2020

DOI: <https://doi.org/10.21203/rs.3.rs-30895/v1>

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Title page

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ORIGINAL ARTICLE

Enhanced Load-Sharing Technology of Toroidal Drive

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Abstract: In the manufacture and assembly of a toroidal drive mechanism, errors caused have a great influence on the load sharing of the mechanism. To ensure favourable load-sharing characteristics of the mechanism and to compensate for inaccuracies in the manufacture and assembly of the mechanism parts, and based on the transmission characteristics and operating states characteristics of the mechanism, a floating oil film structure was designed to support the planetary gear. The elasticity and hydrodynamic effect of the floating oil film allow the planetary gear to achieve its own small floatation and produce a certain axial displacement, which compensates for the influence of error and achieves load sharing. To examine the effect of the floating oil film structure, the floating oil film bearing was simulated by FLUENT, the characteristics of the floating oil film were analyzed, and the stiffness and damping coefficients of the floating oil film were calculated. In ADAMS, the method of equivalent replacement of the floating oil film with spring damping was adopted to conduct a dynamic analysis on the toroidal drive mechanism with the floating oil film load-sharing structure, and the effectiveness and feasibility of this structure were verified by comparing the changes in transmission performance of the system over time.

Keywords: Toroidal drive • Load sharing • Floating oil film • Load-sharing structure • Dynamic analysis • Load-sharing effect

1. Introduction

The toroidal drive mechanism is a typical spatial multi-conjugate transmission mechanism; certain planetary gears, a centre worm, and a ring gear transfer motion and power by meshing in this mechanism, achieving a ‘power split’ structure. This mechanism has the characteristics of a large bearing capacity, high transmission efficiency, large transmission ratio, and compact structure, and is a new type of transmission mechanism with good transmission performance[1]. However, owing to the complicated transmission structure, manufacture of the centre worm and ring gear is difficult, and errors in processing and assembly are therefore readily produced. These errors may lead to an unbalanced loading between the planetary gears, resulting in an eccentric load phenomenon and reductions in the transmission performance and service life of the mechanism[2]. Therefore, the addition of an appropriate load-sharing structure can compensate for the influence of error, solve the problem of uneven load distribution, and play a

key role in excellent transmission performance of the toroidal drive mechanism.

In the field of toroidal load sharing, scholars have conducted in-depth theoretical and experimental research on toroidal drive load sharing, and have obtained significant research results. Tooten[3] studied the eccentric load problem of a cylindrical tooth-type toroidal drive mechanism and proposed improvements. Jarchow and Vonderschmidt[4] designed components as floating support structures to achieve load sharing, measured the load-sharing coefficient of a system with floating sun gear, planetary gear, and planetary frame, and compared and analysed the load-sharing performance of different floating component systems. Xu et al.[5-7] considered the eccentric load problem of components along the axis, and deduced three more practical formulas for calculating the load distribution of toroids with different tooth shapes. Yuan et al.[8] optimised the dynamic equilibrium of the planetary gear power split. Shang[9] and Li[10] studied the load-sharing characteristics of the system when each component floated. Jiang[11] calculated and analysed the parameters of the planetary gear with a thick oil film load-sharing system.

However, the above studies have not included in-depth analysis on the characteristics of a floating oil film load-sharing structure. In this study, based on the characteristics of the toroidal drive mechanism and the influence of error, the support of the planetary gear was designed as a floating oil film load-sharing structure, and the hydrodynamic effect of the floating oil film was used to make the planetary gear shift, compensate for the influence of error, and achieve the effect of load sharing. To verify the effect of the floating oil film structure, the floating oil film bearing was simulated using FLUENT, and the stiffness and damping coefficients of the floating oil film were calculated. In ADAMS, the method of equivalent replacement of the floating oil film with spring damping was adopted to conduct a dynamic analysis of the toroidal drive mechanism with the floating oil film load-sharing structure, and the effectiveness and feasibility of this structure were verified by comparing the changes in transmission performance of the system over time.

2. Planetary Gear Supports with Load-sharing Structure

2.1 Introduction to Toroidal Drive Mechanism

Figure 1 shows a toroidal drive retarder, which is mainly composed of a centre worm, planetary gears, planetary frame, ring gear, and cylindrical rollers[12]. The centre worm is for power input, and a number of free-rolling cylindrical rollers are evenly distributed on the planetary gear; the cylindrical rollers mesh with the centre worm and ring gear respectively, to achieve rotation. When the ring gear is fixed as the frame, the planetary gear rotates while driving the revolution of the planetary frame, and finally the planetary frame outputs motion and power[13]. The mechanism adopts the form of a spatial pure-rolling drive with multiple cylindrical rollers participating in the driving motion and power, which is a new and efficient driving structure[14].

Figure 1 Diagram of toroidal drive retarder

2.2 Design of Floating Oil Film Load-sharing Structure

Referring to the design of a hybrid sliding bearing[15], the purpose of the floating oil film load-sharing structure is to serve as intermediate ring between the planetary gear and the outer ring (or shaft) of the planetary gear bearing. A certain radial clearance is left between the intermediate ring and the inner hole of the planetary gear, and oil is supplied to the clearance through the oil duct. During the transmission process, the planetary gear drives the lubricating oil into the gap to rotate in the same direction and at the same speed, and the planetary gear and lubricating oil bear the load in the same direction. A thick oil film is then formed, a hydrodynamic effect is produced, and load sharing of the planetary gear is achieved, as shown in Figure 2.

Figure 2 Floating oil film load-sharing structure of planetary gear

The main parameters of the intermediate ring of the planetary gear are listed in Table 1.

Table 1 Design parameters of intermediate ring

Parameters	Symbol	Value
Inner diameter of intermediate ring	D	22 mm
Outer diameter of intermediate ring	d	24 mm
Width of intermediate ring	l	24 mm
Radial clearance	φ	0.05 mm
eccentricity	ε	0.3
Width to diameter ratio	l/D	0.5
Surface roughness	Re	1.6 μm

Figure 3 shows the centre displacement track of the planetary gear. The oil film formed between the intermediate ring and the planetary gear in the rotation of the planetary gear. When the planetary gear is overloaded, the thickness of the oil film decreases and the oil wedge angle also decreases; under a light load, the thickness of the oil film increases proportionally and the oil wedge angle also increases accordingly. In other words, each planetary gear produces different displacements of the planetary gear shaft to adjust the load and achieve load sharing. In other load-sharing methods, the elastic characteristics of the components are affected by the load of the planetary gear, the relative stiffness of the elastic components is larger, and the effect of damping is smaller; deformation then readily leads to fatigue wear, reducing the service life[16]. The floating oil film load-sharing structure uses the stiffness and damping effect of the oil film, is independent of the properties of the component, and can automatically adjust the thickness of the oil film according to load variation to achieve load sharing. At the same time, the oil film has the function of lubrication and damping, which

reduces the vibration and friction of the system[17].

Figure 3 Centre displacement track of planetary gear

2.3 Calculation of floating oil film load-sharing coefficient

When the ring gear is fixed, the centre worm is connected to the motor to input torque; after the load is applied, the centre worm engages with one of the four planetary gears. Manufacturing and assembly errors cause a certain backlash where the other three planetary gears engage with the centre worm. When the input torque is further increased, and under the influence of the engaging force, elastic deformation will occur at the support of the cylindrical roller and planetary shaft to compensate for the meshing backlash of the other planetary gears, and the remaining planet gears are in the meshed state[18]. When the centre worm meshes with the i th planetary gear ($i = 1, 2, 3, 4$), the angles created by the elastic deformation are set as θ_s and θ_{pi} , and the following expressions for load on the tooth surface can be obtained:

$$\begin{cases} F_{spi} = K_{spi}(r_s\theta_s - r_{pi}\theta_{pi} - \Delta_{spi}) \\ F_{pil} = K_{pil}(r_{pi}\theta_{pi} - r_l\theta_l - \Delta_{pil}) \end{cases} \quad (1)$$

F_{spi} is the load on the tooth surface when the centre worm meshes with the i th planetary gear, and F_{pil} is the load on the tooth surface when the i th planetary gear meshes with the ring gear.

When the ring gear is fixed, $\theta_l = 0$, and the sum of the two sides of Eq. (1) gives us the following for total load:

$$\begin{cases} \sum_{i=1}^n F_{spi} = nK_{spi}(r_s\theta_s - r_{pi}\theta_{pi}) - K_{spi}\Delta_{spi} \\ \sum_{i=1}^n F_{pil} = nK_{pil}r_{pi}\theta_{pi} - K_{pil}\Delta_{pil} \end{cases} \quad (2)$$

According to the power split characteristics of the centre worm and planetary gear,

$$T = r_s \sum_{i=1}^n F_{spi} \quad (3)$$

The theoretical average load of the planetary gear is known to be

$$F_{spi} = T/nr_s \quad (4)$$

At equilibrium, the forces on the planetary gear are balanced,

$$F_{pil} = F_{spi} \quad (5)$$

The load-sharing coefficient refers to the ratio between the maximum load of the planetary gear and the centre worm or ring gear and the theoretical average load. The load-sharing coefficient of planetary gear can be obtained from Eqs. (1) to (5):

$$K_{spi} = \frac{F_{spi_{max}}}{T/nr_s} \quad (6)$$

3. Numerical Simulation of Planetary Gear Floating Oil Film

3.1 Establishment and Meshing of Floating Oil Film Model

A three-dimensional (3D) model of the toroidal floating oil film bearing was established with a width to diameter ratio of 0.5; the inside of the floating oil film bearing was a fixed wall surface, the outside was a rotating wall surface, and the bearing had an oil inlet hole at the top. The basic parameters of the floating oil film bearing were as follows: diameter of bearing pad $D = 48$ mm, bearing width $B = 24$ mm, oil inlet diameter and height $h_l = 0.5$ mm, and eccentricity $\varepsilon = 0.3$.

The 3D model of the floating oil film bearing was imported into ANSYS ICEM modelling software using hexahedral mesh for the mesh division; the result is shown in Figure 4. The value of the mesh determinant $2 \times 2 \times 2$ was greater than 0.7 and the internal angle was greater than 18° , which led to a high partition quality and met the requirements of mesh quality.

Figure 4 Overall oil film mesh and quality

3.2 Steady-state Flow Field Calculation of Floating Oil Film

The steady-state flow field of the floating oil film was solved using FLUENT 14.0 software, the SIMPLE algorithm was used as the pressure–velocity coupling algorithm, Presto software was used to interpolate a discrete form of the pressure equation, and a second-order upwind method was used to interpolate a discrete form of the momentum equation. The initial rotation rate of the planetary gear was set to 1000 rpm, the eccentricity was 0.3, and the oil pressure was 0.2 MPa. The resulting pressure contours are shown in Figure 5.

Figure 5 Pressure contours of floating oil film

Based on the obtained pressure contours of the floating oil film, the following characteristics of the floating oil film can be obtained:

- (1) The floating oil film structure creates stable oil pressure. The pressure is highest at the right-side inlet, with a value of 0.2 MPa. The pressures in the upper and lower oil chambers of the inlet gradually decrease, because when the floating oil film bearing operates, it moves down owing to gravity, which leads to a reduction in the oil film clearance of the lower oil chamber and an increase in that of the upper oil chamber. Positive and negative pressure regions are formed, which conforms to hydrodynamic principles.
- (2) In the axial direction, because the floating oil film bearing discharges oil at both ends and these ends are at atmospheric pressure, the pressure of the oil film decreases in the axial direction and is at atmospheric pressure at the boundary.
- (3) In the circumferential direction, the oil film

rotates in the same direction as the planetary gear, and the lubricating oil is throughout the entire clearance. When the lubricating oil moves from the oil inlet into the oil cavity, the pressure is at a maximum. With the rotation of the planet gear, the pressure of the oil film gradually decreases axially along the neck surface until the negative pressure zone is burst; the supply of lubricating oil then continues and a new pressure distribution of the oil film develops.

3.3 Force Field Characteristic Analysis of Floating Oil Film

The pressure and bearing capacity are the key characteristic parameters of the floating oil film and determine the resistance of the planetary gear to external load, that is, the effectiveness of the floating oil film load-sharing structure[19]. The pressure and bearing capacity of the floating oil film are influenced by the design parameters, the working conditions, and many other factors, among which eccentricity is one of the most important[20]; the eccentricity is therefore selected in this analysis as the main factor influencing the floating oil film. Owing to the eccentricity, the state of the floating oil film in each direction will change, producing new dynamic characteristics. However, an excessive eccentricity will produce strong hydrodynamic effects, and the lubricating oil in the clearances will be prone to backflow, which may cause the rupture of the oil film[21]. Therefore, the influence of eccentricity between 0.3 and 0.6 was first studied, at a rotational speed of 2000 rpm and oil inlet pressure of 0.2 MPa. The pressure distribution of the oil film for different eccentricity s is shown in Figure 6.

(a) $\varepsilon = 0.3$

(b) $\varepsilon = 0.4$

(c) $\varepsilon = 0.5$

(d) $\varepsilon = 0.6$

Figure 6 Pressure distribution of oil film for different eccentricity

As shown in Figure 6, the pressure of the oil film positive-pressure zone increases with increasing eccentricity, but the pressure of the oil film negative-pressure zone decreases. This is because, with the increase in eccentricity, the oil film produces a larger wedge clearance, the positive-pressure oil chamber above the oil film increases gradually, the negative-pressure oil chamber under the oil film decreases gradually, the oil film is significantly squeezed, the hydrodynamic effect is more pronounced, and the pressure of the oil film increases.

As indicated in Table 2, the bearing capacity of the oil film increases with increasing eccentricity. When the eccentricity increases, the wedge angle between the clearances then increases, the squeezing action of the oil film intensifies, and the maximum pressure of the oil

film increases, such that the oil film can bear more load.

Table 2 Bearing capacity of oil film for different eccentricity

Eccentricity ε	0.3	0.4	0.5	0.6
x Direction W_x (N)	162	187	236	280
y Direction W_y (N)	753	1,125	1,402	2,187
Bearing capacity W (N)	770	1,140	1,422	2,205

In the design of the floating oil film load-sharing structure, the eccentricity should therefore be increased appropriately, because that can improve the pressure and the bearing capacity of the floating oil film and enhance the resistance of the planetary gear to external load. However, too large an eccentricity will cause oil backflow. The pressure and temperature of the oil film should also be controlled to avoid rupture of the oil film caused by excessive pressure and temperature.

3.4 Calculation of equivalent stiffness and damping of floating oil film

3.4.1 Difference calculation model of stiffness and damping

When the oil film bearing is disturbed by displacement or velocity in the static equilibrium position, the reaction force by the oil film on the shaft neck will change[22]. When the disturbance is small and with small amplitude, the relationship can be regarded as linear and the oil film can be regarded as having linearised stiffness and damping, as shown in Figure 7.

Figure 7 Linearised floating oil film mechanical model

Expanding the oil film force as a Taylor series and retaining the linear term, we obtain[23]

$$\begin{cases} F_x = F_{x0} + \left(\frac{\partial F_x}{\partial x}\right)_0 \Delta x + \left(\frac{\partial F_x}{\partial y}\right)_0 x\Delta y + \left(\frac{\partial F_x}{\partial \dot{e}}\right)_0 \dot{x} + \left(\frac{\partial F_x}{\partial \dot{y}}\right)_0 \dot{y} \\ F_y = F_{y0} + \left(\frac{\partial F_y}{\partial x}\right)_0 \Delta x + \left(\frac{\partial F_y}{\partial y}\right)_0 x\Delta y + \left(\frac{\partial F_y}{\partial \dot{e}}\right)_0 \dot{x} + \left(\frac{\partial F_y}{\partial \dot{y}}\right)_0 \dot{y} \end{cases} \quad (7)$$

The oil film force is approximated as a linear function of the minute displacement and velocity of the shaft neck. In Eqs. (7), the subscript 0 indicates that the derivative is taken at the static equilibrium position. The oil film force increment caused by the disturbance displacement is defined as the stiffness coefficient of the oil film, that is,

$$\begin{aligned} k_{xx} &= \frac{\partial F_x}{\partial x} & k_{xy} &= \frac{\partial F_x}{\partial y} & k_{yx} &= \frac{\partial F_y}{\partial x} \\ k_{yy} &= \frac{\partial F_y}{\partial y} \end{aligned} \quad (8)$$

The oil film force increment caused by the speed displacement is defined as the damping coefficient of the oil film, that is,

$$\begin{aligned} c_{xx} &= \frac{\partial F_x}{\partial \dot{x}} & c_{xy} &= \frac{\partial F_x}{\partial \dot{y}} & c_{yx} &= \frac{\partial F_y}{\partial \dot{x}} \\ c_{yy} &= \frac{\partial F_y}{\partial \dot{y}} \end{aligned} \quad (9)$$

In the formulae, the first subscript represents the direction of the force, and the second represents the direction of the displacement or velocity. Thus, the dynamic characteristics of the oil film depend directly on the disturbance pressure.

3.4.2 Calculation of stiffness and damping of floating oil film

A disturbance velocity of $v = 25 \mu\text{m/s}$ and disturbance displacement of $s = 25 \mu\text{m}$ was applied to the floating oil film bearing, the planetary gear rotational speed was set to 1000 rpm, and the oil inlet pressure was set to 0.2 MPa. The oil film force before and after the disturbance was calculated by FLUENT post-processing, and the stiffness and damping coefficients of the oil film were calculated by substituting the oil film force into Eqs. (8),(9). The coefficients are listed in Table 3.

Table 3 Stiffness and damping coefficients of floating oil film

Parameter	Value	Parameter	Value
Direct stiffness	1.354	Direct damping	12.5
k_{xx}		c_{xx}	
Direct stiffness	8.765	Direct damping	24.0
k_{yy}		c_{yy}	

4. Dynamics Analysis of Floating Oil Film Load-sharing Structure

4.1 Establishment of Floating Oil Film Rotor Model

Considering the complexity of the toroidal drive mechanism model[24], the 3D model was first established using the 3D software UG, and the exported Parasolid file was imported into ADAMS to establish the simulation model. The established virtual prototype model is shown in Figure 8.

Figure 8 Virtual prototype model

Referring to the linearised floating oil film mechanical model, the floating oil film was modelled as an equivalent spring; circumferentially, the floating oil film was equivalent to four rotating spring dampers, as shown in Figure 9. The spring attribute parameters, listed in Table 3, were input into the model to complete the rotor dynamics simulation model.

Figure 9 Equivalent spring model of floating oil film

4.2 Analysis of Dynamic Simulation Results

To verify the effectiveness of the floating oil film load-sharing structure and analyse the influence of the structure on the transmission performance of the mechanism, the ADAMS simulation data with and without the load-sharing structures of the toroidal drive mechanism were compared.

4.2.1 Analysis of planetary gear speed

Figure 10 shows the change diagram of the angular velocity of planetary gear 1 in the toroidal drive mechanism without and with the floating oil film load-sharing structure. Figure 10 indicates that the fluctuation of the angular velocity of the planetary gear with the load-sharing structure is smaller than without. The oil film therefore has a certain buffering and damping effect on the system; it can balance the load, stabilise the transmission, and reduce the planetary gear velocity mutation, and therefore improves the load-sharing performance of the system. The change diagrams of the angular velocity of other planetary gears are similar to that of planetary gear 1, with only small differences in values, and therefore they require no elaboration.

(a) Angular velocity of planetary gear 1 without load-sharing structure

(b) Angular velocity of planetary gear 1 with floating oil film load-sharing structure

Figure 10. Comparison of angular velocity of planetary gear

Figure 11 shows the change diagram of angular velocity of the planetary frame in the toroidal drive mechanism without and with the floating oil film load-sharing structure. The diagram indicates that the angular velocity of the planetary frame with the load-sharing structure has a smaller fluctuation range and the output speed of the system is more stable. This indicates that the floating oil film load-sharing structure can effectively improve the transmission stability of the system and is conducive to a stable output power of the mechanism.

(a) Angular velocity of planetary frame without load-sharing structure

(b) Angular velocity of planetary frame with floating oil film load-sharing structure

Figure 11 Comparison of angular velocity of planetary frame

4.2.2 Analysis of planetary gear engaging force

Two of the four planetary gears were compared to observe the changes in engaging force. Figure 12 shows the engaging force changes between the planetary gear and the centre worm in the

toroidal drive mechanism without and with the floating oil film load-sharing structure. As indicated in the figure, owing to systematic errors and other factors, the engaging forces of the two planetary gears vary in different manners, and unbalanced loading occurs. The engaging force of each planetary gear without the load-sharing structure fluctuates significantly over time, which is not conducive to an even and stable load transfer for the planetary gear. However, after the floating oil film load-sharing structure is adopted, the engaging force of each planetary gear decreases with time, the fluctuation amplitude of the engaging force decreases, and the eccentric loading of planetary gear decreases, except for the sudden change of load when a few rollers are engaging in and engaging out. This indicates that the floating oil film load-sharing structure has a certain effect on the load sharing of the planetary gear that is conducive to even load transfer and a more stable system drive.

(a) Engaging force of planetary gear 1 without load-sharing structure

(b) Engaging force of planetary gear 1 with floating oil film load-sharing structure

(c) Engaging force of planetary gear 4 without load-sharing structure

(d) Engaging force of planetary gear 4 with floating oil film load-sharing structure

Figure 12 Comparison of engaging force of planetary gear and centre worm

Table 4 shows a comparison of the mean value of the external engaging force without and with the floating oil film load-sharing structure. The external engaging forces of the mechanism with the load-sharing structure are similar and relatively concentrated, indicating that the mechanism with the floating oil film load-sharing structure can achieve load sharing.

Table 4 Comparison of mean value of external engaging force without and with floating oil film load-sharing structure

Contact pairs	Mean value of external engaging force(N)	
	No load-sharing structure	Floating oil film load-sharing structure
Planetary gear 1 and centre worm	2,758	2,567
Planetary gear 2 and centre worm	2,428	2,480
Planetary gear 3 and centre worm	2,655	2,523
Planetary gear 4 and centre worm	2,406	2,515

Figure 13 shows comparison diagrams of the forces acting on the planetary gear and the ring gear in the toroidal drive mechanism without and with the floating oil film load-sharing structure. As indicated in the figure, after the floating oil film load-sharing structure is adopted, the fluctuation range of the engaging force between the planetary gear and the ring gear decreases, and the planetary gear can adjust the meshing position with the ring gear, which compensates for the influence of the machining and assembly errors of the planetary gear on the driving and

meshing and improves the system load-sharing performance.

(a) Engaging force of planetary gear 1 without load-sharing structure

(b) Engaging force of planetary gear 1 with floating oil film load-sharing structure

(c) Engaging force of planetary gear 4 without load-sharing structure

(d) Engaging force of planetary gear 4 with floating oil film load-sharing structure

Figure 13 Comparison of engaging force of planetary gear and ring gear

Table 5 shows a comparison of the mean value of the internal engaging forces between the planetary gear and ring gear in the toroidal drive mechanism without and with the floating oil film load-sharing structure. The transmission load of the mechanism with the load-sharing structure tends to be balanced; in contrast, without the load-sharing structure the force of the planetary gear in the toroidal drive mechanism is either too large or too small.

Table 5 Comparison of mean value of internal engaging force without and with floating oil film load-sharing structure

Contact pairs	Mean value of internal engaging force(N)	
	No load-sharing structure	Floating oil film load-sharing structure
Planetary gear 1 and ring gear	2,561	2,494
Planetary gear 2 and ring gear	2,358	2,418
Planetary gear 3 and ring gear	2,535	2,488
Planetary gear 4 and ring gear	2,322	2,406

4.2.3 Analysis of planetary gear axial displacement

The axial motion of planetary gear was simulated using ADAMS. Figure 14 shows the centre displacement diagram of planetary gear 1 in the toroidal drive mechanism with the floating oil film structure. The centre displacement diagrams of other planetary gears are similar to that of planetary gear 1, with only small numerical differences. As indicated in the figure, when the mechanism runs, the floating oil film produces a hydrodynamic effect that allows the planetary gear to resist the external load. The axis of the planetary gear moves above and below the centre position, which plays a role in adjusting the meshing position, compensating for the influence of error, and balancing the load.

(a) Displacement of planetary gear centre in x direction

(b) Displacement of planetary gear centre in y direction

Figure 14 Centre displacement as a function of time

4.3 Dynamic Force Analysis of Floating Oil Film

Figure 15 shows the change in the equivalent oil film force of planetary gear 1 in the transmission process. The changes for other planetary gears are similar to that of planetary gear 1 with only small differences in value, and therefore do not require an explanation.

The figure shows that, in the initial stage, the speed of the planet gear is low, the transmission is not stable, and the bearing capacity of the oil film fluctuates significantly. As planetary gear speed improves, the system runs in a stable manner, and the fluctuation in the bearing capacity of the oil film is reduced. When the system reaches a stable operating condition, the floating oil film forms a stable hydrodynamic effect and stable bearing capacity controlled at 3,000 N, which is greater than the average load of the planetary gears in Tables 4 and 5. Therefore, the oil film structure can withstand the impact of the load in the operating range, assist the planetary gear in resisting the displacement of the centre caused by external loads, allow the position of the meshing point of the planetary gear with the centre worm and ring gear to adjust by itself, compensate for the influence of machining and assembly errors on the load transmission, and achieve load-sharing effect.

Figure 15 Change in equivalent oil film force of planetary gear1

4.4 Analysis of Dynamic Load-sharing Performance of Floating Oil Film

The load-sharing coefficient is an index to evaluate the load-sharing performance of the system[25]. The closer the coefficient value is to 1, the better the load-sharing performance. With the data of the planetary gear force extracted from ADAMS, the load-sharing coefficient solved using MATLAB, and data fitting performed using Eq. (6), curves of the load-sharing coefficient of the toroidal drive mechanism without and with the floating oil film load-sharing structure over a period of time were obtained.

(a) Load-sharing coefficient of toroidal drive mechanism without load-sharing structure

(b) Load-sharing coefficient of toroidal drive mechanism with floating oil film load-sharing structure

Figure 16 Load-sharing coefficient of system

The comparison shows that when the floating oil film load-sharing structure is adopted, the maximum load-sharing coefficient of the system decreases from 1.31 to 1.20, the minimum load-sharing coefficient increases from 0.72 to 0.83, and the fluctuation range of the load-sharing coefficient of each planetary gear decreases. Therefore, the floating oil film

load-sharing structure of the planetary gear can adjust the loading situation of each planetary gear, reduce the influence of error on the transmission load, and effectively improve the load-sharing performance of system.

5. Conclusion

- (1) Comparisons of the angular velocity of the planetary gear, the output angular velocity of the planetary frame, and the engaging force of the planetary gear with the centre worm and ring gear demonstrate that the transmission performance of the system with the floating oil film structure is improved, the rotations of the planetary gear and the planetary frame are smoother, the fluctuation range and the mutation of the meshing force of the planetary gear decrease, and the load-sharing performance is improved.
- (2) When the centre displacement curve of the planetary gear is simulated under stable working conditions, the hydrodynamic effect of the floating oil film produces a floating displacement in the planetary gear of approximately 0.06 mm in the x and y directions to adjust the loading of the planetary gear, effectively compensating for the influence of error and achieving load sharing.
- (3) The simulation and analysis of the bearing capacity of the floating oil film indicate that, after stable operation of the floating oil film, the bearing capacity of the oil film fluctuates at 3 kN, and the planetary gear can resist the central displacement caused by the external load; thus, load sharing is achieved.
- (4) The engaging force of the planet gear was extracted and the load-sharing coefficient of system with the floating oil film load-sharing structure was calculated using MATLAB. Compared with the system without the load-sharing structure, the system with the floating oil film load-sharing structure has a maximum load-sharing coefficient that is 0.11 lower and a minimum coefficient that is 0.11 higher, the fluctuation of the coefficient is reduced, and the value of the coefficient is closer to 1. These demonstrate that the floating oil film load-sharing structure of planetary gear is effective and that

improving the load-sharing performance of the system is feasible.

6. Declaration

Acknowledgements

Not applicable

Funding

Supported by Natural Science Foundation of Hunan (Grant No. 2018JJ4006), National Independent Innovation Demonstration Area Foundation of Changsha-Zhuzhou-Xiangtan (Grant No. 2018XK2302), and Hunan Special Foundation for the Construction of Innovative Provinces (Grant No. 2019GK4025).

Availability of data and materials

The datasets supporting the conclusions of this article are included within the article.

Authors' contributions

The author's contributions are as follows: Mei-Jiao Mao was in charge of the whole trial; Meng-Yu Dong wrote the manuscript; Mei-Jiao Mao and Shi-Ping Yang assisted with sampling and laboratory analyses.

Competing interests

The authors declare no competing financial interests.

Consent for publication

Not applicable

Ethics approval and consent to participate

Not applicable

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Figures

Figure 1

Diagram of toroidal drive retarder

Figure 2

Floating oil film load-sharing structure of planetary gear

Figure 3

Centre displacement track of planetary gear

Figure 4

Overall oil film mesh and quality

Figure 5

Pressure contours of floating oil film

(a) $\varepsilon = 0.3$

(b) $\varepsilon = 0.4$

(c) $\varepsilon = 0.5$

(d) $\varepsilon = 0.6$

Figure 6

Pressure distribution of oil film for different eccentricity

Figure 7

Linearised floating oil film mechanical model

Figure 8

Virtual prototype model

Figure 9

Equivalent spring model of floating oil film

(a) Angular velocity of planetary gear 1 without load-sharing structure

(b) Angular velocity of planetary gear 1 with floating oil film load-sharing structure

Figure 10

Comparison of angular velocity of planetary gear

(a) Angular velocity of planetary frame without load-sharing structure

(b) Angular velocity of planetary frame with floating oil film load-sharing structure

Figure 11

Comparison of angular velocity of planetary frame

(a) Engaging force of planetary gear 1 without load-sharing structure

(b) Engaging force of planetary gear 1 with floating oil film load-sharing structure

(c) Engaging force of planetary gear 4 without load-sharing structure

(d) Engaging force of planetary gear 4 with floating oil film load-sharing structure

Figure 12

Comparison of engaging force of planetary gear and centre worm

(a) Engaging force of planetary gear 1 without load-sharing structure

(b) Engaging force of planetary gear 1 with floating oil film load-sharing structure

(c) Engaging force of planetary gear 4 without load-sharing structure

(d) Engaging force of planetary gear 4 with floating oil film load-sharing structure

Figure 13

Comparison of engaging force of planetary gear and ring gear

(a) Displacement of planetary gear centre in x direction

(b) Displacement of planetary gear centre in y direction

Figure 14

Centre displacement as a function of time

Figure 15

Change in equivalent oil film force of planetary gear1

(a) Load-sharing coefficient of toroidal drive mechanism without load-sharing structure

(b) Load-sharing coefficient of toroidal drive mechanism with floating oil film load-sharing structure

Figure 16

Load-sharing coefficient of system