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Noriaki Ishii
Osaka Electro-Communication University

Tatsuya Oku
Osaka Electro-Communication University

Keiko Anami
Ashikaga Institute of Technology

Akinori Fukuda
Matsushita Home Appliance Company

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LUBRICATION MECHANISM AT THRUST SLIDE-BEARING OF SCROLL COMPRESSORS
(EXPERIMENTAL STUDY)

Noriaki Ishii¹, Tatsuya Oku², Keiko Anami³, Akinori Fukuda⁴

¹Profesor, Department of Mechanical Engineering, Osaka Electro-Communication University
18-8 Hatsu-chou, Neyagawa-city, Osaka, Japan
Tel: +81-720-20-4561; Fax: +81-720-20-4577; E-mail: ishii@isc.osakac.ac.jp

²Doctoral Course Student, Division of Mechanical and Control Engineering,
Graduate School, Osaka Electro-Communication University, 18-8 Hatsu-chou, Neyagawa-city,
Osaka 572-8530, Japan
Tel: +81-720-20-4565; E-mail: d04201@isc.osakac.ac.jp

³Assistant Professor, Department of Mechanical Engineering, Ashikaga Institute of Technology,
268-1 Ohmae-chou, Ashikaga-city, Tochigi, Japan,
Tel: +81-284-62-0609 (Ext.241); Fax: +81-284-62-9802; E-mail: anami@ashitech.ac.jp

⁴Chief Engineer, Air Conditioning Devices Division, Matsushita Home Appliance Company,
Matsushita Electric Industrial Co., Ltd., 5-1-5 Sakuragaoka, Kusatsu Shiga, Japan
Tel: +81-77-567-9829; Fax: +81-77-561-3196

ABSTRACT

This study focuses on a significant effect of the pressure difference between the outside space pressurized by the intermediate pressure and the inside suction chamber upon the improvement in lubrication at the thrust slide-bearing of the scroll compressors. A thrust slide-bearing model submerged in the refrigerant oil was operated in a pressurized circumstance with R-22, where the pressure difference was adjusted at zero to 1.0 MPa and the frictional coefficient at the thrust slide-bearing was measured at a series of the orbiting speed. As a result, the outstanding improvement of lubrication at the thrust slide-bearing, caused by the pressure difference, was confirmed. The improvement of efficiency was caused by formation of a wedge at the sliding surfaces of the thrust slide-bearing.

1. INTRODUCTION

The scroll compressors, widely used for room air-conditioners, have the thrust bearing, where the orbiting thrust plate is firmly pressed on the fixed thrust plate, for its stable orbiting motion. There are several types of the thrust bearing, such as the slide bearing type, the ball bearing type, and so on. Among them, the thrust slide-bearing is the most common type for its better performance of low mechanical friction loss and low noise generation. In the thrust slide-bearing, the orbiting flat plate orbits on the fixed plate, being firmly pushed against the fixed one by an intermediate pressure between the discharge and suction pressures, and a special device such as an oil pump with high power is never used for lubrication of the sliding surface. Nevertheless, the thrust slide-bearing never induces any serious troubles in lubrication, such as a seizure of the sliding surfaces. The thrust slide-bearing exhibits rather better performance, than expected.

In the thrust slide-bearing, since the moving scroll orbits, the oil particles once trapped in the sliding space never flow-out in principle, being heated by shearing stress. It can be roughly explain, that’s why the thrust slide-bearing never induces a seizure. However, this explanation is surely inadequate in answering why the thrust slide-bearing can exhibit its better performance of low mechanical friction loss. A few studies for the thrust slide-bearing of the scroll compressors have been reported theoretically by Kulkarni (1990a, b), and experimentally by Nishiwaki et al.
(1996), but the present significant subject has not been solved at all. The scroll compressor efficiencies have been computer-simulated by Ishii et al. (2002a, 2002b, 2000, 1996, 1994, 1992), for its optimal design in efficiency, where the refrigerant leakage flow has been fundamentally examined in detail, but the insight of the frictional state at the thrust slide-bearing has not been well studied. It seems that some studies for the lubrication mechanism at the thrust slide-bearing are absolutely needed for developing better performance scroll compressors.

This study presents lubrication test results for the thrust slide-bearing of the scroll compressors, to address the key factor inducing the better performance in lubrication. The thrust slide-bearing was equivalently replaced by a simple model, composed of a cylindrical thrust plate representing the orbiting scroll and a flat thrust plate representing the fixed scroll. This study especially focuses on the pressure difference between the out- and in-sides of the thrust slide bearing. The outside is pressurized at an intermediate pressure between the discharge and suction pressures, while the inside is at the suction pressure. Caused by this pressure difference, the oil at the thrust slide-bearing flows from the outside space into the inside space and in addition a wedge is formed between the friction surfaces. In order to examine the effect of this pressure difference upon the lubrication performance, the thrust slide-bearing model was submerged in the refrigerant oil pressurized in the closed vessel with R-22, and the inner space of the cylindrical thrust plate was released to the atmospheric pressure. The lubrication tests were conducted, where the pressure difference was adjusted at zero to 1.0MPa, and the thrust force and the orbiting speed were varied in the range up to 9200N and 3600rpm, respectively. The frictional force and the temperature at the sliding surface were carefully measured. Finally, the friction surfaces of the thrust slide-bearing, after tested, were carefully observed to examine the state of wear.

2. SCROLL COMPRESSOR AND ITS THRUST SLIDE-BEARING

![Diagram of scroll compressor](image)

Figure 1: High-pressure type scroll compressor.

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A high-pressure type scroll compressor is shown in Figure 1a, which is the cross-sectional view of the whole compressor. The compression mechanism is at the upper portion of the closed vessel, and is driven by the motor at the middle portion of the closed vessel. As shown in Figure 1b, the compressed gas is discharged upward from the center port of the compression mechanism, and once flows downward through the motor rotor, and then flows upward, passing the outside of the motor stator. The refrigerant oil collected on the bottom of the closed vessel is pumped by an oil pump attached at the bottom end of the crankshaft, upward through a vertical hole inside the crankshaft.

As shown in Figure 1b, the oil pumped up to the top end of the crankshaft lubricates the eccentric bearing, and presses up the orbiting thrust plate and the tip seal on the orbiting wrap. Furthermore, the oil passes through the needle valve into the intermediate pressure space, where the pressure is adjusted at an optimal intermediate pressure, by the control vale releasing to the suction chamber. The intermediate pressure presses the orbiting thrust plate up to the fixed thrust plate. Thus, the orbiting thrust plate slides on the fixed thrust plate, thus forming a thrust slide-bearing, which is the shaded area in Figure 1c.

It should be noted here that the outside of the thrust slide-bearing is pressurized at the intermediate pressure and the inside is at the lower suction pressure, and thereby the pressure difference has arisen. This pressure difference will induce an oil flow from the outside to the inside of the thrust slide-bearing and in addition a wedge formation between the friction surfaces, thus yielding an outstanding improvement of the thrust slide-bearing performance in lubrication. It is the major purpose of the present study to reveal this significant effect of the pressure difference upon the performance in lubrication at the thrust slide-bearing.

### 3. TRIBO-TESTER FOR LUBRICATION TESTS OF THRUST SLIDE-BEARING

For convenience and simplification in performing lubrication tests of the thrust slide-bearing, the orbiting scroll thrust plate was replaced by a cylindrical thrust plate, which was arranged to the up side and fixed, while the fixed scroll thrust plate was replaced by a flat thrust plate, which was arranged to the down side and driven by the motor.
for orbiting motion, as shown in Figure 2. The test pieces of the thrust slide-bearing have the specifications shown in Table 1. The material is Aluminum alloy for the fixed cylindrical thrust plate and Cast iron for the orbiting thrust plate. The initial roughness is 0.7 and 3.0 µm, for the fixed and orbiting thrust plates, respectively.

In the lubrication tests, the thrust slide-bearing should be pressurized in the oil and the inside should be released to a lower pressure, and in addition the fixed cylindrical thrust plate should be loaded by the axial force in addition to the gas thrust force. In order to realize such a lubrication test, the thrust slide-bearing model was confined in the pressurized closed vessel, so-called a special “Tribo-tester,” as shown in Figure 3.

The thrust slide-bearing model is submerged in the refrigerant oil VG-56 for the refrigerant R-22 which is stored in the tank outside the closed pressure vessel. The tank is heated and R-22 gas is fed into the closed pressure vessel to adjust its pressure, while the inside space under the fixed thrust plate is released through the capillary tube to the atmospheric pressure, outside the closed pressure vessel. The control valve attached to the end of the capillary tube can adjust the pressure of the inside space of the thrust slide-bearing. The fixed thrust plate is axially loaded by the axial load shafts and the spring in the axial load cylinder. This axial spring force, represented by $F_s$, can be controlled from the outside of the closed pressure vessel. This thrust force $F_s$ was measured with the strain gauges attached on the lower axial load shaft. In addition, the fixed thrust plate is pressed downward by the pressure force represented by $F_p$, due to the pressure difference between the outside and inside spaces of the thrust slide-bearing.

The orbiting thrust plate is driven by the motor, outside the closed pressure vessel, thus dragging the fixed thrust plate. Its drag force 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 with the bottom end of the axial load shaft. Therefore, the frictional force $F$ can be correctly measured by the strain gauges on the axial load shaft. The friction temperature was measured by a thermocouple, attached closer to the friction surface, as shown in Figure 3.

### 4. LUBRICATION TEST RESULTS

The major specifications of the present lubrication tests are shown in Table 2. The inside of the tribo-tester was pressurized at 1.0MPa. First, the pressure difference control valve was entirely closed, where the inside space of the
thrust slide-bearing was at 1.0MPa and hence the pressure difference between the outside and inside spaces, $\Delta p$, was zero. Under such a condition, the lubrication tests were conducted, where the axial spring force $F_s$ and the orbiting speed at an orbiting radius of 3.0mm were varied from 0 to 1000N and 300 to 3600rpm, respectively. Representative results of lubrication tests at $\Delta p = 0$MPa & $F_s = 800$N is shown by blank circles in Figure 4, where the frictional force $F_f$, the friction coefficient $\mu (=F_f/F_s)$ and the friction surface temperature are presented versus the orbiting speed. $F_f$ takes 30N at 300rpm and decreases gradually with increasing the orbiting speed. Correspondingly $\mu$ takes 0.043 at 300rpm and decreases with increasing the orbiting speed, to 0.033 at 3000rpm, while the friction surface temperature increases from 32 to 37°C with increasing the orbiting speed.

Second, the pressure difference control valve was entirely opened, where the inside space of the thrust slide-bearing was at the atmospheric pressure, and hence the pressure difference $\Delta p$ was 1.0MPa. Under this condition, similar lubrication tests were conducted, thus resulting in the representative data, shown by black circles in Figure 4, where the gas thrust force $F_p$ was 8600N in addition to the axial spring force $F_s$ of 600N, and hence the resultant thrust force $F_t$ was 9200N. In calculating the gas thrust force $F_p$, it was assumed that the pressure acting on the friction surface changes linearly from 1.0MPa at the periphery to 0MPa at the inner circumference, as shown in Figure 5. Since the friction surfaces of the cylindrical fixed and orbiting plates are initially the flat plane of high accuracy, this pressure distribution along the friction surface is basically correct, and hence this upward gas forces were deducted from the downward ones acting on the upper surfaces. As a result, the friction coefficient $\mu$ at the thrust slide-bearing can be calculated by dividing the frictional force $F_f$ by the resultant thrust force $F_t$: $\mu = F_f/F_t$.

The frictional force $F_f$ increases to 70 to 80N, as shown in Figure 5a, whereas the friction coefficient $\mu$ outstandingly decreases. $\mu$ does not change with increasing the orbiting speed and takes a constant value of 0.008, about 1/5 to 1/4 of the friction coefficient at zero pressure difference. The friction surface temperature $T_f$ exhibits higher values than at zero pressure difference, from 42 to 53°C.

<table>
<thead>
<tr>
<th>Pressure difference $\Delta p$ [MPa]</th>
<th>0~1.0</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axial spring force $F_s$ [N]</td>
<td>0~1000</td>
</tr>
<tr>
<td>Gas thrust force $F_p$ [N]</td>
<td>0~8600</td>
</tr>
<tr>
<td>Resultant thrust force $F_t = (F_s + F_p)$ [N]</td>
<td>0~9600</td>
</tr>
<tr>
<td>Orbiting speed [rpm]</td>
<td>300~3600</td>
</tr>
<tr>
<td>Orbiting radius [mm]</td>
<td>3.0</td>
</tr>
<tr>
<td>Refrigerant oil</td>
<td>VG56</td>
</tr>
<tr>
<td>Refrigerant R-22</td>
<td></td>
</tr>
</tbody>
</table>

Figure 4: Lubrication test results of thrust slide-bearing with and without pressure difference, versus orbiting speed.
Finally, the pressure difference control valve of the capillary tube was adjusted step-by-step, so that the inside space pressure decreases from 1.0MPa to zero, that is, the pressure difference $\Delta p$ increases from 0 to 1.0MPa. Similar lubrication tests were conducted to obtain the representative results, shown in Figure 6, where the abscissa is the pressure difference $\Delta p$. As shown in Figure 6a, with increasing $\Delta p$, the resultant thrust force $F_t$ increases linearly, and correspondingly the frictional force $F_f$ also increases but shows its maximum value of 85N at $\Delta p=0.3$MPa ($F_t=3180N$) and once decreases and then increases again. As a results, with increasing $\Delta p$, the friction coefficient $\mu$ linearly decreases in its initial stage and then approaches to a constant value of about 0.01, about 1/4 of the value at zero pressure difference. The friction surface temperature $T_f$ increases from 30 to 48°C with increasing $\Delta p$.

6. WEAR STATE OF FRICTION SURFACE

The friction surface of the fixed cylindrical thrust plate (Aluminum alloy), after lubrication-tested, is shown in Figure 8, where a special attention should be paid to a clear difference of wear state along the radial direction. As observed in the enlarged photo of Figure 7b, the inside area indicated by “A” is like a mirror plane, and the middle
area indicated by “B” shows severe abrasive scratches describing a circle of orbiting motion, which disappears at the outside area “C”. Thereupon, the surface roughness of these areas was measured, as presented in Table 3, where the roughness distinctively decreases from 0.27 at the outside area “C” to 0.056 at the inside area “A”. 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 on the contrast the outside “C” had come floating.

7. CONCLUSIONS

Lubrication tests of the thrust slide-bearing of the scroll compressors were conducted in the closed vessel pressurized with the refrigerant R-22 gas, focusing on the effect of the pressure difference between the outside and inside spaces upon the lubrication performances, such as the frictional forces, the friction coefficients, the friction surface temperature and the wear state of the friction surface. As a result, the outstanding improvement in lubrication at the thrust slide-bearing, induced by the pressure difference, was confirmed: at the pressure difference of 1.0MPa, the friction coefficient decreased to about 1/5 on low orbiting speed operation and to about 1/4 on high speed operation, compared with the friction coefficient at zero pressure difference. The observation of wear state of the friction surface, after tested, suggested that such an outstanding improvement of lubrication performance was caused by a wedge formation between the friction surfaces, in addition to the oil flow into the inner space with lower pressure.

A wedge between the friction planes will be caused by the elastic deformation mainly due to high pressure loads of the orbiting scroll made of Aluminum alloy. It is thought that the measured data of the friction surface temperature were not correct in its absolute value, and the high temperature at the friction surface also will cause the elastic deformation of the orbiting scroll. Such a wedge formation at the friction planes seems to be a key factor of the outstanding improvement of lubrication performance at the thrust slide-bearing of the scroll compressor. Since the wedge formation will depend upon various factors, such as the back pressure on the orbiting scroll, the oil film pressure at the friction surface, the friction temperature, the inertial forces and so on, it is indeed a difficult subject to reveal the qualitative effect of the wedge upon the lubrication improvement. In order to solve such a difficult but serious subject, a certain theoretical approach is surely desired.

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