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Study on Bearing Characteristics of Reciprocating Compressor for Refrigerator

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ABSTRACT

In recent years, in order to reduce the energy consumption of refrigerators through reduction of input power and improved refrigeration capacity, reciprocating compressors are gradually converted to inverter type. Correspondingly, we have studied the sliding bearing characteristics of the crankshaft journal at variable speed which is the key to improving the efficiency of reciprocating compressor.

Generally, the characteristics of a sliding bearing are explained with a Stribeck curve. In the fluid lubrication regime, there is sufficient lubricant between metal surfaces. As the rotational speed decreases, the viscous resistance is reduced and it will result in the reduction of friction coefficient. Conversely, it is widely known that further reduction of rotational speed will shift to boundary lubrication regime where the friction coefficient increases due to metal to metal contact.

In the inverter reciprocating compressor, it is presumed that the sliding of crankshaft on bearing at high rotational speed is in fluid lubrication regime. The input power can be reduced with gradual decrease of friction coefficient by lowering rotational speed, but then the sliding characteristic of the crankshaft at each rotational speed was unclear such that when the speed is too low it will progress into the boundary lubrication regime.

This paper explains our experimental study to obtain a Stribeck curve at low load, low speed, and low viscosity oil in inverter operation using a bearing tester that estimate sliding crankshaft on bearing. The results obtained enable the prediction of sliding characteristic of crankshaft and optimization of the bearing property in reciprocating compressor.

1. INTRODUCTION

The Paris Agreement which is an international framework for combating global warming was signed in 2016. Its content is to reduce the emission of CO_2 and other greenhouse gases. Japan has set a goal to reduce greenhouse gas emissions by 26% compared to 2013 by 2030 in the INDC submitted to UNFCCC. On the other hand, EU has pledged to achieve 40% reduction as compared to 1990 by 2030. The INDC clearly showed that countries in the worldwide are setting high targets to achieve the aim of Paris Agreement. In this effort, there are two solutions for reducing greenhouse gases for household refrigerators. Firstly, it is the conversion to hydrocarbon refrigerants with low global warming potential such as R600a. Secondly it is to reduce power consumption by improving the energy efficiency as most of the power consumption is from the compressor. Hamaoka *et al*. (1996, 1997) studied energy saving in refrigerators by optimizing the rotational speed of an inverter reciprocating compressor according to the usage of the refrigerator. In recent years, works to further reduce power consumption were shown through development of improved heat insulation performance of refrigerators and reciprocating compressor that operates at rotational speed below 1200rpm. In order to achieve reduction of power consumption under such operating conditions, it is important to grasp sliding characteristics to predict and minimize sliding loss. This study focus on the plain bearing characteristics of the crankshaft and the bearing, which account for most of the sliding loss in reciprocating compressors. Conventionally, the characteristics of plain bearing under high rotational speed or fluid lubrication regime can be described by Reynolds equation. However, in the low rotational speed, theoretical calculation is complex when metal surface comes into contact. Kobayashi *et al*. (1998) reported the bearing characteristics of a reciprocating compressor for R134a using a journal bearing tester. In this test, the load was larger than that of R600a now widely used in refrigerators. Besides, the test was conducted on induction compressor operating at constant speed of maximum 3500rpm with a higher viscosity grade oil than current mainstream. Thus, it was challenging to accurately predict the sliding loss on inverter compressors. In this paper we discuss the experimental studies on bearing characteristics at low load, low speed, and low viscosity oil under variable speed operation by using a journal bearing tester. As we were able to estimate the reduction of the sliding loss in the reciprocating compressor accurately, the results from this study were applied in compressor development.

2. SLIDING BEARING THEORY

The general plain bearing characteristic is described by a Stribeck diagram as shown in Figure 1. The horizontal axis is the Sommerfeld number (S), a dimensionless quantity for evaluating the sliding state, and the vertical axis shows the friction coefficient. The equation for the Sommerfeld number is shown in equation (1).

$$S = \frac{\eta N}{P}\left(\frac{r}{c}\right)^2 \tag{1}$$

In above formula, η is oil viscosity [Pa·s], N is rotational speed [rps], P is bearing pressure [Pa], r is shaft radius [mm] and c is radial clearance [mm]. The right side of the Stribeck curve with larger Sommerfeld number describes the fluid lubrication regime where adequate oil film separates the metal surfaces to prevent wear and failure. The friction coefficient decreases with reduction of viscous resistance from the effect of lowering rotational speed or oil viscosity. The midsection of the Stribeck curve around the transition point is the mixed lubrication regime in which metal contacts occurs and sliding loss increases as the rotational speed is lowered. The lubrication shifts towards boundary lubrication regime on left side of the transition point as rotational speed is reduced further. The thin oil film is inadequate to prevent metal contact resulting in the increase of friction coefficient. In order to reduce the sliding loss, it is vital to clearly understand factors affecting the friction coefficient property in the mixed lubrication regime.

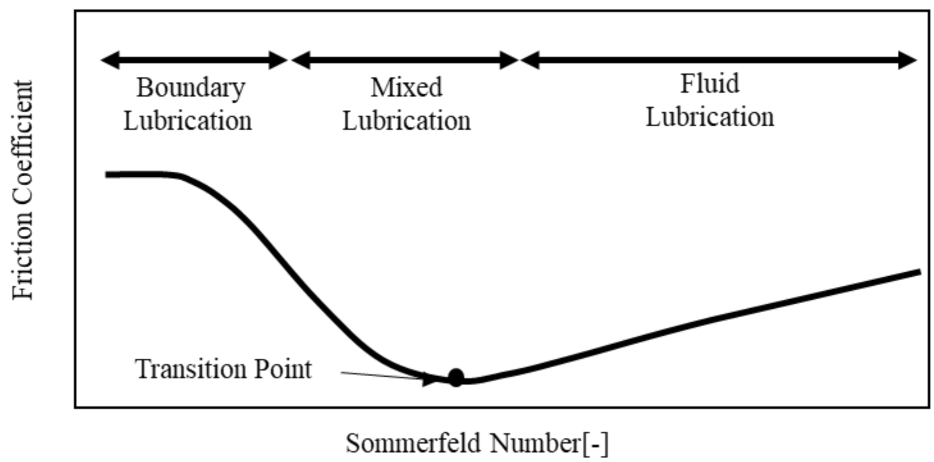

Figure 1: General Stribeck Diagram

3. EXPERIMENTAL STUDIES

3.1 Test Method

Figure 2 is the schematic diagram of the journal bearing tester used in this study. In this tester, the shaft is placed horizontally and driven by an inverter motor. The sliding length is defined by the bearing length. The bearing which is mounted on a housing is replaceable. The calculation of frictional force is explained as follows. A load (F) is applied to the bearing from the bottom of the housing. The rotational force (F_l) of the housing attached with the bearing is measured at a distance (l) from the center of the shaft to the measurement point by load cell as shown in Figure 2. F_l is generated as the shaft rotates and it is used to obtain the frictional force (F_r) in equation (2).

$$F_r = F_l \cdot \frac{l}{r} \tag{2}$$

The friction coefficient (μ) is derived from equation (3) using F_r and F.

$$\mu = \frac{Fr}{F} = \frac{F_l}{F} \cdot \frac{l}{r} \tag{3}$$

The shaft is lubricated by oil supplied to the oil pool on the housing which flows through the oil groove and fills the sliding portion of shaft and bearing.

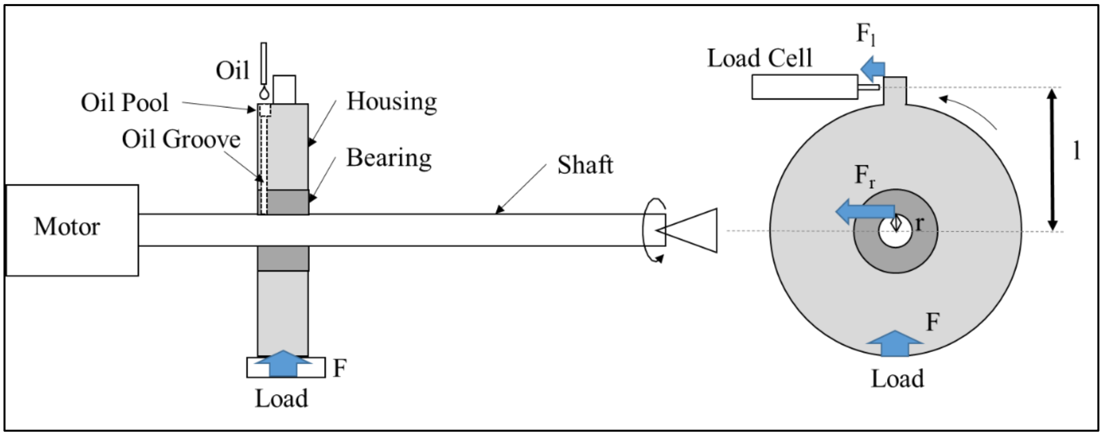

Figure 2: Schematic Diagram of Journal Bearing Tester

3.2 Specimen & Test Condition

The shaft and the bearing is fabricated from gray cast iron material. A surface modification is applied on the shaft by chemical conversion treatment. The shaft has a diameter of 16mm and the bearing length is 13mm. The surface roughness of the shaft and bearing after finishing is Ra 0.16µm and Ra 0.24µm respectively. Table 1 shows the test conditions of the journal bearing tester.

Table 1: Journal Bearing Test Condition

Item	Unit	Value
Load	N	98, 147, 196
Rotational Speed	rpm	100 ～ 2400
Oil Viscosity Grade (Mineral)	-	VG5, VG8

4. RESULTS & DISCUSSIONS

4.1 Effects of Load on Friction Coefficient

Firstly, it is important to verify that the change of friction coefficient can be measured even at low load condition using the journal bearing tester. Figure 3 shows the result for VG8 lubricating oil. The friction coefficient is expressed in ratio after normalized with respect to the coefficient at 196N applied load. The test was conducted at rotational speed of 200rpm and 800rpm. It was confirmed that at higher rotational speed of 800rpm there is fluid lubrication where the friction coefficient ratio increases as the load decreases. The result at low rotational speed of 200rpm shows mixed or boundary lubrication where the friction coefficient ratio decreases with load.

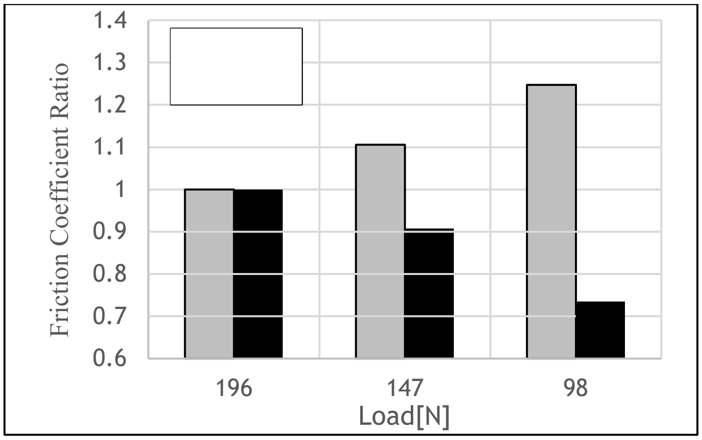

Figure 3: Friction Coefficient for Each Loads

4.2 Effects of Rotational Speed on Friction Coefficient

Next, it was determined whether a friction coefficient change in a low rotational speed regime can be evaluated by the bearing tester. Figure 4 shows the relationship between the rotational speed and the friction coefficient. The rotational speed was varied between 100rpm to 1200rpm. Both the rotational speed and friction coefficient are normalized with respect to the minimum friction coefficient obtained. The effect of rotational speed was investigated at 98N and 196N applied load with VG8 lubricating oil. Result in Figure 4 shows that when the speed ratio is greater than 1, there is the fluid lubrication regime where the friction coefficient ratio increases with rotational speed ratio for both applied load condition. It was also confirmed that as the speed ratio falls below 1, there is the mixed or boundary lubrication regime where the friction coefficient ratio increases with reduction of the rotational speed. Therefore, it was verified that the friction coefficient change in a low rotational speed regime can be measured by using the tester.

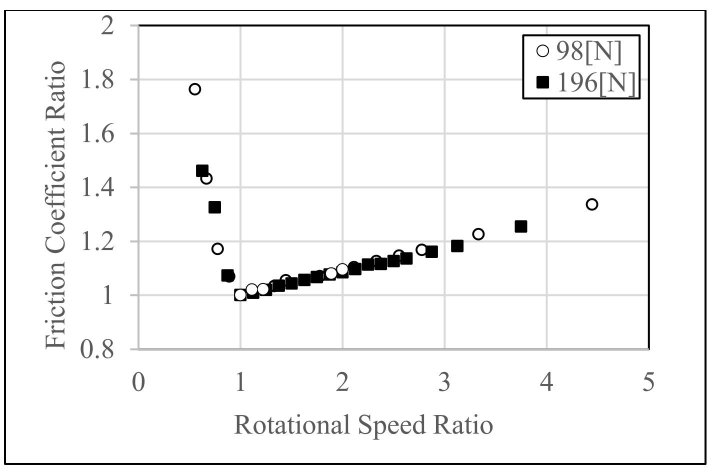

Figure 4: Relationship between Rotational Speed Ratio and Friction Coefficient Ratio

4.3 Effects of Oil Viscosity on Friction Coefficient

Next, we clarify whether a friction coefficient change using lower viscosity oil can be measured using the tester. Figure 5 shows the relationship between the rotational speed ratio and the friction coefficient ratio using lubricating oil viscosity grade VG5 and VG8. The test was performed at minimum 98N applied load and the rotational speed was varied in the range of 100rpm to 1200rpm. The result obtained shows similar trend as previous section, such that as the rotational speed ratio is higher than 1, there is the fluid lubrication regime where friction coefficient ratio increases with speed ratio for both oil. Similarly, as the speed ratio is less than 1, mixed or boundary lubrication regime appears where the friction coefficient ratio increases with reduction of rotational speed for both oil. Thus, it was verified that change of friction coefficient using low viscosity oil can be evaluated by using the tester.

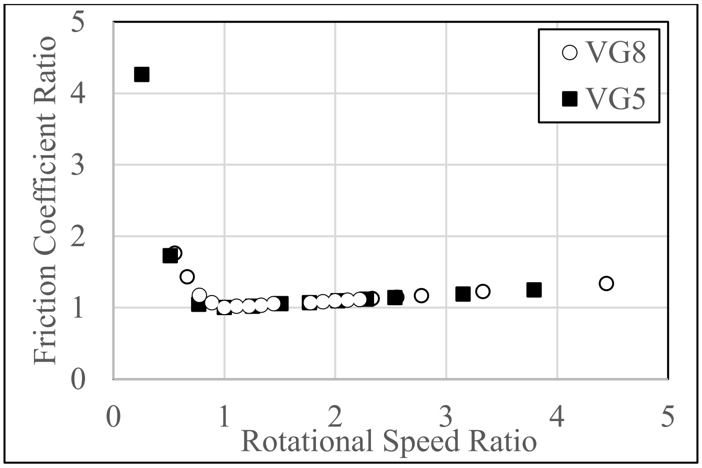

Figure 5: Relationship between rotational speed ratio and friction coefficient ratio due to oil viscosity grade

4.4 Sommerfeld Number

The results from the preceding tests show that the friction coefficient in the low load, low speed, and low viscosity condition can be evaluated with the bearing tester. The friction coefficient was plotted against the Sommerfeld number as shown in Figure 7. A Stribeck curve with a minimum transition point where mixed lubrication regime shifts to boundary lubrication regime is evident. Furthermore, it was able to detect the small variation of the friction coefficient in the region near the transition point. Therefore, this bearing test method is effective for the analysis of bearing characteristics in the low load, low speed, and low viscosity. Using the results from this test, it is possible to predict the sliding loss in the reciprocating compressor by an approximation equation in which the friction coefficient is a function of the Sommerfeld number. The prediction of sliding loss results using a reciprocating compressor are described in the next section.

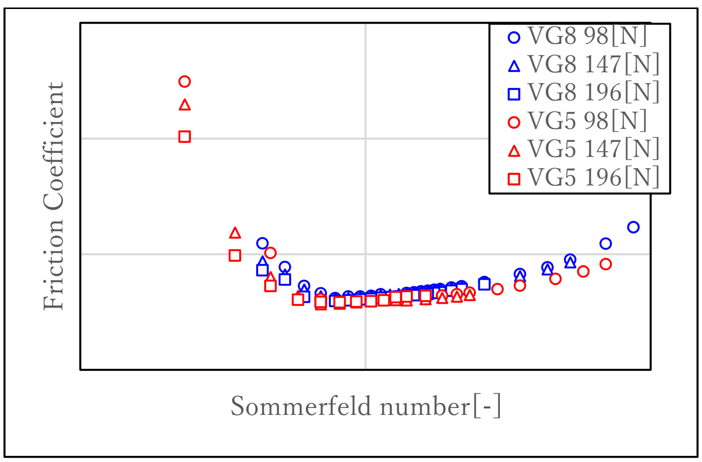

Figure 7 Relationship between Sommerfeld Number and Friction Coefficient

5. VALIDATION TEST ON RECIPROCATING COMPRESSOR

The sliding loss in a reciprocating compressor was calculated from the friction coefficient obtained in the experimental studies, and a validation test was performed. Table 2 shows the compressor test conditions using crankshafts with 4 different sliding length. The result is shown by comparing the actual input with estimated input.

Table 2: Compressor Validation Test Conditions

Item	Unit	Value
Cylinder Capacity	cc	9.1
Refrigerant	-	R600a
Oil Viscosity Grade	-	VG5 (Mineral)
Operating Frequency	rpm	1020, 1320, 1620,2220
Evaporator temperature	℃	-23.3
Condenser temperature	℃	38

Figure 8 displays the relationship between the calculated input value and the actual measured input value normalized to the average input value of each operating frequency. The result has a good correlation based on the coefficient (R) 0.90. This validates that the sliding loss of the reciprocating compressor can be estimated accurately based on experimental results from the journal bearing tester.

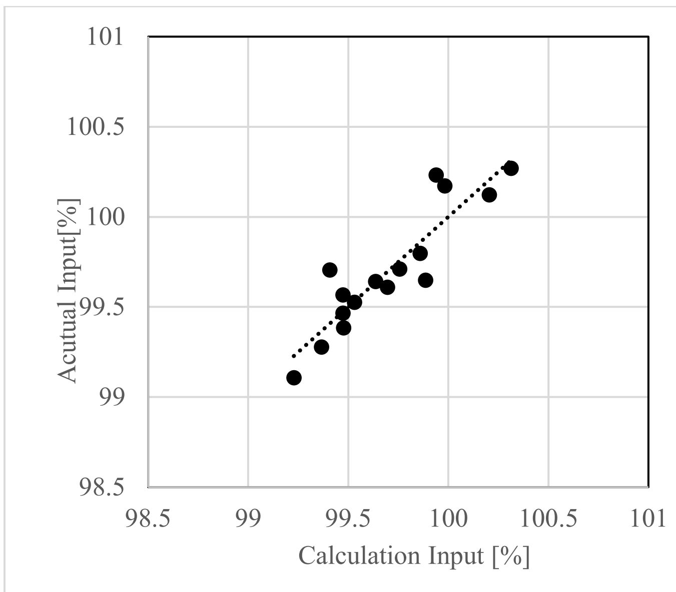

Figure 8 Relationship between predicted input value and measured input value

Using the estimation above, a reciprocating compressor with improved crankshaft design was developed. The reduction of sliding loss at low speed has enable us to achieve improvement in the inverter compressor COP up to 3%.

6. CONCLUSIONS

The following results were obtained from this study.

- We verified the plain bearing characteristics in the low load, low speed, and low viscosity regimes using the journal bearing tester.
- Sliding loss in the operating conditions of reciprocating compressor for household refrigerators could be accurately predicted.
- Using this prediction, reduction in sliding loss and improvement of COP up to 3% were achieved in the development of inverter reciprocating compressor

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