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Tatsuya Oku
Osaka Electro-Communication University

Keiko Anami
Ashikaga Institute of Technology

Noriaki Ishii
Osaka Electro-Communication University

Kiyoshi Sano
Matsushita Electric Industrial Co.

LUBRICATION MECHANISM AT THRUST SLIDE-BEARING OF SCROLL COMPRESSORS (THEORETICAL STUDY)

Tatsuya Oku[1], Keiko Anami[2], Noriaki Ishii[3], Kiyoshi Sano[4]

[1]Doctoral Course Student, Division of Mechanical and Control Engineering,
Graduate School, Osaka Electro-Communication University, 18-8 Hatsu-chou, Neyagawa-city,
Osaka 572-8530, Japan   Tel: +81-720-20-4565; E-mail: d04201@isc.osakac.ac.jp

[2]Assistant Professor, Department of Mechanical Engineering, Ashikaga Institute of Technology,
268-1 Ohmae-chou, Ashikaga-city, Tochigi , Japan,
Tel: +81-284-62-0609 (Ext.241); Fax: +81-284-62-9802; E-mail: anami@ashitech.ac.jp

[3]Professor, Department of Mechanical Engineering, Osaka Electro-Communication University,
Tel: +81-720-20-4561; Fax: +81-720-20-4577; E-mail: ishii@isc.osakac.ac.jp

[4]Senior Staff Engineer, Air Conditioning Research Laboratory,
Matsushita Home Appliance Company, Matsushita Electric Industrial Co., Ltd., 2-3-1-1
Nojihigashi, Kusatsu, Shiga 525-8520, Japan   Tel: +81-77-567-9801; Fax: +81-77-561-3201

ABSTRACT

This study presents a theoretical explanation for the excellent lubrication at the thrust slide-bearing of scroll compressors, caused by elastic deformation of the thrust plate, due to large loads. In theoretical calculations, the thrust slide-bearing surfaces are treated as a rough sliding one, and the average Reynolds equation by Patir & Cheng for the rough surfaces is applied to analyze the fluid lubrication at the thrust slide-bearing, while the Solid Contact Theory by Greenwood & Williamson is applied to analyze the plastic and elastic contacts between the orbiting and fixed thrust plates. For given values of the wedge angle between the sliding surfaces, the oil film pressure, the solid contact force, the fluid frictional force and the solid shearing drag force are calculated to determine the resultant friction coefficient at the thrust slide-bearing. As a result, the theoretical calculations show a good agreement with the lubrication test results, thus unveiling the excellent fluid lubrication at the thrust slide-bearing, caused by the wedge formation due to large thrust loads.

1. INTRODUCTION

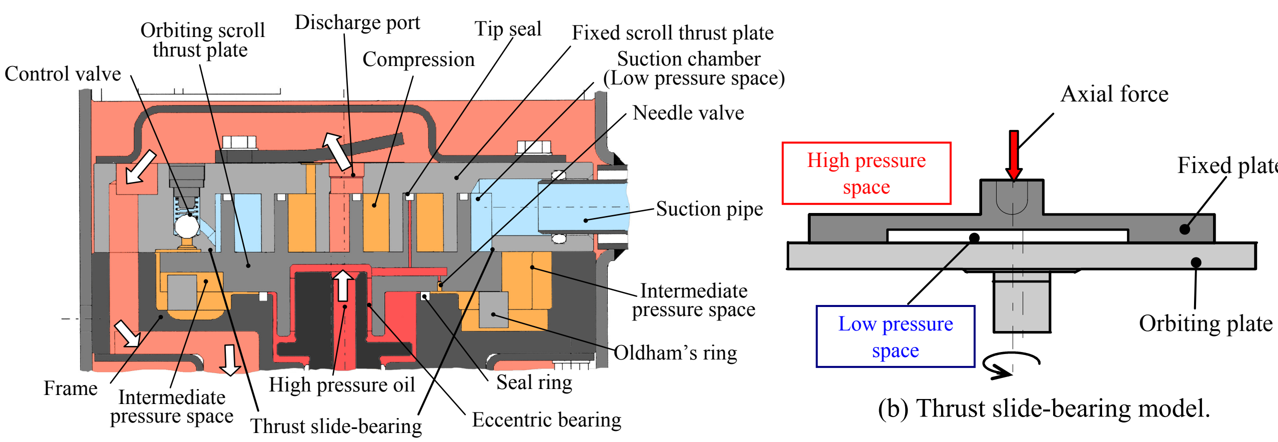
The common type of thrust bearing of the scroll compressors, widely used for room air-conditioners, is the sliding type for its high performance in lubrication and for its low noise generation, where the orbiting thrust flat plate is firmly pressed on the fixed one. The thrust slide-bearing supports a large thrust force and is not lubricated by a special device like an oil pump with high power, but the thrust slide-bearing never induces any serious troubles in lubrication, such as a seizure of the sliding surfaces, and rather exhibits a better performance in lubrication. A few theoretical studies for the thrust slide-bearing of the scroll compressors have been reported by Kulkarni (1990a, b), and an experimental study by Nishiwaki et al (1996), but the essential characteristics of lubrication at the thrust slide-bearing have not been revealed at all. Thereupon, as the first step, lubrication tests for the thrust slide-bearing in the closed vessel pressurized with the refrigerant R-22 gas were conducted by Ishii et al (2004). As a result, it was shown that the pressure difference between the outside and inside spaces of the thrust slide-bearing made a significant role to improve the performance in lubrication: the friction coefficient from 0.043 to 0.033 at zero pressure difference drastically decreased to about 0.008 at 1.0MPa pressure difference. In addition, the observation of wear state of the friction surface, after tested, suggested that the excellent lubrication was caused by a wedge formation between the friction surfaces, in addition to the oil flow into the inner space with lower pressure.

The major purpose of the present study is to theoretically explain the excellent lubrication at the thrust slide-bearing. The orbiting thrust flat plate, made of Aluminum alloy, is pressed on the fixed one due to high pressure loads on it and induces elastic deformation, thus forming a wedge against the rigid fixed thrust plate. Such a wedge formation surely depends upon various factors, such as the pressure loads on the back and inner sides of the orbiting scroll, the oil film pressure at the friction surface, the friction temperature and so on. Therefore, it is indeed a difficult subject to theoretically determine the wedge angle. Thereupon, for given wedge angles, a theoretical analysis is made for lubrication mechanism at the thrust slide-bearing. The average Reynolds Equation developed by Patir & Cheng (1978, 1979), applicable to any general roughness structure, is used to analyze the fluid lubrication at the thrust slide-bearing, and the Solid Contact Theory by Greenwood & Williamson (1966) is used to analyze the plastic contacts between the orbiting and fixed thrust plates. Accordingly, the oil film pressure, the oil film thrust force, the solid contact force, the fluid frictional force and the solid shearing drag force are calculated to determine the resultant attitude of the orbiting scroll and the resultant friction coefficient for a series of conditions of the lubrication tests.

2. THRUST SLIDE-BEARING AND ITS MODEL FOR LUBRICATION TESTS

The compression mechanism of a high-pressure type scroll compressor is shown in Figure 1a, where the orbiting thrust plate is pressed upward against the fixed thrust plate, by the high and intermediate pressure oil. The orbiting thrust plate, made of Aluminum Alloy, induces an elastic deformation, thus forming a wedge at the thrust slide-bearing. In addition, the pressure difference between the intermediate and suction pressures, across the thrust slide-bearing, induces an oil flow from the outside into the inside.

In order to examine the lubrication performance at the thrust slide-bearing, the simplified model shown in Figure 1b was made, where the orbiting scroll thrust plate was replace by a cylindrical thrust plate, which was arranged to the up side and fixed through a pivot bearing, while the fixed scroll thrust plate was replaced by a rigid flat thrust plate, which was arranged to the down side and driven by the motor for orbiting motion. The outside of the test model is pressurized in a tribo-tester and the inside was released to the atmospheric circumstance to give a pressure difference between the outside and inside spaces.



(a) Thrust slide-bearing.

(b) Thrust slide-bearing model.

Figure 1: Thrust slide-bearing of a scroll compressor and its model for lubrication tests.

3. MODEL DEVELOPMENT FOR THRORETICAL ANALYSES

3.1 Average Reynolds Equation

The thrust slide-bearing model shown in Figure 1b can be represented by Figure 2, for theoretical calculations of the fluid lubrication by the oil film between the cylindrical and orbiting thrust plates. The cylindrical thrust plate with the outer radius r_o and the inner radius r_i has a wedge angle α at its periphery, the center of which is pressed downward at the spring force F_s through the pivot bearing. The cylindrical thrust plate can move about the x and y

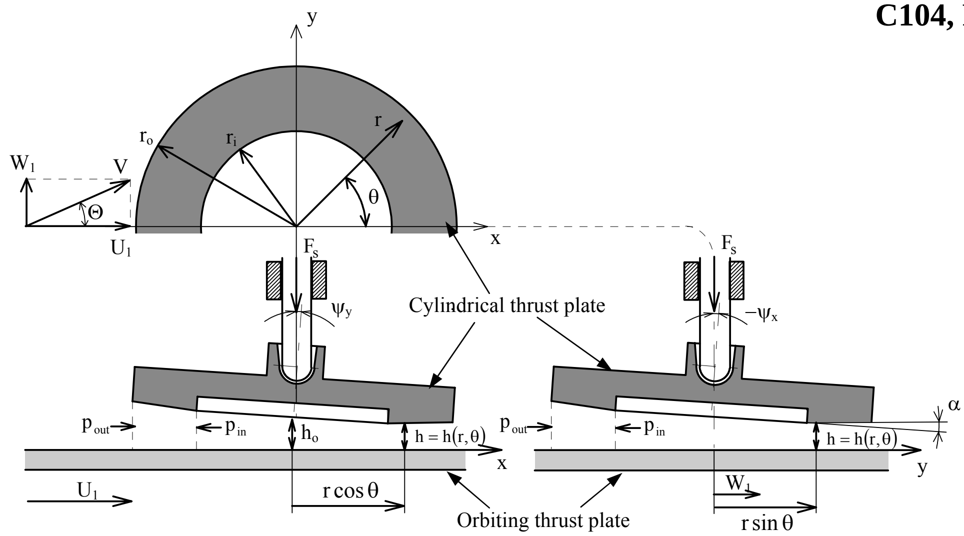


Figure 2: Mathematical model of thrust slide-bearing for theoretical analysis of fluid lubrication.

axes, as represented by rotations ψ_x and ψ_y, respectively. The x and y axes are the Cartesian coordinates on the orbiting thrust plate surface, with the origin at the center of the cylindrical thrust plate.

With the average clearance height between the cylindrical and orbiting thrust plates, h_o, the oil film thickness h is given by a function of the polar coordinates with radius r and angle θ:

$$h(r,\theta) = h_0 + (r - r_i)\tan\alpha - r\cos\theta\cdot\psi_y + r\sin\theta\cdot\psi_x \quad \text{where} \quad r_i \le r \le r_o\,. \tag{1}$$

The boundary pressure on the oil film is indicated by p_{out} at the periphery and by p_{in} at the inner circumference. The boundary velocity on the orbiting thrust plate is indicated by U_1 and W_1 in the x and y directions, respectively:

$$U_1 = V\cos\Theta\,,\, W_1 = V\sin\Theta\,, \tag{2}$$

where V and Θ represent the orbiting velocity and its orbiting angle, respectively.

Here introduce σ for the composite rms roughness of the standard deviations of surface roughness, σ_1 for the orbiting thrust plate and σ_2 for the cylindrical one. For an isothermal, incompressible oil, the pressure p in Elastohydrodynamic lubrication between the rough surfaces is governed by the Average Reynolds Equation relative to the Cartesian coordinates of x and y, derived by Patir and Cheng (1979):

$$\frac{\partial}{\partial x}\left(\phi\frac{h^3}{12\mu^*}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\phi\frac{h^3}{12\mu^*}\frac{\partial p}{\partial y}\right)=\frac{U_1}{2}\frac{\partial \bar{h}_T}{\partial x}+\frac{U_1}{2}\sigma\frac{\partial \phi_s}{\partial x}+\frac{W_1}{2}\frac{\partial \bar{h}_T}{\partial y}+\frac{W_1}{2}\sigma\frac{\partial \phi_s}{\partial y}+\frac{\partial \bar{h}_T}{\partial t}\,, \tag{3}$$

where μ^* represents the oil viscosity. h represents the nominal oil film thickness given by (1), as shown by the dotted line in Figure 3. $\bar{h}_T$ represents the average oil film thickness which can be obtained by integrating the local film thickness h_T, with introduction of a polynomial density function in stead of the Gaussian frequency density function for random surface roughness, as given by the following expressions:

$$\bar{h}_T = h \qquad (H_r \ge 3)\;,$$

$$\bar{h}_T = \frac{3\sigma}{256}\left\{35 + Z\left(128 + Z\left(140 + Z^2\left(-70 + Z^2\left(28 - 5Z^2\right)\right)\right)\right)\right\} \qquad (H_r < 3), \tag{4}$$

where H_r represents the oil film thickness to surface roughness ratio:

$$H_r \equiv h/\sigma\;. \tag{5}$$

Z represents one third of H_r ($Z \equiv H_r/3$). ϕ is the pressure flow factor for isotropic surfaces, representing the effect of surface roughness on the oil flow due to mean pressure difference:

$$\phi = 1 - 0.90\exp(-0.56H_r)\;. \tag{6}$$

ϕ_s is the shear flow factor, representing the effect of the surface roughness on fluid transportation, as given by

$$\phi_s = V_{r1}\Phi_s(H_r) - V_{r2}\Phi_s(H_r), \tag{7}$$

where V_{r1} and V_{r2} are the variance ratio, defined by

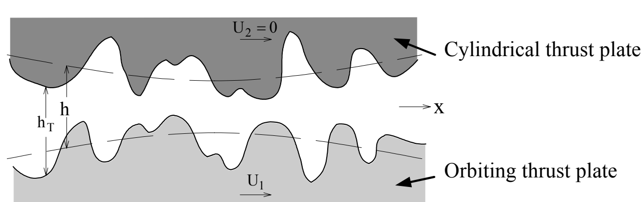


Figure 3: Film thickness function.

$$V_{r1} = \left(\frac{\sigma_1}{\sigma}\right)^2,\ V_{r2} = \left(\frac{\sigma_2}{\sigma}\right)^2. \tag{8}$$

Φ_s is a function of H_r, given by

$$\Phi_s = A_1 H_r^{\alpha_1} e^{-\alpha_2 H_r + \alpha_3 H_r^2} \quad (H_r \le 5)\,, \qquad \Phi_s = A_2 e^{-0.25 H_r} \quad (H_r > 5)\,, \tag{9}$$

where A_1, A_2, α_1, α_2 and α_3 are given in a Table by Patir & Cheng (1979), for isotropic surface roughness :

$$A_1{=}1.899, \qquad A_2{=}1.126, \quad \alpha_1{=}0.98, \quad \alpha_2{=}0.92, \quad \alpha_3{=}0.05\ . \tag{10}$$

Here introduce the following non-dimensional variables:

$$R \equiv r/r_o, \qquad P \equiv p/p_a, \qquad H \equiv h/h_{ref}, \qquad \overline{H}_T \equiv \overline{h}_T/h_{ref}, \qquad \tau \equiv \omega t\ . \tag{11}$$

where p_a represents the atmospheric pressure, h_{ref} an arbitrary standard film thickness and ω the angular orbiting velocity. With these variables, equation (3) relative to the Cartesian coordinates (x, y) can be transformed to the following non-dimensional expression relative to the polar coordinates (R, θ):

$$\frac{1}{R}\frac{\partial}{\partial R}\left(\phi R H^3 \frac{\partial P}{\partial R}\right) + \frac{1}{R^2}\frac{\partial}{\partial \theta}\left(\phi H^3 \frac{\partial P}{\partial \theta}\right) = \lambda \cdot \frac{1}{R}\left\{\frac{\partial}{\partial R}\left(\overline{H}_T \cos(\theta - \Theta)\right) - \frac{\partial}{\partial \theta}\left(\overline{H}_T \sin(\theta - \Theta)\right)\right\}$$
$$+ \lambda\sigma \frac{1}{R}\left\{\frac{\partial}{\partial R}\left(\phi_s R \cos(\theta - \Theta)\right) - \frac{\partial}{\partial \theta}\left(\phi_s \sin(\theta - \Theta)\right)\right\} + \sigma_s \frac{\partial \overline{H}_T}{\partial \tau}, \tag{12}$$

where λ is the bearing number determining the bearing load capacity and σ_s is the squeeze number representing the squeeze film works, as defined by

$$\lambda = \frac{6\mu^* V}{r_o p_a}\left(\frac{r_o}{h_{ref}}\right)^2, \quad \sigma_s = \frac{12\mu^* \omega}{p_a}\left(\frac{r_o}{h_{ref}}\right)^2. \tag{13}$$

3.2 Oil Film Thrust and Viscous Forces

The Average Reynolds Equation (12) can be numerically solved to calculate the oil film pressure. Integrating p(r,θ) over the whole bearing surface, the resultant oil film force F_{OIL} can be calculated:

$$F_{OIL} = \iint p(r,\theta)\, r d\theta dr\ . \tag{14}$$

In addition, as given by Patier & Cheng (1979), the oil film viscous force F_{vs} on the bearing surface with random roughness, due to oil viscosity, can be calculated by

$$F_{vs} = \iint \frac{\mu^* V}{h}\left[(\phi_f + \phi_{fs}) - 2V_{r2}\phi_{fs}\right] r d\theta dr \quad , \tag{15}$$

where ϕ_f and ϕ_{fs} are called a “shear stress factor”, correcting the effect of the surface roughness on the oil film shearing force. These factors are given by the same expressions as expressions (7) to (10) (refer to Patier & Cheng 1979 for details). For smooth and parallel friction surfaces, ϕ_f and ϕ_{fs} approach 1.0 and 0, respectively, and then expression (15) is reduced to a well-known oil film shearing force.

3.2 Solid Contact Force and Its Shearing Force

The solid contact theory by Greenwood & Williamson (1966) assumes a plastic contact of the hemispherical projection and the flat plane, thus deriving the local solid contact ratio α*(r,θ) of the local real contact area dA to the local nominal contact area rdθdr, in the following expression:

$$\alpha^*(r,\theta)\left(\equiv \frac{dA}{r d\theta dr}\right) = \pi\eta\beta\sigma \int_h^\infty (s - h)\phi^*(s) ds, \tag{16}$$

where the standardized height distribution ϕ*(s) is empirically given by Gaussian to a very good approximation:

$$\phi^*(s) = \frac{1}{\sqrt{2\pi}} e^{-\frac{1}{2}s^2}\ . \tag{17}$$

η represents the surface density of asperities and β the asperity summits radius. Accordingly, the solid contact force F_{sc} and the solid shearing force F_{ss} are given by

$$F_{sc}\left(\equiv \int p_c \cdot dA\right) = \iint p_c \alpha^*(r,\theta) \cdot r d\theta dr, \quad F_{ss}\left(\equiv \int \tau \cdot dA\right) = \iint \tau\alpha^*(r,\theta) \cdot r d\theta dr, \tag{18}$$

where p_c and τ represent the plastic flow pressure and the shearing strength of the friction surface with softer material, respectively.

3.3 Friction Coefficient

The resultant frictional force F_f is given by the sum of the oil film shearing force F_{vs} and the solid shearing force F_{ss}, and the resultant thrust force F_T is given by the sum of the axial spring force F_s and the nominal gas thrust force F_p. Ultimately, the frictional coefficient μ is calculated by

$$\mu \ \left(\equiv \frac{F_f}{F_T}\right) = \frac{F_{vs} + F_{ss}}{F_s + F_p} \ , \qquad (19)$$

where the nominal gas thrust force F_p, due to the pressure difference between the outer and inner spaces of the thrust slide-bearing, is calculated by

$$F_p = \pi r_i^{\,2}\left(p_{out} - p_{in}\right) + \pi\left(r_o^{\,2} - r_i^{\,2}\right)\left(\frac{p_{out} - p_{in}}{2}\right). \qquad (20)$$

3.4 Attitude of Cylindrical Thrust Plate

The resultant frictional force F_f, the solid contact force F_{sc}, the oil film force F_{OIL} and the spring forces F_s, the gas forces F_{po} and F_{pi} exert on the cylindrical thrust plate, as shown in Figure 4, where the equilibrium equations of forces and moments are given by

$$F_{pi} + F_{OIL} + F_{sc} - F_s - F_{po} = 0 \ ,$$

$$\iint p(r,\theta)\cdot r\sin\theta\cdot rdrd\theta + \iint \alpha^*(r,\theta)\cdot p_c\cdot r\sin\theta\cdot rdrd\theta - Lpiv\cdot F_f\sin\Theta = 0 \ ,$$

$$-\iint p(r,\theta)\cdot r\cos\theta\cdot rdrd\theta - \iint \alpha^*(r,\theta)\cdot p_c\cdot r\cos\theta\cdot rdrd\theta + Lpiv\cdot F_f\cos\Theta = 0 \ , \qquad (21)$$

which determine the attitude of the cylindrical thrust plate, that is, the average clearance height h_o, the rotations ψ_x and ψ_y.

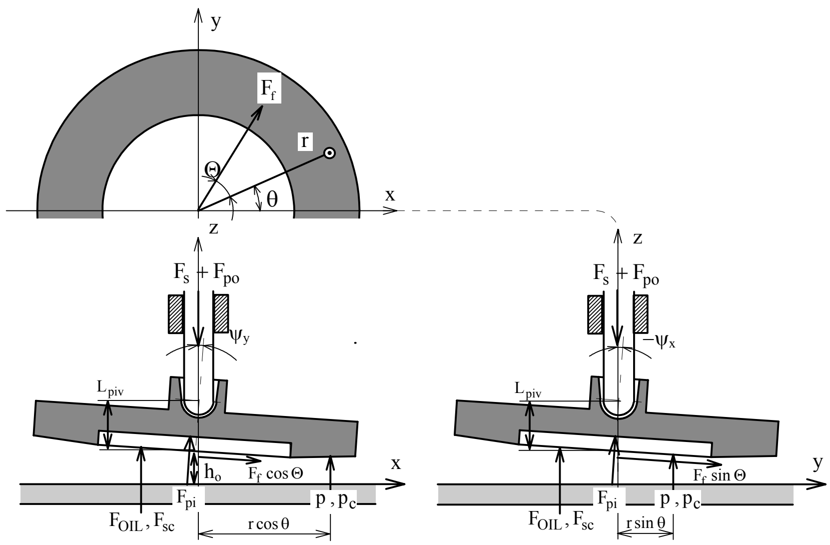


Figure 4: Forces on the cylindrical thrust plate.

4. MAJOR CHARACTERISTICS OF LUBRICATION

4.1 Numerical Calculations

First, the attitude of the cylindrical thrust plate is determined by the equilibrium equations given by (21). Second, the Average Reynolds Equation (12) is solved to determine the oil film forces, and then the solid contact forces are calculated. Calculated results are fed back to equations (21) to determine more correct attitude of the cylindrical thrust plate. For numerical calculations, the equation (12) can be reduced to the following difference approximation based on the central difference, as shown in Figure 5.

$$
\begin{aligned}
&\left[\frac{1}{R_P}\frac{1}{\Delta R^2}\left\{(\phi R H^3)_n+(\phi R H^3)_s\right\}+\frac{1}{R_P^2}\frac{1}{\Delta\theta^2}\left\{(\phi H^3)_e+(\phi H^3)_w\right\}\right]P_P\\
&-\frac{1}{R_P}\frac{1}{\Delta R^2}(\phi R H^3)_n\cdot P_N-\frac{1}{R_P}\frac{1}{\Delta R^2}(\phi R H^3)_s\cdot P_S\\
&-\frac{1}{R_P^2}\frac{1}{\Delta\theta^2}(\phi H^3)_e\cdot P_E-\frac{1}{R_P^2}\frac{1}{\Delta\theta^2}(\phi H^3)_w\cdot P_W\\
&\quad=\lambda\frac{1}{R_P}\frac{1}{\Delta R}\cos(\theta_P-\Theta)\left\{(H_T R)_n-(H_T R)_s\right\}\\
&\quad-\lambda\frac{1}{R_P}\frac{1}{\Delta\theta}\left\{(H_T\sin(\theta-\Theta))_e-(H_T\sin(\theta-\Theta))_w\right\}\\
&\quad+\lambda\sigma\frac{1}{R_P}\frac{1}{\Delta R}\cos(\theta_P-\Theta)\left\{(\phi_s R)_n-(\phi_s R)_s\right\}\\
&\quad-\lambda\sigma\frac{1}{R_P}\frac{1}{\Delta\theta}\left\{(\phi_s\sin(\theta-\Theta))_e-(\phi_s\sin(\theta-\Theta))_w\right\}\ ,
\end{aligned}
\qquad (22)
$$

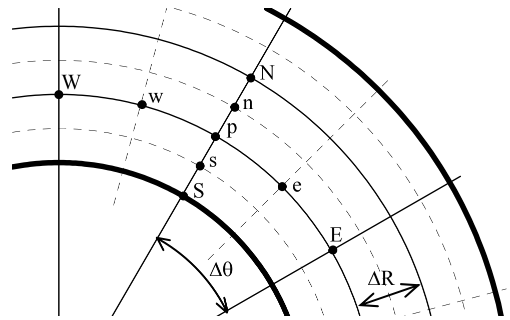


Figure 5 : Lattice division of oil film at the thrust slide-bearing.

where the pressure at point “p” is approximated from the pressures at points “W”, “E”, “N” and “S”. Application of this difference approximation to all the lattice points derives simultaneous linear equations, which can be numerically solved for given boundary conditions by using the SOR(Successive Over-Relaxation) method, to ultimately calculate the pressure P at all the lattice points.

Numerical calculations were made for the specifications shown in Table 1 and Figure 6, for the test peaces of lubrication tests by Ishii et al (2004). The standard deviation of roughness was σ_1=2.62μm for the orbiting thrust plate made of Cast Iron, and σ_2=0.784μm for the cylindrical thrust plate made of Aluminum Alloy with outer diameter r_o=130.0mm and the inner one r_i=75.7mm. The asperities of the Aluminum Alloy with the plastic flow pressure p_c=1600MPa were carefully examined to find the surface density η=150mm^{-2} and the summits radius β=2μm. The refrigerant oil VG-56 has the viscosity μ^*=0.051 Pa·s. The pressure difference at the thrust slide-bearing, Δp, was adjusted at 0 and 1.0MPa. Correspondingly, the resultant thrust force F_T was 800 and 9200N, respectively. The orbiting speed N was varied from 300 to 3600rpm, with the orbiting radius of 3.0mm, and hence the resultant sliding velocity V was varied from 0.0942 to 1.13m/s after all. The number of lattice division was fixed at 180 in the radial direction and at 24 in the tangential direction, as its permissible maximum value for our computer (Pentium4 2.6GHz, Memory 512MB, HDD 60GB, Compaq Visual Fortran Professional Edition 6.6.0).

The formation of wedge at the thrust slide-bearing is caused by the elastic deformation of the thrust plate, which depends upon the excessive oil film pressure also, in addition to the gas thrust and axial spring forces. It is indeed difficult to perfectly solve the present subject, taking all factors into exact considerations. Thereupon, as the first approach, the wedge inclination tanα was assumed, based on detailed examinations, as shown in Figure 6, where the dotted line is for Δp =0MPa and the solid line is for Δp =1.0MPa. The wedge at Δp =0MPa increases in inclination, since the oil film hydrodynamic pressure increases with increasing the orbiting speed. On the contrast, the wedge at

Table : Major specifications for calculations.

Standard deviation of surface roughness	Orbiting thrust plate σ_1 [μm]	2.62	
	Cylindrical thrust plate σ_2 [μm]	0.784	
Bearing dimension	Outer diameter r_o [mm]	130.0	
	Inner diameter r_i [mm]	75.7	
Plastic flow pressure p_c [MPa]		1600	
Surface density of asperities η [mm^{-2}]		150	
Asperity summits radius β [μm]		2.0	
Lubricant viscosity μ^* [Pa·s]		0.051	
Boundary pressure	Inside p_{in} [MPa]	0.1	
	Outside p_{out} [MPa]	0.1	1.1
Pressure difference Δp [MPa]		0	1.0
Nominal gus thrust force F_p [N]		0	8600
Axial spring force F_s [N]		800	600
Resultant thrust force F_T [N]		800	9200
Orbiting speed N [rpm]		300～3600	
Orbiting radius [mm]		3.0	
Sliding velocity V [m/s]		0.0942～1.13	
Number of lattice division	Radial	180	
	Tangential	24	

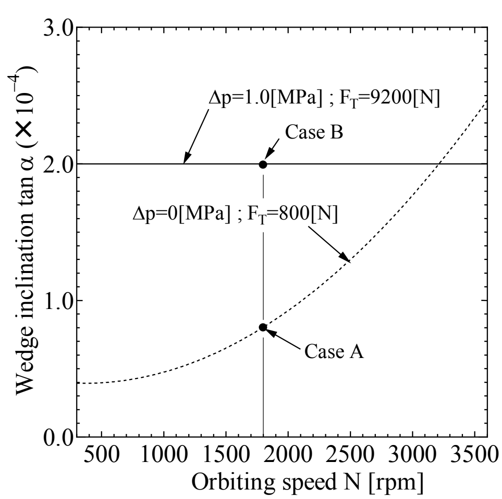


Figure 6 : Wedge inclination, assumed for calculations.

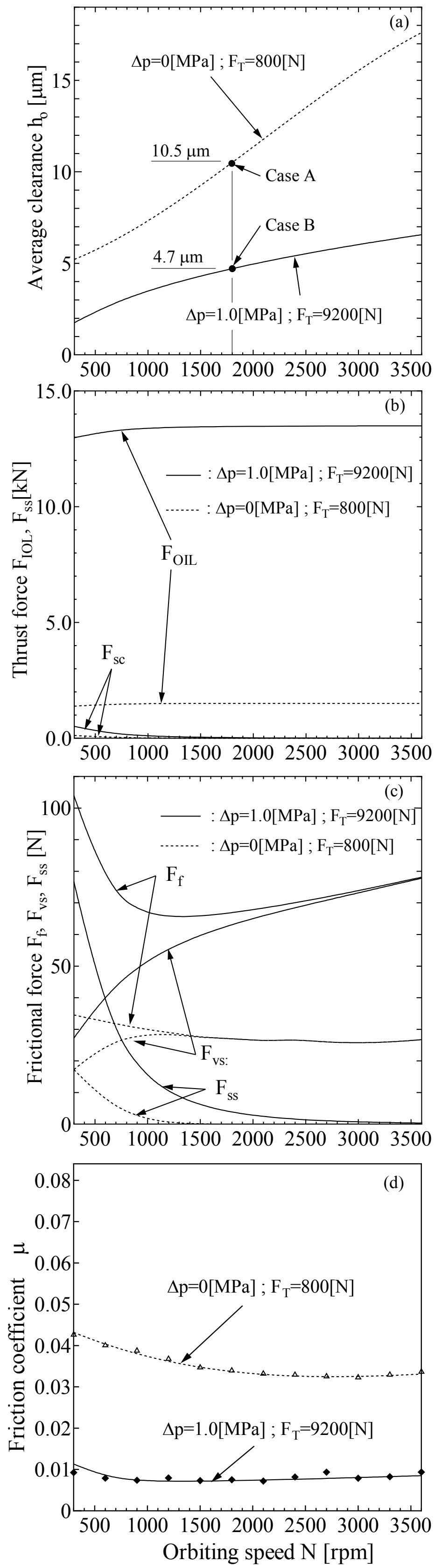


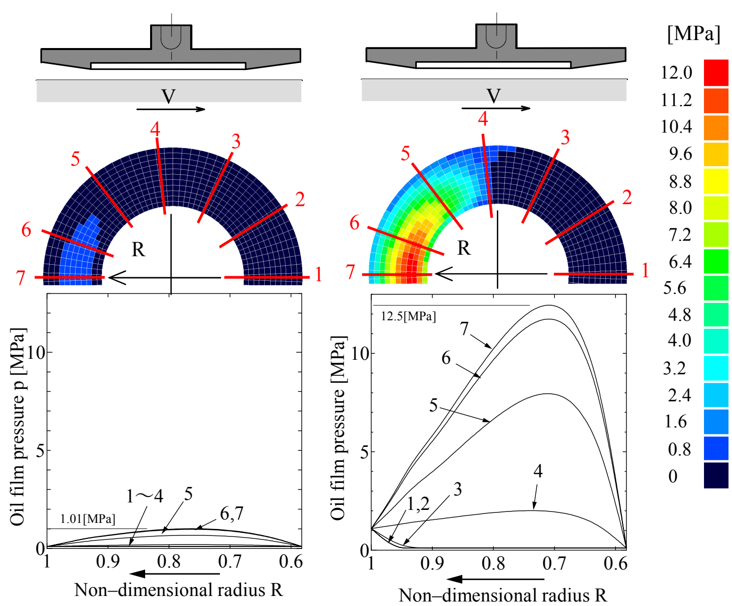
Figure 7: Calculated results.

Δp=1.0MPa takes a constant inclination even if the orbiting speed increases, since the large gas thrust force due to the pressure difference holds down the effect of the oil film hydrodynamic pressure.

4.2 Calculated Results

Major calculated results of lubrication at the thrust slide-bearing are shown in Figure 7, where the dotted line is at Δp=0MPa and the solid line is at Δp=1.0MPa. The attitude of the cylindrical thrust plate is shown in Figure 7a. The average height of floating on the oil, h_o, increases from 5.2 to 17.6μm at Δp=0MPa and from 1.8 to 6.6μm at Δp=1.0MPa, as the orbiting speed N increases from 300 to 3600rpm, while the attitude rotations were so small that it could be disregarded.

Accordingly, the oil film thick enough to achieve the excellent fluid lubrication is formed at the thrust slide-bearing, as shown in Figures 7b and 7c. As N increases, the solid contact force F_{sc} of a small value decreases more, whereas the oil film force F_{OIL} shows comparatively large value, rather increasing gradually. Therefore, most the thrust loads on the bearing is supported by the oil, even when Δp=1.0MPa. Furthermore, as N increases, the solid shearing force F_{ss} rapidly decreases, and hence most the resultant frictional force F_f depends upon the oil film viscous force F_{vs}. Therefore, even when Δp=1.0MPa, F_f never increases so much as in proportion to the nominal thrust load F_T. As a result, the friction coefficient μ at Δp=1.0MPa shows very small values less than 0.01, compared with those at Δp=0MPa, as shown in Figure 7d, where the theoretical results shown by the dotted and solid lines show a good agreement with the plotted data from lubrication tests. For reference the oil film pressure is shown in Figure 8, where its maximum value reaches 12.5MPa at Δp=1.0MPa and N=1800rpm (V=0.57m/s).



(a) Case A (Δp=0[MPa], F_T=800[N], $\tan\alpha = 0.77\times10^{-4}$)

(b) Case B (Δp=1.0[MPa], F_T=9200[N], $\tan\alpha = 2.0\times10^{-4}$)

Figure 8: Oil film pressure(without squeeze effect) at V=0.57[m/s] (N=1800[rpm]).

5. CONCLUSIONS

This study presented theoretical calculations for the lubrication at the thrust slide-bearing of scroll compressors, based on Average Reynolds Equation and Solid Contact Theory for random roughness surfaces. As a result, it is concluded that a wedge is formed at the sliding area, caused by elastic deformation of the orbiting scroll made of Aluminum Alloy, due to high pressure loads on it, and its wedge with small inclination of the order 10^{-4} can induce the oil film , thick enough to achieve the excellent fluid lubrication at the thrust slide-bearing.

In the present study, calculations were made for given wedge inclinations which essentially depend upon the structure of the orbiting scroll, in addition to the oil film pressure at the sliding area and the high pressure loads on the orbiting scroll. Further theoretical calculations taking all the factors into exact considerations are intensively desired, to seek for the fundamental best design of the thrust slide-bearing of the scroll compressors, for its optimal performance.

NOMENCLATURE

dA	local real contact area	(m^2)
F_f	resultant frictional force	(N)
F_{OIL}	oil film force	(N)
F_p	nominal gas thrust force	(N)
F_{pi}, F_{po}	gas 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	(m)
h_o	average clearance	(m)
h_{ref}	arvitrary standard film	(m)
H_r	oil film thickness to surface roughness ratio	(-)
$\bar{h}_T$	average oil film thickness	(m)
N	orbiting speed	(rpm)
p	oil film pressure	(Pa)
p_a	atmospheric pressure	(Pa)
p_c	plastic flow pressure	(Pa)
p_{out} , p_{in}	boundary pressure	(Pa)
r_o, r_i	bearing radius	(m)
U_1, W_1, V	boundary velocity	(m/s)
V_{r1}, V_{r2}	variance ratio	(-)
α	wedge angle	(rad)
α^*	local solid contact ratio	(-)
β	asperity summits radius	(m)
Δp	pressure difference	(Pa)
η	surface density of asperities	(m^{-2})
Θ	orbiting angle	(rad)
λ	bearing number	(-)
μ	frictional coefficient	(-)
μ^*	oil viscosity	($Pa \cdot s$)
σ, σ_1, σ_2	standard deviations of surface roughness	(μm)
σ_s	squeeze number	(-)
τ	shearing strength	(Pa)
ϕ	pressure flow factor	(-)
ϕ_s	shear flow factor	(-)
ϕ_f, ϕ_{fs}	shear stress factor	(-)
ψ_x , ψ_y,	rotation angle	(rad)
ω	angular orbiting velocity	(rad/s)

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