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FRICTIONAL CHARACTERISTICS OF THRUST BEARING IN SCROLL COMPRESSOR

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ABSTRACT

This paper presents frictional characteristics of thrust bearing in scroll compressor. The coefficient of friction in thrust bearing was measured by both elemental friction test and loss classification based on the performance test of actual scroll compressor, and investigated frictional characteristics focusing on the behavior of sliding portion which affects the generation of oil film by experimental and analytical approach.

Through measurements of the coefficient of friction, it was found that the coefficient of friction increases with decrease of contact pressure in the region that contact pressure is relatively low, and the coefficient of friction and the behavior of sliding portion have a strong correlation. In addition, these characteristics were supported by mixed lubrication analysis.

Based on experimental and analytical results obtained here, the technique of optimization of thrust bearing was shown. The optimization of thrust bearing achieved approximately 2% improvement of total efficiency.

1. INTRODUCTION

Frictional loss in thrust bearing occupies a large part of total mechanical loss in scroll compressor. Therefore, the accurate prediction of frictional loss in thrust bearing is one of the most important tasks for design of scroll compressor, and it is necessary for performance improvement to decrease the frictional loss in thrust bearing.

In the past, many researches about reduction of thrust bearing loss in scroll compressor by reduction of thrust load have been made. For example in recent work, application of static pressure assist bearing to CO_2 scroll compressor(1), analysis of back pressure control method for optimization of thrust load over a wider operating condition(2), and minimize of thrust load with new compliant mechanism(3) are reported. On the other hand, there are some reports mentioned the coefficient of friction in thrust bearing. Nishiwaki et al.(4) measured the coefficient of friction of the sliding portion under orbiting motion by elemental friction test. Tsubono et al.(5) measured the coefficient of friction in thrust bearing from the relationship between backpressure and mechanical loss. Ashitani et al.(6) reported the frictional loss in thrust bearing decreased by optimization of contact area. However, these studies are not enough to clarify frictional characteristics of thrust bearing, so further studies are required in order to obtain a deeper understanding of frictional characteristics.

In this study, we measured the coefficient of friction in thrust bearing by both elemental friction test and loss classification based on the performance test of actual scroll compressor. And frictional characteristics are investigated focusing on the behavior of sliding portion which affects the generation of oil film by experimental and analytical approach.

2. MEASUREMENT OF THE COEFFICIENT OF FRICTION IN THRUST BEARING

2.1 Friction Test

In order to investigate frictional characteristics of sliding portion under orbiting motion, the frictional force was measured by elemental friction test simulating actual thrust bearing in scroll compressor. Schematics of test apparatus and specimens are shown in Figure 1 and 2 respectively. Lower specimen, calls orbiting specimen hereafter, mounted the moving stage, makes orbiting motion. Upper specimen, calls fixed specimen hereafter, is fixed and loaded in the direction of orbiting specimen by weights. Axial force and frictional force between both specimens are measured with the 3-component force sensor on upper side of the fixed specimen. The sliding surface is placed in the oil bath and the oil temperature is controlled with the heater and the circulation pump. The rotating speed is controlled by the inverter. In addition, The behavior of sliding portion can be measured with displacement sensors mounted in the fixed specimen. Specifications of specimens and test conditions are shown in Table 1 and 2 respectively. Two materials, aluminum and cast iron, are adopted, and orbiting specimens have similar surface treatments as actual thrust bearings. Moreover, all friction tests are carried out after sufficient running-in to eliminate the change of frictional force by running time[4].

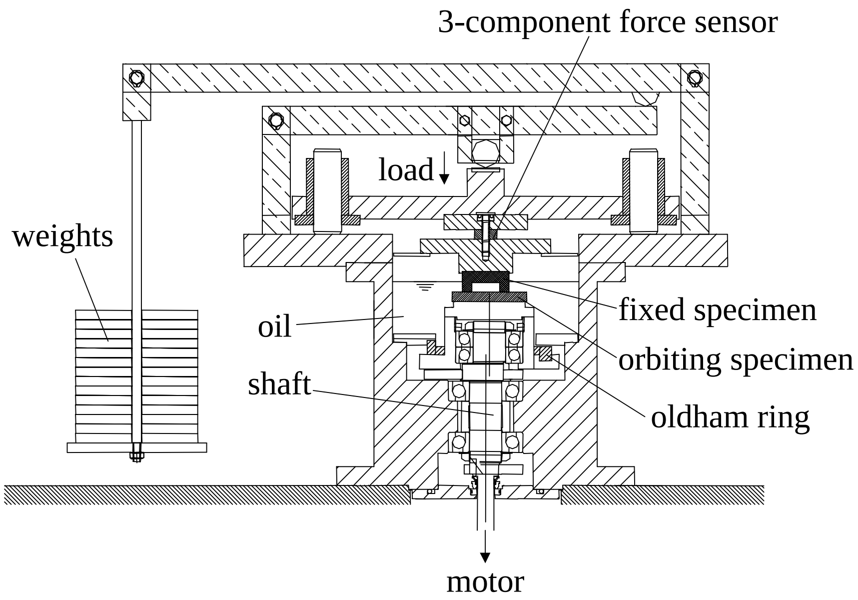

Figure 1 Schematic of test apparatus

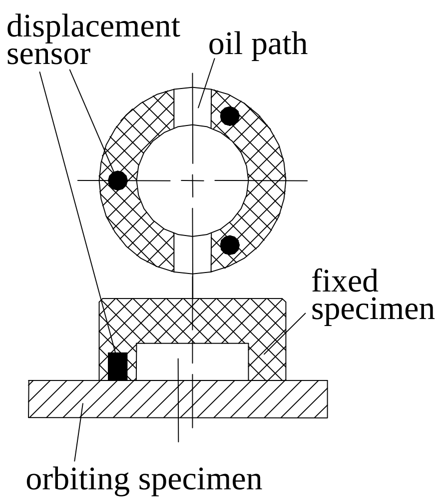

Figure 2 Schematic of specimens

Table 1 Specifications of specimens

	Fixed	Orbiting
Shape	Cylinder	Plate
Diameter [mm]	Outer: 50 Inner: 30	80
Material	(a) Aluminum (b) Cast iron	(a) Aluminum* (b) Cast iron**
Surface roughness Ra*** [μm]	(a) 0.09 (b) 0.21	(a) 0.21 (b) 0.24
Contact area [mm^2]	1057	

* Lubrication hardening anodic oxidation treatment
** Phosphating chemical treatment
*** The value after testing

Table 2 Test conditions

Orbiting radius	2.5mm
Rotating Speed	20 to 70 rps
Load	300 to 3000 N
Oil	POE
Oil viscosity	5 to 20 mPa·s

2.2 Evaluation of Frictional Loss in Actual Thrust Bearing

Frictional loss in actual thrust bearing can be evaluated by following procedure. Firstly, Total input power is separated into the indicative power and the sum of mechanical loss and motor loss by the measurement of cylinder pressure. Secondly, total mechanical loss is obtained by separating the motor loss with the measurement of the motor efficiency. Finally, frictional loss in thrust bearing is obtained by separating mechanical losses of each parts such as journal bearings, oldham ring, etc. from total mechanical loss. Here, the motor efficiency and each mechanical losses, above-mentioned, have already been measured by other experiments respectively. Using the thrust bearing loss W, The coefficient of friction m can be obtained as follows.

$$m = \frac{W}{F_z \cdot \left(2 \cdot p \cdot r \cdot N\right)} \tag{1}$$

Where, F_z is the thrust bearing load obtained by the measurement of cylinder pressure. r is the orbiting radius, and N is the rotating speed.

3. RESULTS AND DISCUSSIONS

3.1 Friction Test Results

Figure 3 shows an example of the frictional force measured by friction test. Here, the shaft rotates counterclockwise, and rotating angle f is defined as 0deg. when the direction of eccentricity accords with the x-axis. The position and the radius of each trajectory indicate the direction and the magnitude of frictional force respectively. That is, the greater the radius of trajectory, the larger the frictional force is. Since the frictional force acts in the opposite direction of sliding direction, the frictional force at f =0deg. is plotted on the negative region of y-axis. The measured data at f =0deg. was also located on the y-axis approximately.

Figure 4 shows the variation of coefficient of friction with contact pressure. Here, the coefficient of friction is the average value per rotation. In the region that the contact pressure is relatively high, the coefficient of friction decreased with decrease of contact pressure. And in this region, the variation of coefficient of friction was independent of rotating speeds. On the other hand, in low contact pressure, the coefficient of friction rose with decrease of contact pressure. Moreover, as the rotating speed increased, the point that the coefficient of friction converted to increase shifted to higher side of contact pressure and the coefficient of friction became higher.

In order to investigate the cause that the coefficient of friction rose in low contact pressure, the behavior of sliding portion and the frictional force were measured simultaneously. Figure 5 shows the variation of the coefficient of friction and tilt angle between both sliding surfaces obtained from outputs of three displacement sensors. From the result of displacement measurement, it was found that the tilt increased in less than a certain value of contact pressure although tilt angle is kept low in higher contact pressure. Since the direction of tilt agreed with that of eccentricity, it is considered that the tilt occurred by the centrifugal force on eccentric parts. Taking notice of the variation of coefficient of friction and tilt angle on each rotating

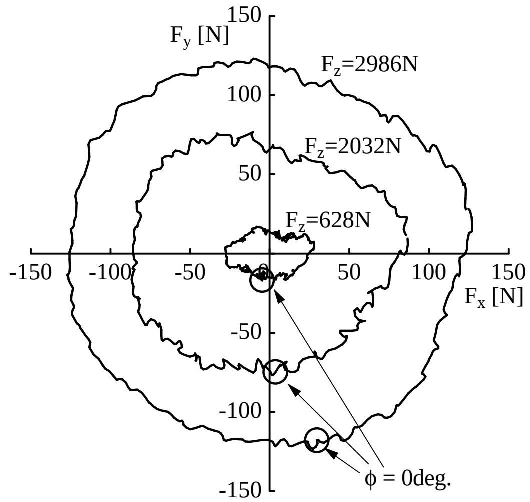

Figure 3 Trajectories of the frictional force
Aluminum, N=30rps, η=5-9mPa·s

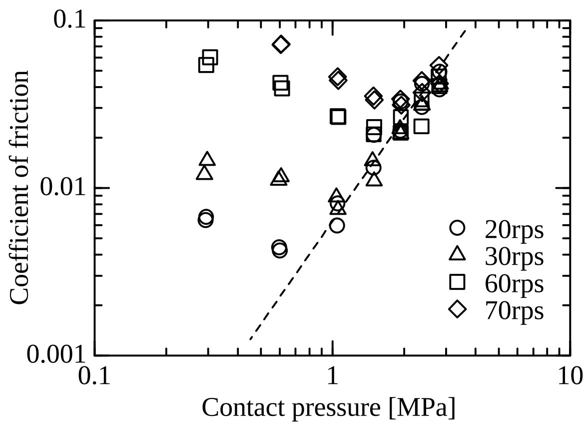

Figure 4 The variation of coefficient of friction with contact pressure
Aluminum, η=5-9mPa·s

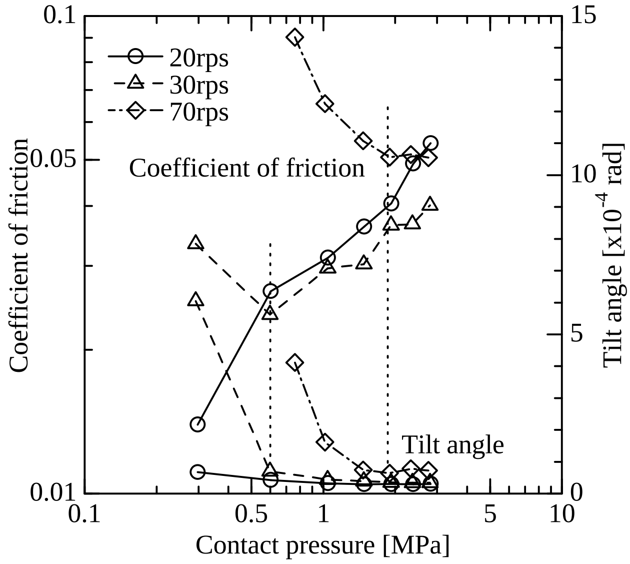

Figure 5 The variation of coefficient of friction and tilt angle with contact pressure
Aluminum, η=5-9mPa·s

speed, it was found that the point that the coefficient of friction converted to increase is correlated with the point that tilt angle became rising. That is to say, it is considered that the increase of behavior in sliding portion is one of the causes that the coefficient of friction rose in low contact pressure.

3.2 Analysis with Mixed Lubrication Theory

For theoretical explanation of the relationship between the tilt of sliding surface and the coefficient of friction obtained from friction test, mixed lubrication analysis on sliding surface was carried out. In this study, the analysis was based on Soda's theory(7) that can obtain the frictional characteristic with comparatively simple way.

According to Soda's theory, the coefficient of friction under mixed lubrication is determined by the proportion of fluid and boundary lubrication. That is, using the coefficient of friction under boundary lubrication m_b and fluid lubrication m_f, the coefficient of friction under mixed lubrication m is given by the following equation.

$$m = c \cdot m_b + (1 - c) \cdot m_f \qquad (2)$$

Where, c is the share of direct contact part in total bearing area obtained from the oil film pressure analysis in the sliding surface.

Analytical model and coordinates are shown in Figure 6. Inner and outer radii of contact area are defined as R_{in} and R_{out} respectively. The orbiting specimen makes a orbiting motion in radius r, and is supposed to be tilted at an angle of a in the direction of eccentricity according to the result of friction test. The sliding surface is assumed not to be deformed by load or heat. Clearance between both specimens is h, and the minimum clearance h_0 occurs in opposite direction of eccentricity.

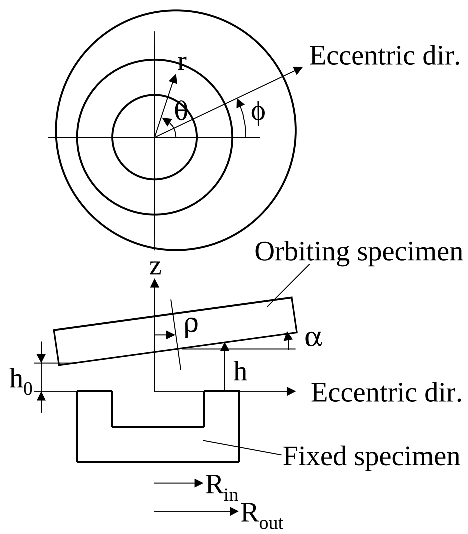

Figure 6 Analytical model and coordinates

Oil film pressure P in the sliding surface can be calculated by solving Reynolds equation expressed as below.

$$\frac{\partial}{\partial r}\left(rh^3\frac{\partial P}{\partial r}\right)+\frac{\partial}{\partial q}\left(\frac{h^3}{r}\frac{\partial P}{\partial q}\right)$$
$$=6m\left[\frac{\partial}{\partial r}(hrV_r)+\frac{\partial}{\partial q}(hV_q)-2rV_r\frac{\partial h}{\partial r}-2V_q\frac{\partial h}{\partial q}+2rV_z\right] \quad (3)$$

In equation (3), h is the viscosity, V_r, V_q, and V_z are velocity components of orbiting specimen in directions of r, q, and z respectively expressed as below.

$$V_r = rw\sin(q-f), \qquad V_q = rw\cos(q-f), \qquad V_z = -w\frac{\partial h}{\partial q} \quad (4)$$

Where, w is the angular velocity. Clearance h is given as follows using the minimum clearance h_0 and the tilt angle a.

$$h = h_0 + R_{out}\tan a\left[1+\frac{r}{R_{out}}\cos(q-f)\right] \quad (5)$$

The distribution of oil film pressure can be obtained by solving the equation (3) numerically. Boundary condition are given by

$$P(R_{in},q)=P(R_{out},q)=0, \qquad P(r,q)=P(r,q+2p) \quad (6)$$

Reynolds boundary condition

$$P(r,q^*)=\left.\frac{\partial P}{\partial q}\right|_{q=q^*}=0 \quad (7)$$

is also considered with negative pressure. In equation (7), q^* should be determined by boundary condition during iteration.

Once the tilt angle a is given, The clearance distribution h can be obtained as equation (5) with the balance between the oil film pressure and the external force. Both sliding surfaces are supposed to be contact when the clearance h is less than a value determined from the sum of fixed and orbiting surface roughness.

Figure 7 shows the variation of coefficient of friction and share of direct contact part with load under constant tilt angle. Oil film is generated by the squeeze action with variation of the clearance occurred from the tilt of sliding surface. When the load is low, it is operated under conditions of fluid lubrication which is kept sufficient thickness of oil film. In this region, the coefficient of friction is small and decreased with the increase of load. As the load become higher, the thickness of oil film reduced, and part of the sliding surface come into contact. In this region, called mixed lubrication, the share of direct contact part and the coefficient of friction increased with the increase of load.

Figure 8 shows the variation of coefficient of friction with tilt angle under constant load. Although the

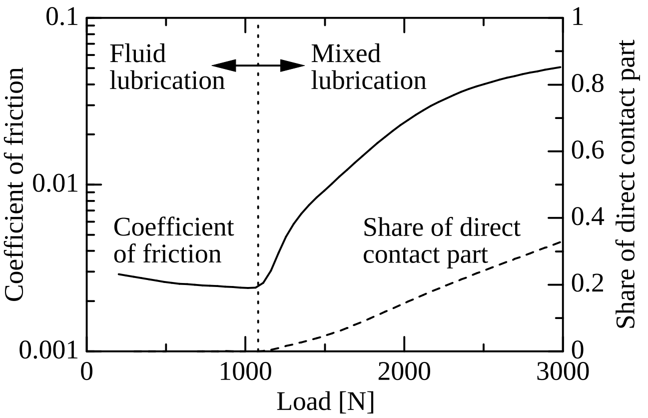

Figure 7 The variation of coefficient of friction and share of direct contact part with load
N=30rps, $\alpha=10^{-4}$rad, η=5mPa·s

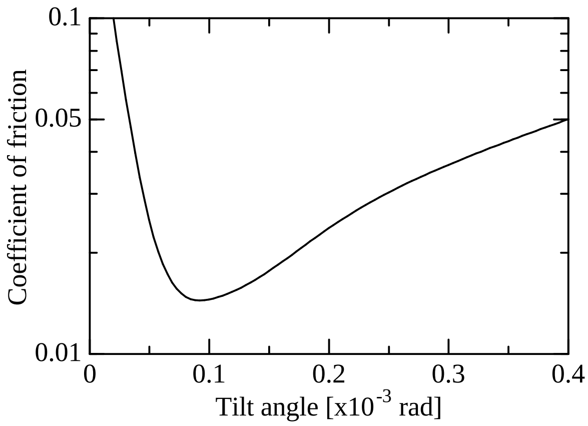

Figure 8 The variation of coefficient of friction with tilt angle
N=30rps, F_z=2000N, η=5mPa·s

coefficient of friction decreased because of squeeze action as the tilt angle increased, the coefficient of friction turned to increase with reduction of load capacity. That is, The optimum value of tilt angle exists in terms of minimization of coefficient of friction.

In order to compare the analysis result with the result of friction test, additional calculations in the same conditions of friction test were carried out. Here, tilt angles measured in friction test were applied to the calculation, and the coefficient of friction under boundary lubrication was determined by fitting to the result of friction test. Comparison between the analysis and the friction test is shown in Figure 9. It was found in Figure 9 that the analysis is in agreement with results in friction test quantitatively with respect to the characteristic that the coefficient of friction increases in low contact pressure. Therefore, it is considered that the characteristic that the coefficient of friction increases with increase of behavior in sliding portion was supported analytically.

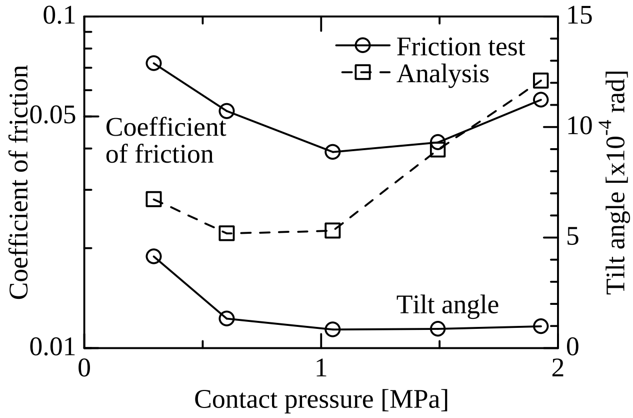

Figure 9 Comparison between analysis and friction test
Cast iron, N=60rps, η=5-8mPa·s

Table 3 Specification of scroll compressor

Displacement	39ml
Refrigerant	R410A
Oil	POE
Motor	DC Brushless

3.3 Relationship between the Coefficient of Friction and the Behavior of Orbiting Scroll in Actual Thrust Bearing

Through the friction test and the analysis with mixed lubrication theory mentioned above, it was found that the frictional characteristics of sliding portion under orbiting motion is strongly related to the behavior of sliding portion. For the next step, the coefficient of friction and the behavior of sliding portion in the actual thrust bearing are measured. The scroll compressor for commercial air conditioner with R410A is used for the measurement. Specifications of the compressor are shown in Table 3. The tilt angle of orbiting scroll is obtained by measuring axial displacements at the end plate of orbiting scroll with displacement sensors, and the coefficient of friction is obtained by the procedure stated in 2.2. The contact pressure is set with the control of driving pressure.

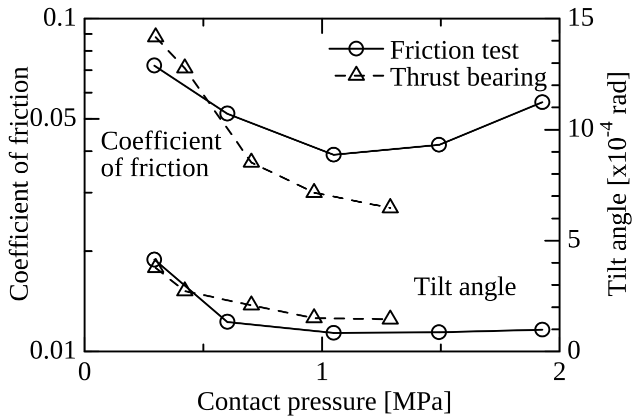

Figure10 Comparison between thrust bearing and friction test
Friction test: Cast iron, N=60rps, η=5-8mPa·s
Thrust bearing: N=56rps

The comparison between the test result in actual thrust bearing and in friction test is shown in Figure 10. The tilt angle of orbiting scroll increased with the decrease of contact pressure. Since the direction of tilt agreed with that of radial gas force approximately, it is considered that the tilt occurred by the moment from radial gas force. Moreover, the coefficient of friction was also increased with decrease of contact pressure. That is, the coefficient of friction increased with increase of tilt between sliding surfaces which corresponds with the result of friction test, though the mechanism how the tilt occurs is different from that in friction test.

From the results presented above, It is effective for reduction of coefficient of friction to keep the tilt of orbiting scroll as small as possible. Furthermore, it is considered that the optimization of contact pressure of thrust bearing by adjusting the bearing area can be effective for reduction of the thrust bearing loss, since the coefficient of friction has minimum value to contact pressure.

4. REDUCTION OF FRICTIONAL LOSS BY OPTIMIZATION OF THRUST BEARING

In order to confirm the effect of frictional loss reduction by optimization of thrust bearing based on the frictional characteristics mentioned above, compressor performance test was carried out. The improved scroll compressor for commercial air conditioner with R410A was examined. In the improved compressor, thrust load is supported at the outer side of the end plate of orbiting scroll to keep the tilt of sliding surface as small as possible. And the contact pressure of the thrust bearing is optimized by reducing the thrust bearing area by 17% as against conventional type. The performance test resulted in approximately 2% improvement of total efficiency at rated condition. A similar optimization has already been applied to scroll compressor for electric vehicle(8) and for gas engine heat pump (GHP)(9) as well as for commercial air conditioner mentioned here. We try to apply the optimize technique obtained in this study to wider variety of scroll compressors so as to reduce the energy consumption.

5. CONCLUSIONS

Frictional characteristics of thrust bearing in scroll compressor are investigated by both experimental and analytical approach focusing on the behavior of sliding portion. The following conclusions are obtained from this study.

- In the region that contact pressure is relatively low, the coefficient of friction rose with decrease of contact pressure, although the coefficient of friction decreased with decrease of the contact pressure in high contact pressure.
- The coefficient of friction and the behavior of sliding portion have a strong correlation. The coefficient of friction increased as the behavior of sliding portion increased.
- The characteristic that the coefficient of friction rose with increase of behavior in sliding portion was supported by mixed lubrication analysis.
- The optimization of thrust bearing achieved approximately 2% improvement of total efficiency in rated condition.

NOMENCLATURE

F_z	: thrust load	a	: tilt angle between sliding surfaces
h	: clearance of sliding surface	c	: share of direct contact
h_0	: minimum clearance of sliding surface	f	: rotating angle
N	: rotating speed	h	: viscosity
P	: contact pressure	m	: coefficient of friction
r	: radial coordinate	m_b	: coefficient of friction under boundary lubrication
R_{in}	: inner radius of thrust bearing	m_f	: coefficient of friction under fluid lubrication
R_{out}	: outer radius of thrust bearing	q	: angular coordinate
V_r ,V_q ,V_z	: velocity components of orbiting specimen	r	: orbiting radius
W	: thrust bearing loss	w	: angular velocity
z	: axial coordinate		

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