An Investigation into the Mechanical Properties of Hydrostatic Thrust Bearing for Constant Flow Circular Ring Oil Pad

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Abstract **- With the weight and volume increasing of the heavy/large equipment parts, the heavy CNC vertical lathe has become the ideal processing equipment for these parts. The main factor which affects the machining quality and efficiency of heavy CNC vertical lathe is the mechanical properties of the hydrostatic thrust bearing. This article do the research based on the large size circular ring oil pad's lubrication performance of the hydrostatic thrust bearing in the heavy/large equipment, establishing the lubrication performance distribution mathematical model of the velocity field, flow field, pressure field , etc., analysing the bearing behaviour of the large size circular ring oil pad. The results show that when the rotational speed of the hydrostatic thrust bearing is higher than a certain speed ω, the oil flow which is generated by the centrifugal force will be greater than that generated by the pressure difference. The friction heat is not easy to transfer out in the above situation, which restrains the bearing rotating speed of the hydrostatic thrust bearing. The paper author proposes that the temperature rise can be controlled by increasing the oil pressure or adding the cooling system which makes the oil in the oil-returning groove inflow to the resistive oil edge after cooling to the oil-returning groove in the third and fourth quadrant junction. The research provides powerful theoretical foundation for practical application of the large size circular ring oil pad hydrostatic thrust bearing, its structure design and operating reliability, realizing the lubrication performance prediction of the large size hydrostatic thrust bearing.**

Keywords - vertical lathe, constant flow, hydrostatic thrust bearing, circular ring oil pad, lubrication performance

I. INTRODUCTION

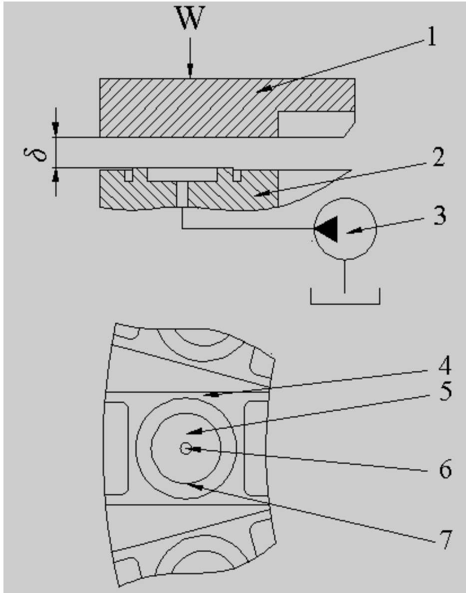
With the rapid development of modern industry, heavy/large CNC machining equipment are to the direction of high speed, high precision, high efficiency, high stability, high compound and high power, and the heavy/large hydrostatic thrust bearing whose performance directly affect the machining quality and efficiency of the equipment is the major part of the heavy/large numerical control processing equipment. The theory and application research of hydrostatic bearing technology got rapid development at home and abroad in recent years. In terms of the oil film temperature field, Gu adopted the digital particle image velocity measurement (DPIV)/temperature measuring experiment to study the small vertical gap's heat convection, measuring the fluid flow field of two-dimensional space and thermal field distribution [1]. Shul' Zhenko described the thermal field of thermal units in heater surface and derived the corresponding equations with parametric matrix structure [2]. Zhang do the research about the three-dimensional stress field and temperature field of water-lubricated high-speed dynamic and static pressure bearing, obtaining three-dimensional pressure distribution, temperature field distribution and dynamic characteristic parameters of dynamic and static pressure bearing under water lubrication [3]. Shao do the temperature field simulation and analysis of fan cavity and circular ring chamber heavy hydrostatic bearing [4]. Li do the characteristic analysis on heavy hydrostatic bearing of DVT500 vertical lathe [5]. In the fields of oil film pressure and oil film thickness, Zhu simplified N-S equation, calculated oil film bearing capacity and did the optimum design of oil recess used in hydrostatic bearing based on ANSYS[6]. Shao do the research about the oil film pressure field and oil film thickness affecting the lubrication performance of hydrostatic bearing under different overloading situations and different oil pad shape through the fluid finite element analysis software [7]. The research about hydrostatic thrust bearing lubrication performance which was done by Yu shows that oil cavity area and oil cavity depth have little influence on lubrication performance, but have certain influence on the temperature field[8]. In the aspect of the hydrostatic bearing performance, Ghosh discovered that the fluid inertia force and the fluid compressibility in oil chamber have certain influence on hydrostatic thrust bearing performance [9]. Fazil analysed the leakage through the neural network method and simulated the fluid static pressure bearing system with the fluid finite element software [10]. In the aspect of others, Li proposed using heat pipes as heat dissipation component of hydrostatic thrust bearing in high heat flux area to control thermal deformation of hydrostatic thrust bearing[11]. Wei got the vibration frequency of work turntable under different oil cavity number and spring stiffness through numerical simulation of the hydrostatic bearing supporting system work turntable[12]. Tang Put forward a new type oil-returning groove structure to solve the heavy single oil cavity hydrostatic thrust bearing's being unable to bear bias load of the heavy CNC vertical lathe work turntable[13]. Novikov established the annular hydrostatic thrust bearing's mathematical model and did the numerical analysis on its performance by using the low viscosity ecological clean fluid as lubricating oil[14]. Crabtree analysed the performance difference of the shallow oil cavity and the deep oil cavity[15].

II. HE WORKING PRINCIPLE OF CIRCULAR RING OIL PAD HYDROSTATIC THRUST BEARING

The working principle of circular ring oil pad hydrostatic thrust bearing is shown as Fig.1. The shunt puts oil into the same flow pressure oil. The same flow pressure oil that is put into the rectangular pad oil cavity pushes up the workbench and forms a layer oil film of hydrostatic bearing to realize the liquid lubricant. The clearance between the oil resistive edge of oil cavity and the workbench makes the oil pressure drop into environmental pressure. The workbench does rotation operation under the pressure oil film's supporting and the external force's driving. The hydrostatic thrust bearing has many advantages such as wide working speed range, no creeping phenomenon under low speed, high motion precision, low friction coefficient, low driving power, long working life, high static and dynamic stiffness, vibration-absorbing performance and high stability.

III. MECHANICAL PROPERTIES OF THE CIRCULAR RING OIL PAD HYDROSTATIC THRUST BEARING

The circular ring oil pad is shown as Fig.2. After the constant flow pressure oil is put into the sector pad oil cavity, the oil pressure is dropped into environmental pressure under two factors' comprehensive interaction that are the gap pressure difference of the circular ring oil pad hydrostatic thrust bearing between the workbench and the sealing side and the centrifugal force generated by workbench rotation. Therefore, the whole working status of the oil pad can be simplified as the following two conditions' combination(the gap flow generated by the pressure difference between the workbench and the circular ring oil pad and the gap flow generated by the sealing side's rotation whose rotation center is the hydrostatic thrust bearing rotation centre).



1- worktable; 2- guide; 3- pump; 4- oil-return slot; 5- oil cavity; 6- oil entrance hole; 7- oil resistive edge;
W- external load; δ- oil film thickness

Fig.1 The working principle of hydrostatic thrust bearing

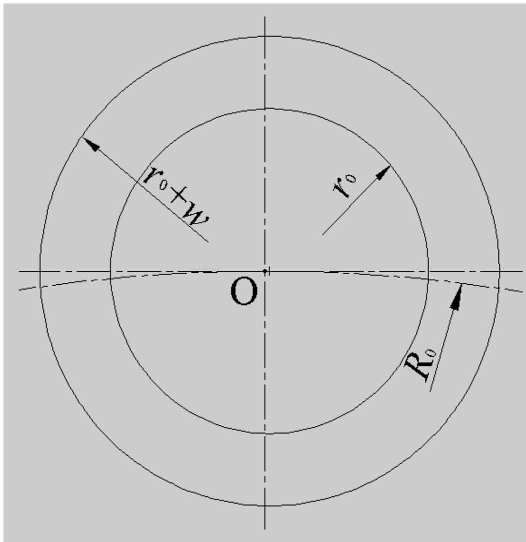


Fig.2 The circular ring oil pad structure

The fluid motion in hydrostatic thrust bearing is in line with the incompressible viscous flow law, so we can describe viscous incompressible fluid momentum conservation motion equation by using the expression of incompressible viscous flow according to the Newton second law, which is the Navier-Stokes equation, usually called N-S equation which is shown as Formula 1.

$$\begin{cases} \rho a_x = \mu\left[\frac{\partial^2 u_x}{\partial x^2} + \frac{\partial^2 u_x}{\partial y^2} + \frac{\partial^2 u_x}{\partial z^2}\right] - \frac{\partial p}{\partial x} + \rho g_x \\ \rho a_y = \mu\left[\frac{\partial^2 u_y}{\partial x^2} + \frac{\partial^2 u_y}{\partial y^2} + \frac{\partial^2 u_y}{\partial z^2}\right] - \frac{\partial p}{\partial y} + \rho g_y \\ \rho a_z = \mu\left[\frac{\partial^2 u_z}{\partial x^2} + \frac{\partial^2 u_z}{\partial y^2} + \frac{\partial^2 u_z}{\partial z^2}\right] - \frac{\partial p}{\partial z} + \rho g_z \end{cases} \tag{1}$$

A. The Gap Flow of Pressure Difference of Circular Ring Oil Pad

The working state is gap flow generated by the circular ring pressure difference between the workbench and the sealing side, when the hydrostatic thrust bearing workbench is lifted up by the pressure oil, which is shown as Fig.3. We suppose that the radius of the inner ring is r0, the radius of the outer ring is r0+w, gap height (Oil film thickness) is δ, oil pad cavity pressure is p0, the pressure of oil pad export is 0. we can deduce from the continuity equation, because l»δ, b»δ, ,uy=uz=0 and the Y direction length w is big. The oil is belong to incompressible fluid, so we can ignore its mass force and the N-S equation of the plate pressure gas flow can be inferred as follow.

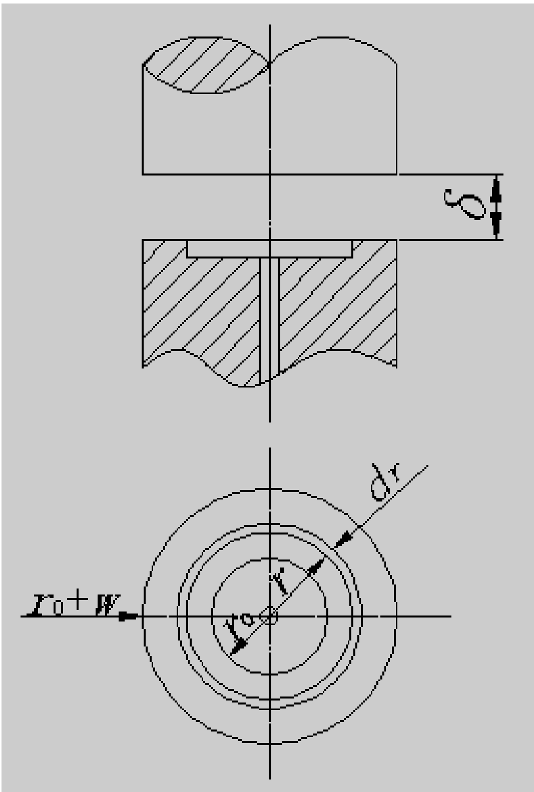


Fig.3 The gap flow of pressure difference of circular ring oil pad

$$\begin{cases} \mu\frac{\partial^2 u_x}{\partial z^2} - \frac{\partial p}{\partial x} = 0 \\ \frac{\partial p}{\partial y} = 0 \\ \frac{\partial p}{\partial z} = 0 \end{cases} \tag{2}$$

The formula (2) shows that the pressure p is only related to x, ux is only with z, so formula (2) can be simplified as follow. associated

$$\mu\frac{d^2 u_x}{dz^2} = \frac{dp}{dx} \tag{3}$$

After doing the quadratic integral for formula (3), we can deduce formula (4) with the boundary conditions that are ux=0 under the condition that z is 0 and ux=0 under the condition that z is δ.

$$u_x = \frac{(z-\delta)z}{2\mu}\frac{dp}{dx} \tag{4}$$

The flow qx through the whole plate clearance can be inferred as follow.

$$q_x = \int_0^\delta u_x w dz = -\frac{w\delta^3}{12\mu}\frac{dp}{dx} \tag{5}$$

The flow velocity(ur) equation of the radius r and the flow (qr) equation of the radius r can be inferred as formula (6) and formula (7) for the gap flow of pressure difference of circular ring oil pad.

$$u_r = \frac{(z-\delta)z}{2\mu}\frac{dp}{dr} \tag{6}$$

$$q_r = \int_0^\delta u_r b dz = -\frac{\pi r\delta^3}{6\mu}\frac{dp}{dr} \tag{7}$$

After doing the integral for formula (7), we can deduce the pressure p at any radius as formula (8) with the initial conditions that are r=r0 and p=p0.

$$p = p_0 - \frac{6\mu q_r}{\pi\delta^3}\ln\frac{r}{r_0} \tag{8}$$

When r is r0+w, p is 0, formula (8) can be inferred to formula (9).

$$q_r = \frac{\pi\delta^3 p_0}{6\mu\ln\left(1+\frac{w}{r_0}\right)} \tag{9}$$

Combined with formula (6), formula (8) and formula (9), we can deduce the flow velocity(ur) equation of the radius r and the pressure (pr) of the radius r as follow.

$$u_r = \frac{p_0(\delta - z)z}{2\mu r \ln\left(1+\frac{w}{r_0}\right)} \tag{10}$$

$$p_r = p_0 \frac{\ln \frac{r_0 + w}{r}}{\ln\left(1+\frac{w}{r_0}\right)} \tag{11}$$

The direction of the flow velocity(ur) equation of the radius r and the flow (qr) of the radius r is normal of oil cavity outline contour outward. The pressure (pr) of the radius r decreases linearly along to the oil cavity outline contour direction.

B. The Gap Flow of the Relative Rotation Movement

When the hydrostatic thrust bearing workbench is rotating, the sealing oil edge does the relative rotating gap flow taking the centre of the hydrostatic thrust bearing as the rotary centre. The total gap flow consists of the two parts which are the gap flow generated by the relative motion between the workbench and the parallel plate of the sealing side and the gap flow generated by the centrifugal force of the workbench's rotating. We put the rotary centre O of the hydrostatic thrust bearing workbench as the coordinate origin, the counter clockwise rotary angular velocity of the hydrostatic thrust bearing is ω, the gap height is δ, which is shown as Fig.4. Because uz is 0, p is 0, we can deduce , by the continuity equation. The size of x and y direction is large, therefore, we can suppose , . We can ignore the quality of the oil, because the oil belongs to the incompressible fluid. We can deduce the N-S equation under the relative rotation movement and the centrifugal force action according to the foresaid conditions.

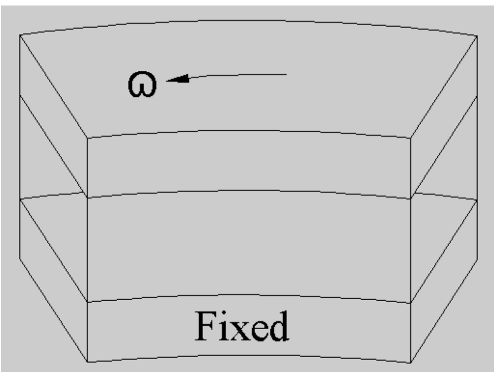


Fig.4 The gap flow of the relative rotation movement

$$\begin{cases} \rho a_x = \mu \dfrac{\partial^2 u_x}{\partial z^2} \\ \rho a_y = \mu \dfrac{\partial^2 u_y}{\partial z^2} \end{cases} \tag{12}$$

We can draw the conclusion that ux and uy are only related to z from formula (12), so we can simplify formula (12) as formula (13).

$$\begin{cases} \rho a_x = \mu \dfrac{d^2 u_x}{dz^2} \\ \rho a_y = \mu \dfrac{d^2 u_y}{dz^2} \end{cases} \tag{13}$$

Take a micro fluid whose distance to the centre of the thrust bearing is R, the angle with the vertical centre-line is φ (counter clockwise direction is positive), then the centrifugal acceleration of micro fluid is shown as formula (14).

$$\begin{cases} a_x = \omega^2 R \sin\varphi \\ a_y = -\omega^2 R \cos\varphi \end{cases} \tag{14}$$

After substituting formula (14) to formula (13) and doing the quadratic integral for formula (30), we can deduce the velocity equations with the boundary conditions that are z=0, ux=0, uy=0, z=δ, ux=-ωRcosφ, uy=-ωRsinφ.

$$\begin{cases} u_x = \dfrac{\rho\omega^2 R\sin\varphi}{2\mu} z^2 - \left(\dfrac{\rho\omega^2 R\delta\sin\varphi}{2\mu} + \dfrac{\omega R\cos\varphi}{\delta}\right) z \\ u_y = -\dfrac{\rho\omega^2 R\cos\varphi}{2\mu} z^2 + \left(\dfrac{\rho\omega^2 R\delta\cos\varphi}{2\mu} - \dfrac{\omega R\sin\varphi}{\delta}\right) z \end{cases} \tag{15}$$

Do the integral for formula (15) (z as the variable). The flow equations of the micro fluid through tiny width dy in the x direction and the micro fluid through tiny width dx in the y direction are shown as formula (16).

$$\begin{cases} q_x = -\left(\dfrac{\rho\omega^2 R\delta^3\sin\varphi}{12\mu} + \dfrac{\omega R\delta\cos\varphi}{2}\right) dy \\ q_y = \left(\dfrac{\rho\omega^2 R\delta^3\cos\varphi}{12\mu} - \dfrac{\omega R\delta\sin\varphi}{2}\right) dx \end{cases} \qquad 16)$$

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C. The Gap Flow in The Comprehensive Effect

Combined with formula (10), formula (15), formula (17) and Fig.5, we can deduce the speed of the arbitrary point of the circular ring oil pad, which is shown as formula (18).

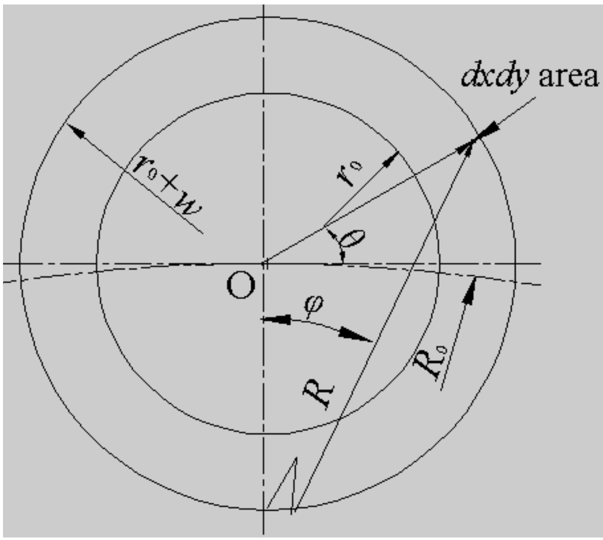

Fig.5 The dimension parameter schematic diagram of circular ring oil pad

$$\begin{cases} r\sin\theta = R\cos\varphi - R_0 \\ r\cos\theta = -R\sin\varphi \end{cases} \tag{17}$$

$$\begin{cases} \mathrm{u}_x = \dfrac{\rho\omega^2 R\sin\varphi}{2\mu}z^2 - \left(\dfrac{\rho\omega^2 R\delta\sin\varphi}{2\mu} + \dfrac{\omega R\cos\varphi}{\delta}\right)z - \dfrac{p_0(\delta - z)zR\sin\varphi}{2\mu\left(R^2 + R_0^2 - 2RR_0\cos\varphi\right)\ln\left(1 + \dfrac{w}{r_0}\right)} \\ \mathrm{u}_y = -\dfrac{\rho\omega^2 R\cos\varphi}{2\mu}z^2 + \left(\dfrac{\rho\omega^2 R\delta\cos\varphi}{2\mu} - \dfrac{\omega R\sin\varphi}{\delta}\right)z + \dfrac{p_0(\delta - z)z(R\cos\varphi - R_0)}{2\mu\left(R^2 + R_0^2 - 2RR_0\cos\varphi\right)\ln\left(1 + \dfrac{w}{r_0}\right)} \end{cases} \tag{18}$$

Combined with formula (9), formula (16) and formula (17), we can deduce the flow of any micro area δdxdy of the circular ring oil pad, which is shown as formula (19).

$$\begin{cases} \mathrm{q}_x = \left(\dfrac{\rho\omega^2\delta^3 r\cos\theta}{12\mu} - \dfrac{\omega\delta(R_0 + r\sin\theta)}{2} + \dfrac{\delta^3 p_0\cos\theta}{12\mu r\ln\left(1 + \dfrac{w}{r_0}\right)}\right)dy \\ \mathrm{q}_y = \left(\dfrac{\rho\omega^2\delta^3 (R_0 + r\sin\theta)}{12\mu} + \dfrac{\omega\delta r\cos\theta}{2} + \dfrac{\delta^3 p_0\sin\theta}{12\mu r\ln\left(1 + \dfrac{w}{r_0}\right)}\right)dx \end{cases} \tag{19}$$

Doing the integrals whose integral interval separately is 0～π/2, π/2～π, 3π/2～2π and 3π/2～2π after substituting the boundary conditions (dx=-rsinθdθ and dy=rcosθdθ) into formula (19), the flow of four quadrants can be obtained as follow.

The total flow of X direction and Y direction in the first quadrant can be inferred as formula (20):

$$\begin{cases} \mathrm{Q}_x = \dfrac{\pi\rho\omega^2\delta^3 r^2}{48\mu} - \dfrac{\omega\delta r(2R_0 + r)}{4} + \dfrac{\pi\delta^3 p_0}{48\mu\ln\left(1 + \dfrac{w}{r_0}\right)} \\ Q_y = \dfrac{\rho\omega^2\delta^3 r(4R_0 + \pi r)}{48\mu} + \dfrac{\omega\delta r^2}{4} + \dfrac{\pi\delta^3 p_0}{48\mu\ln\left(1 + \dfrac{w}{r_0}\right)} \end{cases} \tag{20}$$

The total flow of X direction and Y direction in the second quadrant can be inferred as formula (21)

$$\begin{cases} \mathrm{Q}_x = -\dfrac{\pi\rho\omega^2\delta^3 r^2}{48\mu} - \dfrac{\omega\delta r(2R_0 + r)}{4} - \dfrac{\pi\delta^3 p_0}{48\mu\ln\left(1 + \dfrac{w}{r_0}\right)} \\ Q_y = \dfrac{\rho\omega^2\delta^3 r(4R_0 + \pi r)}{48\mu} - \dfrac{\omega\delta r^2}{4} + \dfrac{\pi\delta^3 p_0}{48\mu\ln\left(1 + \dfrac{w}{r_0}\right)} \end{cases} \tag{21}$$

The total flow of X direction and Y direction in the third quadrant can be inferred as formula (22)

$$\begin{cases} \mathrm{Q}_x = -\dfrac{\pi\rho\omega^2\delta^3 r^2}{48\mu} - \dfrac{\omega\delta r(2R_0 - r)}{4} - \dfrac{\pi\delta^3 p_0}{48\mu\ln\left(1 + \dfrac{w}{r_0}\right)} \\ Q_y = \dfrac{\rho\omega^2\delta^3 r(4R_0 - \pi r)}{48\mu} - \dfrac{\omega\delta r^2}{4} - \dfrac{\pi\delta^3 p_0}{48\mu\ln\left(1 + \dfrac{w}{r_0}\right)} \end{cases} \tag{22}$$

The total flow of X direction and Y direction in the forth quadrant can be inferred as formula (23)

$$\begin{cases} \mathrm{Q}_x = \dfrac{\pi\rho\omega^2\delta^3 r^2}{48\mu} - \dfrac{\omega\delta r(2R_0 - r)}{4} + \dfrac{\pi\delta^3 p_0}{48\mu\ln\left(1 + \dfrac{w}{r_0}\right)} \\ Q_y = \dfrac{\rho\omega^2\delta^3 r(4R_0 - \pi r)}{48\mu} + \dfrac{\omega\delta r^2}{4} - \dfrac{\pi\delta^3 p_0}{48\mu\ln\left(1 + \dfrac{w}{r_0}\right)} \end{cases} \tag{23}$$

IV. THE MECHANICAL PROPERTY ANALYSIS OF THE HYDROSTATIC TRUST BEARING OF CONSTANT FLOW CIRCULAR RING OIL PAD

The article analyses the working process of the hydrostatic thrust bearing in clockwise. when the hydrostatic thrust bearing is rotating in anticlockwise, substitute the constant value ω with the constant value –ω into formula (20) , formula (21) and formula (23) and the flow of four quadrants in anticlockwise can be calculated.

(1) Formula (20) to (23) indicate that because of the relative rotation between workbench and base, influence of relative movement of workbench and sealing side's parallel plate and gap flow generated by centrifugal force of workbench's rotation is different in the four quadrants. All horizontal and vertical flow rates are different in the four quadrants.

(2) As is known from formula (20), for the first quadrant, as the rotating speed increasing, at first, the flow generated by pressure difference between workbench and sealing side is bigger than that generated by relative movement of parallel plate between workbench and sealing side. Then, gradually, the flow generated by centrifugal force of workbench's rotation becomes bigger than that generated by relative movement of parallel plate between workbench and sealing side and throughout the process, the flow of first quadrant is from oil chamber to oil-returning slot.

(3) As is known from formula (21), for the second quadrant, as the rotating speed increasing, the flows generated by pressure difference between workbench and sealing side, relative movement of parallel plate between workbench and sealing side, and centrifugal force of workbench' s rotation are all from oil chamber to oil-returning slot.

(4) As is known from formula (22), for the third quadrant, as the rotating speed increasing, at first, flow generated by pressure difference between workbench and sealing side and flow generated by relative movement of parallel plate between workbench and sealing side are bigger than that generated by centrifugal force of workbench's rotation. Then, gradually, the flow generated by centrifugal force of workbench's rotation becomes bigger the other two. So, in the process, flow of the third quadrant flows from oil chamber to oil-returning slot at first. Then as the rotating speed increasing, the flow flows back to oil chamber from oil-returning slot.

(5) As is known from formula (23), for the forth quadrant, as rotating speed increasing, at first, flow generated by pressure difference between workbench and sealing side is bigger than that generated by relative movement of parallel plate between workbench and sealing side and by centrifugal force of workbench's rotation. Then, gradually, flow generated by centrifugal force of workbench's rotation becomes bigger than the other two. So in this process, flow of forth quadrant flows from oil chamber to oil-returning slot at first. Then as the rotating speed increasing, the flow flows back to oil chamber from oil-returning slot.

(6) In the third and forth quadrant the flow flows from oil-returning slot to oil chamber. This is mainly because that in boundary area of third and forth quadrant, with the increasing of workbench's rotating speed, flow generated by centrifugal force is bigger than that generated by pressure difference between workbench and sealing side. In addition, the area of which flow flows from oil-returning slot to oil chamber becomes bigger. It causes the temperature rise become larger, especially when the oil level is lower than relative rotation acting surface, because the oil flow cannot flows from oil-returning slot to oil chamber, the temperature rise of this area would be larger. To decrease the temperature rise efficiently, cooling system could be added at the border of the third and forth quadrant to cool down oil in the oil-returning slot before it flows to oil chamber in order to restrict the temperature rise of this area.

(7) When the hydrostatic thrust bearings rotate clockwise, the results of first to forth quadrant are same as the second, first, forth, third quadrant when it rotate counter clockwise.

V. CONCLUSIONS

The oil flow of hydrostatic thrust bearing is driven by centrifugal force, pressure difference and the shear force between oil and plate caused by relative movement between them. So, for hydrostatic thrust bearing with circular oil pad, there exists a bottleneck speed ϖ. If the rotating speed of the hydrostatic thrust is higher than it, the oil flow generated by centrifugal force would be bigger than that generated by pressure difference. Thus, the heat caused by friction is difficult to sent out and it would restrict the rotating speed of hydrostatic thrust bearing. To increase the rotating speed of hydrostatic thrust bearing, oil pressure could be increased or cooling system could be added in oil-returning slot at border of the third and forth quadrant to cool down the oil before it flows to sealing side in order to restrict temperature rise in this area. This paper provides theoretical basis for the structure design, operating reliability and practical application of the large hydrostatic thrust bearing with rectangular oil pad. In addition, it could realize the lubrication performance prediction of the large hydrostatic thrust bearing.

CONFLICT OF INTEREST

The authors confirm that this article content has no conflicts of interest.

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