Study to Improve the Spindle Bearing Stiffness of Medium External Cylindrical Grinding Machines Based on Numerical Simulation of Hydrostatic Lubrication

Van-Hung Pham**, Tuan-Anh Bui, Thuy-Duong Nguyen
Hanoi University of Science and Technology, No. 1, Dai Co Viet, Hai Ba Trung, Hanoi, Viet Nam
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Abstract

Medium-sized cylindrical grinding machines are fine-machining machine tools, which are widely used in the manufacturing industry. Accuracy of grinding process depends mainly on the precision and working quality of the spindle unit, in which the stiffness of the spindle bearings plays an important role. On the spindle bearings of external cylindrical grinding machine, the machine is usually uses hydrodynamic bearing with three self-aligning pad, which allows the machining workpiece to achieve a rounded and accurate diameter up to 0.005 mm. Based on the hydrostatic lubrication theory as well as the current fabrication and fabrication capabilities of the self-aligning pad of hydrodynamic bearing spindle, this paper presents the results of calculation and numerical simulation to improve the bearing film stiffness of the medium cylindrical grinding machines with a reasonable hydrostatic parameters.

Keywords: External cylindrical grinding machine; Spindle bearing stiffness; Hydrostatic lubrication.

1. Introduction

The spindle unit of medium-sized external circular grinding machine is a key unit that directly affects machine precision, especially in fine and finishing machining processes. Most of medium external grinding machines are using a hydrodynamic bearing spindle with three self-aligning pads (shown in Fig. 1). In comparison to the self-aligning pad bearing spindle, the hydrostatic bearing drives in more stable spindle center when the radial load is changed (the cutting depth is reduced by machining time) and it is in accordance with the operating condition of the external grinding machine with a stable speed of rotation.

One of the important technical requirements of external cylindrical grinding machines is the stiffness of the spindle unit.

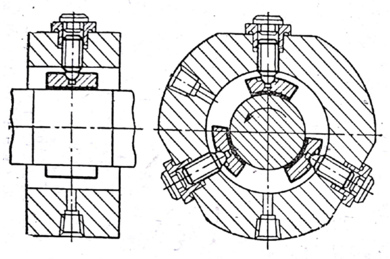


Fig. 1. Cross section of a self-aligning hydrodynamic bearing on an external cylindrical grinding machine [1].

* Corresponding author: Tel: (+84) 913359081
Email: hung.phamvan@hust.edu.vn

One of the important technical requirements of external cylindrical grinding machines is the stiffness of the spindle unit. The spindle unit stiffness is determined by the ratio of the force causing displacement and the displacement value of the spindle. A greater stiffness results in a higher precision level of the machine, and a higher precision machining of workpiece. Therefore, enhancement of the spindle stiffness of the external grinding machines is the regular desire of researchers. With self-aligning pad, hydrodynamic spindle clamping units on existing cylindrical grinding machines, the continuous improvement of stiffness has certain difficulties due to the bearing film stiffness depending on the film thickness. Besides, it varies according to machining conditions and depends on the mechanical structure of the self-aligned hemispherical joints, which wear out over time

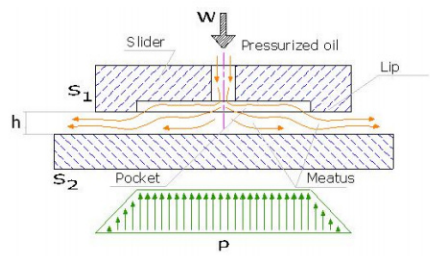


Fig. 2. Hydrostatic Lubrication with High Pressure Oil Film [2]

The hydrostatic bearing shown in Fig. 2 has structural features with fixed high-pressure oil chambers. It shows a more stable film thickness and a higher rigidity in comparison with a hydrodynamic bearing, and it does not depend on the bearing structure. Thus, the hydrostatic spindle bearing is one of the most suitable solutions for the purpose of improving the spindle unit stiffness of the grinding machine with the efficient aid of precision machining of the oil chambers by CNC machines.

At present, there are many researches on the application of hydrostatic bearing spindle for machine tools. The popular studies are upgrading and conversing spindle units using ball bearings into hydrostatic spindle units. The results show that there is a good dynamic response both in simulation and experiment. In 2016, He Qiang et al. presented a numerical and experimental method to select the parameters and fabricate the hydrostatic spindle unit to replace the ball bearing spindle in a vertical machining equipment, whose operating speed is 800 rpm [3]. S. Uberti et al. presents a study on design and manufacture of testing benches for inspection and assessment of a hydrostatic bearing applied in a linear moving spindle, which enables to carry out the tests to reduce vibration and determine the stiffness of the hydrostatic bearing and to improve machining accuracy [2].

W. Chen et al. designed a hydrostatic bearing for a spindle milling machine from the dynamic point of view. The conducted machining experiments shows a correspondence between the spindle structure and the dynamic parameters, including the stiffness [4].

In 2012, Dong ju Chen et al. presented a study on static and dynamic behavior of a shaft supported by hydrostatic bearings. They analyzed the effect of imbalanced vibration on the machining accuracy. The imbalance-induced force in two directions is derived from the dynamic results [5]. This research shows that the location and stiffness of the bearing affect the machining accuracy of workpiece.

2. Improvement of spindle stiffness of medium-size external circular grinding machine with hydrostatic lubrication solution

The structure and principle operation of the hydrostatic spindle unit used for the medium-sized outer grinding machine is shown in Fig. 3 [3]. The wet lubrication was performed by the lubricant in four high pressure recesses provided by a pump system through a restrictor that completely separates the shaft and bearing surfaces. The center of the spindle coincides with the center of the bearing. With respect to the weight of the shaft, the external load (P), the effective area of the oil recess F, and the eccentricity e, the equilibrium force equation can be written as:

$$P = \left(p_3 - p_1\right).F \tag{1}$$

where p_1 and p_3 present the oil pressure of the recess 1 and 3, respectively.

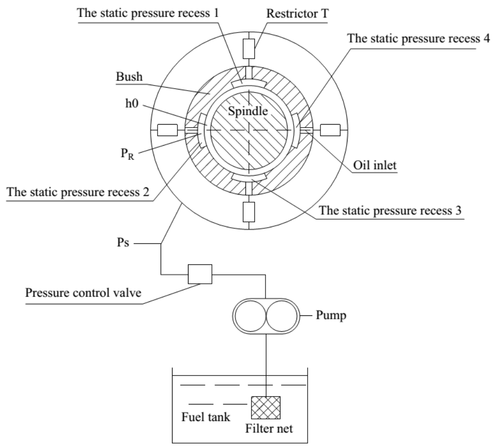


Fig. 3. The structure of a constant pressure oil supply hydrostatic bearing on machine tools [3]

In fact, with hydrostatic spindle bearing and the Reynold's assumption that *e* is very small.

Based on the law of conservation of mass, conservation of energy, the Reynold equations for radial and axial drive are given as follows:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial z}\right)=6r\frac{\partial h}{\partial x} \tag{2}$$

And the dimensionless equation

$$\frac{\partial}{\partial \varphi}\left(\frac{\overline{h}^3}{\overline{\mu}}\frac{\partial \overline{p}}{\partial \varphi}\right)+\lambda^2\frac{\partial}{\partial \overline{y}}\left(\frac{\overline{h}^3}{\overline{\mu}}\frac{\partial \overline{p}}{\partial \overline{y}}\right)=3\overline{\mu}\overline{\omega}\frac{d\overline{h}}{d\varphi} \tag{3}$$

where λ = L/D: ratio of length and diameter of the bearing

φ: angle coordinates;

p- oil film pressure, *y*-radial coordinates,

$\overline{y},\overline{p},\overline{h},\overline{\mu},\overline{\omega}$: dimensionless parameters.

The lubricant film thickness is determined by equation [5]:

$$h=h_o(1-\mathcal{E}\cos\varphi) \tag{4}$$

where, h – film thickness,

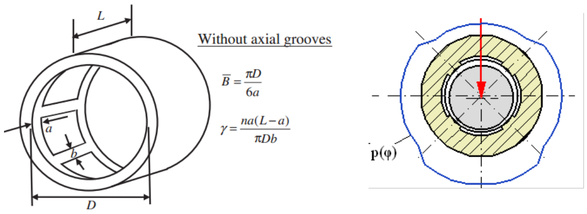
h_o – film thickness under line eccentricity,

φ - angular position from the line of eccentricity,

$\mathcal{E} = e/h_o$ – *eccentricity ratio*.

2.1 Basic parameters of hydrostatic spindle bearing of medium-sized grinding machine with D=70 mm and n= 3000 rpm

Select the hydrostatic bearing structure without oil drainage, having four oil recess as shown in Fig. 4.



(a) (b)

Fig. 4. (a) Hydrostatic bearing structure diagram without oil drainage; (b) Load diagram of hydrostatic bearing with 4 oil recess

Based on the geometrical geometry of the hydrodynamic bearing of the external circular grinding machine, the dimensions of length and distance between two adjacent recesses *b* are respectively chosen as follows:

$$L = 0.8D;\ b = a = 0.25L$$

To ensure the load bearing capacity of the spindle when upgrading of using hydrostatic bearing, the technological ability of the grinding wheel being suitable with the current machine structure, the same diameters of the shaft and grinding wheel with the hydraulic bearing must be chosen. To ensure a completely wet lubrication with the bearing parameters including the spindle diameter of 70 mm, operating at rotary speed of 3000 rpm, and total shaft length of 535 mm, the parameters *a* and *b* must be chosen as the same 14mm [6].

The shape and size of the oil chamber have a certain influence on bearing working ability with high rotational frequency, whereby the lubricant can be swept out of the oil recess to create turbulent effects or hydrodynamic effects resulting in large losses in the oil recess. To overcome this issue, the recess depth must be increased larger than 20Δ. Thus, the oil recess is set below the eccentric (e), the net hydrodynamic flow [5] is zero.

Using hydrostatic bearing for the grinding machine spindle, the minimum oil film thickness to ensure a completely wet lubrication condition is calculated by equation as follows:

$$h_{omin} = R_{zT} + R_{zB} + \gamma_b \leq \Delta_{min}/3, \qquad (5)$$

where: R_{zT} - surface roughness of bearing journal;

R_{zB} – surface roughness of bearing;

$\gamma_b = 2 - 3\ \mu m$ – Reserve distance to keep the oil film uninterrupted.

The bearing and bearing journal surfaces are fine machined with surface roughness class of 8 -10, which is according to the TCVN 2511, so the surface roughness is about 3.2 – 0.8 µm. Thus, the minimum oil film thickness is about 9 µm. Hence, Δ_{min} can be chosen as 27 µm. The limited relative eccentricity ε of a hydrostatic bearing is usually 0.4. The relative eccentricity of the grinding machine hydrostatic spindle can be chosen as $\mathcal{E} = 2e/\Delta = 0.3$

So, the eccentricity e of the grinding machine hydrostatic spindle can be calculated as 4.5 µm. And we can write:

$$h_o(L) = \Delta/2 = 15\ \mu m$$

$$h_o(U) = 1.5\, h_o(L) = 22.5\ \mu m \qquad (6)$$

where: $h_o(L)$ - Lower limit of clearance (h_{min}) corresponding to β=0.4

$h_o(U)$ - Upper limit of clearance corresponding to $\beta=0.7$ [5]

$\beta = p_r/p_s$ - Ratio of oil chamber and pump pressure

In addition, the power ratio can be written as

$K = H_f/H_p$, or $H_f = H_p(1+K)$, and $K = 3$ with spindle machine tool. (7)

where: H_t – total power,

$H_f = \eta.\ A_f U^2/h$ - friction power,

$H_p = p_s.q$ - hydrostatic pumping power,

$A_f = A - 3/4A_r$ - friction area recessed bearing,

A_r – *recess area*, $A = \pi DL$ - total area of the sliding contact.

The pump pressure of the hydrostatic system can be estimated as:

$$p_s = \frac{W}{LDW_n} \qquad (8)$$

where: W_n – loading factor,

According to the experience of manufactures, the oil recess pressure p_r in accordance with the ability of manufacturing technology is in the range of 1-5 MN/m^2, and the best pressure is in the range of 1-2 MN/m^2. Hence, the lubricant recess pressure of the spindle bearing of external cylindrical grinding machine is chosen as 2 MN/m^2.

For the lower limit of clearance about 15 µm, the pump pressure p_s is determined as 5MN/m^2.

Similarly, the pump pressure approximates 2.85 MN/m^2 is also calculated with the upper limit of clearance about 22.5 µm.

The lubricant viscosity η is determined by equation as: $\eta = S_o / (n/p_s. (D/2h_o)^2)$ (9)

Thus, the viscosity changes in the range of 1.25 $x10^{-3}$- 1.60 Pa.S

The concentric flow q_{oL} is estimated by equation as follows: $q_{oL} = p_s.h_o^3. \beta.B_n/\eta$ (10)

and it is about 14.148x$10^{-6}$$m^3/s$ for the designed system.

For the bearing with four recesses, the lubricant flow per restrictor q_{res} can be expressed as follows:

$$q_{res} = q_{oL}/4 \qquad (11)$$

In the case of upper limit of clearance, the flow q_{oU} is estimated about 37.211x10^{-6} m^3/s. So, the flow per restrictor q_{res} about 9.302x10^{-6} m^3/s is also indicated.

Besides, the hydrostatic pumping power H_p can be calculated as: $H_p = p_s.q$

And with $K = 3$, the total power is

$$H_t = H_p(1+K) = 0.3\ kW$$

For light machine oil, maximum temperature rise is increased according to the power ratio, and it is said to be as

$$\Delta T = 2.2 \times 10^{-6} \times p_s = 11^o\ C \qquad (12)$$

2.2 Numerical simulation of spindle bearing stiffness changing of the medium-sized grinding machine by the main parameters

General equation for hydrostatic film stiffness presented by Rowe [7,8] with any number of recesses and control structure is written as follows:

$$J = \frac{p_s.L.D}{h_0}.\frac{3N^2}{2\pi}.\frac{\beta\left(1-\frac{a}{L}\right)\sin^2\left(\frac{\pi}{N}\right)}{z+1+2\gamma.\sin^2\left(\frac{\pi}{N}\right)} \qquad (13)$$

or $$J = \frac{p_s L.D}{h_0}J_n \qquad (14)$$

where, N - Number of recesses,

$\gamma = \frac{N.a(L-a)}{\pi.D.b}$ - Bearing shape factor,

z = ½ . $[\beta/(1-\beta)]$ – for orifice control,

$z = [\beta/(1-\beta)]$ – for capillary control.

The bearing film stiffness at eccentricity ratio of 0.3 with orifice control and other main parameters for the simulation in this study include $N = 4$, $D=70$ mm, $a = 14$ mm, $\theta = 30^0$, $L = 56$ mm, $h_o = 15 – 22.5$ µm, $P_s = 2.85 – 5\ MN/m^2$, $\beta = 0.4 – 0.7$, $\eta = 1.25x10^{-3} – 1.60x10^{-3}$ Pa.S, $q_o = 14.148\ x10^{-6} – 37.211x10^{-6}\ m^3/s$.

With four recesses and orifice control:

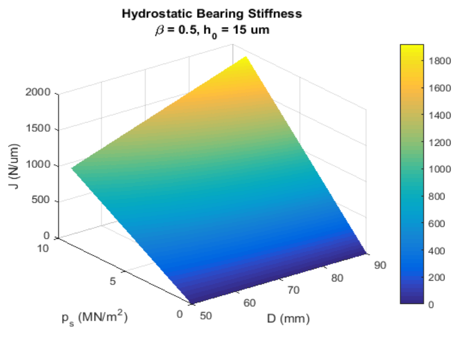
$$J_n = 7.65K_{bs}\frac{\beta(1-\beta)}{2-\beta+2\gamma(1-\beta)} \qquad (15)$$

where $K_{bs} = (1-a/L) = 0,75$.

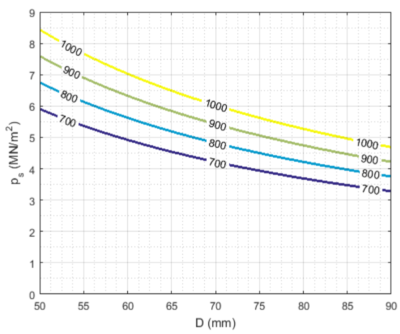
Hence, $$J_n = 5.74\frac{\beta(1-\beta)}{2-\beta+1.52(1-\beta)}$$

The numerical simulations were carried out by the MATLAB software to show the relationship between the spindle bearing stiffness and the main parameters of the machine

2.2.1 The hydrostatic film stiffness J depends on p_s and D with $\beta = 0.5$, $h_{oL} = 15$ µm and $h_{oU} = 22.5$ µm



(a)



(b)

Fig. 5. Hydrostatic film stiffness J with β = 0.5, h_{oL} = 15 µm

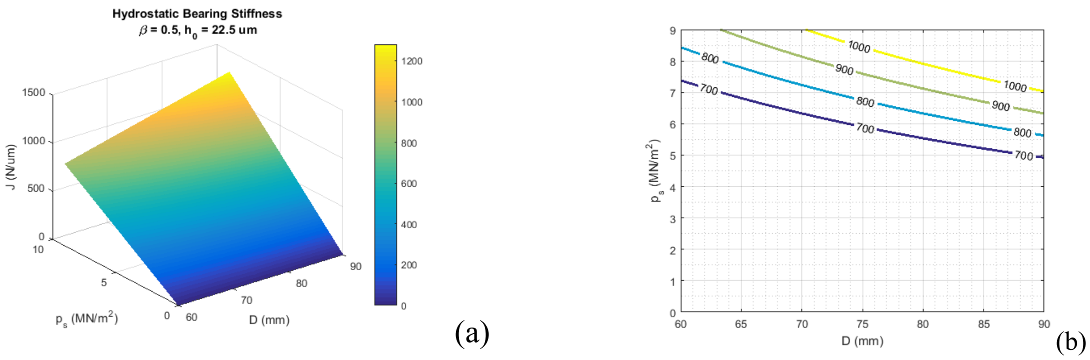


Fig. 6. Hydrostatic film stiffness J with β = 0.5 and h_{oU} = 22.5 μm

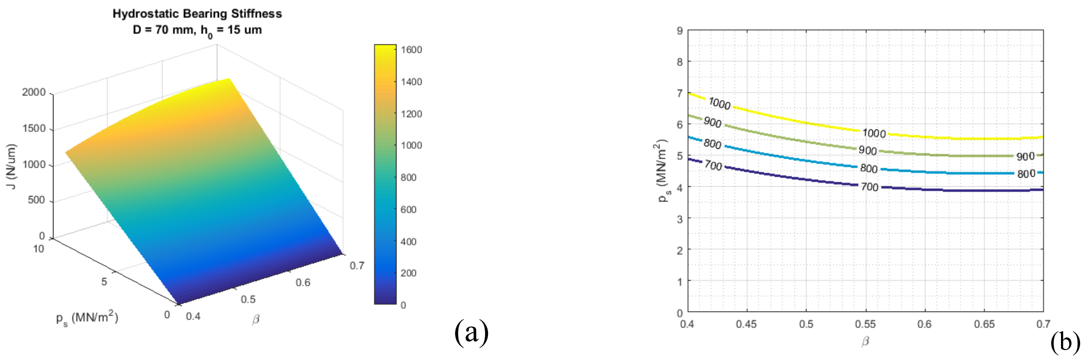


Fig. 7. Hydrostatic film stiffness J with D = 70mm and h_{oL} = 15 μm

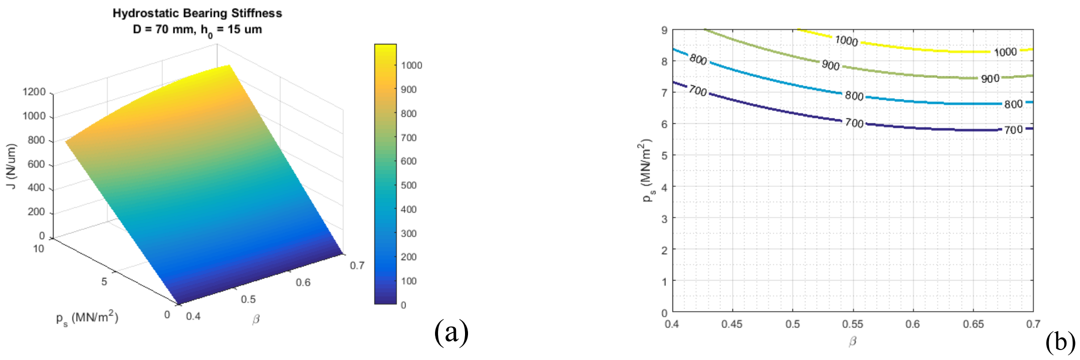


Fig. 8. Hydrostatic film stiffness J with D = 70 mm and h_{oU} = 22.5 μm

Changing the clearance from the lower limit to upper limit corresponding to 15 μm and 22.5 μm, respectively. The simulation results are shown in Fig. 5 and Fig. 6. Hence, Fig. 5 (a) and Fig. 6 (a) indicate that with the same spindle diameter, the hydrostatic bearing stiffness much depends on the lubricant film pressure and the radial clearance. Increasing the oil film pressure results in increasing the bearing stiffness. However, in both cases the radial clearance of 15 and 22 μm with other similar conditions, the stiffness of the hydrostatic bearing is greater in the case of small radial clearance. The results are clearly shown in Fig. 5(b), Fig. 6(b).

2.2.2 The hydrostatic film stiffness J depends on p_s and β with D = 70mm, h_{oL} = 15 μm and ho_U = 22.5 μm

The simulation results of the dependence of hydrostatic bearing stiffness on oil recess pressure and β on the change in radial clearance from 15 to 22.5 μm are shown in Fig. 7 and Fig. 8. The results indicate that for a given value of β, the hardness of the hydrostatic bearing depends on the change in the recess pressure. Indeed, increasing the lubricant recess pressure leads to increasing the bearing stiffness. However, with the same oil recess pressure, the hydrostatic bearing stiffness in the case of radial clearance of 15 μm is greater than that of the radial gap of 22.5 μm (Fig. 7(b), Fig. 8(b)).

2.2.3 The hydrostatic film stiffness J depends on h_0 and p_s with D = 70mm, β = 0,5

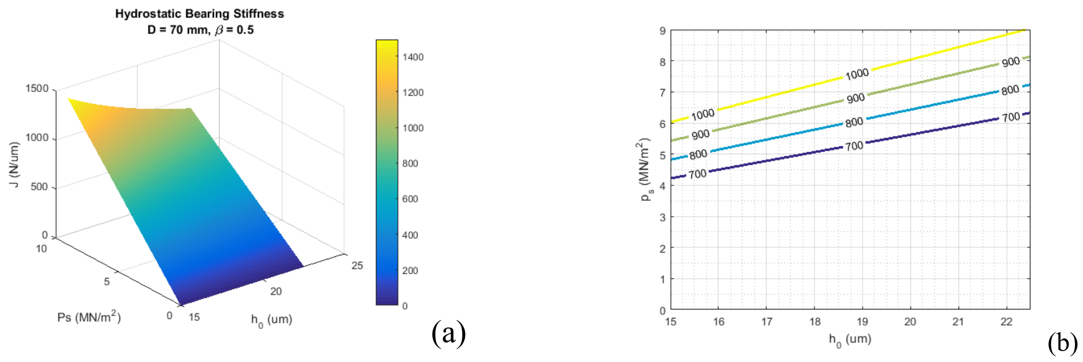


Fig. 9. Hydrostatic film stiffness J with D = 70 mm and β = 0,5

The simulation results of the change in the hydrostatic bearing stiffness depends on the radial clearance change and the oil recess pressure as shown in Fig. 9. This result also shows that the stiffness of the hydrostatic bearing is larger when the clearance radius is smaller with the same oil recess pressure (9.b). On the other hand, the stiffness value of the hydrostatic bearing is proportional to the pressure change in the oil recess

The stiffness of the machine tool spindle is commonly in the range of 250 - 500 N/µm. For medium – sized external cylindrical grinding machines, the total hydrostatic spindle unit stiffness J_Σ should be reached 300 - 500 N/µm. Thus, the hydrostatic bearing stiffness J should be larger 900 N/µm, then the total stiffness is about 320 N/µm.

Based on the simulation results, with the usual method of pressure selection and the ability of machining processing with h_0 in a range of 15-22.5 µm, the total stiffness J_Σ of the hydrostatic spindle unit does not satisfy the working requirements.

To achieve a total stiffness J_Σ about 300 -500 N /µm in the current machining processing capacity with h_0 in a range of 15-22.5 µm, it is necessary to increase the pump pressure p_s about 1.5 times. The main parameters of hydrostatic bearing can be estimated according to j_{no} (β = 0.5).

The total clearances of 30 µm and 45 µm are used to calculate a series of machining tolerances. With the single-workpiece machining for the purpose of upgrading the external grinding machine, we orient the machining process to achieve a radial clearance of 30 µm or increase the lubricant recess pressure

3. Conclusion

This paper presents a study on calculation and numerical simulation of hydrostatic bearing spindle stiffness of the medium-sized external cylindrical grinding machine. The spindle diameter, rotary speed of grinding wheel, and power of machine in this study are 70mm, 3000 rpm and 5.5 kW, respectively. The results of calculating and simulating the hydro-static bearing stiffness of the grinding machine can be concluded as follows:

- There is a direct relationship between the hydrostatic bearing stiffness and the main bearing parameters including oil pump pressure, clearance, machining tolerances, pressure coefficients, spindle diameters. Therefore, the selection of some parameters in accordance with the actual machining conditions to estimate the remaining parameters have different points of view, but not from the spindle unit stiffness to determine related parameters.
- To achieve the required the spindle stiffness of the medium-sized external cylindrical grinding machine, besides determining the basic hydrostatic parameters to ensure a completely wet lubrication, it is necessary to adjust the previous mentioned parameters in the allowable range for the better stiffness. Hence, the lubricant film pressure and total clearance must be larger 7.5 MPa and approximate 30 µm, respectively.
- The simulation and computation results have indicated a feasible selection range for the relevant hydrostatic parameters to obtain the required stiffness as required by the external cylindrical grinding machine.
- In the selection of parameters that satisfy the required stiffness, an attention should be paid to adjusting the parameters related to the roughness, tolerances, and oil pump pressure accordingly as it relates to the costs of machining processing and procurement as well as equipment adjustments.

Acknowledgments

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