2008

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ABSTRACT

This study presents optimal design guidelines for the thrust slide-bearing in scroll compressors to yield maximum efficiency. The average Reynolds equation analysis by Patir & Cheng and the solid contact theory by Greenwood & Williamson are applied to calculate the resultant lubrication performance for a given value of the friction area of the thrust slide-bearing, thus determining the friction coefficient for the thrust slide-bearing. With increasing friction area, the friction coefficient was found to decrease gradually, due to the larger oil film force. However, the influence of the oil viscous force was found to increase for large friction areas, thus increasing the friction coefficient. As a result of these two counteracting effects, optimal performance was found at a certain intermediate value of the friction area. This optimal friction area was determined for a range of orbiting speeds, thrust loads, oil viscosities and wedge angles, in order to provide optimal performance design guidelines for thrust slide-bearings in scroll compressors.

1. INTRODUCTION

In the design of the thrust slide-bearing in a scroll compressor, the specification of the friction surface with oil grooves and surface finishes is normally performed by experience, but the bearing dimensions are usually not considered. These bearing dimensions are usually determined based on the size of the compressor, having been confirmed only by durability tests. Lubrication analysis considering solid contact has not been carried out at all. Design guideline for high performance thrust slide-bearings is lacking. Based on our experience, there must be an optimal design for these thrust slide-bearings in scroll compressors that currently are designed based on experience.

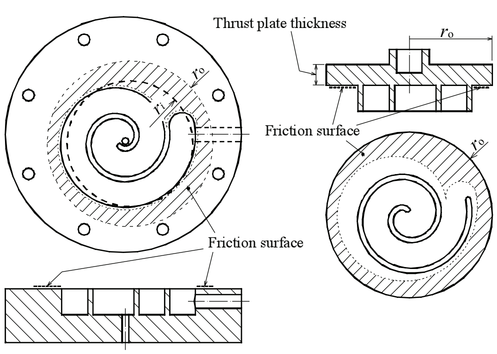
Previous experimental and theoretical studies (Ishii, *et al.*, 2004; Oku, *et al.*, 2004, 2006; Ishii *et al.*, 2007, 2008a; Oku, *et al.*, 2008b) for high performance lubrication of thrust slide-bearings in scroll compressors have identified the wedge formation at the friction planes, due to the elastic deformation of the orbiting thrust plate, as the key to outstanding improvement in lubrication performance. Based on this new concept of wedge formation, the oil film force exerted on the orbiting scroll will increase with increasing bearing friction area. As a result, the floating height of the orbiting scroll above the oil film will increase and the influence of solid contacts will decrease, yielding a significant decrease in the friction force. In the limit of very large friction areas, however, the increased oil viscous force results in a significant increase in the friction force. As a result of these two counteracting trends, the friction force at the thrust slide-bearing will exhibit a minimum at an intermediate value of friction area.

On the basis of these foregoing considerations, the present study provides a theoretical calculation methodology for the determination of the optimal outer radius of thrust slide-bearing for a fixed inner radius. The inner radius is determined through other optimal design simulations (Oku *et al.*, 2006) of the major dimensions of compressor configuration. Coupled with this earlier work, the present study provides optimal design calculations for thrust slide-bearings in scroll compressors. In the calculations, the friction surface temperature, needed to determine the oil viscosity, was assumed to be 70^{o}C (as measured in our experimental study). The pressure difference between the inside and outside of the thrust slide-bearing, significant in determining the wedge angle between the friction planes, was assumed to be 0.6 MPa (an actual working condition value). In order to determine the sensitivity of the optimal bearing area, similar lubrication performance calculations were conducted for variations in the oil viscosity (*i.e.*, friction surface temperature), the thrust plate thickness and the pressure difference through its effect on the wedge angle.

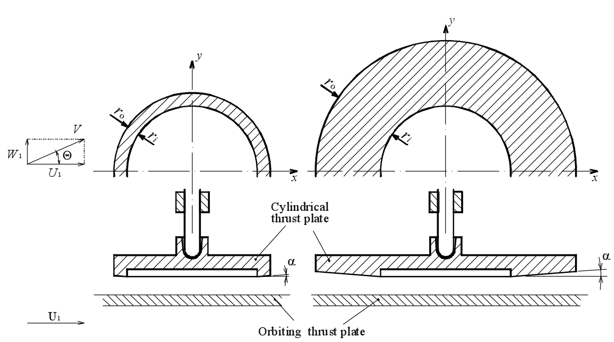
2. MODEL DEVELOPMENT OF THRUST SLIDE-BEARING FOR THEORETICAL ANALYSIS

The friction surfaces of thrust slide-bearing are shown in Figure 1 as the hatched area in the top views. The friction surface on the fixed thrust plate is essentially identical with that on the orbiting plate. The outer boundary of friction surface is the periphery of orbiting thrust disk with radius r_o, while the inner boundary is more complicated. To simplify the problem, the complicated inner boundary is represented by an equivalent circle with the radius r_i, such that the annular friction area enclosed by the dashed line in Figure 1a is the same as the actual friction area. This simplified friction surface is termed the "cylindrical thrust plate model," and is shown schematically in Figure 2, where the orbiting flat thrust plate drags the cylindrical thrust plate.

The inner radius r_i is basically determined by the refrigeration power of compressor, while a remaining significant concern for the engineer is how to determine the outer radius r_o. The cylindrical thrust plate models with different outer radii are shown in Figure 2, where the wedge shaped gap between the friction surfaces is represented by the rigid thrust plate with a slope of α. It is assumed that the wedge angle does not change with increasing outer radius. It should be noted here that the wedge angle α changes with the pressure difference between the inner and outer region of thrust slide-bearing and with the thickness of orbiting thrust plate. Using this optimal calculation model, friction loss energy can be calculated by theoretical calculations of solid contacts and fluid lubrication.



(a) (b)
(a) Fixed scroll and thrust plate
(b) Orbiting scroll and thrust plate
Figure 1 Thrust slide bearing surface of current scroll compressors.



(a) (b)
(a) Small friction surface area
(b) Large friction surface area
Figure 2 Thrust slide-bearing models for optimal design.

Table 1 Major specifications for calculations.

Surface roughness R_a	Orbiting thrust plate [μm]	3.0
	Cylindrical thrust plate [μm]	0.056(in) ~ 0.27(out)
Standard deviation of surface roughness	Orbiting thrust plate σ_1 [μm]	1.458
	Cylindrical thrust plate σ_2 [μm]	0.188(in) ~ 1.15(out)
Bearing dimension	Radius ratio γ	1.2 ~ 3.0
	Outer radius r_o [mm]	36 ~ 90
	Inner radius r_i [mm]	30
Wedge angle tanα ($\times 10^{-6}$)		80
Pivot height L_{piv} [mm]		15.0
Plastic flow pressure p_c [MPa]		1600
Shearing strength τ [MPa]		240
Surface density of asperities η [mm^{-2}]		400
Asperity summits radius β [μm]		8.0
Inner pressure p_{in} [MPa]		0.5
Outer pressure p_{out} [MPa]		1.1
Pressure difference Δp [MPa]		0.6
Axial spring force F_s [N]		600
Orbiting speed N [rpm]		300～6000
Orbiting radius r_{obt} [mm]		3.0
Oil viscosity μ^* [Pa s]		0.013
Sliding velocity V [m/s]		0.0942～2.26
Number of grids	Radial	30
	Circumferential	180

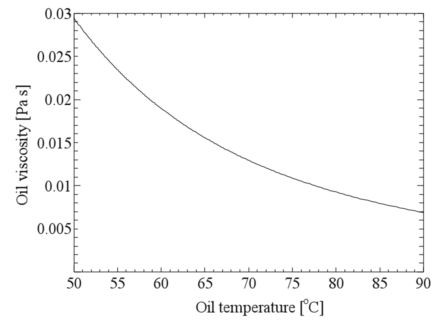


Figure 3 Oil viscosity (VG-56).

3. OPTIMAL PERFORMANCE DESIGN SIMULATIONS FOR THRUST SLIDE-BEARING

3.1 Calculated Conditions

Numerical calculations were made for the major specifications of an actual scroll compressor, shown in Table 1. The inner radius r_i was fixed at 30 mm and the outer radius r_o varied from 36 mm to 90 mm. The radius ratio γ ($=r_o/r_i$) varied from 1.2 to 3.0. The pressure difference Δp was 0.6 MPa, and the measured friction surface temperature T_f was at about 70°C, for which the oil viscosity μ takes on a value of 0.013 Pa·s (see Figure 3). The wedge angle α is fixed at 80×10^{-6} rad based on FEM analysis of the elastic deformation of orbiting thrust plate for Δp=0.6 MPa. The orbiting speed N was varied from 300 rpm to 6000 rpm with an orbiting radius of 3.0 mm, resulting in a bearing surface sliding speed, V, from 0.0942 to 2.26 m/s. The surface roughnesses on the fixed and orbiting plates are assumed to have a Gaussian distribution and are characterized by a factor σ which was introduced by Oku *et al.* (2008b), to which the reader is referred for further details. There were 180 lattice divisions in the tangential direction and 30 in the radial direction in the numerical calculations.

3.2 Calculation Procedure

When orbiting thrust plate is driven, the cylindrical thrust plate floats upward in the axial direction and becomes inclined about x- and y-axes by then unbalanced oil film force distribution, as shown in Figure 4, where the inclination angles about the x- and y-axes are represented by ψ_x and ψ_y, respectively. The cylindrical thrust plate is pressed downward by axial thrust force F_s through the pivot bearing. Considering the attitude of the thrust plate and the wedge, the oil film thickness between the cylindrical and flat thrust plates is given by a function of the polar coordinates with radius r and angle θ as:

$$h(r,\theta) = h_0 + (r - r_i)\tan\alpha - r\cos\theta\cdot\psi_y + r\sin\theta\cdot\psi_x \qquad (r_i \le r \le r_o) \qquad (1)$$

The oil-film pressure $p(r,\theta)$ generated in the thrust bearing surface can be numerically calculated from the average Reynolds equations by Patir and Cheng (1978, 1979) developed for a rough surface slide-bearing with isothermal and incompressible fluid. Integrating $p(r,\theta)$ over the whole bearing surface, the resultant oil film force F_{OIL} can be calculated. In addition, the oil film viscous force F_{vs} on the bearing surface with random roughness, due to oil viscosity, can be calculated. On the other hand, using the solid contact theory by Greenwood and Williamson (1966), the local real contact area A between the sliding surfaces can be calculated from the clearance height distribution. As a result, the solid contact force F_{sc} and the solid shearing force F_{ss} can be calculated. The resultant frictional force F_f

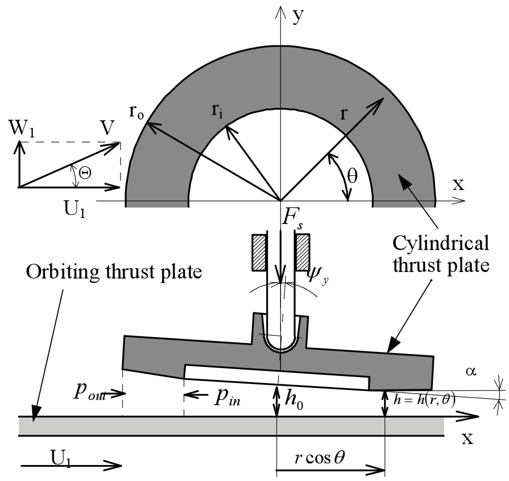


Figure 4 Mathematical model of thrust slide-bearing for theoretical analysis of fluid lubrication.

is given by the sum of oil film shearing force F_{vs} and solid shearing force F_{ss}, while the resultant thrust force F_T is given by the sum of axial spring force F_s and nominal gas thrust force F_p.

First, assuming an initial attitude of the cylindrical thrust plate for the clearance height given by Eq. (1), the solid contact and friction forces are calculated. Secondly the average Reynolds equations are solved numerically by the method of Successive Over-Relaxation (SOR) to determine the oil film forces for given boundary conditions. Calculated results are then fed back to the equilibrium equations for the forces and moments on the cylindrical thrust plate to determine a more correct attitude of the cylindrical thrust plate.

In order to carry out performance optimization of the thrust slide-bearing in a scroll compressor, it is necessary to minimize the friction power loss, W_f, which can be calculated by multiplying the obtained friction force F_f by the orbiting radius r_{obt} and the orbiting angular velocity ω. This is the most significant index for maximizing bearing performance.

3.3 Calculated Results

Calculated results of the friction forces and the average clearance are shown in Figures 5(a) to 5(e), where the abscissa is the radius ratio γ from 1.2 to 3.0. The left ordinate represents the friction force, while the right represents the oil film thickness. The orbiting speed N is from 1200 rpm to 6000 rpm. The solid shearing force F_{ss}, oil viscous

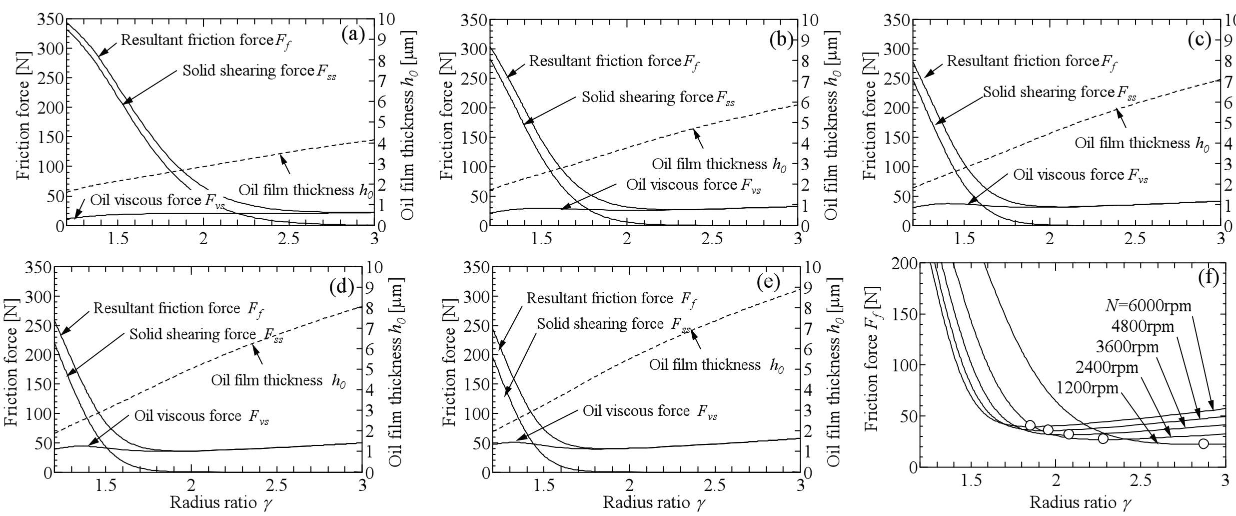


Figure 5 Calculated results of friction force F_{ss}, F_{vs}, F_f and oil film thickness h_0 vs. radius ratio γ (Δp=0.6 MPa): (a)N=1200 rpm; (b) N=2400 rpm; (c) N=3600 rpm; (d) N=4800 rpm; (e) N=6000 rpm; (f) Resultant friction force F_f for various N.

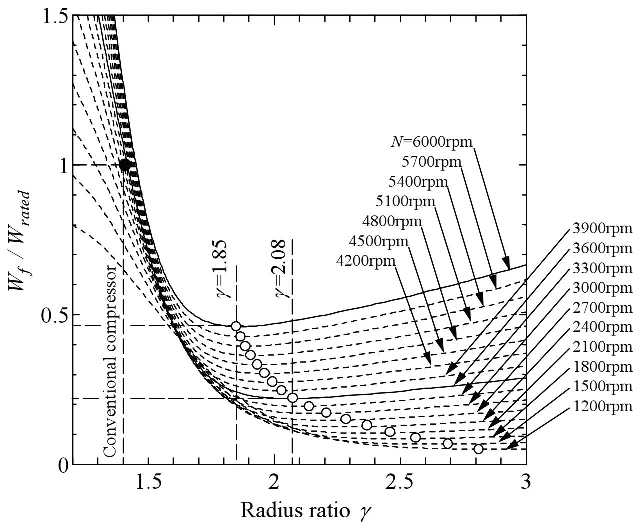


Figure 6 Non-dimensional energy loss W_f / W_{rated} (Δp=0.6 MPa).

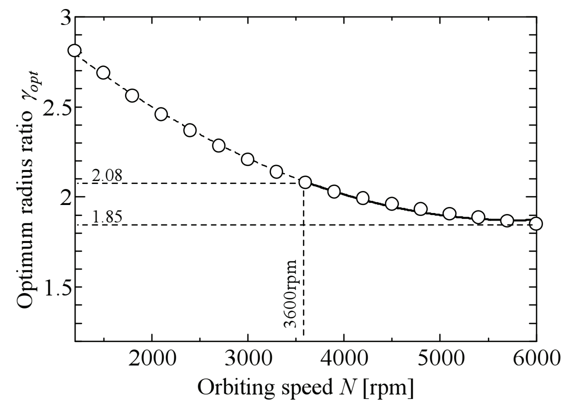


Figure 7 Optimum radius ratio γ_{opt} vs. orbiting speed N (Δp=0.6 MPa).

shearing force F_{vs} and resultant friction force F_f are shown by solid lines, while the oil film thickness is indicated by the dashed lines. The calculated results can be summarized, independently of the orbiting speed: with increasing radius ratio γ, the oil film thickness h_0 increases and consequently the solid shearing force F_s rapidly decreases, while the oil viscous force F_{vs} increases. As a result, the resultant friction force F_f rapidly decreases initially and then increases with increasing radius ratio γ. The resultant friction forces for various orbiting speed N are compared in Figure 5(f), in which it can be seen that the radius ratio for the minimum value of F_f decreases with increasing orbiting speed N, as plotted by open circles.

In order to determine optimal performance design guidelines, the dominant friction power loss was calculated from the resultant friction force F_f, and is shown in Figure 6. In this figure, the ordinate is friction power loss normalized by the rated power loss W_{rated} for the actual scroll compressor with γ= 1.4 and operating at N=3600 rpm. The filled circle is the condition for which the friction loss is equal to the rated loss. The minimum friction power loss is plotted by an open circle for each orbiting speed. At the rated orbiting speed of N=3600 rpm, the minimum friction power loss slightly more than 0.20 occurs at a radius ratio of γ=2.08, indicating a drastic reduction of approximately 80% relative to that in the actual current scroll compressor design.

Figure 7 presents the optimal radius ratio γ_{opt} vs. orbiting speed N, which shows that the optimal radius ratio decreases with increasing orbiting speed. For orbiting speeds greater than 3600 rpm, the radius ratio of the thrust slide-bearing should be designed to be γ=2.08 or less, falling 1.85 at N=6000 rpm.

4. OPTIMAL PERFORMANCE DESIGN GUIDELINES

In order to investigate the sensitivity of these optimal design guidelines for thrust slide-bearings, other calculations were undertaken for variations in the oil viscosity (*i.e.*, friction surface temperature), the thrust plate thickness and the pressure difference through its effect on the wedge angle.

4.1 Oil Viscosity

Since the friction surface temperature changes with the refrigeration cycle load, it is necessary to examine the influence of temperature on the optimal bearing radius ratio. For this purpose, similar simulations were conducted with the oil viscosity μ* varied from 0.019 to 0.009 Pa·s which correspond the friction surface temperatures from 60 to 80 °C, as shown in Figure 2.

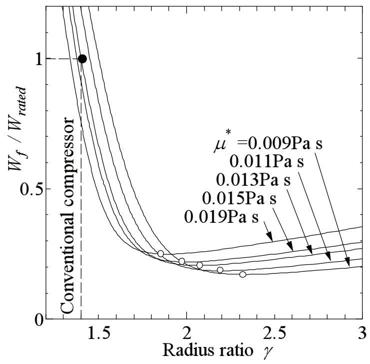


Figure 8 Non-dimensional energy loss W_f/W_{rated} (Δp=0.6 MPa, N=3600 rpm)

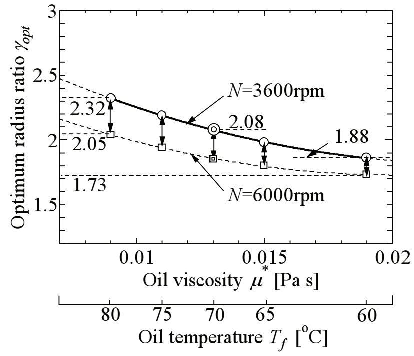


Figure 9 Optimum radius ratio γ_{opt} vs. oil viscosity μ^* (Δp=0.6 MPa).

Only the calculated results for the non-dimensional friction power loss are shown in Figure 8, where the actual scroll compressor working condition is again plotted by the filled circle. With decreasing oil viscosity, the optimal radius ratio γ, plotted as open circles, increases and the friction power loss decreases. Figure 9 shows the optimal radius ratio γ_{opt} vs. the oil viscosity. The data point for γ=2.08 plotted near the center of the plot by a double-circle is for a friction surface temperature of 70°C at an orbiting speed of 3600 rpm. The optimal radius ratio changes from 1.88 to 2.32 with increasing friction surface temperatures from 60 to 80°C. In addition, the optimal radius ratio γ_{opt} at N=6000 rpm is shown by the open squares and dotted line, and is smaller than that at the lower speed, varying from 1.73 to 2.05.

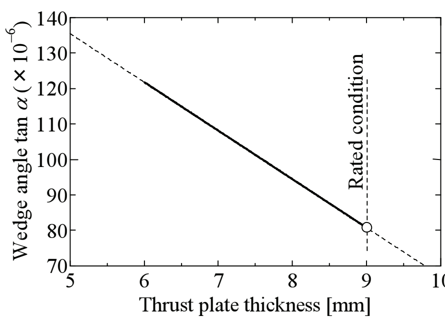


Figure 10 Wedge angle α vs. thrust plate thickness.

4.2 Thrust Plate Thickness

If the thickness of thrust plate is decreased, the wedge angle between the friction surfaces increases, and the lubrication performance is improved, thus changing the optimal radius ratio. The wedge angle can be calculated by FEM analysis (Ishii *et al.,* 2008a) for various thicknesses of the thrust plate. Figure 10 shows the dependence of the wedge angle α on the thrust plate thickness. The wedge angle is seen to increase almost linearly with decreasing thrust plate thickness. The wedge angle of the actual scroll compressor thrust plate with the thickness of 9 mm is plotted by the open circle.

The non-dimensional friction power loss W_f is shown in Figure 11, where the filled circle represents the actual working condition (γ=1.4, thrust plate thickness = 9 mm, oil viscosity μ^*=0.013 Pa·s), while the unfilled symbols are for optimal performance. The optimal radius ratio decreases with increasing the wedge angle. Figure 12 is a

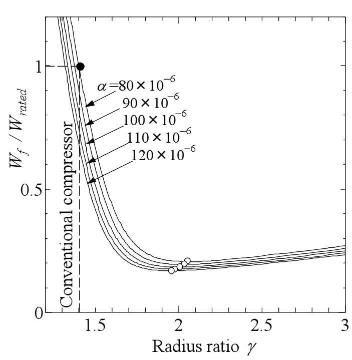


Figure 11 Non-dimensional energy loss W_f/W_{rated} (Δp=0.6 MPa, N=3600 rpm).

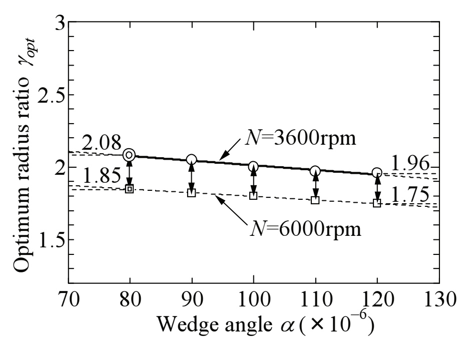


Figure 12 Optimum radius ratio γ_{opt} vs. wedge angle α (Δp=0.6 MPa).

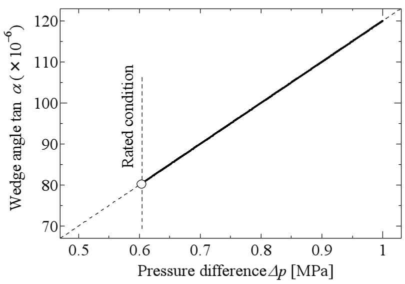


Fig. 13 FEM calculated wedge angle α vs. pressure difference Δp.

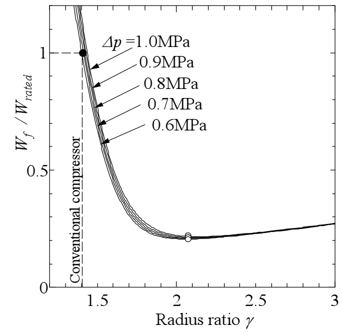


Fig. 14 Non-dimensional energy loss W_f / W_{rated} (N=3600 rpm).

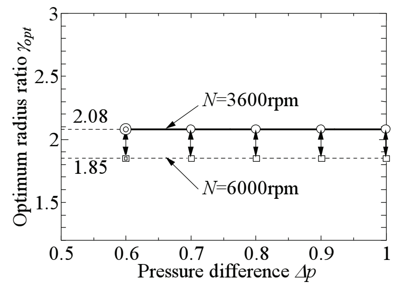


Fig. 15 Optimum radius ratio γ_{opt} vs. pressure difference Δp.

rearranged portrayal of optimal radius ratio as a function of wedge angle α. As shown by solid lines and blank circles, the optimal radius ratio changes only about 5% or less, ranging from 2.08 to 1.96 with increasing the wedge angle. Blank squares and dotted line are the optimal radius ratio for the highly loaded condition of N=6000 rpm. These optimal ratios are about 0.2 less than the corresponding value for N=3600 rpm.

4.3 Pressure Difference

In actual scroll compressors, the orbiting scroll is forced upward against the fixed scroll by the intermediate pressure in order to maintain stable orbiting motion. In order to examine the effect of the pressure difference Δp between the outside intermediate pressure space and inner suction pressure space, the lubrication characteristics were calculated for Δp=0.6 ~ 1.0 MPa, where the wedge angle determined from FEM analysis was assumed to change with the pressure difference ranging from 80×10^{-6} to 120×10^{-6}, as shown in Figure 13.

Similar performance simulations yielded the results in Figure 14 for an orbiting speed of 3600 rpm. For pressure difference changes from 0.6 to 1.0 MPa, the friction power loss shows no notable change. With increasing pressure difference, the oil film pressure increases due to the increased wedge angle, while the oil film thickness decreases due the increased thrust load. These two effects cancel, yielding no net effect.

Figure 15 shows the rearranged optimal radius ratio γ_{opt} as a function of pressure difference Δp, plotted as open circles and the solid line, where γ_{opt} is constant 2.08. The open squares and the dashed line are the optimal ratio for the highly loaded condition of N=6000 rpm. The optimal ratio takes on a constant value of γ=1.85.

5. CONCLUSIONS

Optimal design guidelines for thrust slide-bearings in scroll compressors for maximum lubrication performance were developed from theoretical calculations. Average Reynolds equation and solid contact theory were applied to calculate the resultant friction force and friction power loss for a range of outer-to-inner radius ratios of friction surface. The results are summarized as follows:

(1) Theoretical calculations for the radius ratio varying from 1.2 to 3.0, for a pressure difference of 0.6 MPa and oil viscosity at 70 °C were conducted to verify that the resultant friction force on the friction surface exhibits a minimum value, due to the characteristics of the oil viscous shearing force and the solid shearing force. The friction power loss at the friction surface exhibits a minimum value at the optimal radius ratio of 2.08 at 3600 rpm and of 1.85 at 6000 rpm.

(2) When the inner-to-outer radius ratio of scroll compressor thrust slide-bearing is chosen at 2.08, the friction power loss can be reduced to 20% of that for an actual conventional compressor.

(3) Theoretical calculations were conducted for a variety of oil viscosity corresponding to the friction surface temperature changes from 60 to 80 °C. As a result, the optimal radius ratio changes from 1.73 to 2.32.

(4) Theoretical calculations were conducted for a variety of wedge angles corresponding to a change in thrust plate thickness. The optimal radius ratio was essentially constant at 1.96 to 2.08 at 3600 rpm and at 1.75 to 1.85 at 6000 rpm.

(5) Theoretical calculations were conducted for a variety of pressure differences from 0.6 to 1.0 MPa. The optimal radius ratio was constant at about 2.8 at 3600 rpm and at about 1.85 at 6000 rpm.

It should be finally noted here that the present study shows surprising results for the optimal performance design guideline of thrust slide-bearing in scroll compressors, which currently being confirmation by laboratory tests.

NOMENCLATURE

dA	local real contact are	(m^2)
F_f	resultant frictional force	(N)
F_{OIL}	oil film force	(N)
F_p	nominal gas thrust force	(N)
F_s	axial spring force	(N)
F_{sc}	solid contact force	(N)
F_{ss}	solid shearing force	(N)
F_T	resultant thrust force	(N)
F_{vs}	oil viscous force	(N)
h	nominal oil film thickness	(μm)
h_o	average clearance	(μm)
N	orbiting speed	(rpm)
p	oil film pressure	(Pa)
p_c	plastic flow pressure	(Pa)
p_{out}, p_{in}	boundary pressure	(Pa)
r_o, r_i	plate radius	(mm)
T_f	friction surface temperature	(°C)
U_1, W_1, V	boundary velocity	(m/s)
V_{r2}	variance ratio	(-)
W_f, W_{rated}	friction loss energy	(W)
α	wedge angle	(rad)
α^*	local solid contact ratio	(-)
β	asperity summits radius	(μm)
Δp	pressure difference	(Pa)
γ, γ_{opt}	radius ratio of thrust bearing	(-)
η	surface density of asperities	(m^{-2})
Θ	orbiting angle	(rad)
μ	frictional coefficient	(-)
μ^*	oil viscosity	(Pa·s)
σ_1, σ_2	standard deviations of surface roughness	(μm)
τ	shearing strength	(Pa)
ϕ_f, ϕ_{fs}	shear stress factor	(-)
ψ_x, ψ_y	rotation angle	(rad)

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ACKNOWLEDGEMENT

The authors extend their great thanks to Prof. Hiroshi Yabe and Prof. Toshihiro Ozasa, Osaka Electro-Communication University for their bountiful advice concerning the present theoretical calculations. Furthermore, the authors would like to extend their sincere gratitude to Mr. Masahiro Atarashi, Director, Appliance Development Center, Corporate Engineering Division, and Mr. Shuichi Yamamoto, Director, Air Conditioning Devices Business Unit, Matsushita Home Appliances Company, the Matsushita Electric Industrial Co., Ltd., for their cooperation in carrying out this work and their permission to publish this study.