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Design Of Journal Bearings In Reciprocating Compressors

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ABSTRACT

Journal bearings in hermetic reciprocating compressors are analyzed experimentally as well as numerically. The motor current and the load applied to the bearing are measured to obtain the performance curve of the journal bearing, from which the critical Sommerfeld number is derived.

In the numerical analysis, the Reynolds equation and equilibrium equations of the journal center are solved simultaneously with Newton-Raphson iterative method. Reynolds boundary condition is used for solving the Reynolds equation. Minimum film thickness and friction loss of the journal bearing are calculated under static loading condition.

The universal value of the critical Sommerfeld number is obtained as a design criterion from the experimental result. The numerical analysis shows the way to suggest the optimum design of the journal bearing.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
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<tbody>
<tr>
<td>C</td>
<td>clearance</td>
<td>m</td>
</tr>
<tr>
<td>e</td>
<td>journal center position</td>
<td>m</td>
</tr>
<tr>
<td>H</td>
<td>friction loss</td>
<td>W</td>
</tr>
<tr>
<td>i</td>
<td>motor current</td>
<td>A</td>
</tr>
<tr>
<td>L</td>
<td>bearing length</td>
<td>m</td>
</tr>
<tr>
<td>R</td>
<td>bearing radius</td>
<td>m</td>
</tr>
<tr>
<td>T</td>
<td>torque, N-m</td>
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<tr>
<td>U</td>
<td>surface velocity average</td>
<td>m/s</td>
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<tr>
<td>X, Y</td>
<td>rectangular coordinate system</td>
<td>m</td>
</tr>
<tr>
<td>z</td>
<td>axial coordinate</td>
<td>m</td>
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<tr>
<td>µ</td>
<td>friction coefficient</td>
<td></td>
</tr>
<tr>
<td>D</td>
<td>bearing diameter</td>
<td>m</td>
</tr>
<tr>
<td>F</td>
<td>load</td>
<td>N</td>
</tr>
<tr>
<td>h</td>
<td>film thickness</td>
<td>m</td>
</tr>
<tr>
<td>KT</td>
<td>motor torque-current constant</td>
<td>N-m/A</td>
</tr>
<tr>
<td>p</td>
<td>pressure</td>
<td>Pa</td>
</tr>
<tr>
<td>S</td>
<td>Sommerfeld number</td>
<td></td>
</tr>
<tr>
<td>t</td>
<td>time</td>
<td>s</td>
</tr>
<tr>
<td>ΔU</td>
<td>surface velocity difference</td>
<td>m/s</td>
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<tr>
<td>x</td>
<td>circumferential coordinate</td>
<td>m</td>
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<tr>
<td>η</td>
<td>dynamic viscosity</td>
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<tr>
<td>ω</td>
<td>rotational speed</td>
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INTRODUCTION

The friction loss in the journal bearing comprises a major part of the mechanical loss of the hermetic reciprocating compressor. Therefore, reduction in the friction loss of the journal bearing becomes important to improve the efficiency of the compressor.
As the ways of the friction loss reduction, such as reducing the bearing area or using the lubricant with lower viscosity, have bad influences on the bearing reliability in general, compromise between the efficiency and the reliability is very important in the optimum design of journal bearings.

In this study, experimental and numerical methods are presented to find the ways of designing efficient as well as reliable journal bearings in hermetic reciprocating compressors.

**EXPERIMENTAL ANALYSIS**

**Journal Bearing Test Machine**

Figure 1 shows the journal bearing test machine used in this study. Using this machine, the journal bearing of the crankshaft in the reciprocating compressor can be tested “as is” with slight modification. Crankpin is removed from the crankshaft and then the concentric pin is attached, where the motor is directly connected using a gear coupling. By the pneumatic cylinder, static load is applied to the concentric pin through the ball bearing. Surface temperatures of the bearing are measured by four thermocouples inserted as shown in Fig. 1.

Figure 2 shows manually controlled pneumatic load. The load is increased stepwise when the temperature of the bearing surface is stabilized (Fig. 3). When the applied load exceeds the load-carrying capacity of the bearing, input current to the motor, which represents friction loss of the journal bearing, is increased drastically as shown in Fig. 4, and then the motor is stopped automatically.

To investigate the effect of the bearing dimension and the operating condition on the bearing performance, a series of tests listed in Table 1 is accomplished. Before each test, “running-in” operation is executed to minimize the effect of the surface roughness. After the running-in period, the journal surface is flattened as shown in Fig. 5.

**Test Results**

In this study, a bearing performance curve is used to specify the journal bearing characteristics, which indicates the relationship between the Sommerfeld number and the friction coefficient of the bearing. The Sommerfeld number of the journal bearing is defined as

\[
S = \frac{\eta N}{R} \left( \frac{R}{C} \right)^2.
\]

At each stage of the load, the maximum surface temperature among the four measuring values is used to determine the lubricant viscosity \( \eta \). The bearing load \( F \) is calculated from the pneumatic load using the leverage rule. The friction coefficient of the journal bearing is obtained as

\[
\mu = \frac{TRF}{RF},
\]

where the motor current \( i \) is averaged value over each stage of the load. If the journal bearing is operated in the hydrodynamic lubrication region, the friction coefficient is increased when the Sommerfeld number is increased. But in the mixed or boundary lubrication region, such relationship is no longer valid.

Figure 5 is the bearing performance curve obtained. In spite of having different bearing dimensions and operating conditions, it is found in either case that the lubrication regime is changed at the almost same value of the Sommerfeld number. This universal value of the critical Sommerfeld number can be used as a design criterion of journal bearings.

**NUMERICAL ANALYSIS**

In the numerical analysis, the bearing reliability is represented by the minimum film thickness of the journal bearing. Under static loading condition, the minimum film thickness can be determined from the equilibrium position of the journal center. The friction loss, which indicates the bearing efficiency, is written as (Booker, 1989)

\[
H = \int \left( \frac{h^3}{12\eta} \nabla p \cdot \nabla p + \frac{2}{h} \Delta U \cdot \Delta U \right) dx dz.
\]
As shown in Eq. (3), the pressure distribution in the lubricant film must be specified to estimate the friction loss of the journal bearing. To find the pressure distribution and the journal center position, the following equations are needed.

**Governing Equations**

The Reynolds equation that governs the pressure distribution in the lubricant film is

$$
\nabla \cdot \left( \frac{h}{12\eta} \nabla p \right) - \nabla \cdot (hU) + \frac{\partial h}{\partial t} = 0,
$$

where \( \nabla = \frac{\partial}{\partial x} \mathbf{i} + \frac{\partial}{\partial z} \mathbf{k} \), \( U = \mathbf{i} \omega / 2R \). Film thickness is written as

$$
h = C - e_x \cos \frac{x}{R} - e_y \sin \frac{x}{R}.
$$

The position of the journal center can be determined from equilibrium equations. Ignoring the journal inertia, equilibrium equations of the journal center become

$$
\int \rho \cos \frac{x}{R} \, dx \, dz = 0
$$

$$
\int \rho \sin \frac{x}{R} \, dx \, dz - F = 0.
$$

The journal center position can be determined from the condition that the pressure distribution calculated from the Reynolds Eq. (4) should satisfy the above equilibrium equations.

**Method of Analysis**

The Reynolds equation is discretized by finite element technique. When solving the Reynolds equation, Reynolds boundary condition is applied using Murty’s algorithm (Oh, 1984). The governing equations, the discretized Reynolds equation and the equilibrium equations, are solved simultaneously with Newton-Raphson iterative method (McIvor and Fenner, 1989; Oh and Goenka, 1985). Solution procedure can be found in Fig. 7.

**Simulation Results**

Figure 8 shows the effect of the change in the projected bearing area on the minimum film thickness. Corresponding friction loss change is shown in Fig. 8. The projected area of the bearing can be changed identically by changes in the bearing length or diameter, but the effects on the bearing performance are somewhat different. The bearing length is more effective to increase the minimum film thickness, while the friction loss is very sensitive to the bearing diameter. As a result, to design a reliable as well as efficient journal bearing, bearing length should be increased and the diameter decreased.

The minimum film thickness and the friction loss are also affected by the bearing clearance as shown in Fig. 10. To determine the upper bound of the bearing clearance, which guarantees the operation in the hydrodynamic lubrication region, the minimum film thickness calculated from the numerical analysis must be compared with the surface roughness of the journal and bearing surfaces (Stachowiak and Batchelor, 1993). Comparing the rate of changes of the minimum film thickness and the friction loss, lower bound of the bearing clearance can be determined. In the case of Fig. 10, 0.8 is found to be the lower bound of the normalized bearing clearance. When the bearing clearance is smaller than that, the rate of change of the friction loss become larger than that of the minimum film thickness.

**CONCLUSION**

In this study, journal bearings in hermetic reciprocating compressors are analyzed experimentally as well as numerically. In the experimental analysis, the critical Sommerfeld number is proposed as a design criterion. The method of numerical analysis presented is found to be effective to the optimum design of journal bearings. It is concluded that these results can pay an important role in the optimum design of journal bearings in reciprocating compressors.
REFERENCES


<table>
<thead>
<tr>
<th>Table 1: Test Conditions</th>
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<tbody>
<tr>
<td>Test No.</td>
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<tr>
<td>Lubricant Supply Temperature (°C)</td>
</tr>
<tr>
<td>Bearing Length (mm)</td>
</tr>
</tbody>
</table>

Figure 1: Journal Bearing Test Machine

Figure 2: Pneumatic Load

Figure 3: Bearing Surface Temperature

Figure 4: Motor Current
Figure 5: Surface Profiles

Figure 6: Experimental Result

Figure 7: Solution Procedure

Figure 8: Projected Bearing Area vs. Minimum Film Thickness

Figure 9: Projected Bearing Area vs. Friction Loss

Figure 10: Bearing Clearance vs. Minimum Film Thickness and Friction Loss