New Understanding of Precession-Rolling-induced Oil Film Pressure in Thrust-Slide Bearings in Scroll Compressors

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ABSTRACT

In scroll compressors, the gas compression pressure induces an overturning moment, resulting in a tilted orbiting scroll and the generation of a "wedge oil film force" in the thrust-slide bearing due to its orbiting motion. This study introduces a novel lubrication mechanism, due to the motion denoted as "precession rolling" of the thrust plate, which occurs in addition to the conventional wedge oil film pressure. The focus of this study is on the induction of extremely high oil film pressure due to precession-rolling. The oil film force due to precession-rolling is analyzed using a theoretical analysis of the oil film pressure between scroll wraps. Theoretical analysis reveals that precession-rolling-induced oil film pressure exceeds conventional wedge oil film forces due to sliding motion, indicating the importance of precession-rolling in the lubrication mechanism in scroll compressors.

1. INTRODUCTION

The lubrication mechanism of the thrust slide bearing in scroll compressors is closely related to the development of a fluid wedge between the sliding surfaces of the thrust slide bearing, as was previously considered by Ishii *et al.* (2008) and Oku *et al.* (2008), but was not considered in the previous study Kulkarni (1990). In the previous analysis, the thrust plate is assumed to undergo an elastic deformation due to the pressure difference between the inner and outer regions of the thrust bearing. In many cases, however, the thrust plate is highly rigid so as not to meaningfully elastically deform.

The gas compression pressure and the inertial force in scroll compressors induce an overturning moment that tilts the orbiting scroll, as shown in Figure 1, even if elastic deformation does not occur. As a result, the conventional "wedge oil film force" is generated in the thrust-slide bearing due to its orbiting motion. However, the tilt direction of the orbiting scroll changes depending on the orbiting angle, because the direction of the overturning moment rotates with the orbiting angle. Therefore, the tilted orbiting thrust plate will undergo both sliding and rolling motion against the fixed thrust plate. This rolling motion whirls similar to that of precession, but the thrust plate does not undergo self-rotation, so we denote this process as "precession-rolling" motion.

The precession-rolling-induced oil film pressure occurs in addition to the conventional wedge oil film pressure that effectively supports the large thrust load on the orbiting scroll. The present study focuses on the induction of a large oil film pressure due to precession-rolling.

The oil film pressure due to precession-rolling is analyzed using a theoretical analysis of the oil film pressure between scroll wraps, as previously derived by Anami et al. (2024). In addition, a theoretical analysis of tilted pad bearing was also applied to the analysis of wedge oil film pressure due to sliding, with new consideration of pressure

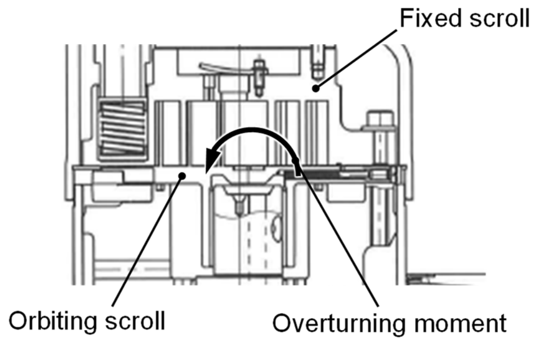
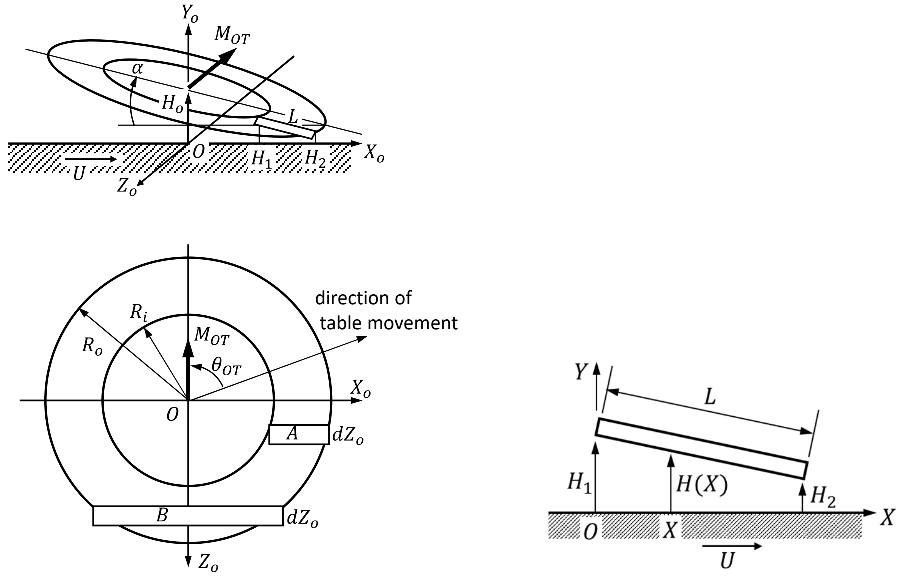


Figure 1: Schematic view of thrust slide bearing of scroll compressor



(a) tilted rigid cylindrical model (b) tilted pad cell model

Figure 2: Analysis model for oil film pressure due to sliding motion of the tilted rigid thrust plate.

differences that has not been previously considered. The theoretical analysis and calculation results of the oil film force due to precession-rolling and the wedge-induced oil film force are developed in the present paper.

2. WEDGE OIL FILM PRESSURE DUE TO SLIDING MOTION

If the thrust plate is flat and rigid, it floats with an appropriate gap balancing the axial load and oil film force. In addition, the tilt of the thrust plate is at an appropriate angle, balancing the moment due to the oil film force and overturning moment. In this study, the thrust plate of the orbiting scroll is replaced with a rigid cylindrical plate, as shown in Figure 2(a). It assumed that the tips of the scroll wraps cannot support the thrust force, thus the complicated shape of the thrust plate is replaced with a concentric ring. The cylindrical plate which represents the orbiting thrust plate is fixed at a tilt direction perpendicular to the overturning moment. Further, the flat plate representing the fixed thrust plate is held horizontal under the cylindrical plate. When the flat plate in Figure 2(b) moves horizontally with velocity U, the oil film pressure is generated by the oil viscosity.

For the analysis, the thrust plate is assumed to consists of a series of small pad cells, denoted as (A) and (B) in Figure 2(a). Focusing on a single small pad cell, with a horizontal flat plate moving under the fixed tilted pad cell, as shown in Figure 2(b). Sufficient oil exists between the tilted pad and flat plate. The analysis method to determine the oil film pressure produced in such cases is known from the analysis of Hori (2002), among others. In the case of the thrust plate of a scroll compressor, a pressure difference exists between the right- and left-hand side of the pad cell. Thus, the oil film pressure was analyzed in the case of this pressure difference.

The Reynolds' equation based on the Navier-Stokes equation is given as follows:

$$\frac{d}{dX}\left(H^3\frac{dP}{dX}\right) = 6\eta U\frac{dH}{dX} \tag{1}$$

The right-hand side represents the wedge effect where oil film pressure is generated as oil is being drawn in. Here, the stretch and squeeze effects are ignored. U denotes the plate velocity in X direction and H is the clearance gap at position X, as shown in Figure 2b.

Solving Equation (1), the oil film thickness H_m resulting from the maximum generated oil film pressure and the oil film pressure distribution P are obtained. Then, the integral constant can be determined by the boundary conditions of $P(0; H_1) = P_1$ and $P(L; H_2) = P_2$.

The nondimensional oil film thickness h_m generated by the maximum oil film pressure, and the nondimensional oil film pressure distribution p are obtained as follows:

$$h_m = \left(p_1 - p_2 + \frac{1}{m}\right)\frac{2m^2}{m+1} \tag{2}$$

$$p - p_1 = \frac{x}{hm}\left\{1 - \frac{h_m(m+h)}{2hm}\right\} \tag{3}$$

in which, the following nondimensionalizations are introduced:

$$x = \frac{X}{L}\ ;\ h = \frac{H}{H_2}\ ;\ \ m = \frac{H_1}{H_2}\ ;\ \ h_m = \frac{H_m}{H_2};\ \ p = \frac{P}{\frac{6\eta UL}{H_2^2}} \tag{4}$$

m represents the gap ratio. $p_1 and\ p_2$ represent the nondimensional pressure of P_1 and P_2, respectively.

3. OIL FILM PRESSURE DUE TO PRECESSION-ROLLING MOTION

The clearance gap thickness varies from that at the outer circumference of the tilted orbiting thrust plate to that of the fixed plate. When the circumferential gap is expressed as a function of mean radius of curvature, R, the oil film pressure due to the precession-rolling can be obtained using the analysis of the oil film pressure between scroll wraps from Anami *et al.* (2024).

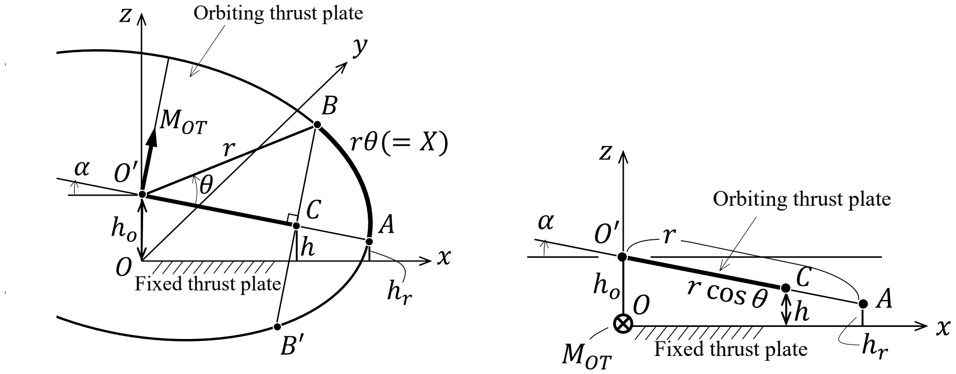
It is assumed that the overturning moment M_{OT} acts on the orbiting thrust plate, and the thrust plate is inclined with the tilt angle α, as shown in Figure 3. The clearance gap at the center of the thrust plate O' is shown in h_o. The minimum gap appears at position "A", which is shown as h_r. The gap at position "B" with the deviation angle of θ can be given by h at position "C". Because the direction of the overturning moment M_{OT} rotates with the rotation of crank shaft, the minimum gap position "A" rolls along the outer circumference of the tilted plate. The rolling velocity V_r is given as:

$$V_r = r\dot{\theta} \tag{5}$$

where, r and $\dot{\theta}$ represent the radius of the rolling position and the orbiting speed, respectively. The oil film pressure induced by rolling depends on the circumferential distribution of the gap. The gap shape along circumferential length X (= $r\theta$) near the position "A" can be expressed as follows:

$$h - h_r = \frac{\sin\alpha}{2r}X^2 \quad (X \ll 1) \tag{6}$$

This expression suggests that the gap shape can be represents by a cylinder with the radius of



(a) schematic view of tilted plate (b) side view of tilted plate

Figure 3: Tilted orbiting thrust plate and clearance gap to the fixed thrust plate.

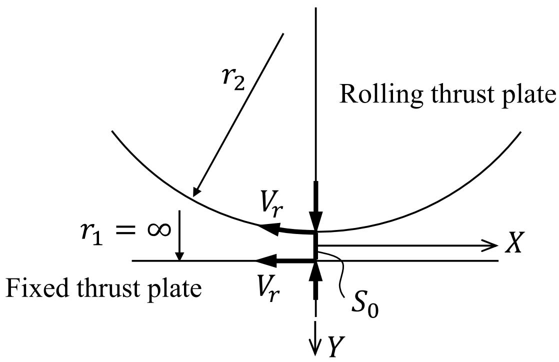


Figure 4: Analytical model of precession-rolling induced oil film pressure

$$r_2 = \frac{r}{\sin\alpha}. \tag{7}$$

Figure 4 shows the analytical model, in which the cylindrical thrust plate with the radius of r_2 undergoes precession-rolling on the flat fixed thrust plate with the radius of $r_1 = \infty$. The axial gap appears between the two thrust plates. The minimum gap is represented by S_0. The mean radius of curvature of two thrust plates, R, can be calculated as follows:

$$\frac{1}{R} = \frac{1}{2}\left(\frac{\sin\alpha}{r} - \frac{1}{\infty}\right), \quad \therefore\ R = \frac{2r}{\sin\alpha} \tag{8}$$

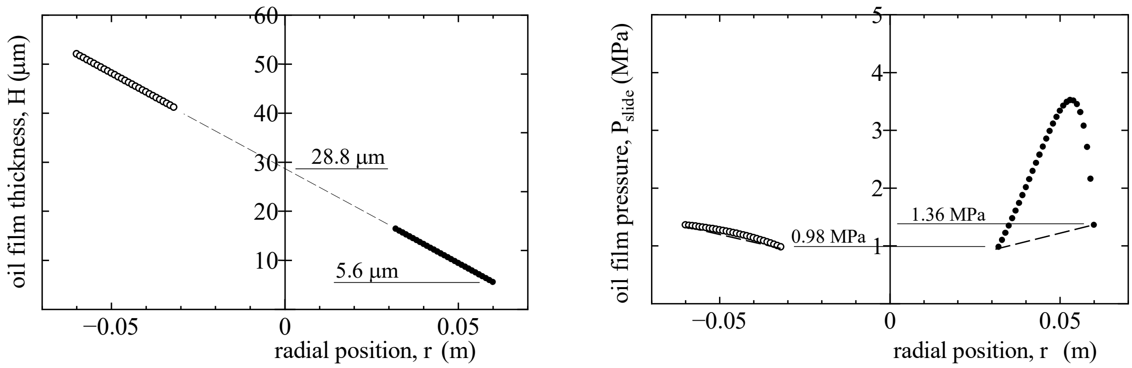
When the tilt angle α takes on a very small value, the mean radius of curvature R approaches infinity.

In Figure 4, both fixed and rolling thrust plates move in the negative X direction with same velocity V_r. Therefore, the rolling motion with the velocity of V_r can be expressed as the stationary state in the $X - Y$ coordinate. As a result, the oil film pressure, generated by this rolling motion, can be obtained by applying the previously developed analysis of oil film pressure between scroll wraps by Anami *et al.* (2021). As a result, the generated maximum oil film pressure can be given by following equation, in which η represents the oil viscosity:

$$P_{max} = 0.7605\eta \times 2V_r\sqrt{\frac{R}{S_0^3}} \tag{9}$$

Table 1: Specifications for calculation

Parameter	Value
inner radius of thrust plate, R_i	32 mm
outer radius of thrust plate, R_o	60 mm
orbiting radius, r_o	3.5 mm
suction pressure, P_s	0.98 MPa
discharge pressure, P_o	2.45 MPa
intermediate pressure, P_m	1.36 MPa
thrust force, F_0	15,500 N
overturning moment, $M_{OT}\big\|_{\theta=0}$	75 Nm
rotation speed, N	3600 rpm
oil viscosity, η	0.008 Pa s



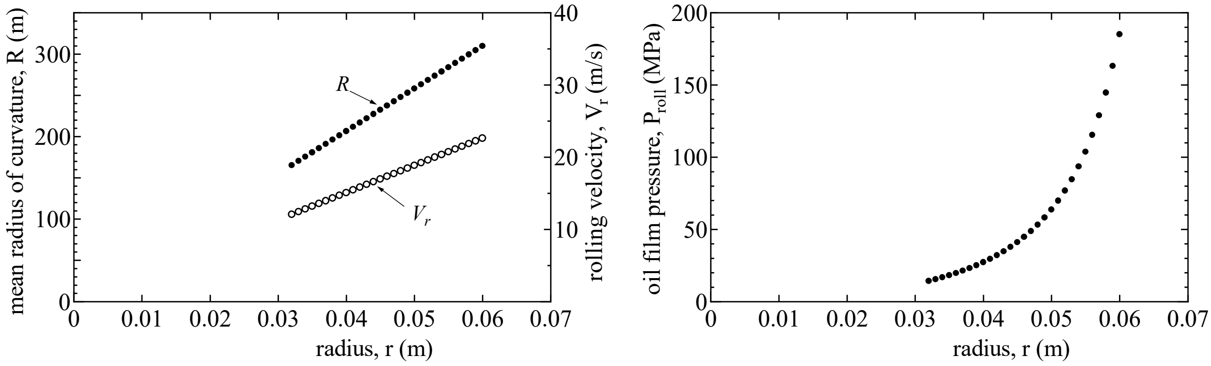
(a) floating height due to wedge oil film pressure (b) wedge oil film pressure distribution due to sliding

Figure 5: Calculating result of wedge oil film pressure due to sliding motion.

4. EXAMPLES OF CALCULATION

The oil film pressure due to sliding motion and precession-rolling motion were calculated. The major specifications of calculation are shown in Table 1. The scroll compressor with a suction pressure of 0.98 MPa and a discharge pressure of 2.45 MPa is assumed. The 15,500 N of thrust force including gas force act on the orbiting scroll and 1.36 MPa of intermediate pressure act on the back side of orbiting scroll. The overturning moment at orbiting angle $\theta = 0°$ takes a value of 75 Nm. The thrust slide bearing of orbiting scroll is replaced by a concentric ring with the inner radius of 32 mm and the outer radius of 60 mm. The viscosity of oil, η, is set to 0.008 Pa s.

The attitude of the thrust plate consists of the floating height and the tilt angle. The floating height and the tilt angle due to the wedge oil film pressure of sliding motion of the cylindrical plate were calculated, using the analysis of the wedge oil film pressure. The appropriate floating height was identified such that the oil film force obtained by integrating the oil film pressure becomes comparable to the thrust force, by changing floating height. Similarly, the attitude of the thrust plate was identified such that the moment of oil film force becomes comparable to the overturning moment by changing tilt angle. The obtained attitude of thrust plate is shown in Figure 5(a). The rigid cylindrical plate floats at 28.8 μm at the center of the ring, and tilt angle is 22.2 mdeg. The oil film pressure generated at this attitude of the cylindrical plate is shown in Figure 5(b), which shows the oil film pressure distribution at $Z_0 = 0$ induced when the flat plate moves to the right under the cylindrical plate which has the attitude as in Figure 5(a). A significantly larger oil film pressure is generated on the front of the cylindrical plate, whereas only a small oil film pressure is



(a) rolling velocity and mean radius of curvature (b) oil film pressure due to precession rolling

Figure 6: Calculating result of oil film pressure due to precession rolling.

generated on the back, compared with the static oil film pressure shown by the dashed line. The maximum oil film pressure appears about 20% inside of the outer circumference and takes a value of about 3.5 MPa.

Thus, the attitude shown as Figure 5(a) represents the minimum gap at the rolling radius. The minimum gap at outer circumference takes a value of 5.6 μm. As shown in Figure 6(a), the rolling velocity V_r for each radius varied from 12.0 to 22.6 m/s, and the mean radius of curvature R of the gap for each radius take a significantly large value of 165 to 310 m. The calculated oil film pressure induced by the precession rolling is shown in Figure 6(b). The maximum pressure caused by the precession rolling at the outer circumference reaches an extremely large value of 185 MPa, far exceeding what was anticipated. The oil film force is about 136 kN, which is also far larger than the thrust load. Of course, practical pressure at the outer circumference will decrease due to spread of flow field. Details of this pressure decrease will be analyze in the next step.

Here, it is assumed that the maximum pressure decreases to 1/2 for all radius, as a crude treatment. The floating height will be 93.4 μm, in which this reduced oil film force is balanced by the thrust force. If the tilt angle does not change at 22.2 mdeg, the minimum gap at the outer circumference takes a comparatively large value of 70.2 μm. In this case, the wedge oil film pressure will be almost 0, which means no wedge pressure will be generated.

5. CONCLUSIONS

It is well known that the wedge oil film pressure occurs due to sliding motion of thrust plate at the thrust slide bearing of the scroll compressors. This study shows a new understanding of oil film pressure due to the precession-rolling in addition to the wedge oil film pressure due to sliding motion of thrust plate. The theoretical analysis shows that the oil film pressure due to the precession-rolling is far larger than by the sliding motion. It clearly suggests that the precession-rolling-induced pressure is an extremely important factor in the lubrication of the thrust-slide bearing in scroll compressors. Important findings based on new ideas of the motion of the thrust slide bearing in scroll compressors have been obtained.

In the scroll compressor, the orbiting scroll will be floating, supported by the high oil film force. If the wrap tip of orbiting scroll comes into contact with the fixed scroll, a contact force will occur. The actual attitude of the thrust plate can be accurately obtained by including the tip contact force in addition to the compression gas force in the calculation.

NOMENCLATURE

H, H_1, H_2, H_m	clearance gap (oil film thickness)	(m)
h, h_m	reduced oil film thickness	(–)
kx, kx_m	dimensionless position	(–)
L	pad length	(m)

m	gap ratio	(–)
P, P_1, P_2	pressure	(Pa)
P_{slide}	sliding oil film pressure	(Pa)
P_{roll}	precession rolling oil film pressure	(Pa)
p, p_1, p_2	nondimensional pressure	(–)
R	mean radius of curvature	(m)
r	radius of rolling	(m)
r_1, r_2	radius of curvature	(m)
S_0	minimum oil film thickness	(m)
U	moving velocity of plate	(m/s)
V_r	rolling velocity	(m/s)
X, Y, Z	coordinate	(–)
x	nondimensional position	(–)
α	tilt angle	(deg)
η	oil viscosity	(Pa s)
θ	orbiting angle	(deg)

REFERENCES

Anami, K. et al. (2024), New Understanding of Higher Oil Film Pressure between Scroll Wraps of Scroll Compressors Due to Rolling and Sliding, Proc. of the International Compressor Engineering Conference at Purdue, West Lafayette, IN, ID:1167 (to be presented).

Ishii, N., Oku, T., Anami, K., Knisely, C.W., Sawai, K., Morimoto, T., Iida, N. (2008), Experimental Study of the Lubrication Mechanism for Thrust Slide Bearings in Scroll Compressors, *Int. J. of ASHRAE*, 14(3), 453-465.

Kulkarni, S.S. (1990), Thrust Bearing Design with Rigid Body Dynamics of the Runner Plate, Proc. of the International Compressor Engineering Conference at Purdue, West Lafayette, IN, 333-344.

Oku, T., Ishii, N., Anami, K., Knisely, C.W., Sawai, K., Morimoto, T., Hiwata, A. (2008), Theoretical Model of Lubrication Mechanism in the Thrust Slide-Bearing of Scroll Compressors, *Int. J. of ASHRAE*, 14(2), 239-358.

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