Determining a Feasible Working Condition for Hydrostatic Spindle Bearings of the External Circular Grinding Machine 3K12

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Abstract

In a hydrostatic spindle of machine tools, the bearing structure parameters, lubrication characteristics and working conditions are factors affecting the spindle stiffness and the machining quality. Besides some known geometrical parameters and oil viscosity, other factors such as lubricant pressure and loading capacity need to be determined to find a feasible working condition for a machine tool. This study presents the experimental results of the effect of pump pressures and loads on the stiffness of the spindle integrated a new designed and fabricated hydrostatic bearing. The experiment results show that, with a known oil viscosity of 0.002 PaS, a pump pressure of 5 MPa and a load in a range 500 – 1000 N are the most feasible working condition of the medium – sized external circular grinding machine 3K12.

Keywords: Medium-sized circular grinding machine, pump pressure, loading, total stiffness, hydrostatic spindle bearing.

1. Introduction

Parameters of a machine tool such as structure, stiffness, and vibration of spindle bearing will affect the quality of machining process. In which, the spindle stability after the start-up is most important factor that affects the machining accuracy directly. Grinding is a fine machining process that determines the dimension precision and the surface roughness of workpieces. Stabilization of the wheel-stone axis when grinding that needs to be achieved quickly after the startup is always concerned by scientists. The stiffness of the machine tool spindle is commonly in a range of 250 - 500 N/μm. For medium – sized external cylindrical grinding machines, the total stiffness of hydrostatic spindle unit should be in a range of 300 - 500 N/μm [1].

In the field of machine tools, several researches on integration of a hydrostatic bearing into a machine and analyzing the characteristics of hydrostatic bearing to precision machining have been presented. In 2015, Bo-Sung Kim et al. presented a study on thermal characteristics of the grinding machine applied hydrostatic bearing. They indicated the effect of thermal deformation of CNC grinding machine integrated a hydrostatic bearing on machining quality. The study indicates that the heat distortion of the grinding machine spindle depends on the hydrostatic bearing temperature and it can be used to evaluate the thermal deformation characteristics of the grinding machine [2]. Hua-Chih Huang et al. (2015) developed a design methodology and tools for analyzing hydrostatic sliding boards using capillary on a high precision grinding machine [3]. V. Srinivasan (2013) analyzed the effect of static and dynamic loads on the hydrostatic bearing when changing the pressure and viscosity of the lubricant in the bearing. The authors studied the Reynolds equation and boundary conditions for analyzing variations in parameters related to hydrostatic bearing such as temperature distribution, oil viscosity changes and radial load. Analyzing the simulation results, the authors found that when increasing the lubricant viscosity in the bearing with the moving pads, the wear reduced and the bearing life increased [4]. Besides, in 2007, K. Wasson pointed out that the spindle structure integrating a hydrostatic bearing is suitable for machine tools that require a high precision in mechanical machining. In particular, the analysis also suggests that the design of the spindle with hydrostatic bearing to replace conventional roller bearings results in a reasonable cost on low and medium speed machines [5]. In 2013, Nirav Doshi & Mehul Bambhania presented a study to optimize the film thickness on the V-25 vertical lathe machine with a hydrostatic spindle bearing. The simulation program was performed with speed, oil viscosity and stiffness parameters when varying film thickness [6]. In 2008, Ryszard Przybyl presented the possibility of increasing the stiffness of machine tools spindle units due to the advantage of a poorly known property of the hydrostatic journal bearings [7]. Chen D. et al. (2012) presented the dynamic and static characteristics of a hydrostatic spindle for machine

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tools. Hence, the authors have shown the influence of the eccentricity ratio on the film thickness, stiffness and deformation of a spindle system. They analyzed the effect of imbalanced vibration on the machining accuracy. The research shows that the location and stiffness of the bearing affect the machining accuracy of workpiece [8]. A thermo-mechanical error model of a hydrostatic spindle for a high precision machine tool was proposed by Chen D. et al. in 2011. The authors have believed that the variation of motion error which was induced by thermal effects on a machine worktable during machining. They have also evaluated the heat generated in the spindle elements and the coefficients of convection heat transfer over its outer surface and the influence of thermal on spindle stiffness [9].

In 2016, He Qiang et al. presented a numerical and experimental method to select parameters and fabricate an hydrostatic spindle unit to replace the ball bearing spindle in a vertical machining equipment with whose operating speed is 800 rpm [10]. 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 [11]. 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 [12].

However, there are not many studies on the effects of pump pressure and load on the stiffness of hydrostatic bearing integrated on medium-sized circular grinding machines. Thus, these will be the objects to be studied and investigated their influence on the stiffness of hydrostatic spindle bearing in this study to find the most feasible working condition associated with the fine-machining process on the 3K12 grinding machine after replacing the hydrodynamic spindle bearing with a hydrostatic one.

2. Hydrostatic spindle bearing

The medium size grinding machine 3K12 uses hydrodynamic bearing for its spindle unit due to the high-speed operation and the little load changes during a working cycle. The hydrodynamic spindle bearing with 3 self-aligning pads has ensured the basic requirements for the dimensional and geometric accuracy of fine finishing workpieces. However, due to the characteristics of hydrodynamic lubrication, the center trajectory of spindle varies with speed and load, which has a certain effect on the stability of spindle center and the improvement of machining accuracy according to the increasing requirements of industry. A cross-section of the spindle unit integrated hydrodynamic bearings of the grinding machine 3K12 is shown in Fig.1.

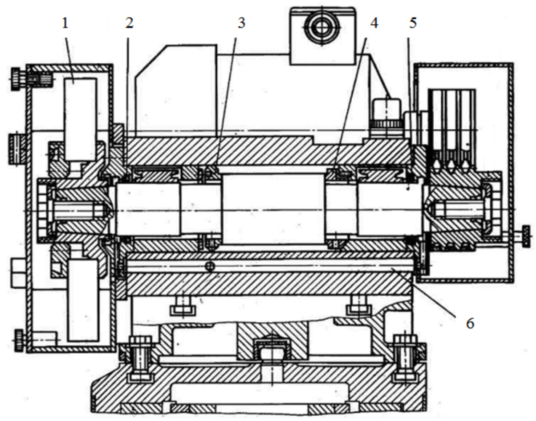

Fig. 1. Spindle unit structure of grinding machine 3K12: 1 – Wheel stone; 2.5 - Oil barrier rings; 3,4 - Hydrodynamic bearings; 6 - Oil return [13].

In this study, hydrostatic spindle was designed to replace the hydrodynamic spindle on the 3K12 external cylindrical grinding machine. The new hydrostatic bearing must ensure the technical requirements as well as the loading capacity of the hydrodynamic bearing that is feasible with working condition of the grinding machine. Accordingly, the dimension of the shaft and bearing case have been designed and machined in the range $\Phi70^{-0.01}_{-0.029}$ and $\Phi70^{0.03}_{0}$ respectively. Thus, the largest clearance h_{0max} is 59µm and the smallest clearance h_{0min} is 10 µm. The designed hydrostatic bearing is composed of 4 oil recesses. The bearing length, recess length and shaft diameter are 56mm, 28mm and 70mm, respectively.

The structure of oil supply system for the hydrostatic spindle unit used in 3K12 grinding machine is shown in Fig.2. 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(MPa),* respectively.

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:

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

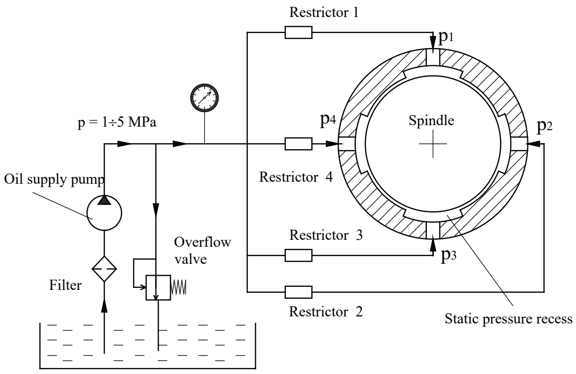

Fig. 2. Structure of oil supply system for hydrostatic bearing on machine tools

And the dimensionless equation is:

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

where $\lambda = L/D$: ratio of length and diameter of the bearing; φ: angle coordinates (rad); *p*- oil film pressure (MPa), *y*-radial coordinates; $\bar{y},\bar{p},\bar{h},\bar{\mu},\bar{\omega}$: dimensionless parameters.

The lubricant film thickness is determined by equation [10]: $h = h_0(1-\varepsilon\cos\varphi)$ (4)

where, h – film thickness (µm); h_o – film thickness under line eccentricity (µm); φ - angular position from the line of eccentricity (rad); $\mathcal{E}$ = e/h_o– eccentricity ratio.

The oil viscosity η is determined by equation as [1]:

$$\eta=\frac{S_h}{\frac{n}{p_s}\left(\frac{D}{2h_0}\right)^2} \qquad (5)$$

where S_h - speed parameter; n – spindle rotation speed (Rad/s); p_s – pump pressure (MPa)

The oil recess pressure p_r in accordance with the ability of manufacturing technology is in a range of 1-5 MN/m^2. For hydrostatic bearing, the ratio of oil chamber and pump pressure $\beta = p_r/p_s$ should be in a range of 0.4 – 0.7 [8]. Hence, the pump pressures chosen for the fabricated hydrostatic bearing in this study are 3, 4, and 5 MN/m^2. In general, the oil viscosity that used for hydrostatic lubrication is chosen low to achieve cooling effect of fluid flow. Therefore, the oil viscosity using in this experiment to investigate the influence on the hydrostatic spindle stiffness were chosen as 0.002 PaS.

3. Experiment setup

To evaluate the actual working ability of the hydrostatic spindle unit, an experiment equipment needs to be developed. A criterion for evaluating spindle unit is the stiffness of hydrostatic bearing. In this study, a system of stiffness testing equipment which is feasible for the grinding machine 3K12 basing on displacement of the spindle under working conditions was developed and shown in Fig.3

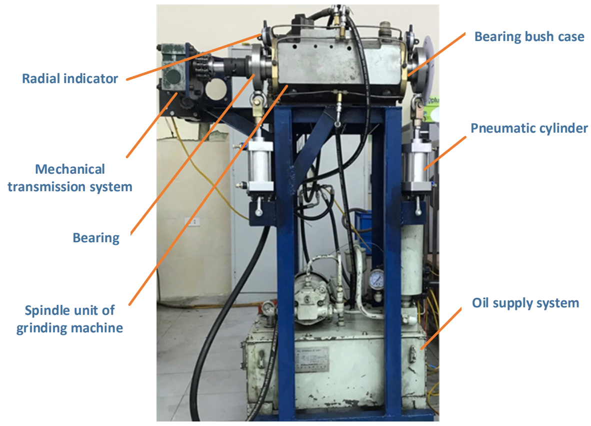

Fig. 3. Hydrostatic spindle testing bench

The pump pressure can be changed in a range 3-5MPa in this experiment. Besides, to obtain the spindle displacement, a load-generating system creating radial forces on both ends of the spindle was built by using two pneumatic cylinders. The radial force is determined by the pressure acting on the pneumatic cylinder. The spindle displacement is monitored by 2 radial indicators (1µm resolution) as shown in Fig.3. These pneumatic cylinders are supplied by 3 separated compressed air sources which are calibrated to corresponding to 3 designed loads that apply to spindle: 500, 1000 and 1500 N. The oil viscosity using in this experiment is 0.002 Pa.S.

Experimental hydrostatic spindle stiffness is expressed as:

$$J=\frac{P}{x} \qquad (6)$$

where: J - hardness of spindle assemblies (N/µm); P - radial load (N); x – radial spindle displacement value (µm).

4. Results and discussions

The experiment has been carried out step by step procedure: pumping a high pressure into bearing; waiting for a minute for stabilization of the pressure oil inside system; applying load on spindle and measuring the displacements of spindle. With each working condition, the displacement of spindle is the average of eight measured values around the circumference of the spindle. The experimental results are presented in *Table 1*. The spindle displacements vs pump pressure at different loads are shown in Fig.4.

Table 1. Displacement and stiffness of spindle unit

Load (N)	Spindle displacement (µm)			Stiffness of spindle unit (N/µm)		
	Pump pressure (MPa)			Pump pressure (MPa)		
	3	4	5	3	4	5
500	2.8	2.1	1.7	185.0	236.8	301.2
1000	5.3	3.9	3.2	187.5	257.1	310.6
1500	8.3	5.9	5.1	180.0	254.7	294.1

As it can be seen, the displacements of the hydrostatic spindle depend on load and pump pressure. Moreover, the spindle displacement also decreases when higher pressure oil is supplied to the system. Within the experimental pressure in a range of 3-5 MPa, the smallest displacement obtained when the pump pressure is 5 MPa. This is consistent with the fact that when the pump pressure increases, the pressure in the oil chambers also increases, creating a greater force to counteract the effect of load, making the system more stable. Therefore, in these experiments, the appropriate pressure for hydrostatic bearing is considered to be 5 MPa. On the other hand, in order to evaluate the effect of pump pressure and load on the spindle stiffness in detail, the calculations based on the experimental data were carried out. This result is also shown in Table 1 corresponding to different loads and pump pressures.

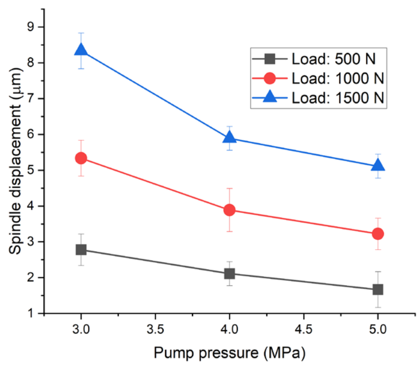

Fig. 4. Spindle displacement vs pump pressure at different loads

The stiffness of the spindle unit is determined based on the changes in pumping pressure with values of 3, 4 and 5 MPa under the loading in a range of 500 – 1500 N as shown in Fig.5 (a). Meanwhile, Fig.5 (b) describes cubic interpolation of the relationship between stiffness vs pumping pressure by Matlab, and load to predict the trend of changing the spindle stiffness according to the load and pressure. In addition, the stiffness of the spindle unit increases in proportion to the pump pressure at all trialed loads. Indeed, at a pressure of 3 MPa, the spindle stiffness reaches the maximum value of approximately 187.5N/µm under a load of 1000 N, while the smaller stiffness about 180 N/µm at a load of 500 and 1500 N is achieved.

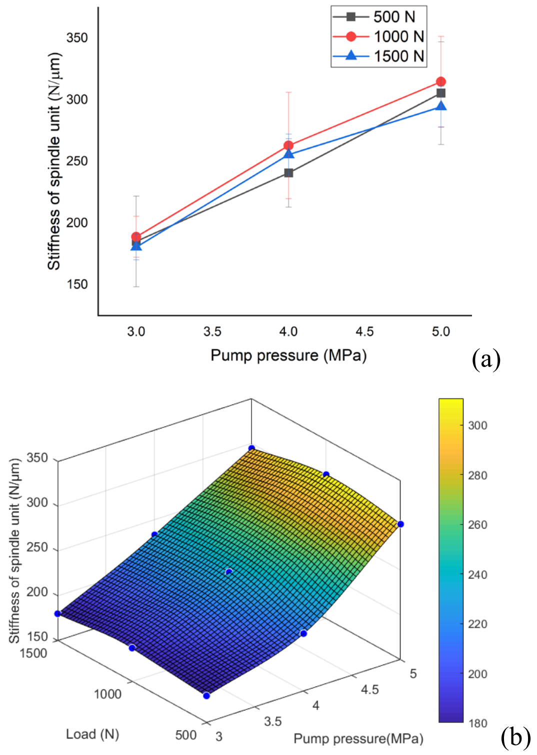

Fig. 5. Total stiffness of hydrostatic spindle unit with different loads and pump pressures

Similarly, the stiffness reaches the maximum values of 257.1 N/µm and 310.6 N/µm corresponding to the pressure of 4 and 5 MPa under a load of 1000N. It also can be seen that, the stiffness of spindle tends to increase when load increases from 500 N to 1000 N, then this value tends to decrease at a load of 1500 N. It may be a basis to recommend that the user needs to adjust the load within a suitable range to achieve the highest stiffness under a given working condition.

It is clear that the stiffness of spindle unit reaches the maximum value in the load range of 500-1000N and decreases at a load of 1500 N. The hydrostatic spindle is upgraded from existed hydrodynamic spindle, new bush case is assembled with existed bush case housing, so the thickness of bush case is limited. On the other hand, under high oil pressure, bush case made of copper would be elastic deformation. Due to this deformation of bush case, when load is increased up to 1500N, the deformation would increase oil leak through surface of bush case lead to decreasing of pressure inside oil chamber then decrease stiffness of oil film in particular and bearing system in general. This experimental result also pointed out that with upgraded hydrostatic spindle and oil viscosity of 0.002 Pa.S, 1000N is limited loading of spindle based on spindle stiffness.

As shown in Fig.5 (a) the spindle stiffness is larger than 300 N/μm when a pump pressure of 5MPa and a load of 500-1000 N are applied. Thus, the suitable load for this hydrostatic spindle bearing is in the range of 500-1000N, this is also feasible for working conditions of the external medium-sized circular grinding machines. The results also recommend that users should set the load in accordance with the working conditions to achieve the best stiffness of the system.

5. Conclusion

The experimental results pointed out that stiffness of hydrostatic bearing seems to proportional to pump pressure changing from 3 – 5 MPa. However, the total stiffness of the spindle is not stable when load is changed. It supposed to be non-regular deformation on circumference of bush case; the copper bronze is also elastic deformed at heavy load those results in reduced stiffness of oil film and spindle unit.

The experimental results pointed out that, with viscosity of 0.002 PaS and pump pressure of 5 MPa, the total stiffness of the hydrostatic spindle unit could reach up to 310.6 N/μm under a load of 500-1000 N, which is feasible for working condition of the medium – sized external cylindrical grinding machine 3K12.

6. Acknowledgement

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