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Simulation and experimental study on lubrication characteristics of vertical hydrostatic guide rails①

Zhang Yanqin (张艳芹)②*, Yu Zeyang*, Chen Yao*, Shao Junpeng*, Yu Xiaodong*, Jiang Hui**
(* College of Mechanical & Power Engineering, Harbin University of Science and Technology, Harbin150080, P. R. China)
(** Qiqihar CNC Equipment CORP, LTD., Qiqihar 161005, P. R. China)

Abstract

In order to improve the cutting stiffness, the paper studies the vertical hydrostatic bearing in the slide when a ram is in feed process. The change of the oil film thickness on hydrostatic guide rail and the curve of the oil film thickness in various cutting forces are calculated and a relation model through theoretical analysis method is derived. The pressure field of the guide rail recess is simulated based on the finite volume method and demonstrated through experiments. The study is of vital theoretical significance for the improvement of machining accuracy of numerical control machines and the entire computer numerical control (CNC) equipment and provides valuable theoretical basis for the design of hydrostatic guide rail in engineering practice.

Key words: hydrostatic bearing, oil pad, supporting characteristics, finite volume method

0 Introduction

The hydrostatic guide rail is widely applied in aerospace industry, vessels, wind power equipment manufacturing enterprises and other fields for advantages of high bearing capacity, low friction coefficient, excellent damping property and strong stiffness. The vertical hydrostatic guide rail is the key part of the modern numerical equipment and the usual lubricating and supporting form of tool holder. Therefore, the performance will influence the machining accuracy of the whole machine.

Recently, most of the domestic and foreign scholars have studied the hydrostatic guide rail and the hydrostatic bearing from various perspectives. Jeon and Kim effectively analyzed the static characteristic and steady state characteristics of the linear hydrostatic guide rail by the software of ABAQUS, established a finite model of the oil film in guide rail to predict structural deformation and changes of the oil film which provide the basis for the theoretical design of the linear hydrostatic guide rail[1]. Okahata and Okuyama proposed that water lubrication is superior to the oil lubrication in precision machinery and constant flux oil supply is better than constant pressure oil supply, simulated the pressure of the oil pad in water hydrostatic bearing through ANSYS and FLUENT, and compared the proposed theoretical analysis model to the numerical results[2]. De Pellegrin, et al. pointed out that the bearing with constant external turbocharged prolonged the service life of the bearing and decreased the friction and wear, and simulated the influence of the size and shape of the groove on the bearing performance[3]. Nicodemus, et al. studied the theoretical influence that micro polar lubricants have on wear properties of hydrostatic bearing with 4-vessel and compensation of capillary resistor, simplified the lubricants with additive into micro polar fluid, and solved the flow equation and the Reynolds equation using the finite element method[4].

Domestic scholars Jiang Yun did simulation analysis of thermal deformation of the moving sideway in various motions and various conditions by the finite element analysis software ANSYS, taking the hydrostatic guide rail in ultra-precision machine tool as the research object. The influence of the guide rail's thermal deformation on the marching accuracy of the machine was studied and the method to decrease the thermal deformation was proposed[5]. Shao and Yu derived viscosity-temperature curve of the oil, compared the performances in different shape through numerical sim-

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② To whom correspondence should be addressed. E-mail: yinsi1016@163.com
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ulation and simulated the temperature field in various depth and velocity using computational fluid dynamics (CFD) and it was verified through experiments. The results shows that the rotation velocity and the viscosity have great influence on the temperature rise of the hydrostatic bearing and the influence of viscosity can not be neglected for the hydrostatic bearing in large linear velocity[6-11]. The author has done a large quantity work of the plane hydrostatic thrust bearing of round guide in the prior period, simulated the bearing characteristics and the lubrication characteristics of the hydrostatic bearing and presented the influence of the viscosity on the performance of the hydrostatic bearing[12,13].

All the research objects are on hydrostatic horizontal guide rail, research contents were emphasized in bearing behavior and fuel supply style from the reference literatures in China and other countries. But few research is available to lubrication characteristics of vertical hydrostatic guide rail, which directly influences the cutting performance of the whole machine. In the study, the vertical hydrostatic guide rail is taken as the research object and the mathematical model of the oil film thickness in various cutting forces is established based on summery of researches on performance of hydrostatic bearing. Then, the pressure field of the oil film is calculated, the curve between the cutting force and the oil film thickness in vertical hydrostatic guide rail is obtained, and the influence of the vertical hydrostatic guide rail's cutting force on the cutting stiffness is studied in order to improve the manufacturing accuracy of the whole machine.

1 Force analysis and the mathematical model of lubrication characteristics of the vertical hydrostatic guide rail

1.1 Force analysis of the vertical hydrostatic guide rail

The head ram moves in the slider and its extended length and cutting force is larger. So the oil film stiffness may be deficient and crushed. And it will cause manufacturing error, decrease the manufacturing accuracy of the whole machine and the product accuracy and surface quality will be lower than the design requirement. The theoretical analysis of force condition of the ram when maximum extended was done to solve the problem.

It is known that the full length of the ram L = 2550mm, the extended length x = 955mm, the maximum cutting force T = 100kN, the size of the cross section 320 × 320mm, the length of the upper cavity 585mm, and the structure and the force analysis when the ram was maximum extended is shown in Fig. 1.

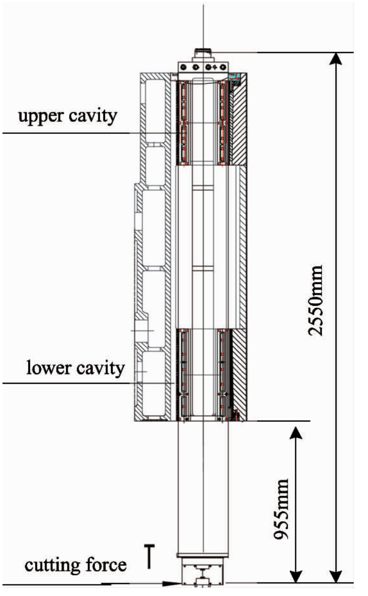

Fig. 1 The free body diagram when maximum extended

Both the upper and the lower have eight cavities. Every two is in a group and the cavities are divided into four sets. There are sixteen cavities in the upper and the lower and the cavity structure is shown in Fig. 2. Fig. 3 shows the effective area of a set of cavities, in which the effective area of a single cavity in the lower is $A_1 = 0.036\text{m}^2$ and the effective area in the upper is $A_2 = 0.033\text{m}^2$. The ram moved up and down in the guide rail driven by ball screw. The cutting force and its offset load is equilibrium to the lower right and the upper left whether the extended is long or not, when the raw moves with the cutting force in right horizontal direction through analysis.

Fig. 2 The oil recess structure

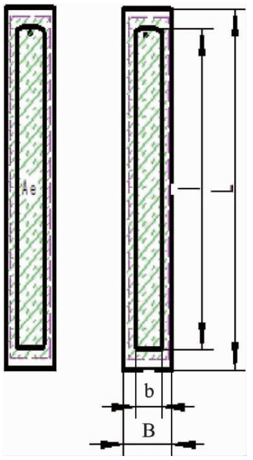

Fig. 3 The effective bearing area of the oil pad

1.2 Mathematical model of the lubrication characteristics

A_e is the effective bearing area of a single oil pad written as

$$A_e = \frac{1}{4}(L + L)(B + B) \tag{1}$$

where L, l, B, and b are the structure parameters of the oil pad in the vertical guide rail.

The load-carrying capacity W of a single pad is

$$W = \frac{3\mu Q[L^2 - l^2] \cdot [B^2 - b^2]}{2h^3[(L^2 - l^2) + (B^2 - b^2)]} \tag{2}$$

where, Q is the total flow of a single oil pad, μ is the lubricating oil viscosity, h is the thickness of the oil film.

In this study, the constant supply flow type is used in vertical hydrostatic guide rail and the relation between the recess pressure and the film thickness is

$$P = \frac{6\mu Q}{h^3\left(\frac{L + L}{B - b} + \frac{B + b}{L - l}\right)} \tag{3}$$

The inner flow can be approximated by a 3D-steady-incompressive flow under low speed motion during the numerical calculation and is laminar flow through calculation of the values of Reynolds number. The film shear heating can be neglected for the uniform and minimal velocity of the slider. It is assumed that there is no heat exchange and when adopting the laminar model, the control equation is:

The continuity equation is:

$$\frac{\partial \rho}{\partial t} + \nabla(\rho u) = 0 \tag{4}$$

The flow is the laminar and is expressed as $\frac{\partial \rho}{\partial t} = 0$, the above formula is

$$\nabla(\rho u) = 0 \tag{5}$$

where, ρ is the density of the flow, u is the velocity vector, ∇ is the divergence, then

$$\nabla(\rho u) = div(\rho u) = \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0 \tag{6}$$

where, u, v and w are the components of the velocity vector u in x, y and z directions respectively.

The momentum equation is

$$\frac{\partial(\rho u)}{\partial t} + div(\rho u u) = div(\mu grad u) - \frac{\partial p}{\partial x} + F_x \tag{7}$$

$$\frac{\partial(\rho v)}{\partial t} + div(\rho v u) = div(\mu grad v) - \frac{\partial p}{\partial y} + F_y \tag{8}$$

$$\frac{\partial(\rho w)}{\partial t} + div(\rho w u) = div(\mu grad w) - \frac{\partial p}{\partial z} + F_z \tag{9}$$

where, p is the pressure on the micro unit of flow, F_x, F_y, F_z, are the body forces of the micro unit of flow, μ is the viscosity of the flow, ρ is the density, u is the velocity vector, $grad(\,) = \partial(\,)/\partial x + \partial(\,)/\partial y + \partial(\,)/\partial z$.

All the above equations can be expanded as formulas in x, y, z directions, only the dead weight of the oil film F_z is considered when the body force is calculated, thus, $F_x = F_y = 0$. The dynamic viscosity μ is set as $0.028\ p_a \cdot s$ and the density ρ is $900\,kg/m^3$.

2 The numerical simulation of the oil film in the vertical hydrostatic guide rail

The study adopts the numerical simulation of the limit working condition of the cutting force $T = 100\,kN$ of the constant supply flow type to study the influence that the changes of the oil film thickness in various cutting forces have on the manufacturing accuracy of the whole machine. The inlet flow of a single oil chamber $Q = 0.52\,L/min$, the thickness of the oil film $h = 0.023\,mm$, the depth of the oil chamber is 2mm, and the pressure distribution of the lubricant oil film are obtained after iteration convergence. The pressure field caused by the cutting force occurrs in the upper and lower guide rail lubricant on the left and right as the cutting force in right horizontal direction. The left is equilibrium to the right after stability, the pressure field on the left and the numerical calculation are shown in Figs 4 and 5.

Both the high pressure region on the upper and lower oil pads is distributed in the oil chamber and shows gradual lowering trend from the cavity to the resistive oil edge, and the pressure on the upper is higher than the lower for the influence of the cutting force.

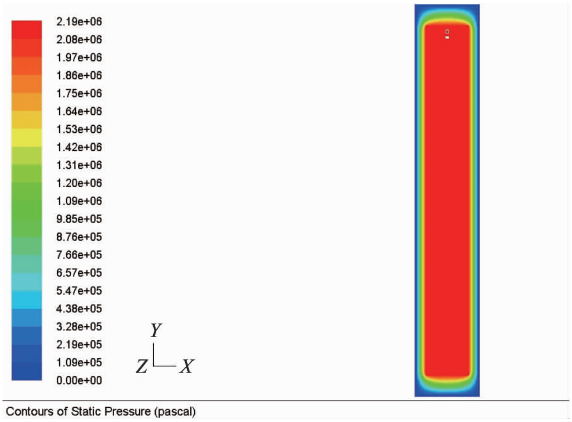

Fig. 4 The oil film pressure fields of the upper lubricant

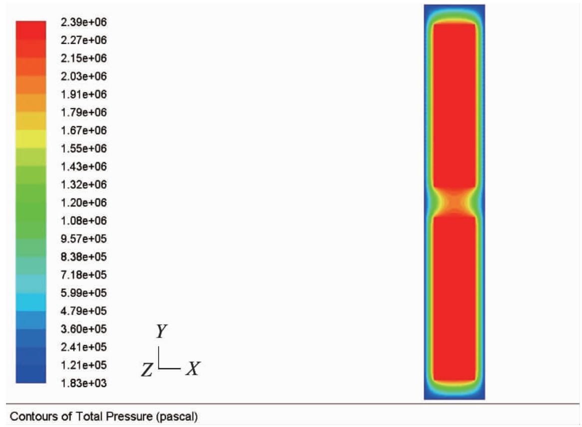

Fig. 5 The oil film pressure fields of the lower lubricant

In addition, the pressure is simulated when the cutting force T is 10kN、20kN、30kN、40kN、50kN、60kN、70kN、80kN、90kN. The simulation values and the theoretical values are shown in Table 1.

Table 1 The theoretical values of the film thickness in various cutting forces

Cutting force (kN)	10	20	30	40	50	60	70	80	90	100
The pressure in the upper cavity (MPa)	0.20	0.43	0.58	0.84	1.08	1.24	1.46	1.70	1.96	2.19
The pressure in the lower cavity (MPa)	0.26	0.52	0.74	0.98	1.19	1.41	1.67	1.92	2.14	2.39
The film thickness (μm)	50.84	40.35	35.25	32.02	29.73	27.97	26.57	25.42	24.44	23.59

The curve between the cutting force and the film thickness is obtained based on the theoritical value in Table 1, and it is shown in Fig. 6.

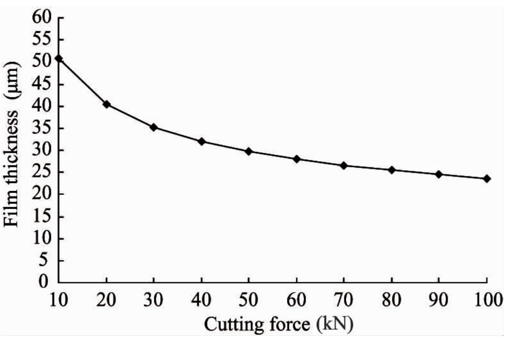

Fig. 6 The theoretical curve between the cutting force and the film thickness

3 Experimental study on the vertical hydrostatic guide rail and comparative analysis of the results

The oil film thickness of the vertical hydrostatic guide rail is measured on the Double-pillar Vertical Lathe DVT500 × 50/100Q-NC when the cutting forces of the ram head are 10kN、20kN、30kN、40kN、50kN、60kN、70kN、80kN、90kN、100kN. The test devices of vertical hydrostatic guide rail are shown in Fig. 7 and Fig. 8 including a micrometer gauge, a high precision level bar and a vertical bar. The level bar and vertical bar are vertically placed with the micrometer gauge fixed on the vertical bar and the probe contacted the slide. Taking the vertical surface as the reference plane, the data read after the slider is stable under each working condition.

Fig. 7 The test device of vertical hydrostatic guide rail

Fig. 8 Testing instrument

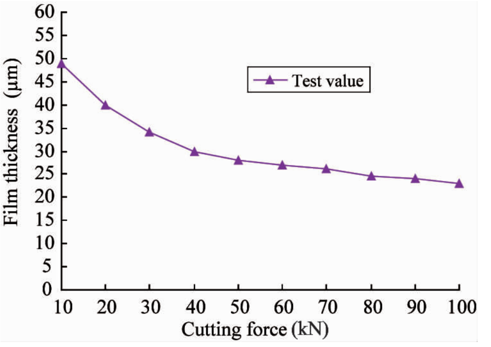

Fig. 9 The experimental curve between the cutting force and the film thickness

The oil film thickness under various cutting forces are obtained based on the test datum under various operation conditions (Table 2) and compared with the theoretical calculation.

The relationship between the cutting force and the oil film thickness in vertical hydrostatic guide rail is drawn according to the data in Table 2 shown in Fig. 9.

The theoretical value and the test values in Tables 1 and 2 show that experimental error is less than 5.0% and the main error sources include pipeline loss and the surface error.

Table 2 The data table of the cutting force and the film thickness under various cutting forces

Cutting force (kN)	10	20	30	40	50	60	70	80	90	100
Film thickness test (μm)	49.1	39.8	34.0	30.2	28.5	27.1	26.1	24.5	24.0	23.0

4 Conclusions

In this paper, the mathematical model of the lubrication characteristics of the vertical hydrostatic guide rail is established and the relationship between the recess pressure and the film thickness is derived. The study provides theoretical basis for the design of hydrostatic guide rail. Experimental results show that when the cutting force is less than 40MPa, with the increase of cutting force the thickness of the hydrostatic oil film changes greatly and the oil film stiffness increases; while when the cutting force is more than 40MPa, with the increase of cutting force the thickness of the hydrostatic oil film changes little, and the oil film has higher anti – compression ability, which results in the higher manufacturing accuracy. This conclusion is consistent with the theoretical calculation.

The inner pressure filed in vertical hydrostatic guide rail and the relationship between the cutting force and the oil film thickness is obtained through the finite volume method and theoretical analysis. The conclusion is of great significance for improving the machining accuracy of numerical control machine.

References

[1] Jeon S Y, Kim K H. A fluid film model for finite element analysis of structures with linear hydrostatic bearings. *Journal of Mechanical Engineering Science*, 2004, 218 (3):309-316

[2] Okuyama O G, Akinori S Y. Numerical study on constant-flow hydrostatic water bearing for a machine-tool table. *Advanced Materials Research*, 2011,325:357-362

[3] Pellegrin De, Dennis V, Hargreaves D J. An isoviscous, isothermal model investigating the influence of hydrostatic recesses on a spring-supported tilting pad thrust bearing. *Tribology International*, 2012,51(6):25-35

[4] Rajasekhar N E, Sharma S C. Influence of wear on the performance of multirecess hydrostatic journal bearing operating with micropolar lubricant. *Journal of Tribology*, 2010,132(2): 1-11

[5] Jiang Y, Hou G A, Sun T. Thermal finite element analysis of hydrostatic guide. *Avlation Precision Manufacturing Technology*, 2011,47(5):23-26

[6] Yu X D, Qiu Z X, Li H H, et al. Lubrication performance and velocity characteristic of a multi-oil-pad hydrostatic thrust bearing with a sector-shape cavity. *Journal of Engineering for Thermal Energy and Power*, 2013, 28 (3):296-300 (in Chinese)

[7] Shao J P. The effect of oil cavity depth on temperature field in heavy hydrostatic thrust bearing. *Journal of Hydrodynamics*, 2011, 123(5):676-680

[8] Yu X D, Meng X L, Li Y H. Lubricating characteristics of high speed circular tilting pad thrust bearing. *Journal of Agricultural Machinery*, 20007, 38(2): 204-207

[9] Shao J P, Zhang Y Q, Li Y H. Influence of load capacity on hydraulic journal support deformation in finite element calculation. *Journal of Central South University of Technology*, 2008, 15(2): 245-249

[10] Shao J P, Zhang Y Q, Dai C X. Numerical simulation analysis of sector and circular oil recess temperature field of heavy hydrostatic. *Journal of Hydrodynamics (Ser. A)*, 2009, 24(1): 119-123

[11] Yu X D, Meng X L, Wu B. Study on the modulus of elasticity of composite pad bushed thrust-bearing. *Journal of Machine Design*, 2007, 24(2): 22-24

[12] Zhang Y Q, Shao J P, Ni S Q. Numerical simulation of temperature field in large size hydrostatic bearing. *China Mechanical Engineering*, 2008, 19: 563-565 (in Chinese)

[13] Zhang Y Q, Yang X D Xu X Q. Viscosity influence research on load capacity of heavy hydrostatic bearing. *Key Engineering Materials*, 2010, 450: 63-66

Zhang Yanqin, born in 1981. She received the Ph. D degree in the specialty of machinery manufacturing and automation from Harbin University of Science and Technology in 2009. She was associate professor. She has been engaged in the teaching and researches in the NC machine key technology and hydrostatic bearing technology.