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Study on Oil Film Bearing and Thermal Characteristics of Spline Hydrostatic Guideway of High-speed Gear Shaper

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Abstract

Aiming at the locking problem of involute spline hydrostatic guideway of YK5115AK2 gear shaper under high speed condition, based on the principle of hydrostatic support technology, the influence of oil film thickness and oil supply pressure design parameters on the bearing capacity and thermal characteristics of guideway is analyzed by finite element steady state analysis, and the influence of viscosity temperature effect on the bearing capacity and thermal characteristics of guideway is analyzed by finite element analysis. The variation trend of bearing capacity and temperature rise under different guide rail design parameters is revealed, and the optimal oil film thickness interval suitable for the guide rail structure is obtained. The research shows that the optimal oil film thickness of the guide rail is between 20μm and 40μm.

Keywords

Spline Hydrostatic Guide Rail; Bearing Characteristics; Thermal Characteristics; Viscous Temperature Effect.

1. Introduction

The hydrostatic guideway technology is to input the lubricating oil with a certain pressure into the oil cavity on the guideway surface through the throttle to form a bearing oil film, so that the guideway surface is in a pure liquid friction state, so as to reduce the friction coefficient of the guideway, and the hydrostatic support technology can effectively control the temperature rise of the guideway oil film and improve the service life of the parts. Aiming at the problems of traditional gear shaper, such as low maximum stroke, small adjustable range of speed, large amplitude of spindle of gear shaper, low efficiency and precision of gear shaping, involute spline hydrostatic guideway is used as the spindle guideway of gear shaper. Involute spline has the characteristics of high centering degree, strong bearing capacity, high precision and long service life. Under the condition of satisfying the stiffness and accuracy of the high-speed round-trip motion of the gear shaper spindle, it can still provide a large circumferential bearing capacity for the spindle to bear the torque generated by the spindle rotation motion.

For the hydrostatic support technology, many scholars at home and abroad have carried out in-depth research on it. In the analysis of the bearing capacity of the hydrostatic guideway and the thermal characteristics of the lubricating oil, Zhao and Gao[1] theoretically analyzed the dynamic and static performance of the guideway for the oil film thickness of the closed hydrostatic guideway, and obtained the influence of the oil film thickness and temperature of the guideway on the power loss, dynamic and static performance of the guideway system. However, the determination of the viscosity of the guide rail oil by Zhao is determined by the dynamic viscosity of the oil grade, which is not brought into the continuous viscosity-temperature equation. Based on the Reynolds equation, Wang[2] calculated and analyzed the effects of main parameters such as oil supply pressure, oil film thickness, liner size, pressure ratio, lubricating oil volume and lubricating oil viscosity on stiffness, damping and damping ratio. The above studies are all general closed hydrostatic guideways, and the oil film thickness can change with the increase of hydrostatic cavity pressure. However, the oil film thickness adjustment of spatial symmetrical hydrostatic guideways (such as involute spline hydrostatic guideways) is very small, and the initial oil film design parameters have a great influence on the bearing performance of the guideways. Lu[3] studied the bearing characteristics of the air-floating hydrostatic guideway, analyzed the influence of the throttling mode and the gas supply pressure on the bearing capacity of the guideway, and designed a kind of precision two-position motion platform than the closed hydrostatic guideway. Du[4] analyzed the influence of different throttling methods on the bearing characteristics of hydrostatic guideway. Based on the principle of rated liquid resistance, the variation trend of oil film stiffness under three different throttling conditions was calculated and simulated, and the influence of different throttling methods and design parameters on the stiffness of hydrostatic guideway was obtained. Du's research is based on the study of the static stiffness of the guide rail, and the dynamic stiffness and other bearing characteristics of the guide rail are not involved. S. Hurley, P. M. Cann, H. A. Spikes[5] studied the high temperature failure of lubricating oil from the direction of thermal aging of lubricating oil caused by high temperature. B.-R. Ho¨ hn and K. Michaelis[6] systematically studied the effects of oil temperature in the lubrication system on typical failures such as wear, scuffing, micro-pitting and pitting of gears. The above two studies are mainly aimed at the problem of lubrication failure caused by the change of the chemical properties of the lubricating oil under high temperature conditions, and there is no specific discussion on the temperature rise caused by high speed. Toshiharu KAZAMA[7] analyzed the optimum temperature and viscosity distribution of lubricating oil by using the viscosity-temperature relationship of lubricating oil. Huang[8] analyzed the thermal analysis of the vertical hydrostatic guideway system by considering the three heat sources of the motor heat, the viscous shear heat generated by the hydrostatic bearing oil film and the friction heat generated by the ball screw nut. The temperature rise of the oil film and the thermal deformation of the guideway were obtained by Ansys simulation, and the experimental platform was built. Nine thermal analysis experiments were carried out at three different feed speeds. It was found that the temperature difference between the finite element model simulation results and the experimental results was less than 3.97 %, which effectively proved the feasibility of using Ansys finite element analysis software in the force and heat analysis direction of the hydrostatic guideway.

Through the study of the above literature, it is found that most of the literature research objects are multi-cavity rectangular guide rails or fan-shaped guide rails, and there are few studies on spatial symmetrical hydrostatic guide rails (such as involute spline hydrostatic guide rails). Because of its excellent bearing characteristics, the spatial symmetrical hydrostatic guideway is mostly used for some bearing conditions with complex stress. Therefore, based on the design of the hydrostatic guideway of the spindle of the involute spline high-speed gear shaper, this paper analyzes the influence of the oil film thickness of the hydrostatic guideway and the oil supply pressure on the bearing characteristics and thermal characteristics of the oil film when the viscosity-temperature effect is introduced. The relationship between the oil film thickness and the bearing capacity and thermal characteristics of the hydrostatic guideway is obtained, which provides theoretical support for the optimal design of the spindle.

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2. Construction of Finite Element Simulation Analysis Model

2.1 Three-dimensional Model Construction of Guide Rail

The main object of this paper is the hydrostatic guideway of the spindle of YK5115AK2 involute spline gear shaper. Because the hydrostatic guideway is connected to the spindle, it has good rigidity and heat dissipation performance, so the simulation study does not consider the physical deformation caused by heat compression. According to the design drawings of the guide rail, the three-dimensional drawing is carried out. Figure 1 is the model of the outer guide rail of the static pressure guide rail, and Figure 2 is the oil film simulation model of the static pressure guide rail.

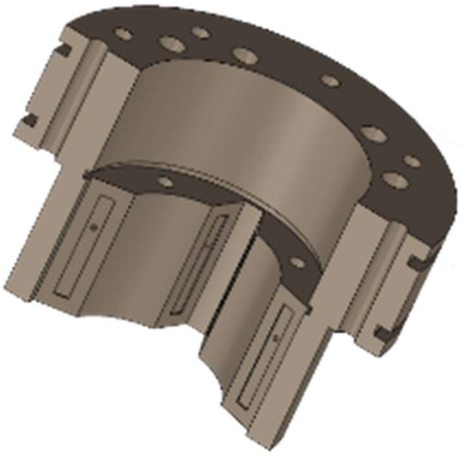

Figure 1. Outer guide rail model

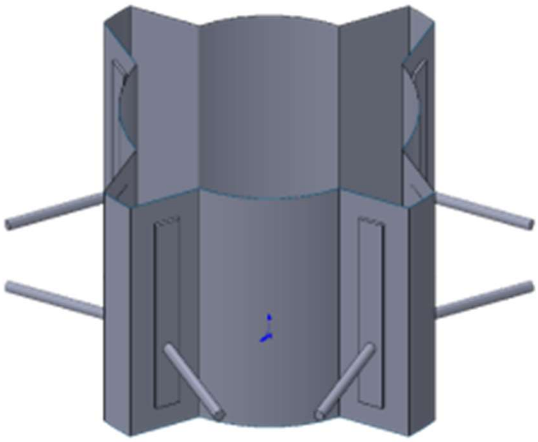

Figure 2. Oil film model

2.2 Finite Element Mesh Generation

The finite element model was constructed by ANSYS WORKBENCH. After completing the pre-processing of the three-dimensional model of the oil film, after many attempts, the sweep and multi-region scheme are used for meshing, and the partial meshing is shown in Figure 3. Because the design thickness of the oil film is small, only 0.01 ~ 0.05mm, and the mesh size is 1mm, the aspect ratio and orthogonal quality are large, but the model can still meet the needs of finite element simulation calculation.

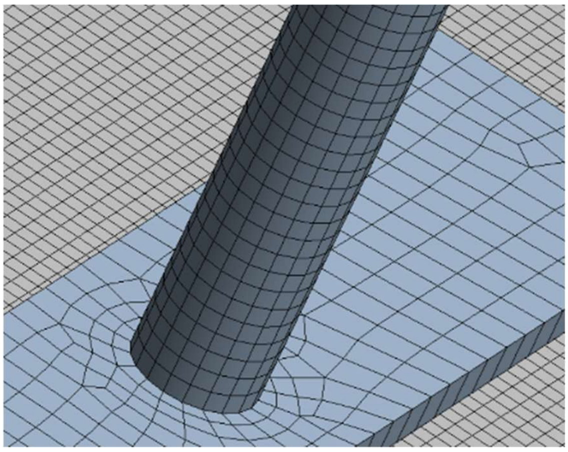

Figure 3. Oil film finite element meshing

2.3 Setting of Boundary Conditions

According to the YK5115AK2 type gear shaper guide rail structure, set the boundary conditions are as follows:

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Table 1. Boundary condition setting of hydrostatic guideway

parameter	numerical value
Oil inlet aperture	3.3mm
Oil inlet length	34mm
Oil film gap of sealing edge	0.01-0.05mm
Slider feed speed	2m/s
Hydraulic oil density	857 kg/m3
Specific heat of hydraulic oil	1955J/(kg K)
Thermal conductivity of hydraulic oil	0.144W/(m K)
Initial viscosity of hydraulic oil	0.005kg/(m s)
Oil inlet oil temperature	300K

The rectangular oil cavity diagram of the guide rail is shown in figure 4. The distance between the oil inlet and the side of the short oil sealing edge is 52 mm, and the whole oil pad is symmetrical about the central axis of the two long oil sealing edges.

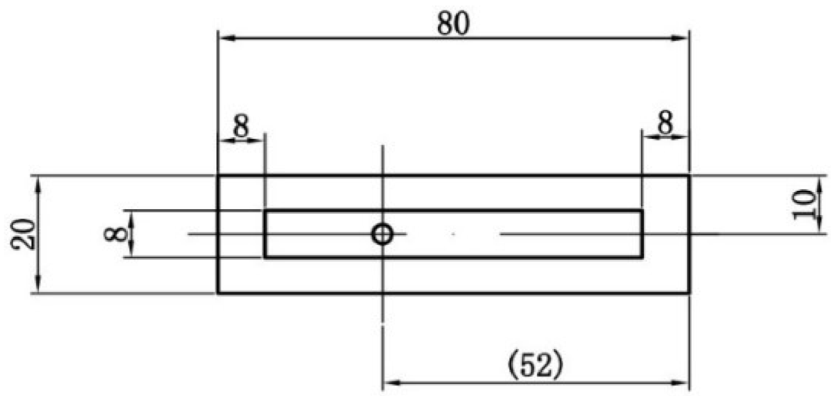
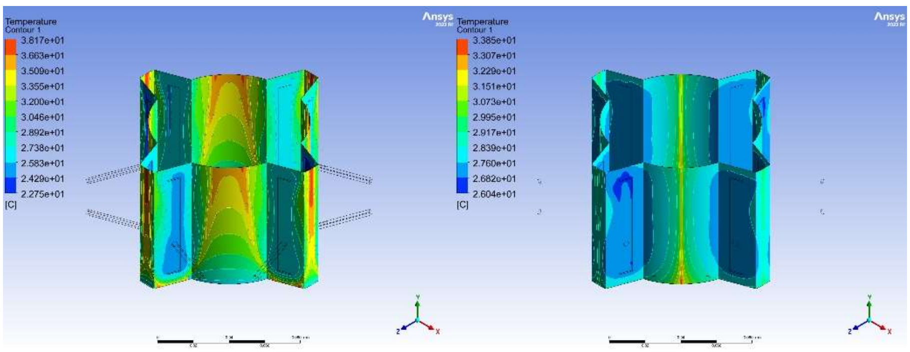


Figure 4. Schematic diagram of guide rail oil cavity

In the process of setting boundary conditions, the solver model is set to select laminar flow, and the fluid is incompressible. The viscous heating module is opened. The inlet is a pressure-type inlet, and the outlet maintains a default atmospheric pressure. The Coupled solution method is used to converge the residual to the e^{-6} level.

3. Thermal Characteristics Analysis of Oil Film



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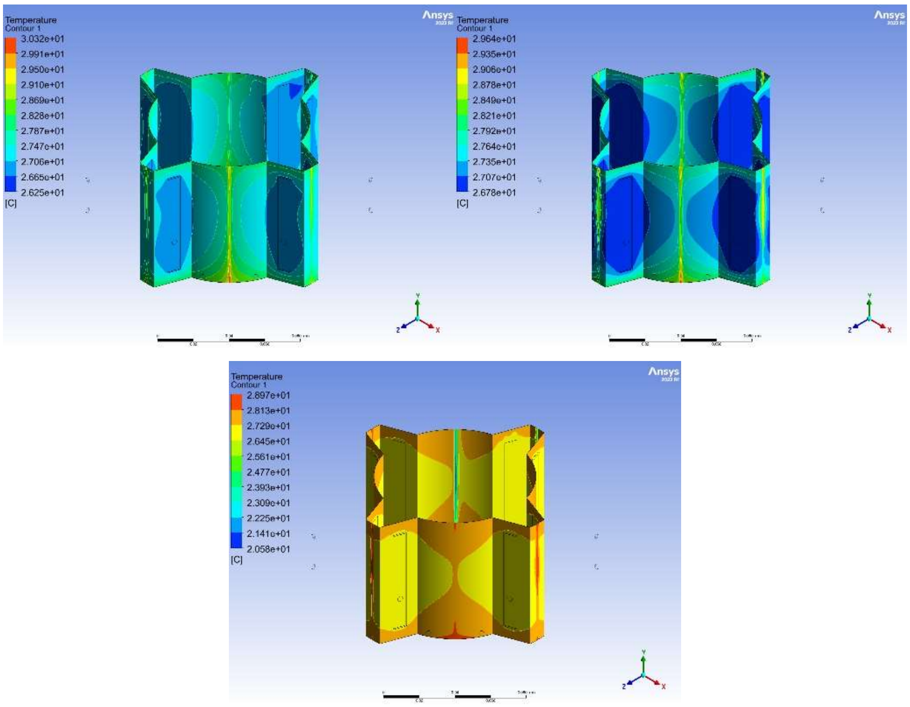

Figure 5. Oil film temperature cloud

In order to analyze the thermal characteristics and bearing characteristics of the oil film, the influence of oil film thickness change and oil supply pressure change on the thermal characteristics and bearing capacity of the oil film is analyzed for the steady state of the machine tool. In this paper, the orthogonal test method is used to test the influence of oil film thickness from 0.01 mm to 0.05 mm and different inlet pressure on the hydrostatic guideway. The bearing capacity of each oil chamber surface of the involute spline hydrostatic guideway and the influence of the temperature rise generated by the oil film on the viscosity of the lubricating oil are obtained by surface integration. After the software solution, the parameters of the static pressure guide oil film are obtained as shown in Figure 5.

Figure 5 shows the oil film temperature cloud map of the oil film gap from 0.01 mm to 0.05 mm thickness. It can be seen that when the oil film gap is between 0.02-0.04 mm, the oil film temperature distribution is more regular. When the oil film gap is low, the oil film appears. The phenomenon of local temperature rise is too high. When the oil film gap is too large, the range of lubricating oil diffusing from the oil cavity to the oil sealing edge is increased, and the low temperature region appears when the pressure is insufficient. The data of the maximum temperature rise of the oil film are sorted out as shown in figure 6 below.

Figure 6 shows the relationship between the maximum temperature of the oil film and the oil supply pressure and the oil film thickness. As the oil film thickness increases, the maximum temperature of the lubricating oil film decreases. When the oil film is thin, the lubricating oil is squeezed, the flow rate increases sharply, the friction intensifies, and a large amount of heat is generated. This is because the oil chamber pressure of the constant pressure oil supply hydrostatic guideway is mainly controlled by the throttle fluid resistance and the oil chamber fluid resistance of the relative sliding gap of the oil chamber sealing surface. When the relative clearance of the moving parts, that is, the thickness of the oil film, decreases, it will lead to an increase in the oil resistance of the oil chamber. When the two moving parts are in direct contact, the oil resistance of the oil chamber approaches infinity[9]. Therefore, too small oil film thickness will increase the liquid resistance of lubricating oil and aggravate the viscous thermal friction of lubricating oil. With the increase of oil supply pressure, the flow rate of lubricating oil continues to increase, the heat generated by viscous heating is taken away, and the temperature of lubricating oil decreases. Luo Zan of Xiangtan University also found this

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phenomenon when he studied the dynamic pressure bearing, and made a theoretical analysis of this phenomenon, which is consistent with the analysis trend of this paper. When the oil film thickness of lubricating oil reaches 0.03 mm, the temperature of lubricating oil shows a steady upward trend.

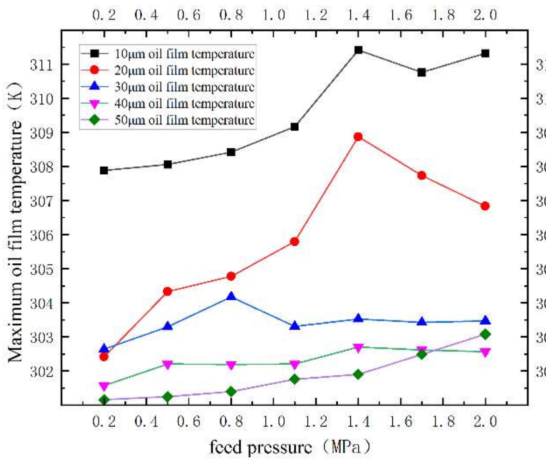

Figure 6. Oil film temperature statistics

4. Analysis of Oil Film Bearing Characteristics

Because the bearing capacity is the product of pressure and surface area, the actual bearing capacity of each oil pad bearing surface can be obtained by integrating the effective bearing surface of the oil pad. Through the surface integral function in ANSYS, 8 pre-set bearing surfaces are selected for Integral surface integral. The surface integral function will automatically calculate the bearing capacity of each surface after integration. It is verified that the bearing capacity calculated by this method is the same as that obtained by the traditional bearing capacity integral formula.

The guide rail has 8 symmetrically distributed oil cavities, so the highest bearing capacity side (A1) is selected for bearing characteristic analysis. The bearing capacity is shown in the following figure.

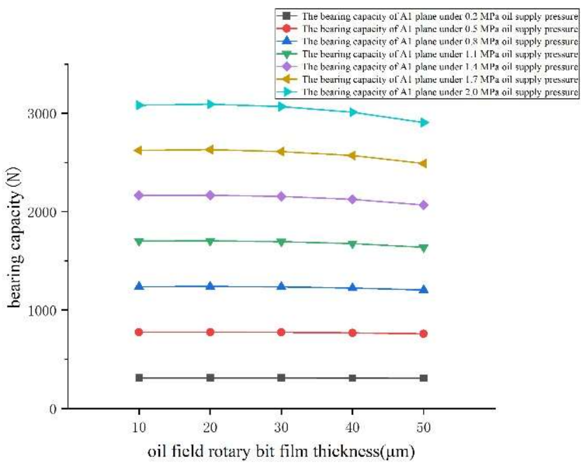

Figure 7. Analysis of bearing capacity of different oil film thickness

It is pointed out in the literature that the bearing capacity of the oil film is mainly determined by the oil supply pressure, and the change of the oil film thickness does not affect the bearing capacity of the oil film. It can be seen from Fig.7 that when the oil supply pressure is 0.2Mpa-1.1Mpa, the bearing capacity of the oil film on the A1 surface basically does not change. When the oil supply pressure rises, the bearing capacity of the oil film on the A1 surface decreases slightly with the increase of the oil film thickness. This phenomenon does not conform to Zhang Yongbin 's description of the change

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trend of the bearing capacity of the hydrostatic guide rail in the literature, so this is theoretically verified. The experimental hydrostatic guideway is symmetrically designed, and eight oil chambers are symmetrically distributed on the involute spline. The surface of each oil chamber can be regarded as a rectangular hydrostatic guideway. The calculation formula of the bearing capacity of the rectangular oil cavity is as follows:

$$W = PA \tag{1}$$

$$A = \frac{1}{4}(L + l)(B + b) \tag{2}$$

In the formula : P - the actual oil pressure on the bearing surface of the oil pad, A - the actual bearing area of a single oil pad.

When the constant pressure oil supply is adopted, the throttle is set before the oil inlet of each oil chamber. Here, taking the capillary throttle as an example, when the guide rail adopts the capillary throttle, under the rated oil supply pressure Ps, the pressure expression of the guide rail oil chamber is as follows:

$$\boldsymbol{p_e = \frac{p_s}{1 + \lambda}} \tag{3}$$

$$\boldsymbol{\lambda = \frac{64 l_e h_0^3}{3\pi d_e^4}\left(\frac{L + l}{B - b} + \frac{B + b}{L - l}\right)} \tag{4}$$

In the formula : h0 - design oil film thickness ; pe - oil chamber pressure ; λ - oil pad liquid resistance ratio ; μ-oil dynamic viscosity ; le - capillary length ; de - Capillary diameter.

In Equation (4), the capillary length, diameter, oil supply pressure and the design parameters of the oil pad are all quantitative, so that the oil chamber pressure and the oil supply pressure form an inverse relationship with the oil film gap. When the oil film thickness increases, the oil pad liquid resistance ratio increases, the oil chamber pressure decreases, and the oil pad bearing capacity decreases. Because the oil film gap is small, the change of oil film thickness has a relatively small effect on the oil cavity pressure.

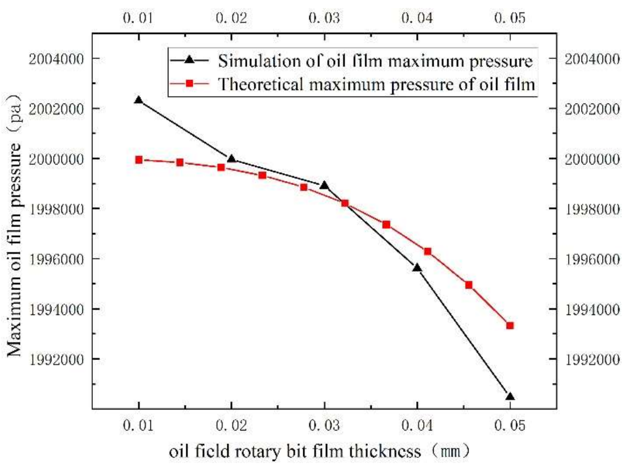

Figure 8. Comparison of maximum oil film pressure between theory and simulation

Through ANSYS simulation, it can also be found that as the oil film thickness increases, the actual lubricating oil pressure on the bearing surface of the oil pad will also decrease accordingly. In view of the 2Mpa oil supply pressure condition model with the most obvious decrease of bearing capacity in figure 8, the simulation experiments of five oil film thickness models are carried out, and the pressure distribution of lubricating oil film on the bearing surface of oil pad with different oil film thickness is obtained. The maximum pressure of theoretical oil film calculated by the theoretical model is compared, as shown in figure 8.

It can be seen from Fig.9 that the fluid simulation model is consistent with the theoretical model. As the oil film thickness increases, the maximum pressure of the oil film after the same oil supply pressure reaches the actual bearing surface of the oil pad becomes smaller, and the larger the oil film thickness, the faster the decreasing trend. As a result, the oil film bearing capacity shows a downward trend with the increase of oil film thickness, and the greater the oil supply pressure and the greater the oil film thickness, the more obvious the downward trend.

5. Conclusion

The setting of the oil film thickness of the hydrostatic guideway has a great influence on the temperature rise of the guideway lubricating oil. Too low oil film thickness will lead to increased viscous heat generation of the lubricating oil. At this time, if the oil supply pressure of the lubricating oil is small and the flow rate is slow, it will lead to a large temperature rise of the lubricating oil and seriously affect the physical properties of the guideway. When the oil film thickness of the lubricating oil increases, the viscous heat of the lubricating oil will be alleviated.

The bearing capacity of each tooth surface of the involute spline hydrostatic guideway can be approximated to the rectangular guideway for design calculation. When the oil film thickness is increased, under the condition of rated oil supply pressure, the increase of oil film thickness leads to the increase of oil liquid resistance ratio, the decrease of oil chamber pressure and the decrease of bearing capacity of oil pad bearing surface. The larger the oil film thickness is, the faster the pressure on the lubricating oil surface decreases, and the greater the influence on the bearing capacity is.

Through the preliminary consideration of the results of the above simulation experiments, the oil film thickness of the involute spline hydrostatic guideway should be designed between 0.02 mm and 0.04 mm, which can make the oil film of the involute spline guideway produce less temperature rise after thermal balance, and the bearing capacity of the oil pad bearing surface decreases less.

Acknowledgments

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