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International Compressor Engineering Conference

School of Mechanical Engineering

2012

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ABSTRACT

The present study focuses on the effect of the thickness and inner form of the thrust plate in a scroll compressor upon the lubrication features. A simplified model of a annular thrust slide-bearing with thinner thrust plate submerged in a refrigerant oil VG-56 was operated under pressure using R-22 as the pressurizing gas, where the pressure difference was adjusted from 0 to 1.0 MPa. The friction force and coefficient of friction were measured over a wide range of orbiting speeds. The wedge angle due to elastic deformation is naturally increased with decreasing thrust plate thickness, resulting in a clear improvement in lubrication characteristics of the thrust slide-bearing. Subsequently, similar lubrication tests were conducted for the thrust plate model with a realistic inner form, one as complicated as that in actual scroll compressors, while maintaining the thickness of the thrust plate as in the previous test. No significant change in lubrication features relative to those from the simplified annular model were identified, confirming the validity of using simplified annular model tests to assess the basic lubrication characteristics of the thrust slide-bearing in scroll compressors.

1. INTRODUCTION

Previous experimental and theoretical studies [1-6] of the high performance lubrication of the thrust slide-bearings in scroll compressors have identified wedge formation at the friction planes, due to the elastic deformation of the orbiting thrust plate, as the key to the outstanding improvement in lubrication performance. The wedge angle of the thrust plate, formed by elastic deformation of the thrust plate, increases with decreasing thrust plate thickness. Therefore, a measurable improvement of the lubrication performance with decreasing thrust plate thickness is expected. From this perspective, previous theoretical studies of the lubrication performance of the thrust slide-bearing were conducted [3-7] to optimize the lubrication performance of the bearing relative to the thrust plate thickness. The theoretical results confirmed that the lubrication performance is certainly improved with decreasing thrust plate thickness [3-7]. Experimental confirmation of this fundamentally significant feature of lubrication performance is undertaken in the present paper.

In this paper initial lubrication tests were carried out using a tribo-tester, in which the thickness of the annular thrust bearing model was decreased from 10.4 mm, as in previous studies [4, 5], to 7.4 mm for the present study. In the present experiments, the thrust plate was supported with a fixed thrust shaft through a pivot bearing, on which strain gauges were securely mounted in order to measure the friction forces at the thrust slide-bearing. The outside of the thrust-slide bearing was at an intermediate pressure, while the inside was at the suction pressure. The resulting pressure difference across the thrust plate was carefully varied in small increments from zero to 1.0 MPa. The friction force and friction coefficient were measured at each pressure difference to experimentally validate the improvement in lubrication performance due to the decreased thickness of the thrust plate.

In our previous experimental and theoretical studies, a comparatively complicated shape of the practical thrust slide-bearing was represented by a simplified ring-shaped-model, so-called “annular thrust bearing”, which has the same friction area as the actual bearing. Therefore, it is appropriate to confirm the validity of our simplified model in representing the lubrication performance of the actual thrust slide-bearing. To verify the model assumptions, lubrication tests were conducted for both the real-shaped thrust bearing and the annular one with the same friction area. In experiments, the scroll wrap portions were entirely removed from the real orbiting and fixed scrolls, since they do not support the thrust load at all, thus exposing only the thrust slide-bearing portion. The thickness of the real orbiting scroll thrust plate was fixed at 10.4 mm, in order to compare with the annular thrust plate model of the same thickness. From direct comparisons of the two sets of tests, the effect of the friction shape can be identified from the experimental results.

Finally, after the tests, the friction surfaces of the thrust slide-bearings were carefully examined in order to examine the state of wear, and thereby identifying the wedge formation at the friction surface for both the real and the annular thrust slide-bearings.

2. THRUST SLIDE BEARING MODEL AND TRIBO-TESTER FOR LUBRICATION TESTS

A cross-sectional view of the compression mechanism in a high pressure scroll compressor is shown in Figure 1. Oil is pumped to the top end of the crankshaft, lubricates the eccentric bearing, and forces the orbiting thrust plate and the tip seal up against the orbiting wrap. Furthermore, the oil passes through a needle valve which regulates the oil pressure at an optimal intermediate pressure as it enters the suction chamber. The intermediate pressure presses the orbiting thrust plate up against the fixed thrust plate. The contact portion at periphery of fixed and orbiting scrolls is called thrust slide-bearing.

The end portions of the fixed and orbiting scroll wraps do not support the thrust load at all, and hence this part is not included to the thrust slide-bearing. For this reason, in the real thrust slide-bearing model, shown in Figure 2(a), the scroll wrap portions are removed. The inner shape basically conforms to the involute curve. The outer boundary is the orbiting circle. The thrust plate has an outer radius r_o of 65 mm and a thickness t of 10.4 mm. A clearance height of 1.25 mm is located at the inner region of the thrust slide-bearing. The bearing has an area A of 8773 mm^2.

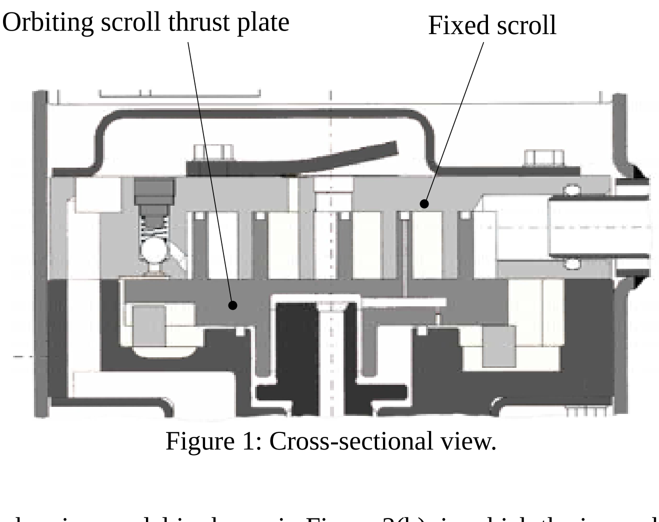
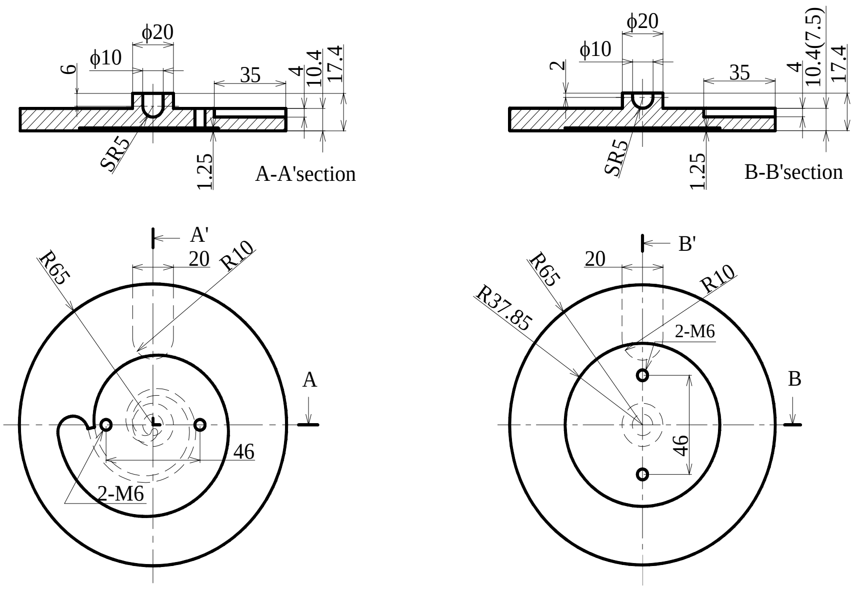


Figure 1: Cross-sectional view.

The simplified thrust slide-bearing model is shown in Figure 2(b), in which the inner shape is approximated as a circle. The outer radius r_o of the thrust plate remains the same 65 mm as in the real model, while the inner radius r_i is selected to be 37.85 mm so that the friction area remains the same as in the real model (A = 8773 mm^2). The same clearance height of 1.25 mm is maintained at the inner portion of thrust slide-bearing. A second annular model was manufactured in which the thickness was decreased to 7.5 mm, but with all other dimensions identical to the first model. Comparing the first and second annular models will permit a quantitative assessment of the improvement in lubrication performance due to the increased wedge angle at the periphery of the thrust



(a) Real shape model (b) Annular model

Figure 2: Fixed thrust plate models.

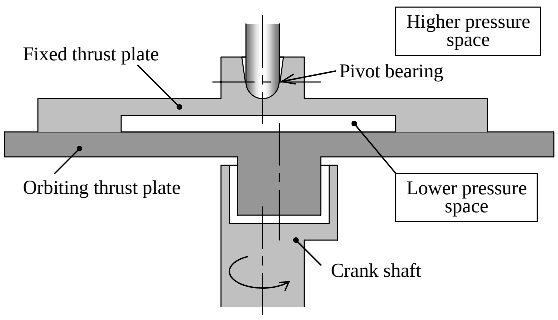


Figure 3: Thrust slide-bearing model.

Table 1: Surface roughness of test model.

		Surface roughness R_a [μm]
Annular model (t=10.4mm)	Fixed plate	0.70
	Orbiting plate	1.42
Real shape model (t=10.4mm)	Fixed plate	0.79
	Orbiting plate	1.39
Reduced thickness cylindrical model (t=7.5mm)	Fixed plate	0.85
	Orbiting plate	1.3

plate resulting from elastic deformation.

The thrust slide-bearing model is fixed with the axial load shaft through the pivot bearing, and pressed against the underside flat thrust plate which is driven by a motor to produce the orbiting motion, as shown in Figure 3. The drag force at the friction surface was precisely measured with strain gauges mounted on the fixed axial load shaft.

The initial surface roughness of both the fixed and orbiting plates is shown in Table 1. The material is an aluminum alloy for the fixed thrust plate and cast iron for the orbiting plate. The average roughness R_a is was 0.7 to 0.85 μm for the comparatively soft fixed thrust plate, while it was 1.3 to 1.42 μm for the comparatively hard orbiting thrust plates.

In the lubrication tests, the thrust slide-bearing model is submerged in refrigerant oil VG-56 for refrigerant R-22. The R-22 is stored in a tank outside the closed pressure vessel. The tank is heated and R-22 gas is fed into the closed pressure vessel to adjust the internal ambient pressure. The pressure in the internal space beneath the fixed thrust plate is regulated through a capillary tube and valve vented to atmospheric pressure outside the closed pressure vessel. The control valve at the end of the capillary tube is used to adjust the pressure of the internal bearing region. The fixed thrust plate is axially loaded by the axial load shaft and a spring in the axial load cylinder. This axial spring force, represented by F_s, can be controlled from outside of the closed pressure vessel. This thrust force, F_s, was measured with strain gauges (KYOWA: KFG-2-120-C1-11) mounted on the leaf spring and oriented along its axis. The strain gages were connected to a dynamic strain amplifier (KYOWA: DPM-6H). In addition, the fixed thrust plate is pressed down by the pressure force, represented by F_p, due to the pressure difference between the external and internal regions of the thrust slide bearing.

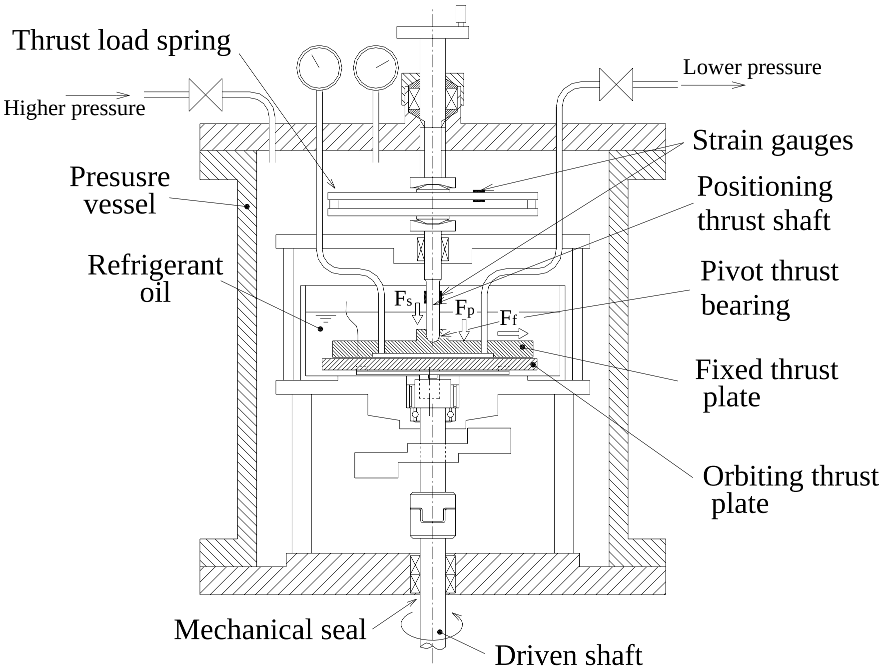


Figure 4: Cross-sectional view of tribo-tester [1-5].

Table 2: Major specifications of lubrication tests.

Pressure difference Δp [MPa]	0 ～ 1.0
Axial spring force F_s [N]	600
Gas thrust force F_p [N]	0 ～ 8887
Resultant thrust force $F_t(=F_s+F_p)$ [N]	600 ～ 9487
Orbiting speed N [rpm]	300 ～ 3600
Orbiting radius r_o [mm]	3.0
Sliding speed V [m/s]	0.0942 ~ 1.13
Refrigerant oil	VG-56
Refrigerant	R-22

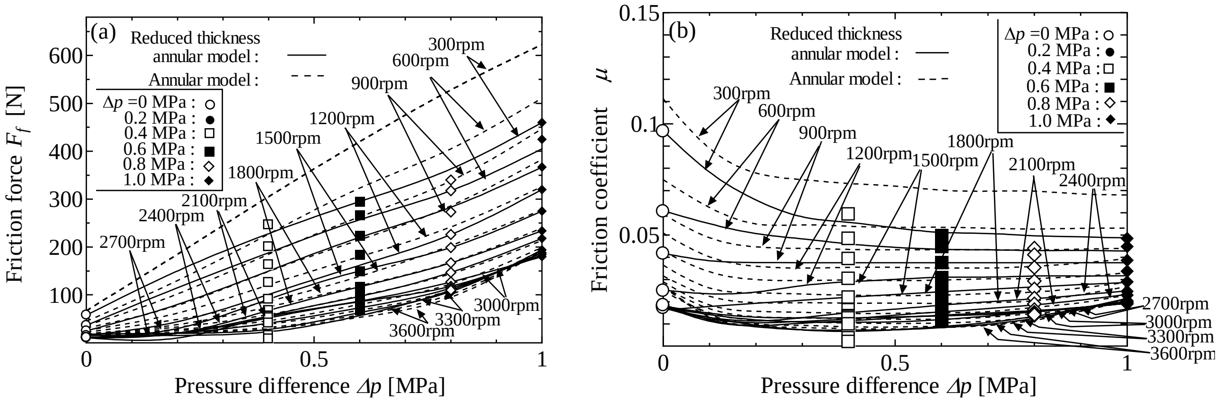
The orbiting thrust plate is driven by a motor located outside the closed pressure vessel, as shown in the cross-sectional view of the tribo-tester in Figure 4. The orbiting thrust plate tries to drag the fixed thrust plate. The drag force exerted on the fixed plate is the frictional force, represented by F_f, between the fixed and orbiting thrust plates. The fixed thrust plate is connected by a pivot coupling to the bottom end of the axial load shaft. Therefore, the frictional force, F_f, can be accurately measured by the strain gauges on the axial load shaft. The crankshaft rotation was measured by a rotary encoder. Measured axial spring force F_s, friction force F_f, friction surface temperature T_f, and crankshaft rotational pulse were recorded using a digital data recorder (ELMEC: EC-2371) and were monitored using a PC (FUJITSU: FMV TO307 & WAAP-WIN).

3. LUBRICATION TEST RESULTS

The major specifications for the present lubrication tests are given in Table 2. The internal pressure of the tribo tester was maintained at 1.1 MPa. the pressure difference control valve of the capillary tube was adjusted step-by-step so that the inside space pressure decreased from 1.1 to 0.1 MPa, that is, the pressure difference Δp increased from 0 to 1.0 MPa. The maximum gas thrust force, F_p, was 8887 N in addition to the axial spring force, F_s, of 600 N, which yielded a maximum resultant thrust force, F_t, of 9487 N. In calculating the pressure thrust force, F_p, it was assumed that the pressure acting on the friction surface varied linearly from 1.1 MPa at the periphery to 0 MPa at the inner circumference. The orbiting speed N was varied from 300 rpm to 3600 rpm with an orbiting radius of 3.0 mm, resulting in a bearing surface sliding speed, V, from 0.0942 to 1.13 m/s. The refrigerant was R22 and the oil was SUNISO VG-56.

3.1 Case of annular thrust plate model with reduced thickness

The lubrication test results for the annular thrust plate with reduced thickness of 7.4 mm are shown in Figure



(a) Friction force F_f (b) Friction coefficient μ

Figure 5: Lubrication test results for a annular thrust-slide bearing model with the thickness of t = 7.4 mm, compared with those for the conventional model with t = 10.4 mm.

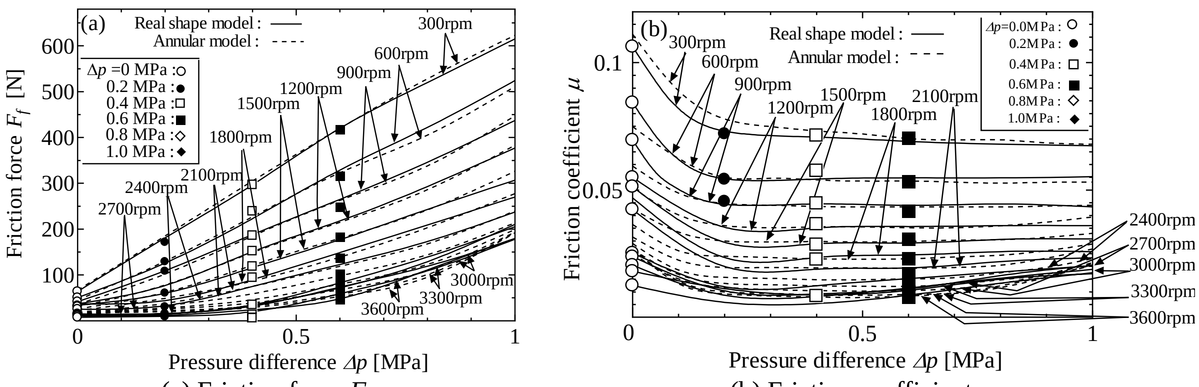
5(a) and 5(b), in which the friction force and the friction coefficient are presented, respectively. The test data are connected smoothly by solid lines. The dashed lines represent the data for the conventional plate thickness t of 10.4 mm. The abscissa is the pressure difference Δp, and the parameter is the orbiting speed N.

It is clear that the friction force for the reduced thickness model decreases over the whole range due to decreased thrust plate thickness. The friction force decrease becomes larger with increasing pressure difference. For instance, it reaches 160 N for N = 300 rpm and Δp = 1.0 MPa. The tendency for deceased friction force appears greatest in the lower range of orbiting speeds, up to 1200 rpm. In the lower orbiting speed range, the increased wedge angle due to elastic deformation of the thinner thrust plate results in a larger oil film pressure due to the dynamic pressure effect than in the thicker plate case. At increased orbiting speeds, for instance, above 1500 rpm, there is little difference between the solid and dashed lines, since a sufficiently large oil film pressure is induced for the both models due to the dynamic pressure effect.

As shown in Figure 5(b), it is also obvious that the friction coefficient decreases significantly over the low range of orbiting speed, up to 1200 rpm. The friction coefficient decrease is approximately constant for pressure differences over 0.4 MPa because both the friction force and the thrust load increase with increasing pressure difference. The difference in the friction coefficient is about 0.02 for the lowest orbiting speed of 300 rpm. With increasing orbiting speed, the difference becomes smaller, for instance, about 0.01 for 600 rpm and about 0.006 for 900 rpm and 1200 rpm.

3.2 Case of thrust plate model with real shape

The lubrication test results for the thrust plate with the real shape is shown in Figure 6, in which the friction



(a) Friction force F_f (b) Friction coefficient μ

Figure 6: Lubrication test results for the real shape thrust-slide bearing model with the thickness of t = 10.4 mm, compared with those for the annular model with t = 10.4 mm.

force and the friction coefficient are presented, respectively, over the pressure difference and with the parameter of orbiting speed. The test data for the real shape are connected smoothly by the solid lines. The dashed line represents the conventional data for the annular thrust plate with the same 10.4 mm thickness. From these test results, one may conclude that there is no significant difference between the solid lines for the real shape model and the dashed lines for the annular model in either the friction force or the friction coefficient. Only at the lowest orbiting speed of 300 rpm does the data for the real plate become slightly lower. For the remaining orbiting speed range, the solid and dashed lines intermingle, making it hard to identify any definitive difference.

Compared with results obtained for varying the thrust plate thickness, the results for the real shape versus the annular shape indicate little change in lubrication performance. This lack of significant difference can be explained as follows: For the simplified annular model, the inner diameter was selected such that the friction area was the same as for the more complicated real model. The average value of friction force over the whole friction area was essentially the same, though there may be slight differences the local values on the friction surface.

4. WEAR STATE OF FRICTION SURFACE

The friction surface of the reduced thickness model, after lubrication testing, is shown in Figure 7. The wear state drastically changes in the radial direction, quite similar to those previously found for the conventional thickness annular model. The inner area indicated by “A” is like a mirror. The central area, indicated by “B”, shows severe abrasive scratches exhibiting circular paths due to orbiting motion. The abrasive scratches disappear in the outer region, indicated by “C”. These observations lead to the measurement of the average surface roughness of these areas. The measured surface roughness is presented in Table 3. The roughness distinctively decreases from 0.848 μm in outer region “C” to 0.337 μm in the inner area “A”. Profiles of the surfaces also are shown in Figure 8. These results of surface roughness significantly suggest that a wedge was formed between the friction surfaces of the thrust slide bearing, so that the inside area “A” was often rubbed and in contrast the outside “C” was floating.

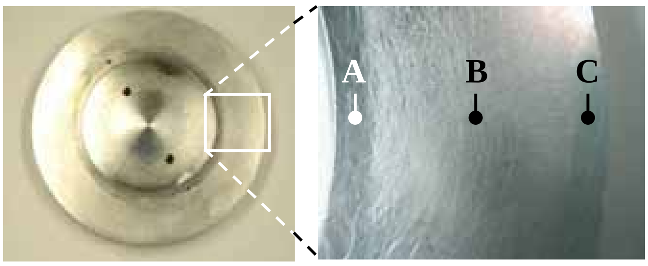


Figure 7: Wear state for reduced thickness annular model after experiments.

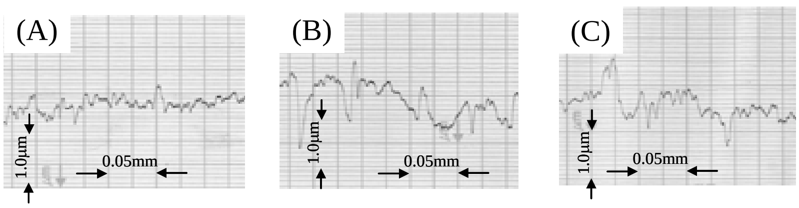


Figure 8: Friction surface profile for reduced thickness thrust plate model.

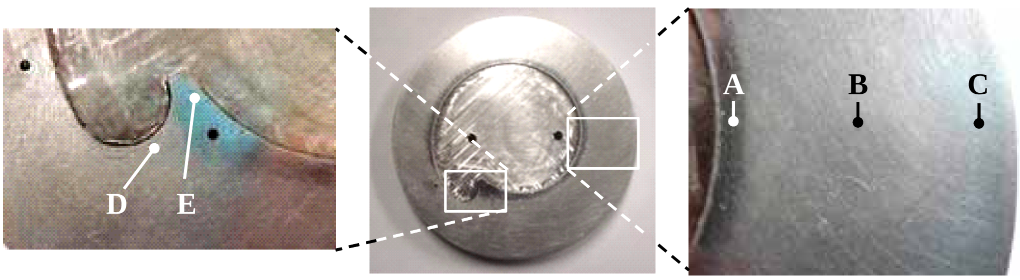


Figure 9: Wear state for real shape model after experiments.

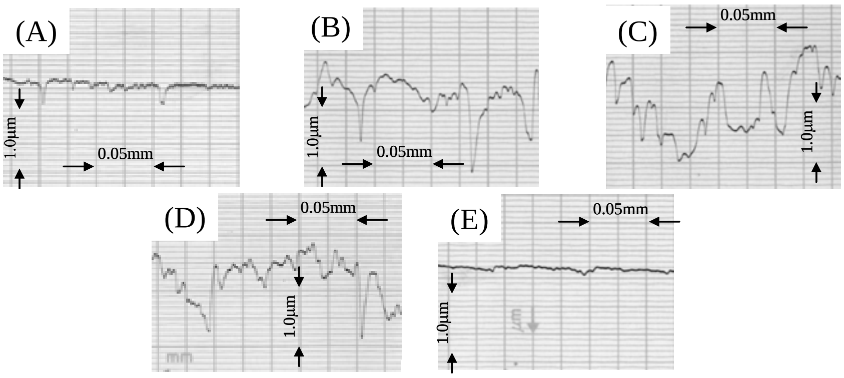


Figure 10: Friction surface profile for real shape thrust plate model.

Table 3: Measured average surface roughness after experiments.

	Surface roughness R_a [μm]				
Measurement position	A	B	C	D	E
Cylindrical model	0.064	0.357	0.628		
Reduced thickness cylindrical model	0.337	0.531	0.848		
Real shape model	0.150	0.510	0.649	0.519	0.130

The friction surface of the real shape thrust plate model is shown in Figure 9. At position “E”, a mirror-like surface was observed as well as at inner area “A”. Profiles of the measured surface of real shape thrust plate model are shown in Figure 10. The surface roughness decreases from the outer area to the inner area. The surface roughness of point “E” is especially small, 0.13 μm, the smallest of all points, as shown in Table 3. As a result, it is confirmed that the real shape thrust plate also forms a wedge which is larger in the outer region.

5. CONCLUSION

Lubrication tests for the thrust slide-bearing of scroll compressors were conducted in a closed vessel pressurized with refrigerant R-22 gas, focusing on the influence of the thrust plate thickness, reduced from 10.4 mm to 7.4 mm, on the lubrication performance. Frictional forces and the friction coefficients were measured for a variety of pressure differences from 0 to 1.0 MPa and for orbital speeds from 300 to 3600 rpm. In addition, the wear state of the friction surface was carefully examined after the lubrication testing. The friction force and the friction coefficient significantly decrease with decreased thrust plate thickness. Lubrication performance was improved with the thinner thrust plate, especially over the lower orbiting speed range. In addition, from the measured roughness of the friction surfaces, it was observed that the surface wear has been reduced by improved lubrication performance.

In addition, similar lubrication tests were conducted for the thrust plate model with a realistic inner form, one as complicated as that in actual scroll compressors, while maintaining the thickness of the thrust plate as in the simplified annular thrust plate test. No significant change in lubrication features relative to those from the simplified annular model were identified, confirming the validity of using simplified annular model tests to assess the basic lubrication characteristics of the thrust slide-bearing in scroll compressors. One may conclude here that the development of high lubrication performance thrust slide-bearings for commercial scroll compressors can be achieved through testing or theoretical calculations using the annular thrust slide-bearing model with its simplified shape of the friction surface.

NOMENCLATURE

F_f : Friction force, N
F_p : Gas thrust force, N
F_s : Spring thrust force, N
F_t : Resultant thrust force, N
N : Orbital speed, rpm
R_a : Surface roughness, m
Δp : Pressure difference, Pa
μ : Friction coefficient, -

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ACKNOWLEDGEMENT

The authors would like to express their sincere gratitude to Mr. ***Shuichi Yamamoto***, Senior Councilor of R&D division, Panasonic Co. Ltd., Mr. ***Kiyoshi Imai***, Vice President, Corporate Engineering Division, Appliances Company, Panasonic Corporation and Mr. ***Masahiro Atarashi***, Director, Appliances Company, Panasonic Corporation for their collaborative support in carrying out this work and their permission to publish this study.