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ABSTRACT

Friction losses in bearings accounts for a large proportion of the losses generated in scroll compressors. Therefore, bearings in scroll compressors for room air conditioners were evaluated by hydrodynamic lubrication analysis and bearing shapes that can reconcile low friction loss with high reliability were investigated.
First, the profile of the main bearing which provides the function of a journal bearing was evaluated by hydrodynamic lubrication analysis, and bearing specifications that achieve a reduction in bearing losses while maintaining reliability were established. Furthermore, performance evaluation and reliability evaluation tests were carried out, and the reduction of friction losses without deterioration in bearing reliability was experimentally confirmed.
Next, the profile of the lower bearing, which functions as both a thrust bearing and journal bearing, was evaluated by hydrodynamic lubrication analysis, and the bearing specifications that achieve a reduction in friction while preserving bearing reliability were established.

1. INTRODUCTION

Due to the recognized need to protect the ozone layer, HCFC refrigerant, which has been conventionally used as a refrigerant for air conditioning equipment, has been changed over to zero-ODP (ozone depletion potential) HFC refrigerants throughout the world, and replacement has been nearly completed in Japan. And now, approaches for prevention of global warming are being pursued. The approaches aim to reduce carbon dioxide emission by improving equipment performance and by using natural refrigerants possessing low GWP (global warming potential). Under these conditions, in room air conditioners for household use, all manufacturers are aggressively developing their own approaches for improving equipment performance, and competition among manufacturers has become quite intense. Consequently, improvement in the efficiency of the compressor, which is a key device in room air conditioners, has become an important goal of manufacturers.

Scroll compressors for room air conditioners are equipped with a main bearing and a lower bearing which support the crankshaft at the top and bottom with a motor in-between, an eccentric bearing which is a fitted section between the crankshaft and the orbiting scroll, and furthermore, a thrust bearing that supports the thrust load that is generated at the compression section. Friction losses in the bearing account for a large percentage of the losses generated in the scroll compressor. Consequently, it is necessary to reduce the friction loss of these bearings to improve the compressor efficiency. In addition, ball bearings were used for the lower bearing of the scroll compressor in our room air conditioners, but in order to improve the reliability and reduce cost, it had been desired to change from ball bearings to a hydrodynamic bearing.

Therefore, we grappled with the problem of reducing the friction loss of the main bearing, while maintaining reliability by enabling the lower bearing to serve as a hydrodynamic bearing. With respect to the main bearing, an investigation was made to reduce friction loss and secure bearing reliability by utilizing hydrodynamic lubrication analyses, and an experiment was carried out (Fumitoshi N., 2001). In addition, with respect to the lower bearing, reduction of friction loss and securing bearing reliability were investigated using hydrodynamic lubrication analyses on a journal-thrust integral-type bearing.

2. SCROLL COMPRESSOR CONFIGURATION AND GUIDELINES FOR INCREASED EFFICIENCY

2.1 Scroll compressor configuration

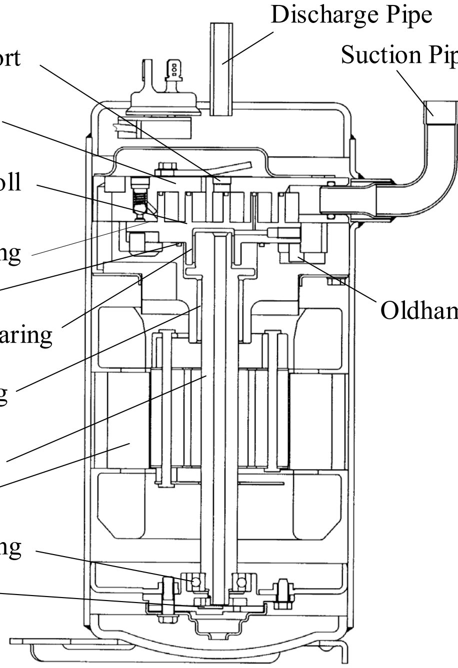

Figure 1: Cross Section of the HFC410A Scroll Compressor

Fig. 1 depicts the configuration of a vertical scroll compressor. The scroll compressor has a compression mechanism disposed at the top and a motor at the bottom and the shell inside is filled with high-side pressure. The compression mechanism comprises a crankshaft that converts the motor rotational motion to an eccentric rotational motion, bearings that support the crankshaft, an orbiting scroll which is driven by the crankshaft and performs orbital motion, an Oldham ring that prevents the orbiting scroll from self-rotation and a fixed scroll that is disposed opposite to the orbiting scroll and forms a compression chamber by being combined with the orbiting scroll.

A thrust bearing is fitted between the orbiting scroll and the fixed scroll, and it receives the thrust load acting on the orbiting scroll. The bearing structural members consist of a main bearing, which is a journal bearing that supports the crankshaft, and an eccentric bearing, which is a journal bearing fitted to the eccentric section of the crankshaft and supports the orbiting scroll. Furthermore, a lower bearing is mounted at the lower end of the crankshaft. The lower bearing provides the functions of both a journal bearing that supports the crankshaft radially and a thrust bearing that supports the thrust load acting on the crankshaft.

At the shell bottom, an oil reservoir is installed, which utilizes a displacement-type pump to deliver stored oil to the compression mechanism for both lubrication as well as for sealing bearings and sliding sections of the orbiting scroll.

2.2 Guidelines for increased efficiency

The overall efficiency (η_{comp}) of a compressor is given by the product of three separate efficiency values: the compression efficiency (η_c), which indicates the ratio of losses due to refrigerant leakage, heating, etc. that occur when the refrigerant is compressed; mechanical efficiency (η_{me}), which indicates the ratio of friction losses generated at each sliding section; and motor efficiency (η_{mo}); which indicates the ratio of losses such as motor heat generation, etc (1).

$$\eta_{comp} = \eta_c \times \eta_{me} \times \eta_{mo} \tag{1}$$

Therefore, the key point is that the overall compressor efficiency can be increased by improving each of these three efficiencies.

This paper reports in detail the approach for increasing mechanical efficiency.

3. ANALYSIS TECHNIQUE

For the hydrodynamic lubrication analysis of a finite-width journal bearing, which can model the main bearing and lower bearing, the approximate theoretical analysis, which introduces a finite-width correction factor (Eiichi N., Hiroshi A., 1965), was used. Note that it has been confirmed that the analysis results of this approximate theoretical analysis are in good agreement with the analysis results presented in the Hydrodynamic Bearing Data of the Japan Society of Mechanical Engineers (1984).

In addition, for the hydrodynamic lubrication analysis of the thrust function of the lower bearing, the commercial software ARMD version 5.4 (Rotor Bearing Technology & Software, Inc.), which can analyze 2-D bearing models, was used.

4. REDUCTION OF MAIN BEARING LOSS

4.1 Examination by analysis

Using the approximate theoretical analysis together with the finite-width correction factor, the relationship between main bearing crankshaft diameter and friction loss as well as the relationship between crankshaft diameter and reliability under the air conditioner operating mode were examined. For the evaluation of bearing reliability, the film thickness parameter Λ was used. The film thickness parameter Λ is given by the ratio of minimum oil film thickness h_{min} to composite surface roughness σ of shaft and bearing ($\Lambda = h_{min}/\sigma$). For $0<\Lambda<1$, the bearing is considered to be in the boundary lubrication condition, for $1<\Lambda<2$ to 3, in the mixed lubrication condition, and for $3<\Lambda$, in the hydrodynamic lubrication condition (Japanese Society of Tribologists, 1987). The scroll compressor main bearing has a composite surface roughness σ of 0.51μm. Table 1 shows the performance evaluation conditions for the scroll compressor.

First, friction loss and reliability of the main bearing are presented with bearing diameter D. Fig. 2 shows the relationship between friction loss and bearing diameter D. Fig. 3 shows the relationship between film thickness parameter Λ and bearing diameter D. Fig. 2 and Fig. 3 indicate that bearing loss decreases significantly with decreasing bearing diameter D, but the minimum oil film thickness, and thus the film thickness parameter Λ, also decreases. Therefore, the reliability tends to lower. When the bearing width L is 27 mm and the radial clearance ratio (radial clearance 2C/bearing diameter D) is 0.0022, a bearing diameter D of 16 mm or greater provides a film thickness parameter Λ of 3 or greater, even during the cooling half-capacity operation mode, where the lubrication condition becomes the severest. Therefore good reliability can be obtained.

Next, the friction loss and the reliability of the main bearing are presented with the bearing width L. Fig. 4 depicts the relationship between the friction loss and the bearing width L. Fig. 5 depicts the relationship between film thickness parameter Λ and bearing width L. It can be seen from these figures that the bearing loss decreases as the bearing width L decreases, but the minimum oil film thickness, and thus the film thickness parameter Λ, decrease as well, the reliability tends to lower. When the bearing diameter D is 16 mm and radial clearance ratio is 0.0022, a bearing width L of 27 mm or greater provides a film thickness parameter L of 3 or more, even during the cooling half-capacity operation mode, where the lubrication condition becomes the severest, and good reliability can be obtained.

The bearing diameter D and the bearing width L were examined analytically, and as a result, it was found that a small diameter is effective in reducing bearing loss. Also, in order to obtain a film thickness parameter Λ greater than about 3, the scroll compressor main bearing should have a bearing diameter D of 16mm, a bearing width L of 27mm, and a radial clearance ratio of 0.0022. The analysis indicates that by achieving the above bearing specifications, it is possible to reduce bearing loss, and at the same time, to maintain the hydrodynamic lubrication condition necessary to obtain reliability.

Table 1: Performance Evaluation Conditions

Case No.	Air Conditioner Operation Mode
1	Heating Rated-capacity Operation Mode
2	Cooling Rated-capacity Operation Mode
3	Heating Half-capacity Operation Mode
4	Cooling Half-capacity Operation Mode

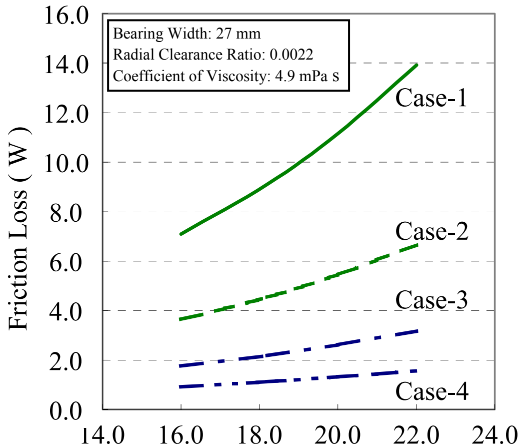

Figure 2: Effects of Bearing Diameter on Friction Loss

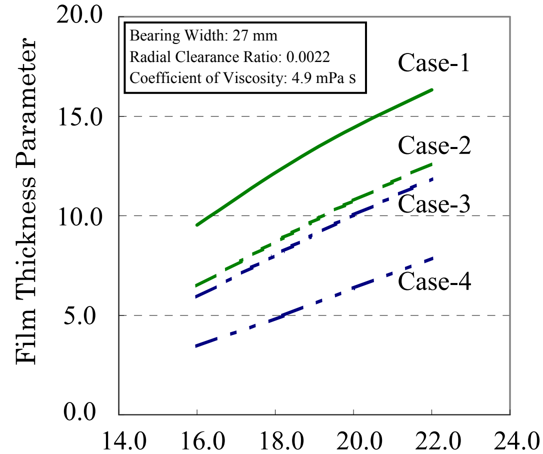

Figure 3: Effect of Bearing Diameter on Film Thickness Parameter

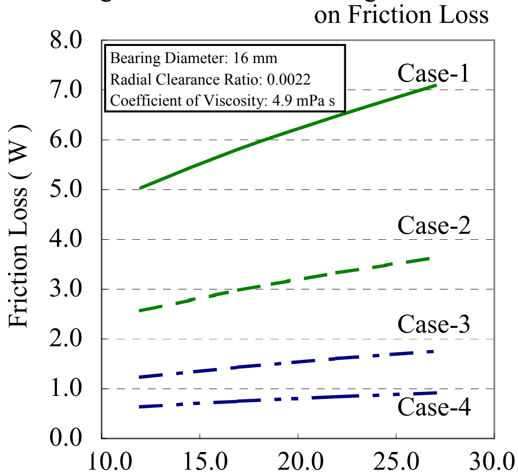

Figure 4: Effects of Bearing Width on Friction Loss

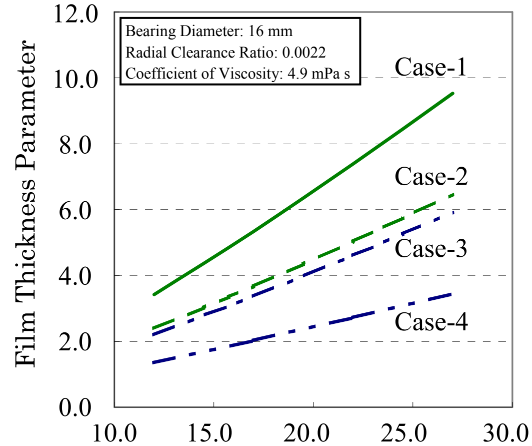

Figure 5: Effect of Bearing Width on Film Thickness Parameter

4.2 Verification by experiments

In order to validate the analysis results, the low-loss bearing specifications (bearing diameter D: 16 mm; bearing width L: 27 mm; radial clearance ratio: 0.0022) were evaluated by a performance evaluation test and lubrication condition confirmation test, using the impressed voltage method.

First, the performance evaluation test was carried out to confirm the main bearing loss reduction rate by changing the main bearing shape.

Fig. 6 shows the comparison of analytical and experimental data for the main-bearing loss reduction rate when the bearing diameter of the scroll compressor was changed from the ϕ33-mm specifications of a conventional model to the ϕ16-mm specifications mentioned above. By utilizing the small-diameter ϕ16-mm specifications, the main bearing loss can be greatly reduced to 17 W (experimental value) under the heating rated-capacity operation mode. The analysis results indicate that the main bearing loss reduction rate under various operating conditions has been predicted qualitatively by the analysis and falls within ±25% of the experimental results.

Next, the reliability of a low-loss bearing with the specifications obtained from the hydrodynamic lubrication analysis, that is, the lubrication condition of a scroll compressor main bearing, was evaluated by the impressed voltage method.

Fig. 7 is a schematic diagram of the impressed voltage method. The bearing bushing is press-fitted and fixed in an electrically insulated condition from the bearing by an insulating-film. 0.1V was impressed between the crankshaft and the bearing bush, and the contact condition between the scroll compressor crankshaft and the bearing bush, that is, the lubricating condition, was evaluated.

In this evaluation system, voltages from 0.09 to 0.1V were indicated under the non-contact condition. Naturally, when the lubrication condition shifts from hydrodynamic lubrication to mixed lubrication and localized contact begins to occur, the measurement voltage decreases.

Fig. 8 shows the voltage between the crankshaft and the bearing bush under various operation modes for the scroll compressor main bearing with the bearing specifications discussed in Section 4.1. The voltage values shown on the y-axis of the graph were obtained by taking a 0.5 second average of the typical voltage waveform between the crankshaft and the bearing under each operation modes, the voltage between the crankshaft and the bearing is 0.09V or higher, and it is assumed that the hydrodynamic lubrication condition has been achieved.

Next, Fig. 9 indicates the relationship between the voltage across the crankshaft/bearing and the bearing width L when operating in the cooling half-capacity operation mode. The voltage between the crankshaft and the bearing significantly decreases as the bearing width L of the main bearing is reduced below 27mm, and it is assumed that the lubrication condition shifts to the mixed lubrication condition. Based on this result, in the case of the above scroll compressor (bearing diameter D: 16 mm; radial clearance ratio: 0.0022), it can be predicted that reducing the bearing width L below 27 mm reduces the reliability. These reliability evaluation results correspond well with the above-mentioned hydrodynamic lubrication analysis results (effect of bearing width).

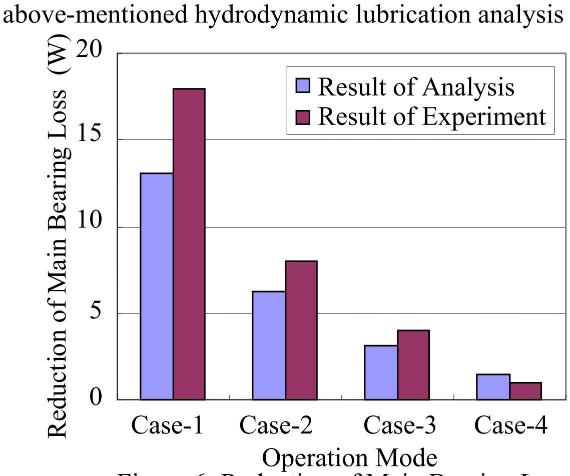

Figure 6: Reduction of Main Bearing Loss

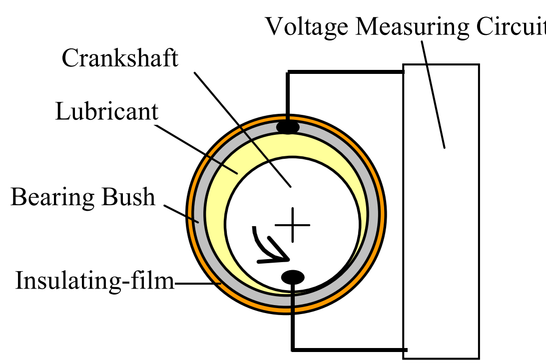

Figure 7: Image of Impressed Voltage Method

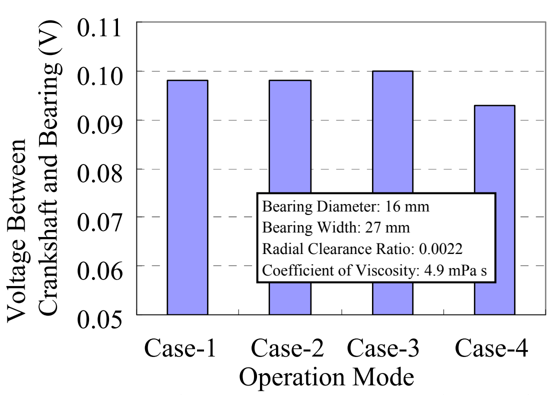

Figure 8: Effect of Operating Mode on Contact Voltage

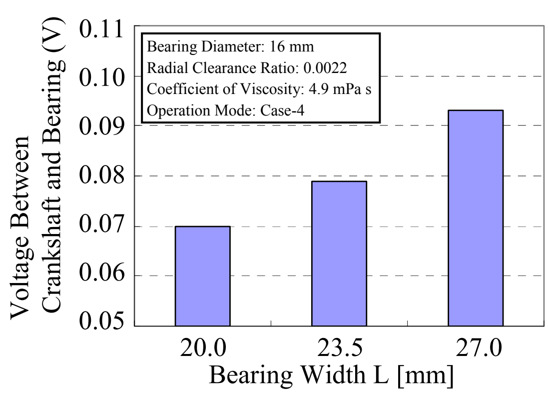

Figure 9: Effect of Bearing Width on Contact Voltage

5. REDUCTION OF LOWER BEARING LOSS

In changing the lower bearing specifications from ball bearing to hydrodynamic bearing, the reliability under the scroll compressor operation modes must be maintained. Consequently, the lower bearing was examined to obtain bearing reliability and at the same time to reduce friction loss.

Fig. 10 is a cross-sectional view of the lower bearing. The crankshaft diameter D_c is $\phi 16$ mm, which corresponds to the outside diameter of the lower thrust bearing.

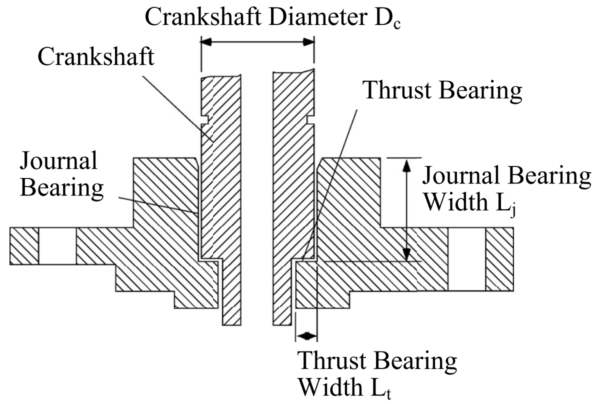

Figure 10: Cross Section of Lower Bearing

5.1 Reduction of the journal bearing loss

Using the approximate theoretical analysis with the finite-width correction factor, an investigation was made of the relationship between bearing width L_j of the scroll compressor lower journal bearing section and the friction loss, and the relationship between bearing width L_j and reliability.

Fig. 11 depicts the relationship between the bearing friction loss and bearing width L_j. Fig. 12 depicts the relationship between film thickness parameter Λ and bearing width L_j. For the cases shown, the scroll compressor lower journal bearing has a composite surface roughness σ of 0.40 μm. Fig. 11 and Fig. 12 indicate that the bearing loss decreases as the bearing width L_j decreases, but the minimum oil film thickness, and therefore the film thickness parameter Λ, decreases as well. This tends to lower the reliability. Under the cooling half-capacity operation mode, which are severe from the viewpoint of lubrication, a value of about 3 is obtained for the film thickness parameter Λ for bearing width L_j of 8 mm.

As a result of analytically examining the bearing width L_j, it was found that shortening the bearing width L_j is effective in reducing bearing loss. Also, to keep the value of the film thickness parameter Λ greater than about 3 in order to obtain high reliability, it was found that the lower journal bearing specifications of the scroll compressor should be 16 mm for crankshaft diameter D_c; 8 mm for bearing width L_j; and 0.0022 for radial clearance ratio. By employing the above bearing specifications, it becomes possible to reduce the bearing loss and at the same time, to ensure hydrodynamic lubrication, that is, to obtain high reliability.

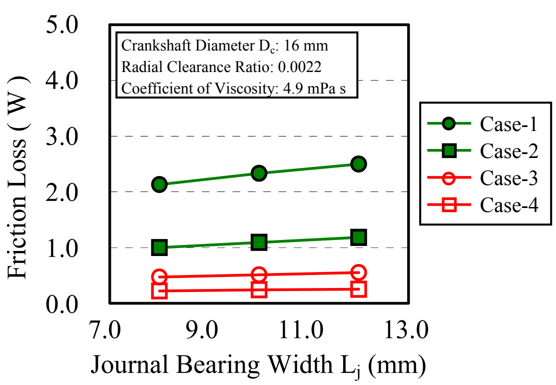

Figure 11: Effect of Journal Bearing Width L_j on Friction Loss

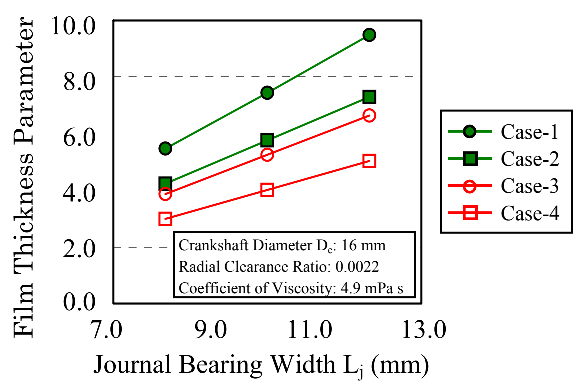

Figure 12: Effect of Journal Bearing Width L_j on Film Thickness Parameter

5.2 Reduction of thrust bearing loss

Using the hydrodynamic lubrication analysis software (ARMD version 5.4), an investigation was made of the relationship between thrust bearing width L_t of the scroll compressor lower bearing and bearing friction loss, as well as the relationship between bearing width L_t and reliability.

Fig. 13 is an analysis model of the thrust bearing section of the lower bearing. The size of the bearing width L_t investigated is expressed by $(D_t - d)/2$ using the outside diameter D_t (=14.8 mm) and the inside diameter d of the thrust bearing surface. The thrust bearing surface roughness can be separated into a short wavelength component and a long-wavelength undulation component. Therefore in the present analysis, the undulation component was modeled using four tapers 5 μm high with the flatness of the processed surface taken into account.

The thrust load that is supported by the lower bearing is the resultant force of the combined empty weight of the motor's rotor and the crankshaft as well as the axial component of the magnetic force generated by the motor's stator and rotor.

Fig. 14 shows the relationship between bearing friction loss and bearing width L_t. Fig. 15 shows the relationship between film thickness parameter Λ and bearing width L_t. For the cases shown, the scroll compressor lower thrust bearing has a composite surface roughness σ of 0.38 μm. As shown in these figures, friction loss slightly decreases as bearing width L_t decreases, but the minimum oil film thickness, and therefore the film thickness parameter Λ, decrease as well, so the reliability tends to degrade. Where the bearing width L_t is 1.7 mm or more, the film thickness parameter Λ is 3 or more, suggesting that the bearing is in the hydrodynamic lubrication condition. However, in the scope of the present analysis, effects of bearing width L_t on friction loss are extremely small and the value is small, too. Therefore, the bearing width L_t was set to 2.3 mm which enables the film thickness parameter Λ to achieve Λ>5 with the reliability and workability taken into account.

It is concluded that a high reliability bearing can be obtained by adopting the above-mentioned bearing specifications.

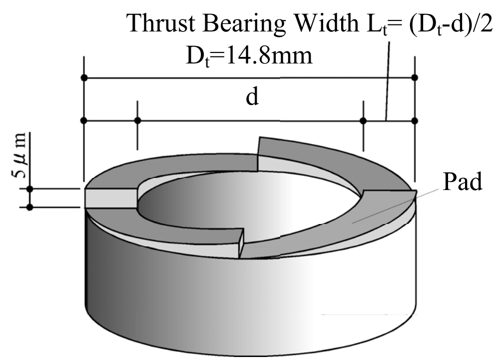

Figure 13: Analysis Model of Thrust Bearing at Lower Bearing

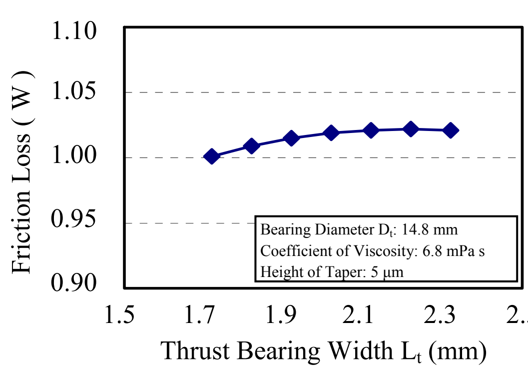

Figure 14: Effect of Thrust Bearing Width L_t on Friction Loss

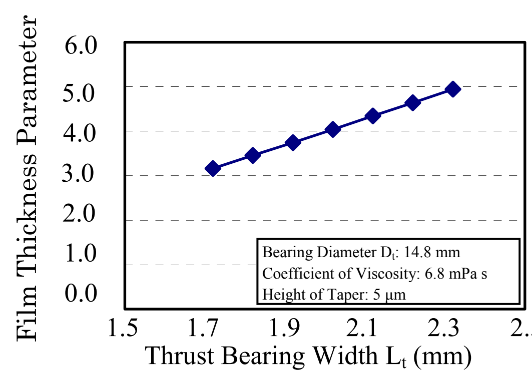

Figure 15: Effect of Thrust Bearing Width L_t on Film Thickness Parameter

6. CONCLUSIONS

In order to reduce the bearing loss in the scroll compressor, hydrodynamic lubrication analyses of the main bearing and the lower bearing were carried out at the design stage, and the relationship between the bearing specifications and bearing loss as well as film thickness parameter Λ was investigated. With respect to the main bearing, friction loss and reliability of low-friction loss bearing specifications obtained by the analysis were experimentally confirmed. The following results were obtained.

- Reducing the diameter of the main bearing results in a reduction of friction losses due to the main journal bearing. It has been confirmed that in the case of our scroll compressor, adopting the main bearing specifications of 16 mm bearing diameter D, 27 mm bearing width L, and 0.0022 radial clearance ratio results in a film thickness parameter value $\Lambda>3$ and minimizes the bearing losses.

- Shortening the bearing width results in a reduction of friction losses due to the lower journal bearing. It has been confirmed that in the case of our scroll compressor, the film thickness parameter Λ becomes $\Lambda>3$ and the minimum bearing loss is achieved when the journal bearing specifications of the lower bearing are set to 16 mm for crankshaft diameter D_c, 8 mm for bearing width L_j, and 0.0022 for radial clearance ratio.

- Increasing the bearing surface results in increased reliability of the lower thrust bearing. In the case of our scroll compressor, when the bearing outside diameter D_t is 14.8 mm and the bearing width L_t is 2.3, the bearing loss only increases slightly compared to the case in which the bearing width L_t is 1.7 mm, but the film thickness parameter Λ increases to $\Lambda>5$ and increased reliability can be achieved.

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