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A Low Friction Thrust Bearing for Reciprocating Compressors

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ABSTRACT

A thrust bearing with a micro texture on its sliding surface that produces hydrodynamic pressure was developed for use in reciprocating compressors. Evaluation using an elemental friction test showed that its friction loss was 20–60 % lower than that of the current design. Measurement of the efficiency of a compressor with the developed thrust bearing showed that the coefficient of performance was 1.4 % higher than that of a compressor with a conventional thrust bearing.

1. INTRODUCTION

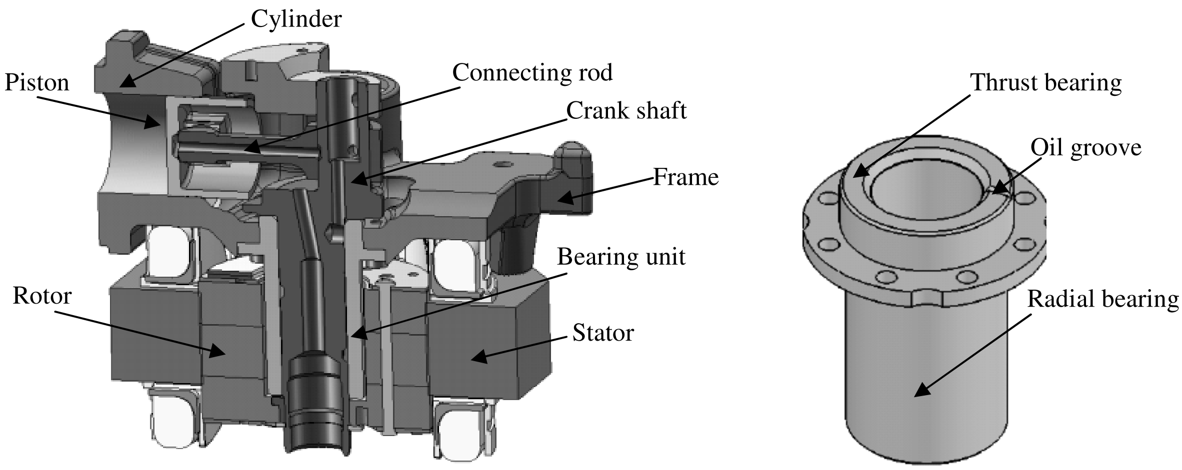
The increased awareness of environmental issues has led to growing consumer demand for electrical appliances that have greater energy-saving properties and less impact on the environment. Refrigerators with low power consumption are also in high demand because the amount of electricity consumed by refrigerators is among the highest for household appliances. It is therefore important to develop a highly efficient compressor to reduce the power consumption of refrigerators.

To develop a highly efficient compressor, it is necessary to reduce the friction loss of the sliding surfaces. A reciprocating compressor has many sliding surfaces such as the outer surface of the piston, the joints of the connecting rod, and the radial and thrust bearings of the crank shaft. In this study, we focused on the friction loss of the thrust bearing and developed an improved thrust bearing structure for reciprocating compressors. The conventional thrust bearing for compressors has a parallel slider with one oil groove. To improve the lubrication property, we investigated three potential design patterns for reducing the friction loss of thrust bearings. These thrust bearings have a micro texture on its sliding surface that produces hydrodynamic pressure. We evaluated the friction loss of these thrust bearings using an elemental friction test and measured the efficiency of a compressor with the developed thrust bearing that was the most effective in an elemental test.

2. EXPERIMENTAL COMPRESSOR

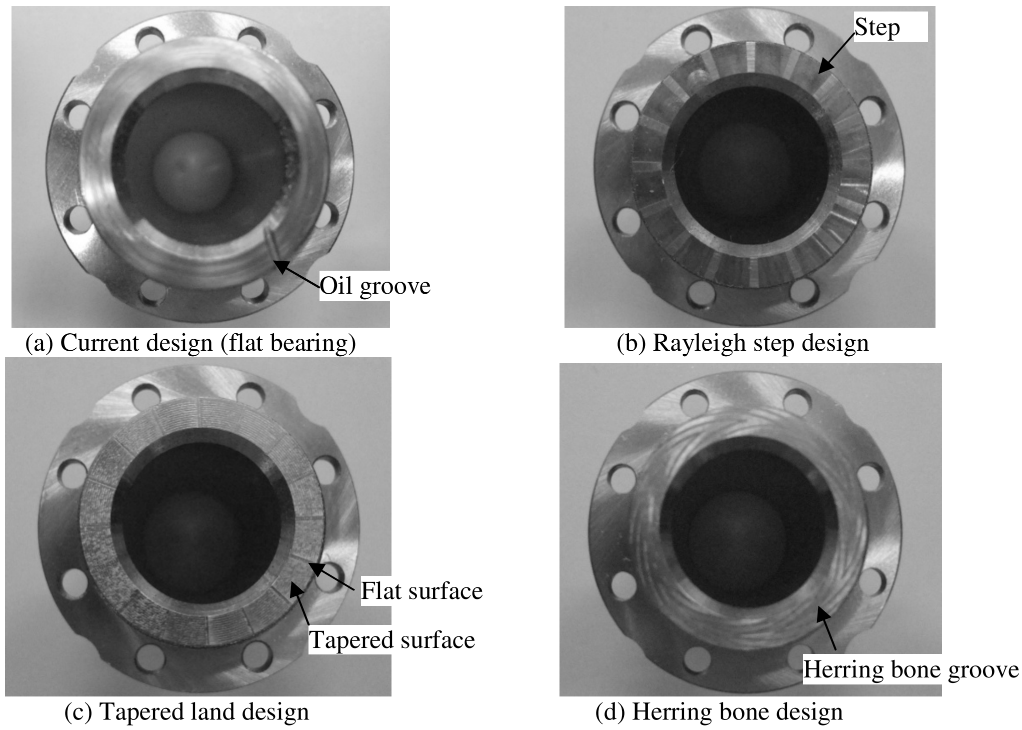
Figure 1 (a) shows a cross section of the experimental compressor we used. The bearing unit with the thrust bearing and a radial bearing was bolted to the frame. We used a bearing unit that could be separated from the frame to facilitate evaluation of the difference in friction loss for several types of thrust bearing. We were able to evaluate the difference in friction loss between thrust bearings by simply changing the bearing unit, without affecting the individual variabilities of the other components such as the crank shaft. Figure 1 (b) shows an oblique perspective view of the bearing unit. The thrust bearing has a flat surface with one oil groove. Its sliding surface is lubricated with oil supplied from this groove.

Figure 2 shows photographs of the thrust bearing test pieces used in the elemental friction test. The current design has a flat surface with one groove for oil supply (Figure 2 (a)). The bearing with a Rayleigh step design has steps on its surface, as shown in Figure 2 (b). The height of this step is less than 100 µm. We used one with 16 steps in this study. The bearing with a tapered land design has tapered and flat surfaces, as shown in Figure 2 (c). We used one with 16 tapered surfaces. The bearing with a herring bone design (Figure 2 (d)) has several micro grooves, which produce hydrodynamic pressure. The micro grooves are aligned in a herring bone configuration.



(a) Cross section of compressor (b) Bearing unit

Figure 1: Experimental compressor



(a) Current design (flat bearing) (b) Rayleigh step design

(c) Tapered land design (d) Herring bone design

Figure 2: Thrust bearing test pieces (bearing units)

3. ELEMENTAL FRICTION TEST AND PERFORMANCE TEST

3.1 Elemental Friction Test

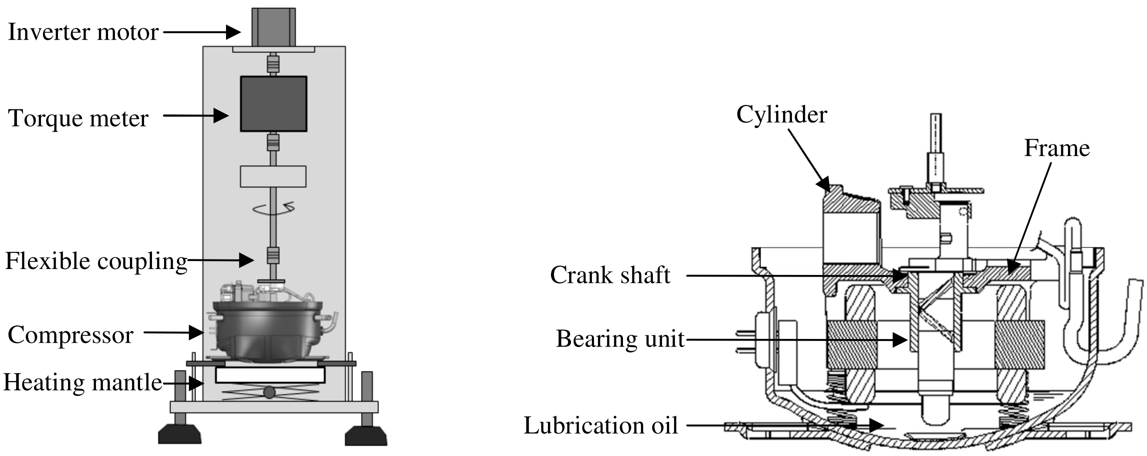
The friction loss of the thrust bearing test pieces shown in Figure 2 was evaluated using an elemental friction test. Figure 3 (a) shows the apparatus for measuring the friction torque of the crank shaft. The crank shaft in the compressor was connected to the torque meter with a flexible coupling. The temperature of the lubrication oil in the chamber was controlled using a heating mantle. The crank shaft was driven by an inverter motor above the torque meter. Figure 3 (b) shows a cross section of the compressor. The piston and connecting rod were removed so that only the friction loss of the bearing unit was measured.

First, the crank shaft was set on the frame of the compressor with the thrust bearing loaded by the weight of the crank shaft and rotor. The friction torque measured with this set-up comprised the torque at the thrust bearing, the radial bearing, and the instruments outside the compressor such as the flexible coupling. Next, the crank shaft was set so as not to touch the thrust surface of the bearing unit so that the friction torque could be evaluated without the torque of the thrust bearing. The difference in torque between this set-up and the previous one was the friction torque of the thrust bearing. Friction loss is the product of the friction torque and rotation speed.

The friction loss of the test pieces at several oil viscosities is shown in Figure 4. The oil viscosity was controlled by adjusting the oil temperature. The friction loss of each thrust bearing with a micro texture was 20-60 % lower than that of the current design. The thrust bearing with the herring bone design had the lowest friction loss that was about 60 % less than that of the current design. This experimental result showed that the micro texture on the sliding surface was very effective to reduce the friction loss of the thrust bearing.

3.2 Performance Test

A compressor with a herring-bone-design thrust bearing was installed in a refrigeration cycle, and the input power and refrigerating capacity of the compressor were measured to evaluate the difference of the performance efficiency between a developed compressor and a conventional compressor. Figure 5 shows the ratio of change in the COP (coefficient of performance) and input power. The horizontal axis is the rotation speed normalized by a certain speed ω_0. The COP of the compressor was 1.4 % higher on average due to a reduction in input power.



(a) Torque meter and compressor (b) Cross section of compressor

Figure 3: Schematics of apparatus for measuring friction torque of crank shaft

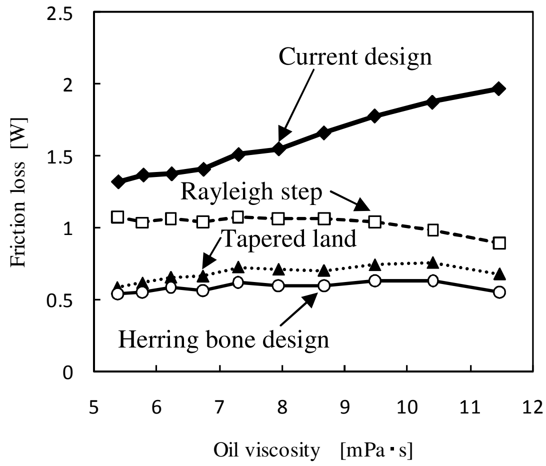


Figure 4: Friction loss of thrust bearings in elemental friction test

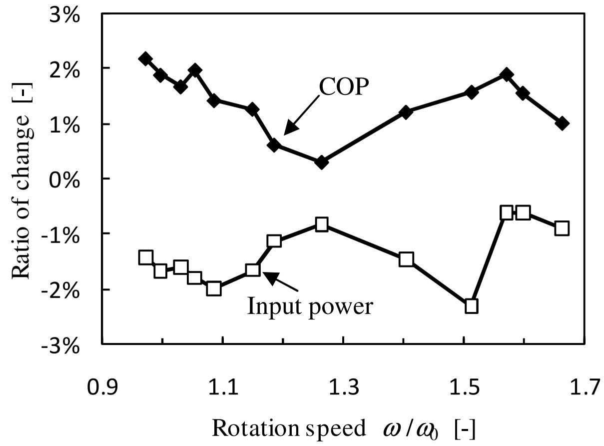


Figure 5: Improvement in efficiency with herring-bone-design thrust bearing

4. THEORETICAL CALCULATION

In the previous chapter, the friction loss of the thrust bearing with a micro texture was evaluated using the elemental friction test. The friction loss of the one with the herring bone design was the lowest and was about 60 % less than that of the current design. In this chapter, we examined the friction loss of the current design and the herring bone design using theoretical calculation.

The friction loss of a thrust bearing is determined by the shape of its surface, its size, the viscosity of the lubricating oil, the thrust load, and the rotating speed of the shaft. With hydrodynamic lubrication, the friction loss of a thrust bearing with parallel flat surfaces is given by

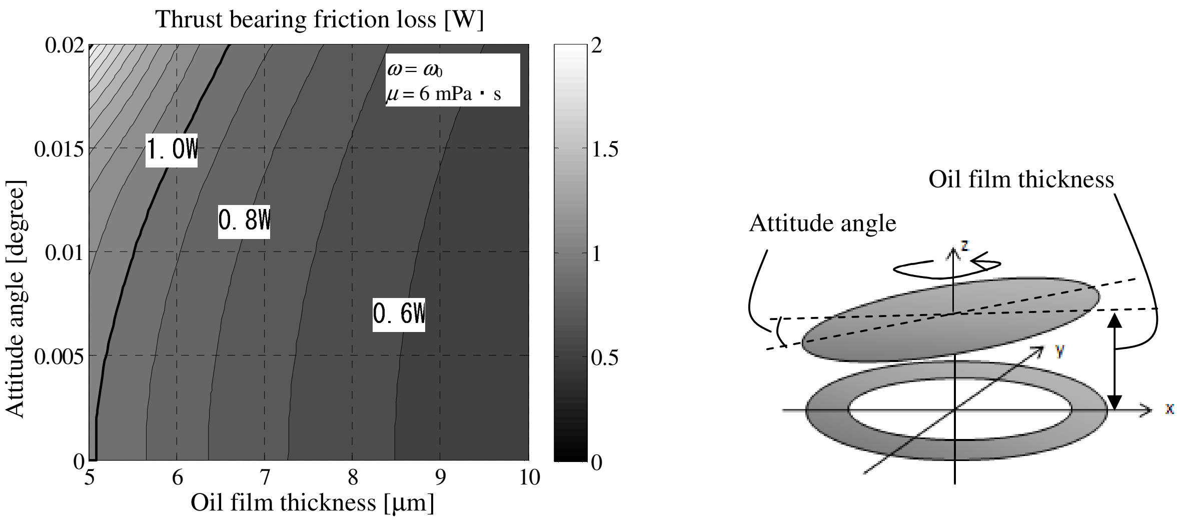
$$L_t = \frac{\mu \pi \omega^2}{2h}\left(r_1^4 - r_0^4\right) \quad , \tag{1}$$

where L_t represents the friction loss, μ is the viscosity of the lubricating oil, ω is the rotating speed, h is the oil film thickness, r_1 is the outer radius of the thrust surface, and r_0 is the inner radius of the thrust surface. However, two opposing thrust surfaces form a tilt angle during compressor operation because the crank shaft is pushed by the compressing gas through the piston. The friction loss of the thrust bearing increases as the tilt angle becomes larger. The friction loss of a thrust bearing with a tilted flat surface is given by a Reynolds equation:

$$\frac{1}{r}\frac{\partial}{\partial \theta}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial \theta}\right) + \frac{\partial}{\partial r}\left(\frac{rh^3}{12\mu}\frac{\partial p}{\partial r}\right) = \frac{r\omega}{2}\frac{\partial h}{\partial \theta} \quad , \tag{2}$$

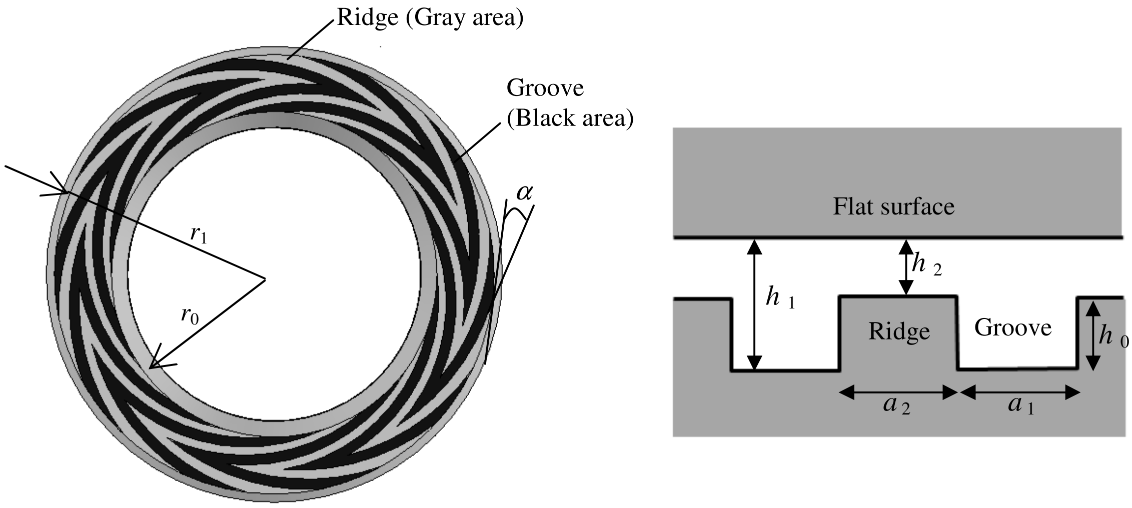
where p represents the oil pressure, r and θ represent the polar coordinates.

Figure 6 (a) shows the friction loss of a flat thrust bearing with a tilted surface at a certain rotation speed ω_0. The horizontal axis is the oil film thickness, which is the clearance between the centers of the tilted thrust surfaces, as shown in Figure 6 (b). The vertical axis is the attitude angle of the thrust surface. The larger the attitude angle and the thinner the oil film, the larger the velocity gradient of the oil between the thrust surfaces. This means that the friction loss is larger when the attitude angle is large and the oil film is thin due to the shear force of the lubrication oil.



(a) Friction loss (b) Model of flat thrust bearing

Figure 6: Theoretical calculation of friction loss of flat thrust bearing



(a) Groove pattern (b) Cross section of groove

Figure 7: Geometry of herring bone grooves on bearing surface

A micro texture, such as herring bone grooves, on the surface produces hydrodynamic pressure. Figure 7 shows the geometry of herring bone grooves on a thrust surface. The load capacity and friction torque of a bearing with a herring bone pattern on its surface was calculated using equations developed by E. A. Muijderman:

$$W_t = \frac{3\pi\mu\omega r_1^{\ 4}}{4h_2^{\ 2}}\left(1-\lambda^2\right)^2 g_1(\alpha, H, \gamma) C_1(\alpha, H, \gamma, \lambda, k) \tag{3}$$

$$M_t = \frac{\pi\mu\omega r_1^{\ 4}}{2h_2}\left(1-\lambda^4\right) g_2(\alpha, H, \gamma) \quad , \tag{4}$$

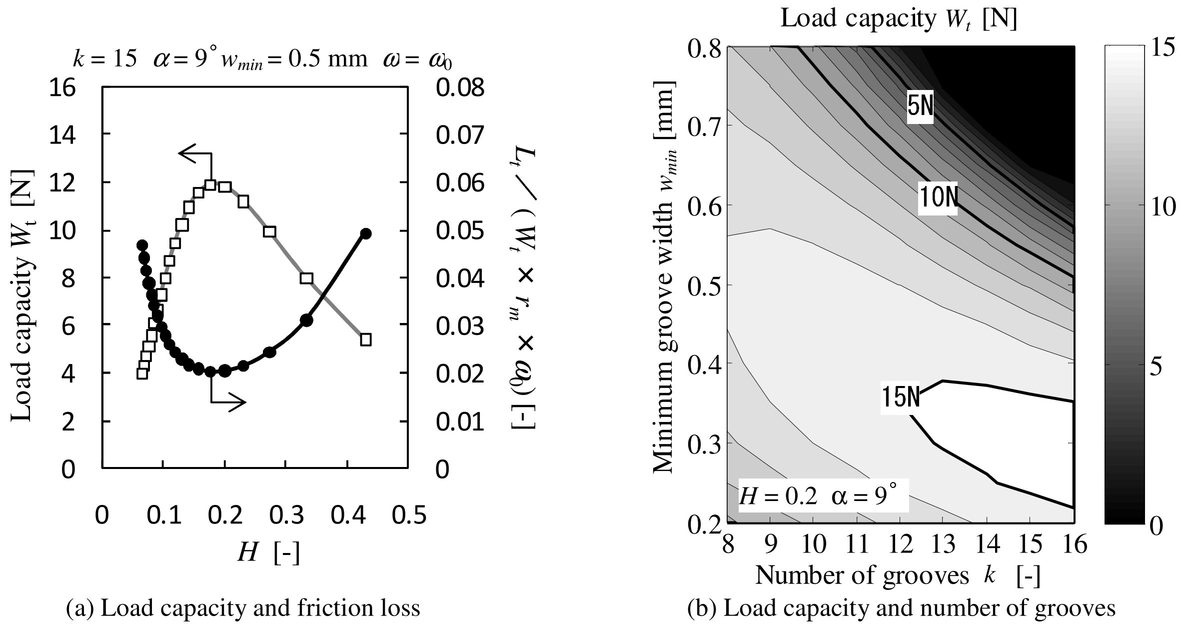
where W_t is the load capacity of the bearing, M_t is the friction torque, μ is the fluid viscosity, ω is the rotational speed, r_0 and r_1 are the inner and outer radii of the thrust surface, h_2 and h_1 are the thrust gap and the gap between the thrust surface of the flat side and the bottom of the grooves, and λ is r_0 / r_1. The variables g_1, g_2 and C_1 are complex geometric parameters based on α (spiral groove angle), H (h_2/h_1), γ (ratio of groove to ridge widths, a_2/a_1), λ, and k (number of grooves). In this case, radius r_m, where the grooves turn sharply, is the root-mean-square of r_0 and r_1. The minimum width of a groove is given by

$$w_{\min} = \frac{2\pi r_0 \sin\alpha}{(1+\gamma)k} \quad . \tag{5}$$

This is the approximate value of the groove width at the edge of the inner radius side on the thrust surface. This minimum width is restricted to the size of the working tool.

The relationship between bearing load capacity and the groove properties calculated using Eq. (5) at a certain rotation speed ω_0 are plotted in Figure 8. The load capacity has a peak value at a specific value of H, as shown in Figure 8 (a). In this calculation, k , w_{min}, and h_2 were set to constant values. The value of the friction loss divided by the load capacity and the sliding speed ($r_m \times \omega_0$, where r_m is the average radius of the inner and outer radii of the thrust surface) corresponds to the friction coefficient. This value has a minimum value at the same value of H, as shown in Figure 8 (a). This means that there is an optimum groove depth for minimizing friction loss. Figure 8 (b) shows the effect of the number of grooves and the minimum groove width on the load capacity. It shows that there is appropriate groove geometry for obtaining sufficient load capacity to retain oil film between the surfaces so as to prevent solid contact. In this study, we designed the geometry of a herring bone bearing for reciprocating compressors using these calculation results.

Figure 9 compares the experiment and calculation results for the current design and the herring bone design. The calculated friction loss was derived assuming hydrodynamic lubrication. Because the current design creates a mixed lubrication condition, the experiment result for the current design was much larger than the calculated one. In contrast, the experiment and calculation results were in good agreement for the herring bone design. This means that the friction surface was in a hydrodynamic lubrication condition due to the effect of the dynamic oil pressure produced by the micro grooves.



(a) Load capacity and friction loss (b) Load capacity and number of grooves

Figure 8: Properties of bearing with herring bone grooves

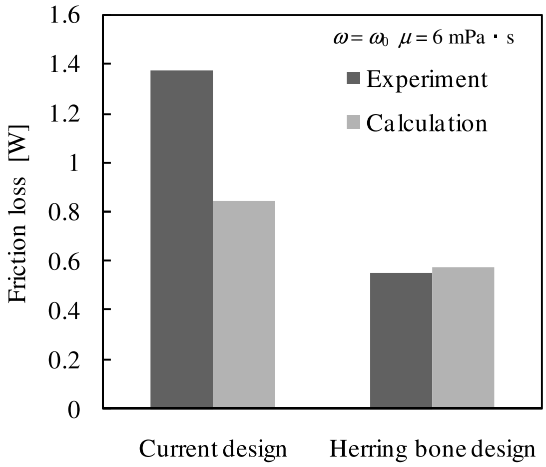


Figure 9: Comparison of experiment and calculation

5. CONCLUSION

Our investigation of three potential design patterns for reducing the friction loss of thrust bearings used in reciprocating compressors showed that a herring bone design was the most effective. A bearing with this design had a friction loss 60 % lower than that of a conventional thrust bearing. Measurement of the efficiency of a compressor with a herring-bone-design thrust bearing showed that its coefficient of performance was 1.4 % higher than that of a compressor with a conventional thrust bearing.

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