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A research of stabilization for high speed air bearing spindles.

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Abstract: The air spindles as well as the hydrodynamic spindles have been very satisfactory solutions for precision drives, which overcome the basic disadvantages of most traditional articulated spindles that represents random errors caused by dynamic gaps. For an air spindle, due to the fact that the friction is small, the abrasion is significantly reduced. Furthermore, the viscosity of the air lubricant is less affected by temperature. Therefore, the lifespan of the spindle is greatly increased, and the device operates reliably in a wide temperature range. Because of those advantages, air spindles are widely used in many fields such as mechanical processing, measurement, aviation, etc. In our particular work, we are focused on applications of air spindles in precision machining of small holes. This paper presents some of the results of our research of stabilization for high speed air spindles. In the research, we have proposed a study model of a high-speed air spindle, thus we study its static and dynamic stability problems and also the solutions for those problems, in order to construct a basis for designing an air spindle with the rotating speed up to 15000 rpm for applications in machining precious holes with the diameter of 0.35 ÷ 2 mm.

Keywords: *air bearing, spindle, stabilization, static, dynamic*

INTRODUCTION

Air bearing is one of fundamental study subjects of air lubrication in the field of friction, wear and tribology. Nowadays, technical requirements of tribology are more rigorous and diverse. The majority of those requirements are satisfied by the completion of the basic structures and by bearing lubricant improvements. Air lubricant has unique advantages which solid or liquid cannot have, such as: viscosity, stability of air properties in a wide range of temperature and relative pressure, reducing the consumption of outflowing oil, cooling, etc. The use of air spindles in high speed machine tools have the following advantages:

- Precision improvement. Use of air bearings increases radial and axial accuracy. Because there is no mechanical contact, wear is minimal, accuracy and reliability are ensured over time. Lathing machines with spindles using air bearings can produce surfaces with the accuracy of 0.05 µm. Even in the case of using diamond tipped heads on porous materials, the precision can be attained by 0.012 µm.

- High stiffness. If the system is designed correctly the radial stiffness on the tool is comparable to that of ceramic ball bearing [1].

- High speed. The fact that the deformation force in the air bearings is extremely low allows the spindle to reach high speeds with little energy generated by the engine and little heat generated. The speed of a spindle using air bearings can be up to 500,000 rpm.

- Improvement of tool life. Low vibration and high rotational accuracy makes the tool life longer - minimizing maintenance and replacement. Longevity of the tool in a lathe machine of West Wind, using air bearings, is four times as long as the similar one without using air bearings.

However, there are certain disadvantages of using air spindles. The general downside is the small load capacity compared to the oil, which is due to the viscosity of air. With the use of thrust air bearings, the load capacity is limited by the air pump system's power. With the use of journal air bearings, the reason is due to the considerable eccentricity occurring during the dynamic process of high speeds. This disadvantage leads to the limited use of air spindles in the case where the load is considerably large.

Another disadvantage of air spindles is that they tend to be unstable compared to similar oil lubricated spindles. Instability modes usually occur with the use of thrust air bearings. To overcome them, it is needed to study new

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structures. The most popular approach today is to conduct air through the capillaries of porous air bearings. However, this method requires the supply of compressed air to be very dry and clean, leading to the need for careful and specific air filtering equipment that cannot always be ensured. Air bearings also have another major issue that is related to the stiffness. With thrust air bearings, one can optimize the design to the allowable static stiffness, but the dynamic stiffness of the spindle is difficult to be changed. Due to its high compressive strength and very low vibration resistance, the air spindle has a very low vibration property. Restraint of regenerative oscillations in the air lubrication equipment requires a good vibration-damping system that also presents a major obstacle to the use of air spindles.

STUDY MODEL

In order to solve the instability problems, we propose a study model (fig. 1) which is used in small hole drilling machines. The loading torque is generated through the supply of high-pressure compressed air to the vortex *8*, which is rigidly connected to the shaft *6* to form the rotor. Two journal air bearings *5* are put together to form a diaphragm air bearing. Air is supplied through the journal air bearings *5* to the rotor and filled into the air lubricating film. The thrust air bearing *9* is to limit the axle dimensions. Balls and bolts are to align the air gap.

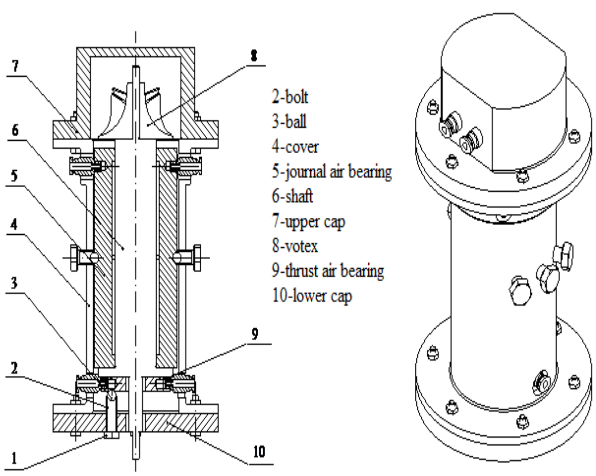

Fig.1. Study model of an air bearing spindle in small hole drilling machines.

The study model is proposed with some given parameters as following: the rotating speed is up to 15000 rpm; the pressure of air supply is not above 4 bar; the radial lubricant thickness is less than 10 μm; the axial lubricant thickness is less than 5 μm.

In this research, we study the instability problems which occur in static and dynamic air bearings and their solutions. With thrust air bearings, the instability is related to the vibration phenomenon of the air bearings and the solution is designing a damping cavity to extinguish vibration. With journal air bearings, the instability is due to the phenomenon of half-speed vortex, and the solution is using air bearings with specific orifices. The designs of our methods are presented in the following content.

STATIC STABILIZATION FOR THRUST AIR BEARINGS.

Static air bearings used in the study model has a function of limiting the axle dimensions, against the effect of gravity and partly cutting force on rotor motion in the working process. Depending on the method of leading and distributing air on the working surface, there may be many types of static air bearings: pocket, groove, etc. and types of porous materials. However, the nature of instability with thrust air bearings is the same. In the particular work, we study a flat-pocket air model (fig. 2), in which instability of static air bearings is the most apparent. After experimenting and analyzing, we have concluded that the instability with thrust air bearings is due to the vibration phenomenon of the air bearings with a specific frequency.

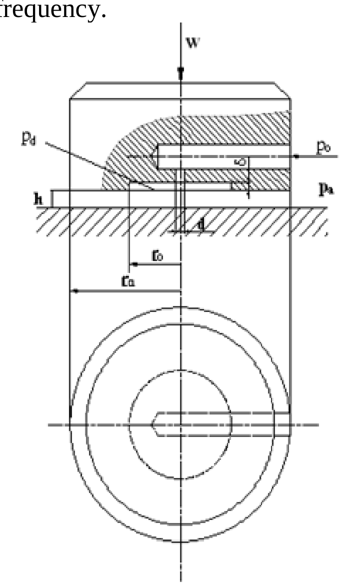

Figure 2. Flat-pocket air model

The cause of the instability in the thrust air bearing, according to C.A.Шенберг, M. Д. Шишеев [2], is due to the existence of the dead volume under the air bearing surface. The very existence of this dead volume causes the pressure in the pocket to react to change of the air gap. In other words, the phase of the air gap change and the phase of the pressure change in the pocket are different. J.W.Powell [3] experimented with only aperture-filled air bearings, without dead volumes under the air bearings, and concluded that most of those bearings were very stable but that such bearings have low stiffness and low loading capacity. Therefore, the compressibility of the air is the cause of the oscillation.

In this research, we figure out that the oscillation around the center of the air bearing is essentially a random oscillation, due to the fact that the air bearing leaves the equilibrium position. Though the air bearing may oscillate with all degrees of freedom, the motive force that causes the instability, referring to a fixed point, must be vertical. The air bearing's oscillation process is a resonant oscillation process whose oscillation frequency coincides with the oscillation frequency of the system. The natural oscillation frequency depends only on the stiffness of the air bearing and the load. The frequency can be defined as:

$$\omega_0 = \sqrt{\frac{C}{m}} \quad (1)$$

where C is the stiffness of the air bearing which appears as a constant in the linear working domain of the air bearing, m - the mass of the air bearing.

The general approach to overcome the vibration of the air bearing is to minimize the volume effect of the pocket, making the pocket pressure response to the air gap variation as fast as possible, while paying attention on designs to maximize the stiffness of the air bearing in order to reduce the inertia effect. There are certain design parameters affecting the vibration of the air bearings such as the relative volume of the air pockets generally defined as the dead volume and the air volume [2]; the oscillated volume; difference between the pocket pressure and the supply pressure [3]; the diameter of the air orifice; etc.

A well-known method of reducing the vibration of the air bearing is given by Lehmann [3], using a method of damping cavity. In this work, we applied this method by constructing a damping cavity as illustrated in the figure 3.

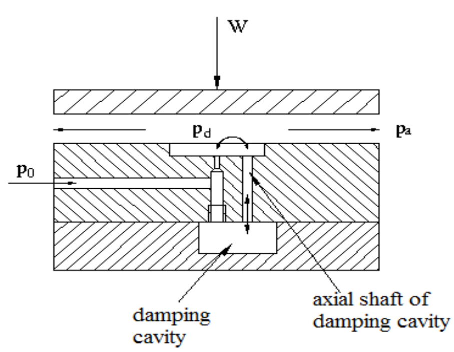

Figure 3. Using a damping cavity to extinguish vibration

Define v1 - volume of the damping cavity without the shaft; p1- pressure of the air is in the damping cavity; d1 - diameter of the axial shaft; l1 - length of the axial shaft; s1 - cross-sectional area of the axial shaft; m1 - mass of the air in the axial shaft; c1 - elastic stiffness of the damping cavity.

At any displacement x, the pressure of the air is in the damping cavity is defined as:

$$p_1 = \frac{m_1^{'}.R.T}{v_1 + x.s_1}. \quad (N/m^2) \quad (2)$$

where R= 287.1(J/ kg.K) - air constant; T- absolute temperature of air (K).
Suppose the stiffness of the damping cavity is the same in the local of the displacement x=0; and the pressure p1 is equal in the damping cavity (including the axial shaft), we have:

$$c_1 = -\left.\frac{dF_1}{dx}\right|_{x=0} = -\left.\frac{s_1.dp_1}{dx}\right|_{x=0} \quad (3)$$

$$\frac{m_1^{'}}{m_1} = \frac{v_1}{s_1.l_1} \quad (4)$$

It is inferred from (2), (3), (4) that:

$$c_1 = m_1.R.T.\frac{s_1}{v_1.l_1} \quad (5)$$

Since the nature oscillation frequency is defined as:

$$f_1 = \frac{1}{2.\pi}\sqrt{\frac{c_1}{m_1}} \quad (6)$$

the volume of the damping cavity is calculated as:

$$v_1 = \frac{R.T.s_1}{4.\pi^2.f_1^2.l_1} = \frac{R.T.d_1^2}{16.\pi.f_1^2.l_1} \quad (7)$$

The method of using a damping cavity is based on the principle of a shock absorber, which is a simple and easy to implement without requirement of high manufacturing precision. However, the fundamental limitation of the introduced method is that vibration resistance can only be achieved in a narrow range of frequencies.

DYNAMIC STABILIZATION FOR JOURNAL AIR BEARINGS

The typical negative aerodynamic effect on the rotating rotor is the half-speed vortex, or half-frequency vortex. This phenomenon has been one of the major obstacles to the widespread applications of air lubrication in high speed spindles.

Consider a long cylindrical spindle, assuming the absolute hard axis is perfectly balanced in the hole. At this point the axis reaches a velocity ω and its center is completely fixed at one point. This state will be maintained if there is no other impact on the system. The center of the shaft is called the static equilibrium position. The aerodynamic force F_0 is generated by the air membrane and balanced by the external force acting on the axis W_0: $\overrightarrow{W_0} - \overrightarrow{F_0} = 0$.

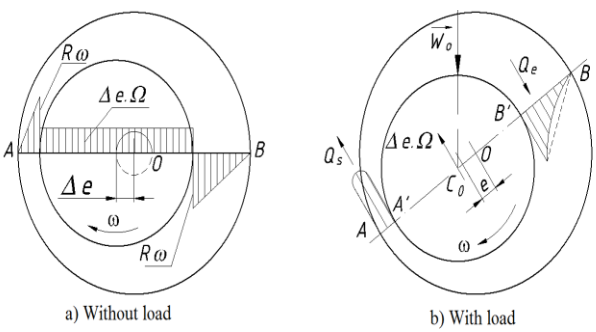

Figure 4. Chart of flow balance.

Consider the axial state displaced from the equilibrium position C_0 to another point C_1 (figure 4) for some reason. In this case, the aerodynamic force is no longer equal to the external force. The secondary aerodynamic force $\vec{f} = \overrightarrow{F_d} - \overrightarrow{F_0}$ is generated by moving the center of the axis to the value Δe.

The preservation of the flow through the AB surface with the center of the shift axis Δe and the vortex Ω perpendicular to AB (fig.4), is characterized by the following formula:

$$2r.\Delta e.\Omega = \frac{r.\omega}{2}(c + \Delta e) - \frac{r.\omega}{2}(c - \Delta e) \quad (8)$$

where r - radius of the axis; c - medium air gap defined as = R-r with R presenting the radius of the hole.

Thus, $\Omega = \omega / 2$ (9).

Actually, due to the finite length of the spindle and the existence of axial flow: $\Omega < \omega / 2$ (10)

The ratio Ω/ω is smaller in the case of shorter holes and larger gaps.

According to [2], assuming $\Omega \approx \omega/2$, we find the condition of dynamic stability:

$$G_N \geq \frac{1}{2} m\Omega^2 \approx m\frac{\omega_{max}^2}{8} \quad (11)$$

where:

- G_N - stiffness of the lubricating layer which is defined as $G_N = \left|\frac{dW}{de}\right|$, with W presenting the lifting force and e - eccentricity.

- ω_{max} - critical angular velocity of the axis.

Stability in this case can be achieved by using air bearings with orifices that significantly increase the stiffness of the air bearings, thus increase the dynamic stability of the spindle. In addition, it also minimizes wear on start and finish of the spindle. However, designers also need to consider and select the appropriate type of air bearings to match the critical angular velocity of the axis ω_{max}.

As the industrial supply pressure, in general, is limited to 4 kgf / cm^2, in order to ensure dynamic stability of air spindles, as discussed in the previous section, the use of orifice effects is limited. Therefore, the stiffness of the dynamic spindle is limited, leading to the limitation of the rotating speed. The direct stiffness coefficients increase with the increasing rotating speed [4]. The shaft mass and the stiffness of air bearing are two major factors which influent the fundamental and second natural frequencies of air spindle [5]. In this paper, the air bearings with orifices are designed in order to overcome the dynamic instability of air spindles.

SIMULATION AND RESULT

In the research we propose a design for journal air bearings with orifices which are located of the spindle (figure 5). Thus, we analyze the effects of those bearings on lifting forces and the stability of the air spindles. Air is supplied into two passive conductors with the dimension of 5 mm, through the four orifices with the diameter of 0.5 mm, then into the H-shaped air grooves and into the air bearing surface. The boundary conditions for input and output pressures are relatively 5 atm and 1 atm. The temperature of the air flow is 25^{o}C.

As the result of applying the finite element method using the software ANSYS, figure 6 illustrates the chart of pressure distribution over the journal air bearing with a particular air gap of 5 µm. The graph of the change of the lift depending on the change of the air gap is illustrated in figure 7. Therefore, the stiffness of the lubricating layer is achieved by calculating the slope of the approximate line. In our work, the stiffness of one air bearing is approximately 32.6 N/µm, thus for two air bearings parallel to each other, the whole stiffness of the spindle is G_N = 65.2 N/µm.

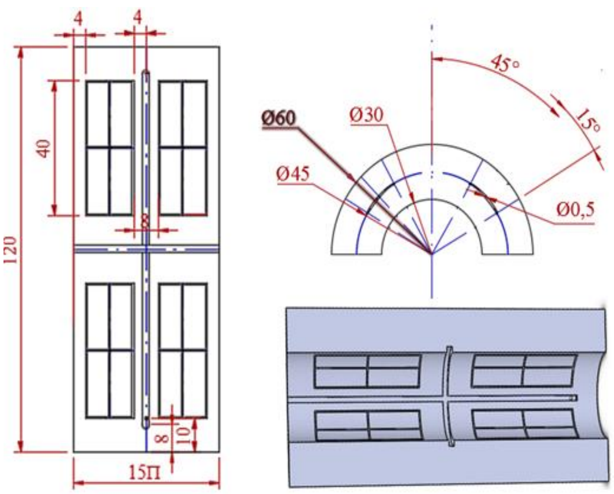

Figure 5. Design of the journal air bearing of the spindle.

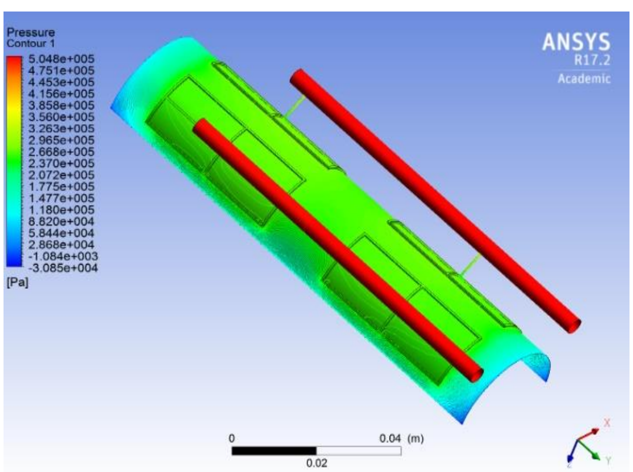

Figure 6. Pressure distribution over the journal air bearing with the air gap of 5 µm.

Using the inequality (11), with the mass of the spindle m = 0.5 kg, the critical rotating speed can be defined as:

$$\omega_{max} \le 2\sqrt{\frac{G_N}{m}} \approx 218\,000\,rpm \qquad (12)$$

As a consequence, with a working speed of 10,000 ÷ 15,000 rpm, the stability of the air bearing spindle is still maintained.

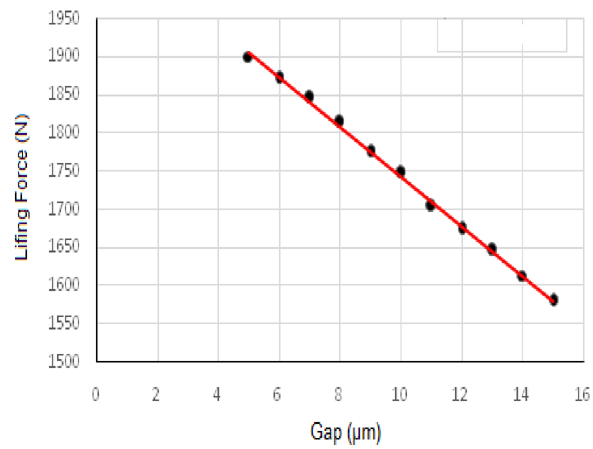

Figure 7. The simulation graph of the change of the lift depending on the change of the air gap.

CONCLUSION

The calculation to ensure the stability of air bearing spindle is a complicated task. Many authors in the world have also researched and acknowledged this. Most of the solutions provide directive designs rather than guidelines on boundary quantification of stability. In addition, manufacturing, assembly and alignment of air bearing spindles is more complicated and difficult in comparison with other types of spindles. The cost of technical maintenance is also more expensive. In this research, we mention some typical contents of those issues when calculating, designing the air spindles. On the other hand, our work has only limited the scope of research in some cases, in order to achieve the basis for designing an air spindle with the speed up to 15000 rpm for applications in machining precious holes with the diameter of 0.35 ÷ 2 mm. We expect that the research would provide more useful information for interested readers and researchers in the related fields.

ACKNOWLEDGMENTS

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