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A Study on Hydrodynamic Coefficient Characteristics of Air Bearing for High Speed Journal

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Abstract: This paper presents the hydrodynamic effect by the journal speed, eccentricity and source positions in order to overcome the defects of air bearing such as low stiffness and damping coefficient. Choosing the two row source position of air bearing is different from existing investigations in the side of pressure distribution of air film because of the high speed of journal and the wedge effects by the eccentricity. These optimal choices of the two row source positions enable us to improve the performance of the film reaction force and loading force as making the high-speed spindle. In this paper, The pressure behavior in theory of air film in high speed region of journal according to the eccentricity of journal and the source positions analyzed. The theoretical analysis has been identified by experiments. The results of investigated characteristics may be applied to precision devices like ultra-precision grinding machine and ultra high-speed milling.

Keywords: Hydrodynamic effect, two row source positions, wedge effects, film reaction force, loading force

Introduction

Externally pressurized air journal bearing is in wide use in high speed rotating machinery and high precision spindle systems because of its advantages such as low friction, low heat generating character and averaging effects of the pressure deviation in the air film in bearing.

The necessity to catch the performance of air bearing is being increased for developments of highly efficient high speed machinery. Therefore, the stiffness and damping values of air bearing is characteristic value to express air bearing's performance. The measuring of air bearing's stiffness coefficient have to be carried out efficiently because it is the greatest important design factor and the load capacity of air bearing is less than any other fluid film bearing because of low viscosity.

An air film in air bearing has the disadvantage that we can not expect the boundary lubrication in a state of emergency. In order to deal with this kind of problem, a number of engineers have investigated the improvement of the loading capacity and high dynamic coefficients [2-7].

But these studies have been analyzed theoretically in the static states or in the air film for single row air source. Up to now, in the case of two row air sources of air bearing, the source position which is the existing design factor have chosen 1/4L of bearing total length from the end of air bearing. A number of theoretical and experimental analysis have been carried out at this position. An international company, Westwind Turbines Ltd., the investigation contents of Poole and J. W. Powell [1] express well Fig. 1.

However, the air bearing clearance have been designed on

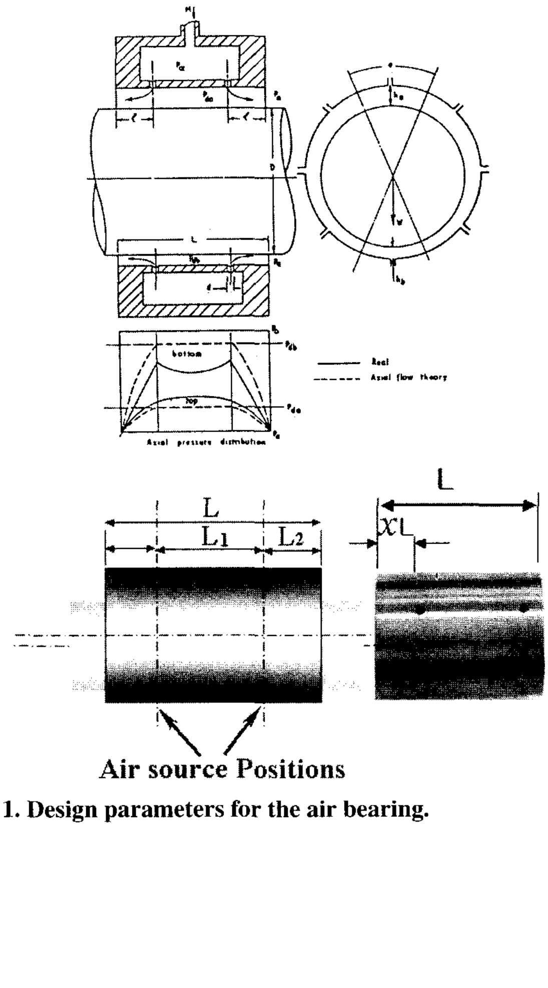

Fig. 1. Design parameters for the air bearing.

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the tight condition of 10-20 μm and we have to always consider hydrodynamic effects into the air film in high speed regions.

The design factors of air bearing have to be reconsidered in the side of load capacity and choose again the source position including the hydrodynamic effects in high-speed regions. Therefore, a final goal of this paper is to find the new boundary condition of the source position through the theoretical analysis according to the source position and experiments loading to the journal.

Numerical Analysis

Reynold equation

The air-film pressure distribution in bearing to have two row sources is determined from following the Reynold equation. In order to solve the loading capacity, air supply source is assumed as a line source and the following assumptions are adopted.

(1) It's possible to disregard the other velocity gradient except the velocity gradient of the film thickness directions.
(2) The surface of the bearing and journal is completely flat.
(3) The air flow in the air film is isothermal, viscous and laminar.
(4) The non-slip condition at the film boundaries applies at the plane surface only.

Therefore, the Reynold equation for the air film under the air bearing and journal is expressed in the following form.

$$\frac{\partial}{\partial x}\left(Ph^3\frac{\partial P}{\partial x}\right)+\frac{\partial}{\partial y}\left(Ph^3\frac{\partial P}{\partial y}\right) = 6\eta\cdot U_j\cdot\frac{\partial(Ph)}{\partial x} \tag{1}$$

where

$$\bar{P} = \frac{P}{P_a},\ \bar{U} = \frac{U_j}{R\omega},\ \bar{h} = \frac{h}{C},\ \theta = \frac{x}{R},\ \zeta = \frac{y}{R},$$
$$\Lambda = \frac{6\eta\omega}{P_a}\left(\frac{D}{C}\right)^2,\ h = C(1-e\cos\theta) \tag{2}$$

By substituting equation (2) into equation (1), the dimensionless Reynold equation can be expressed in the following form.

$$\frac{\partial}{\partial\theta}\left(\overline{Ph^3}\frac{\partial\bar{P}}{\partial\theta}\right)+\frac{\partial}{\partial\zeta}\left(\overline{Ph^3}\frac{\partial\bar{P}}{\partial\zeta}\right) = \Lambda\left(\frac{\partial\overline{Ph}}{\partial\theta}\right) \tag{3}$$

Boundary condition

It is assumed that the pressure from both ends of bearing is 1atm and the supply pressure from the feeding hole is 5atm. The source position on the bearing is expressed in the following form.

$$\bar{P} = 1,\ \text{at}\ \zeta = \pm L/D \tag{4}$$
$$\zeta = \pm L_2/D$$
$$\bar{P} = 5,\ \text{at}\ \left(L_2 = \frac{L}{3}, \frac{L}{4}, \frac{L}{7}, \frac{L}{11}\right)$$
$$\bar{P}(\theta,\zeta) = \bar{P}(\theta\pm2\pi,\zeta)$$

Numerical calculation

In order to solve the Reynold equation, newton-rephton method has been applied because the compressible fluid has non-linear characteristics. The pressure equation can be expressed in the following finite difference form.

$$f_{i,j}(P) = -A\overline{P_{i,j}^2}-B\overline{P_{i,j}^2}+C\overline{P_{i+1,j}^2}-D\overline{P_{i+1,j}^2}+E\overline{P_{i-1}^2}+D\overline{P_{i-1,j}^2}+F\overline{P_{i,j+1}^2}+G\overline{P_{i,j-1}^2} \tag{4}$$

$$A = \left[\frac{1}{\Delta\theta^2}(\overline{h_{i+1,j}^3}+2\overline{h_{i,j}^3}+\overline{h_{i-1,j}^3})+\frac{1}{\Delta\zeta^2}(\overline{h_{i,j+1}^3}+2\overline{h_{i,j}^3}+\overline{h_{i,j-1}^3})\right] \tag{5}$$

$$B = \Lambda\left(\frac{1}{\Delta\theta}\right)(\overline{h_{i+1,j}}-\overline{h_{i-1,j}}) \tag{6}$$

$$C = \frac{1}{\Delta\theta^2}(\overline{h_{i+1,j}^3}+2\overline{h_{i,j}^3}) \tag{7}$$

$$D = \Lambda\left(\frac{1}{\Delta\theta}\right)\overline{h_{i,j}} \tag{8}$$

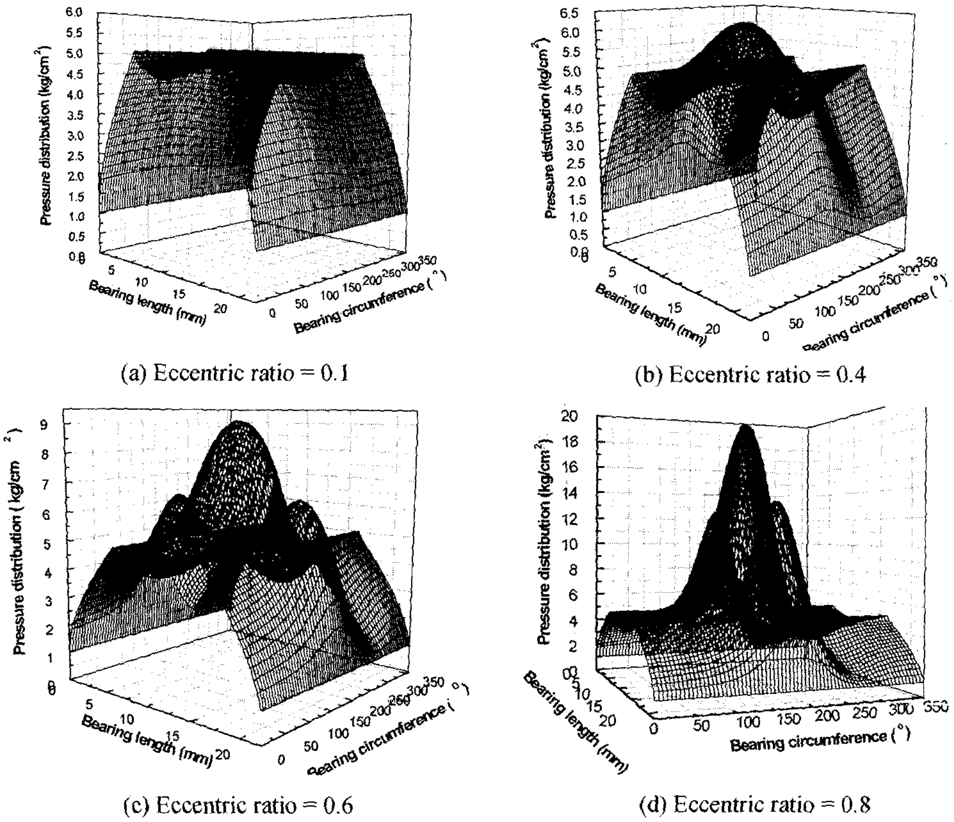
$$E = \frac{1}{\Delta\theta^2}(\overline{h_{i,j}^3}+2\overline{h_{i-1,j}^3}) \tag{9}$$

$$F = \frac{1}{\zeta^2}(\overline{h_{i,j+1}^3}+\overline{h_{i,j}^3}) \tag{10}$$

$$G = \frac{1}{\zeta^2}(\overline{h_{i,j}^3}+\overline{h_{i,j-1}^3}) \tag{11}$$

where, to solve the pressure, Jacobian is expressed in the following form

$$\frac{\partial f_{i,j}(P)}{\partial P} = 2\{-A\overline{P_{i,j}}+C\overline{P_{i+1,j}}+E\overline{P_{i-1,j}}+F\overline{P_{i,j+1}}+G\overline{P_{i,j-1}}\}-B \tag{12}$$



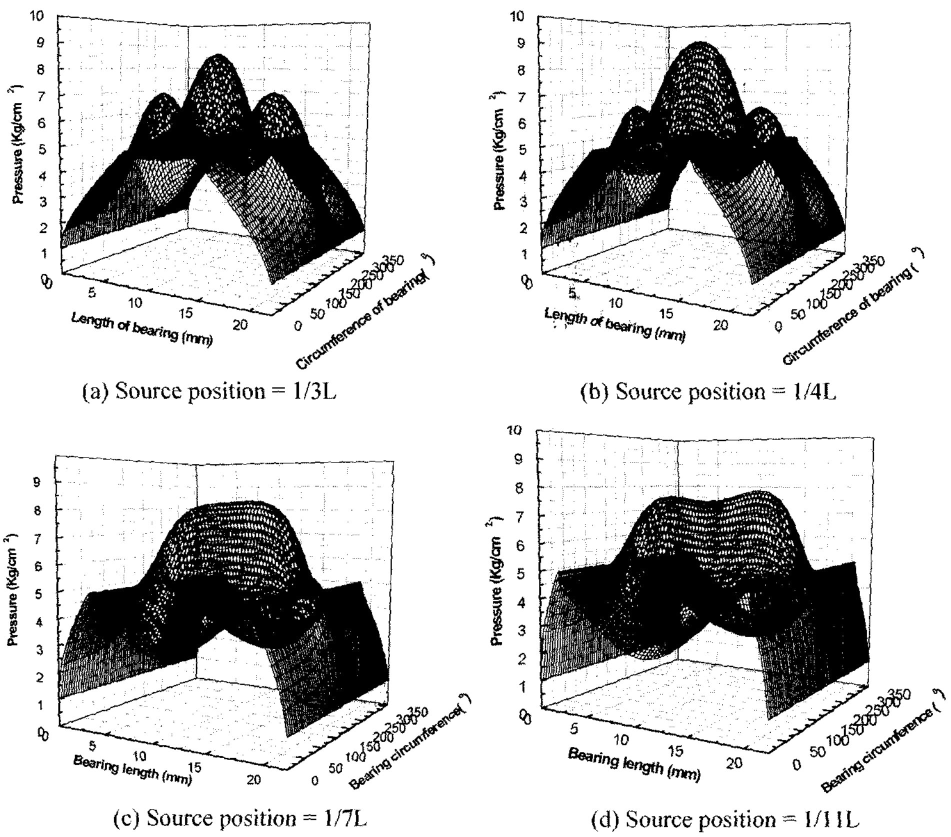
(a) Eccentric ratio = 0.1

(b) Eccentric ratio = 0.4

(c) Eccentric ratio = 0.6

(d) Eccentric ratio = 0.8

Fig. 2. Pressure distributions in air-lubricated journal bearing according to the eccentric ratio (L/D=1, Ps=5atm, DN= 2,000,000, Source position=1/4L).



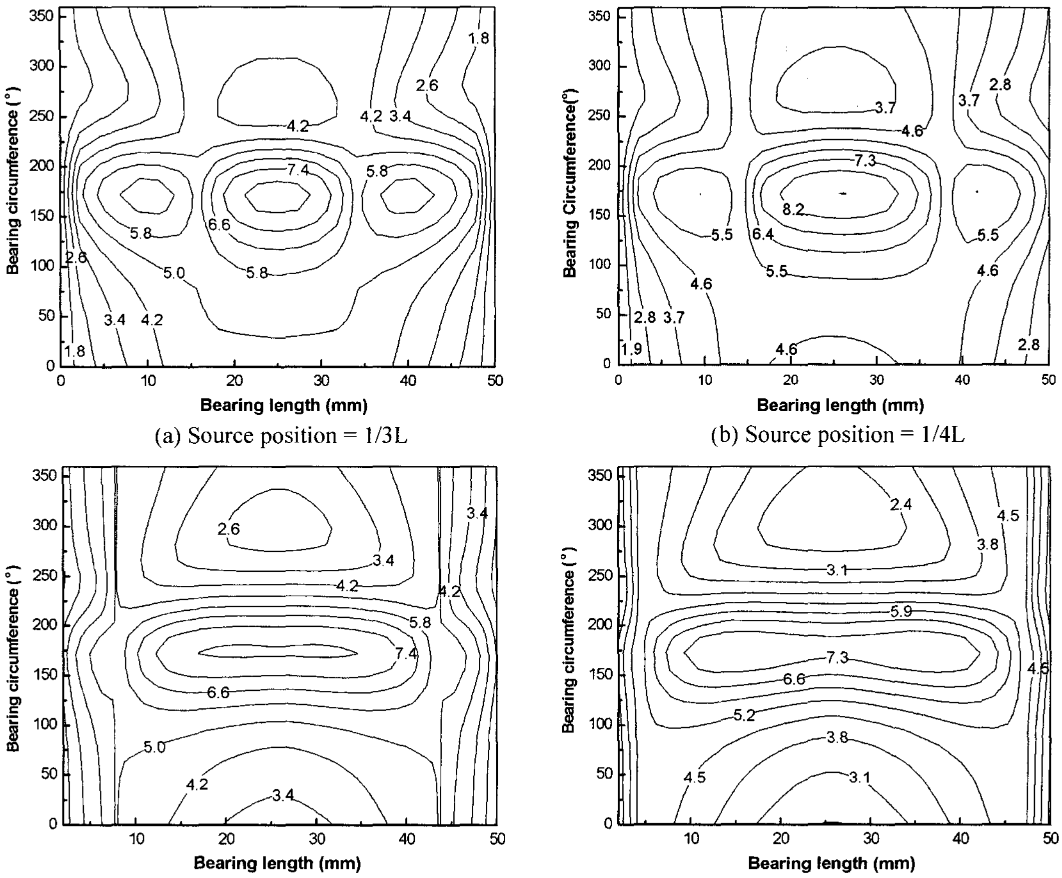
(a) Source position = 1/3L

(b) Source position = 1/4L

(c) Source position = 1/7L

(d) Source position = 1/11L

Fig. 3. Pressure distributions in air-lubricated journal bearing with different the air source positions (L/D=1, Ps=5atm, DN= 2,000,000, Eccentric ratio=0.6).



(a) Source position = 1/3L

(b) Source position = 1/4L

(c) Source position = 1/7L

(d) Source position = 1/11L

Fig. 4. Pressure contours with different air source positions (L/D=1, Ps=5atm, DmN=2,000,000, Eccentric ratio=0.6).

therefore, the iteration step of the air pressure can be expressed in the following equation

$$\Delta P_{i,j} = -\left(\frac{\partial f_{i,j}(P)}{\partial P}\right)^{-1} f_{i,j}(P) \tag{13}$$

$$(P_{i,j})_n = (P_{i,j})_{n-1} + \Delta P_{i,j} \tag{14}$$

The convergence condition is expressed in the following form.

$$\frac{\sum_{i=0}^{m}\sum_{j=0}^{n}\left|(P_{i,j})_n - (P_{i,j})_{n-1}\right|}{\sum_{i=0}^{m}\sum_{j=0}^{n}\left|(P_{i,j})_n\right|} \le 1\times10^{-5} \tag{15}$$

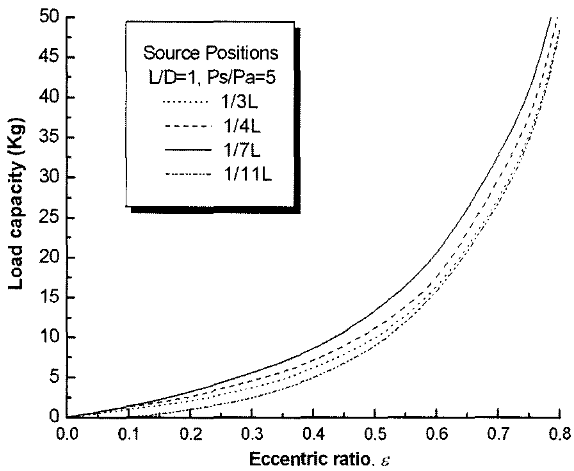


Fig. 5. Load capacity for different source positions (L/D=1, Ps=5atm, DN=2,000,000).

Calculation Results

A number of investigations about the pressure of air films have been explained as an ideal static and steady state film. But we don't have to think about the behaviors of air film in high-speed region of journal. Air film have the possibility that the hydrodynamic effects is created. At the results of that, these hydrodynamic effects have an influence to the bearing stiffness and dynamic coefficients.

Fig. 2 shows the pressure distribution at the source position 1/4L in accordance with the eccentric ratio. The pressure distribution shows that the more the eccentricity increases, the higher the pressure of the circumferential direction at the part of 100-250° by the hydrodynamic effect in high-speed region.

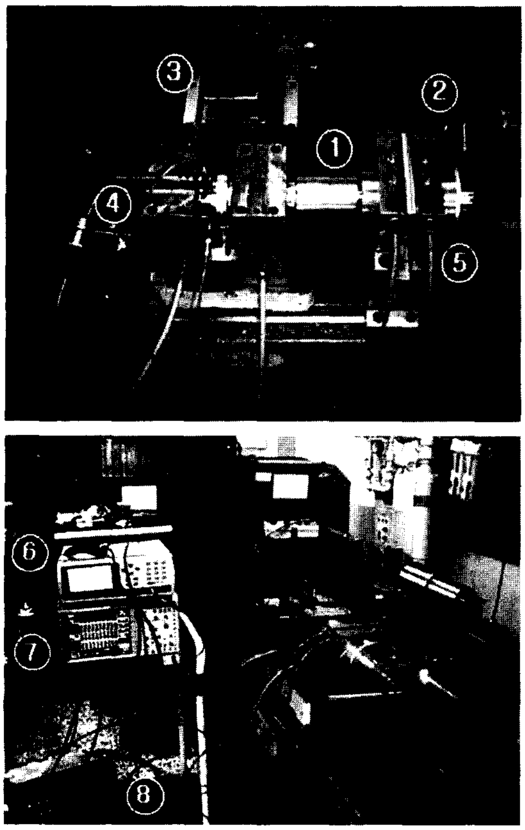


Fig. 6. Experimental set-up. ① Magnetic coupling, ② AC Spindle motor, ③ Dynamometer, ④ Transducer sensor, ⑤ Air spindle, ⑥ Transducer amplifier, ⑦ Digital oscilloscope, ⑧ Charge amplifier.

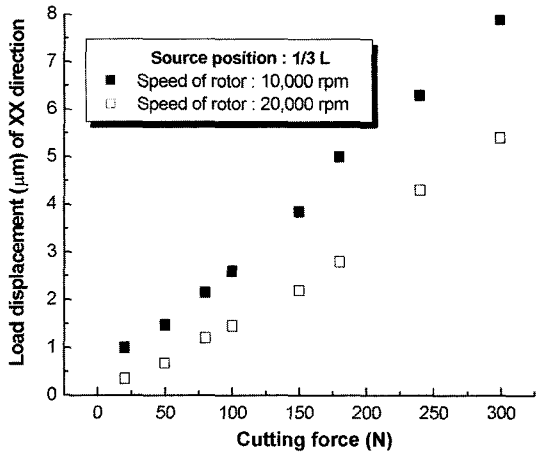
Hydrodynamic effects show the possibility to have an influence on the bearing stiffness. And the source positions too have an influence on the distributions of the pressure in air film and the bearing stiffness.

Fig. 3 shows the pressure profile of the three dimensions according to the source position and this pressure profile is expressed by the contour through Fig. 3. When comparing (a), (b), (c), (d) of Fig. 3, Fig. 4, the local hydrodynamic pressure (a) and (b) is created but the pressure (c) and (d) show distributing widely in the bearing.

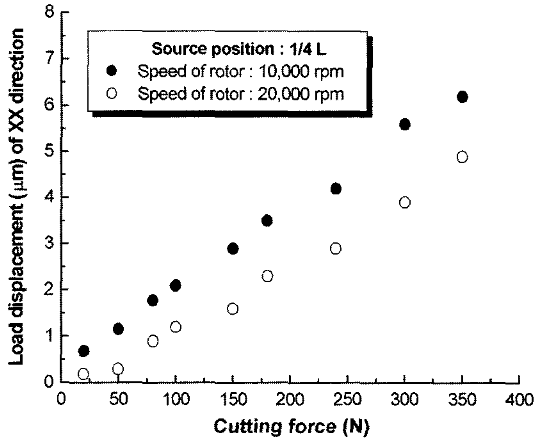
When comparing (c) and (d), the pressure distribution of (d) more appears to come down more than the pressure distribution of (c). As a result, the source position 1/7L has more advantage conditions than other source positions. These results can present the fact that the exiting bearing design of 1/4L in the bearing total length has to be reconsidered.

Fig. 5 show the results for the load capacity which was calculated by the summations of each pressures in air bearing. In general, air source position of air bearing with two row source have designed in 1/4 of bearing length.

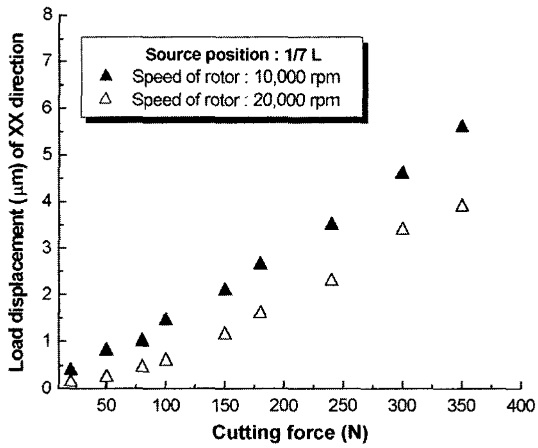
But Fig. 5 appear the different result with existing design value in high-speed region of journal. On the contrary, the load capacity of air source position in 1/7 of bearing length is higher than air source poison in 1/4 of bearing length. This result can be guessed that hydrodynamic effect in high-speed region has



(a) Source position = 1/3L



(b) Source position = 1/4L



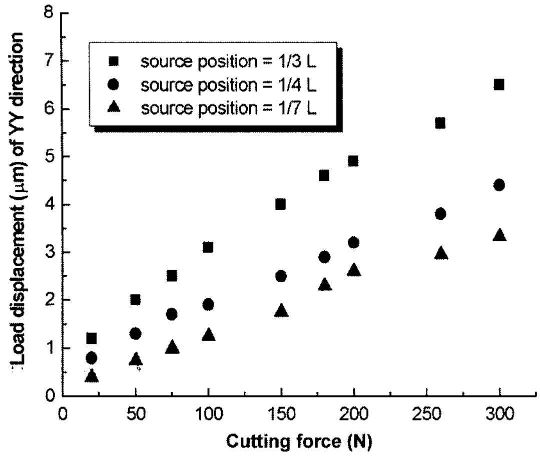
(c) Source position = 1/7L

Fig. 7. Relations between cutting force and displacement

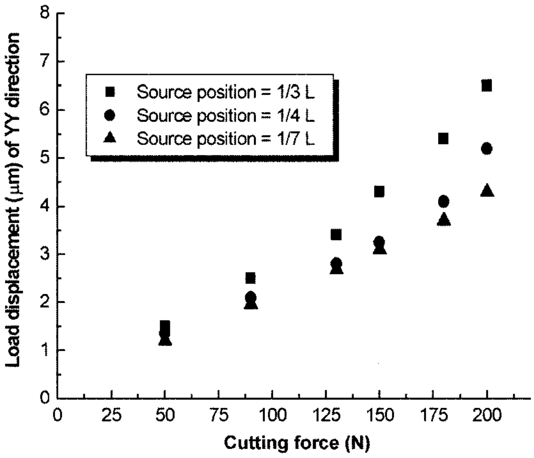
an influence mainly.

Experimental Procedure and Results

Fleming [2] as well as a number of research workers has measured the dynamic stiffness and damping of air bearing. But these studies are limited in application to the industry field because of having a lot of problem such as bearing clearance,



(a) The loading displacements in the xx-direction



(b) The loading displacements in the yy-direction

Fig. 8. Characteristics of loading capacity in accordance with source positions (L/D=1, Ps=5atm, DN=1,000,000, Clearance =20 μm).

the loading method and the special loading devices etc.

In this paper, In order to get practical information for the dynamic coefficient, the experimental method carrying out the cutting force signals in two dimension cutting and getting the displacement signals at this moment was applied. In an old experiment, the dynamic coefficient was measured by rotating the journal on attaching the unbalance mass and getting the displacement signals. But this experimental method has a disadvantage that the exact loading force and the exact loading points can't measure. The experimental method like Fig.6 can get both signals, displacement signal and loading force signal at the same time, as well as get two direction signals, xx direction signal and yy direction signal, at the same time.

At first, dynamic coefficient of air bearing with 2-row source have been analyzed with different journal speed. The experiment of dynamic coefficient was tested in DN 50,000 and DN 100,000. Dynamic coefficient of journal in high-speed region and low-speed of journal have absolutely appeared the difference of journal displacement. The displacement of journal for the same exciting force, as illustrated in Fig.7, can be founded that the displacement in high-speed journal appeared lower than that in low-speed journal.

Fig. 8 shows the difference of loading displacements of the journal in accordance with source positions. Compared with the displacement and cutting force in the cutting feed direction Fig. 8 (a), the displacement value of 1/7L is less than 1/4L at the same cutting force and Fig. 8 (b) shows the similar trends but not clear trends.

This means that the dynamic coefficient of 1/7L bearing is higher than the 1/4L bearing which is an old design values in high speed region.

Conclusion

The characteristics of air-lubricated film and the loading capacity according to the source position have been investigated through the theoretical analysis and the experiments. The theoretical analysis represents that the hydrodynamic effect occurred and source positions cause the hydrodynamic effect of the different shape. At the result, the different pressure-distributions according to the source positions have an influence to the dynamic coefficients and bearing stiffness. The experimental results too have proved these facts showing that the stiffness of 1/7L is superior to the stiffness of 1/4L.

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Nomenclature

$\bar{P}$ = non-dimension pressure ($\mathbf{P}_s/\mathbf{P}_a$)
P_a= atmospheric pressure
P_s = supply pressure
U_j = journal pressure
R = radius of journal
ω = journal angular velocity
h = film thickness
$\bar{h}$ = non-dimension film thickness about journal base circle
e = eccentric ratio
C = mean clearance about journal base circle
η = air viscosity
θ = circumferential direction coordination
ζ =bearing length direction coordination
Λ = bearing number
D =bearing diameter
L = bearing length
L_2 = source position
W_x = film force to circumferential direction
W_y =film force to circumferential direction
W = total film force

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