Verification of High-Performance Lubrication Due to Precession-Rolling in a Fully Loaded Thrust-Slide Bearing in a Large Capacity Scroll Compressor

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ABSTRACT

The large capacity scroll compressor, the subject of this research, adopts an axial compliance structure in which the housing is floated by gas compression pressure to push the orbiting scroll against the fixed scroll, where the small size and high efficiency are achieved. The thrust slide bearing is formed by the housing and the orbiting scroll. This thrust slide bearing has been operated normally under high load conditions under R 32 refrigerant. The lubrication mechanism in the thrust slide bearing was verified, focusing on the oil film pressure due to precession-rolling motion, which has been derived by Anami et al. (2024a).

FEM simulations were conducted on the compression mechanism components that form the thrust slide bearing of the large capacity scroll compressor, under actual contact and gas compression conditions, thus addressing the 3D elastic deformation and orientation of the thrust slide bearing relative to the top surface of the housing. The oil film pressure was then calculated, where the lubrication mechanism due to the precession-rolling motion of the thrust slide bearing was applied. The calculation results were compared with the numerical ones of the conventional wedge oil film pressure due to sliding motion. As a result, it was shown that the lift force due to the precession-rolling motion is far larger than that due to the wedge sliding motion.

1. INTRODUCTION

The actual evaluation tests of a large-capacity scroll compressor have confirmed that the resistance to seizure can be improved by providing a small sloping part on the thrust slide bearing. However, this improvement cannot be addressed by the conventional lubrication theory of the thrust slide bearing, where the oil film force induced by the wedge sliding motion was calculated to find the oil film thickness and any significant difference by adding a slope to the thrust slide bearing was not confirmed. Recently, Anami et al. (2024a) has presented a new understanding of precession-rolling-induced lubrication mechanism at thrust slide bearings, where the theory of oil film pressure generation between scroll wraps by Anami et al. (2024b) is applied.

In this paper, for the thrust slide bearing of the large capacity scroll compressor, the detailed and exact FEM simulations were conducted under realistic contact conditions and gas compression conditions, and the gap between the top surface of the housing and the thrust slide bearing was calculated by considering the 3D elastic deformation, in addition to the attitude of the thrust slide bearing. Then, the oil film force was calculated from the lubrication mechanism by Anami et al. (2024a), induced by both the sliding and precession-rolling motions of the thrust slide bearing, thus being compared with the evaluation results obtained in the actual equipment operation. Moreover, the calculated results were compared with the conventional numerical ones of wedge-shaped induced oil film force, thus examining the difference.

2. LARGE CAPACITY SCROLL COMPRESSOR

It has succeeded in developing a 25 HP constant-speed large capacity scroll compressor with a low-pressure dome structure, responding to R 32 refrigerant environments and achieving small size and high efficiency (Japan Patent JP 6274281 B). Figure 1 shows the schematic view of the developed large-capacity scroll compressor. The features of this compressor are as follows: the high pressure and intermediate pressure from the fixed scroll are introduced into the back of the housing, giving back pressure to the housing. The housing presses the orbiting scroll against the fixed scroll while preventing it from rotating by means of four guide bushes. The housing has a main bearing to support the overturning moment caused by the gas load acting on the orbiting scroll, with the moment caused by the main bearing load acting on the housing. This structure allows the orbiting scroll to float without overturning due to gas compression even at small back pressure. This structure can be pressed against the fixed scroll with a minimum back pressure that exceeds the axial gas load. The structure is designed to press against the fixed scroll with the minimum back pressure. The back pressure chamber can be made smaller to reduce the size of the compression chamber and to reduce the leakage from the compression chamber. Therefore, the scroll compressor is small in size by making the back pressure chamber smaller, achieving the high efficiency by eliminating compression chamber leakage and reducing the sliding losses.

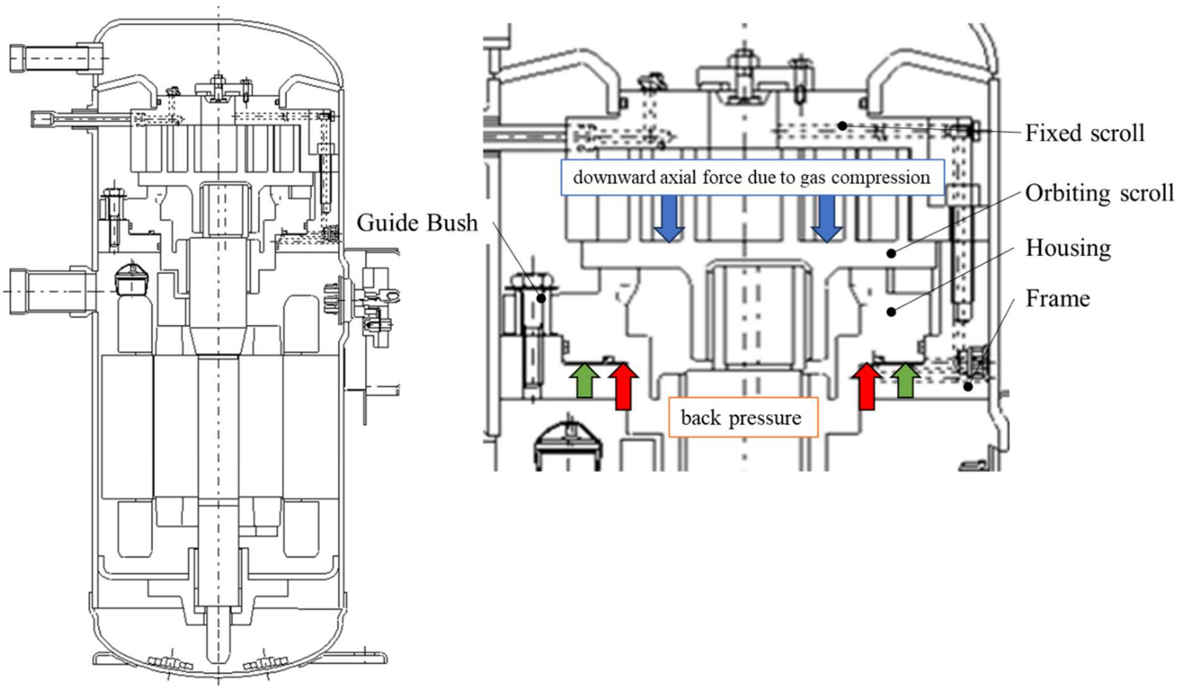


Figure 1: Schematic view of developed large-capacity scroll compressor

For the scroll compressor with this structure, the actual machine operation tests confirmed that the seizure resistance of the thrust slide bearing reduces when the thrust slide bearing is not provided with tapered part, while improves when the thrust slide bearing is provided with tapered part, as shown in Figure 2. The lubrication mechanism of the thrust slide bearing in this structure is verified in the present study.

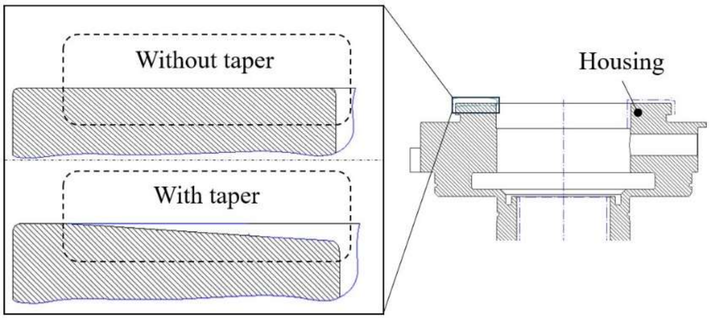


Figure 2: Thrust slide bearing without and with taper of large capacity scroll compressor

3. CALCULATION SCHEME OF OIL FILM FORCES ON THRUST SLIDE BEARING

The large oil film pressure, generated by the sliding and precession-rolling motions at the thrust slide bearing, has been addressed to calculate by Anami et al. (2024a). The floating height and attitudes of the thrust slide bearing can be determined so that the generated oil film forces balance the thrust forces. In this study, the calculations were conducted for the developed 25 HP compressor.

3.1 Wedge Oil Film Force due to Sliding Motion

The thrust plate was treated as consisted of a series of microsection pad, which have pressure difference in right- and left-hand side of the microsection pad.

The gap ratio in each microsection, m, is defined by the following equation.

$$m = \frac{H_1}{H_2}, \tag{1}$$

where H_1 and H_2 represent the gap height of both ends of the microsection, which can be addressed from FEM simulations. With the obtained gap ratio m, the dimensionless oil film force f_{oil} can be calculated by the following expression.

$$f_{oil} = p_1 + \frac{1}{(m-1)^2}\left[\ln m - \frac{m-1}{m}\left\{1 + \frac{h_m}{2m}(m-1)\right\}\right], \tag{2}$$

where h_m represents the dimensionless gap defined by $h_m = (p_1 - p_2 + 1/m) \times 2m^2/(m+1)$. The dimensional oil film force due to the wedge effect in each microsegment can be calculated by

$$dF_{oil} = \frac{6\eta U L^2}{H_2^2} f_{oil} dZ_0, \tag{3}$$

where dZ_o represents the width of each microsegment. The total wedge oil film force due to sliding motion on the thrust slide bearing can be calculated by integrating these values over the entire area of the thrust slide bearing.

3.2 Oil Film Force due to Precession-Rolling Motion

The gap height H and the mean radius curvature R in the circumferential direction can be calculated from FEM simulations for each of the microsections in the radial direction on the thrust slide bearing.

The minimum gap S_0 between the thrust surfaces of the housing and the orbiting scroll can be calculated from the difference in both axial deformations by FEM simulations. The mean radius curvature R can be calculated by the following expression.

$$R = \frac{2r}{\sin\alpha}, \tag{4}$$

where α represents the tilting angle of the orbiting scroll at the radial position r from the center of the thrust slide bearing. With the obtained minimum gap S_0 and the mean radius curvature R, the maximum oil film pressure in each microsection, P_{max}, can be calculated by the following expression, as derived by Anami et al. (2024a, 2024b).

$$P_{max} = 0.7605 \times \eta \times 2V_r\sqrt{\frac{R}{S_0^3}}, \tag{5}$$

where η represents the oil viscosity. The oil film force at each microsegment can be calculated by the following expression with the width of the microsegment $\mathrm{d}r$.

$$dF_{oil} = 1.224 \times \eta \times 2V_r\frac{R}{S_0}\mathrm{dr} \tag{6}$$

3.3 Calculation of Floating Height and Tilting Angle of Thrust Slide Bearing

The calculation flow chart is shown in Figure 3, where the initial gap on the thrust surface and the mean radius curvature are given from FEM simulations for the large scroll compressor. As the first step, the calculation flow assumes an appropriate bearing attitude (mean floating height and tilting angle). Based on the clearance and mean radius curvature of the thrust slide bearing, obtained from the FEM simulation, the oil film forces due to the wedge sliding and precession-rolling motions are calculated. First, the distance to the reaction force generated by the contact of the tip of the wrap of the orbiting scroll with the bottom surface of the fixed scroll is calculated based on the balance of moments in the orbiting scroll for the obtained oil film force and the resulting oil film force moment. Convergence calculations were performed by changing the amount of lift and tilt of the orbiting scroll so that the combined force of this reaction force and the axial gas load at each angle of rotation balance the oil film force.

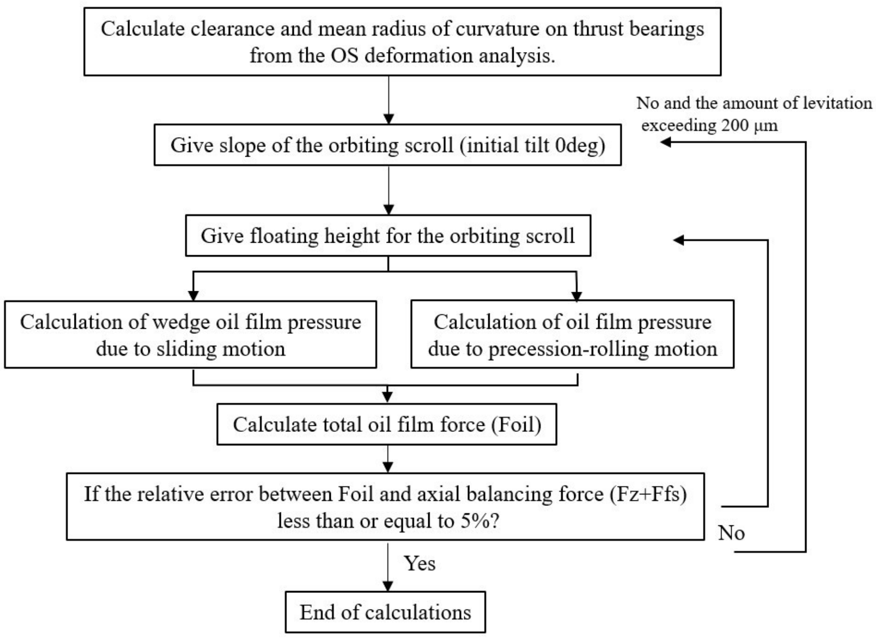


Figure 3: Calculation flow of oil film force by sliding and precession-rolling motions of thrust slide bearing

4. FEM SIMULATIONS

In order to identify the geometrical radius and gap in the thrust slide bearing, the static deformations were calculated by FEM simulations. The analytical models are consisted of the fixed scroll, the orbiting scroll, the housing, and the crankshaft (pin section only), as shown in Figure 4. The simulations were conducted under R 32 refrigerant conditions for the operating conditions of the compressor and the high differential pressure conditions, where the load on the thrust slide bearing is the greatest under the operating conditions. The pressure in the compression chamber was calculated from the area ratio relative to the closed-off suction chamber, with the adiabatic index by internal pressure measurements for the compressor. It was assumed that the tooth tips of the orbiting scroll and the tooth bottom of the fixed scroll contact each other, and the average value of the adjacent compression chamber pressure was given to the tips of the fixed scroll. Back pressure, consisted of high pressure and intermediate pressure, was loaded on the back of the housing, and the centrifugal force was on the orbiting scroll.

The fixed scroll was fully constrained at the fastening surface to the frame section, and the upper convex side was constrained horizontally. For the housing, the portion positioned by the guide bush was constrained horizontally. The orbiting scroll was constrained to prevent from spinning. For the crankshaft (pin section only), the lower end face of the model was fully constrained for simplicity.

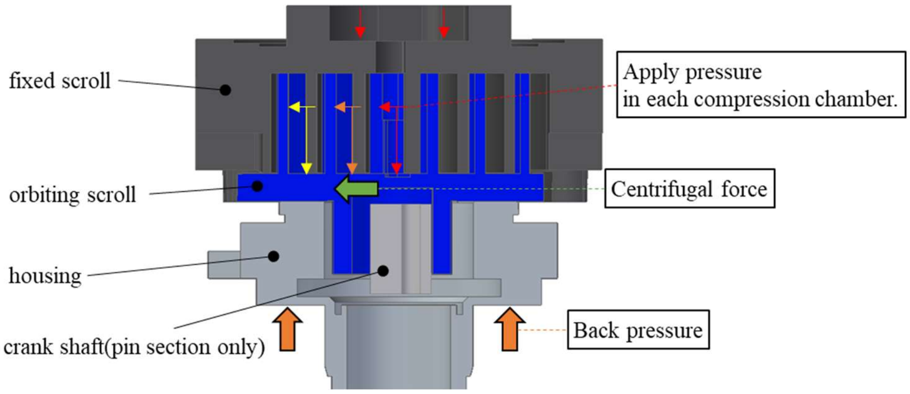
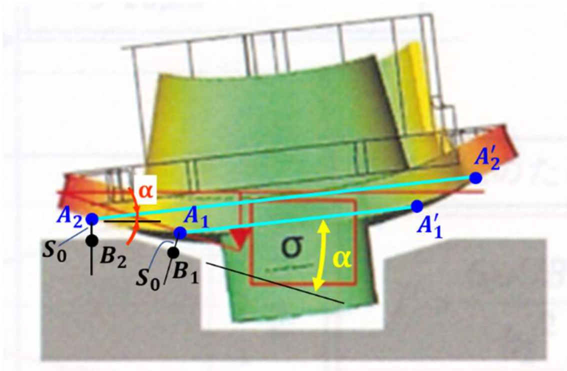


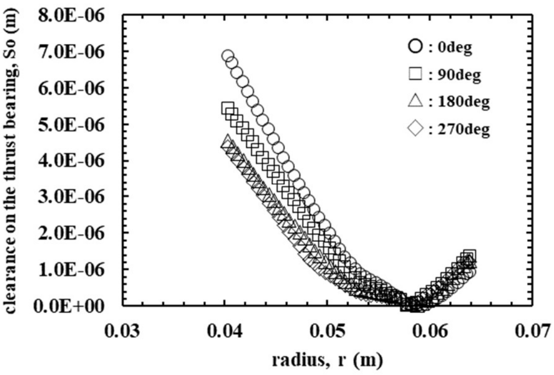
Figure 4: Loading conditions in FEM simulations

Figure 5(a) shows the image to determine the mean radius of curvature R and the clearance gap S_0 at each position of the thrust slide bearing. The radius r at each position was determined by $\overline{AA'}/2$. The clearance gap S_0 at the position was determined by the perpendicular line from A to the housing. Then the tilting angle α was determined at the inclination of $\overline{AA'}$ against the inclination of the position B. From the FEM simulation results, the clearance gap S_0 and tilting angle α are determined for each radius of the orbiting thrust plate. The mean radius of curvature R was calculated by Equation (4).

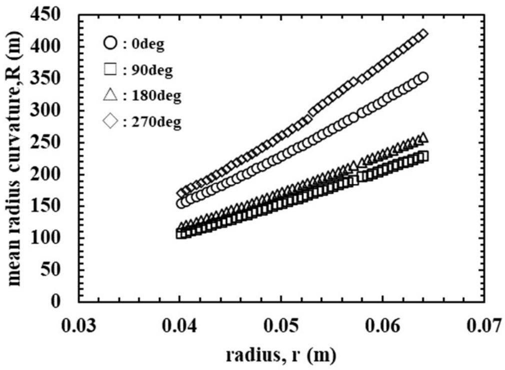
The determined clearance gap S_0 and the mean radius curvature R are shown in Figure 5(b) and Figure 5(c), respectively. The parameter is the orbiting angle for FEM simulation. The compression start angle of the intake chamber is set to 0 deg. The clearance gap takes almost 0 at the radius of 0.058 m from the center of the thrust plate. The mean radius of curvature R takes large value, it reaches a large value of 350 m at the outer circumferential at θ = 0 degree.



(a) Schematic image to determine the mean radius of curvature R and the clearance gap.



(b) Clearance on the thrust slide bearing, S_0



(c) Mean radius of curvature, R

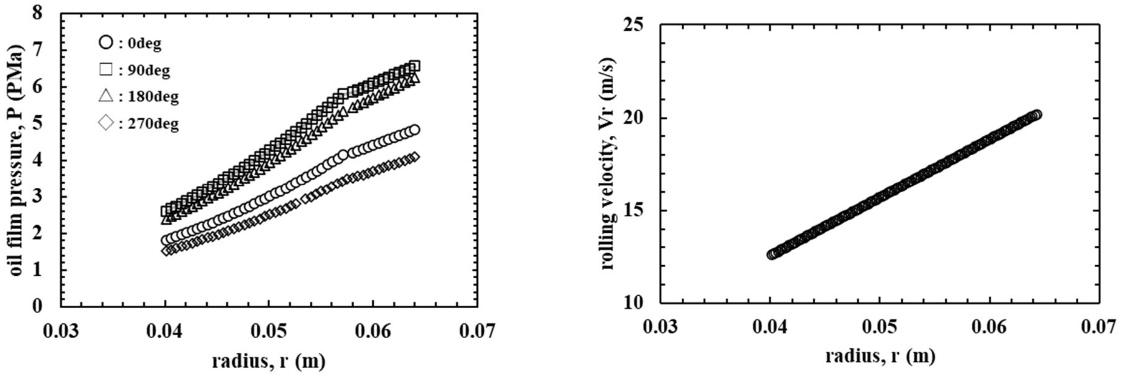
Figure 5: FEM simulation results for S_0 and R at orbiting angle θ = 0, 90, 180, 270 degrees

5. CALCULATED RESULTS

The oil film pressures on the thrust slide bearing of a large-capacity scroll compressor under high-load conditions with R 32 refrigerant was calculated. First, the floating height (oil film thickness) due to the wedge effect of the sliding motion was calculated, where the gaps at both ends on microsection H_1 and H_2 of Expression (1) were obtained by FEM simulation. The oil film thickness where the conventional wedge oil film force balances to the external thrust force is shown in the left column of Table 1. The oil film thickness is very small value of 3.2 μm even if the housing has a taper, showing less difference from without the taper. One may conclude that the thrust slide bearing cannot be sufficiently floated only by the wedge oil film force due to sliding motion.

Table 1: Floating height (oil film thickness) due to wedge sliding and precession-rolling effects.

	averaged floating height, H_0	
	wedge effect	wedge and precession-rolling effects
with taper	3.2 μm	45.1 μm
without taper	2.0 μm	36.4 μm



(a) oil film pressure (b) rolling velocity V_r

Figure 6: Calculated oil film pressure due to precession rolling

Therefore, the floating height induced by both the sliding and precession-rolling motions of thrust plate was calculated by the calculation flow chart shown in Figure 3. The oil film pressure due to the precession-rolling can be calculated by the expression (5), as shown in Figure 6(a), exhibiting sufficiently large value. This is because the rolling velocity V_r, shown in Figure 6(b) takes far larger than the sliding velocity $V_s = r_o\dot{\theta}$ (= 2.14 m/s). Integrating the oil film pressure over the whole thrust slide bearing surface yields the oil film force due to precession rolling.

The floating height H_0 of the oil film thickness and tilting angle were calculated, with considering the sliding and precession-rolling motions. The floating height was determined so that the resultant oil film force balances the thrust force of 18 kN at the rotating speed of 50 s^{-1}. The calculated results of the floating height H_0 are shown in the right column of Table 1.

The resultant floating height, induced by both the sliding and the precession-rolling motions, takes 36.4 μm to 45 μm, which are sufficient for good lubrication of the thrust slide bearing. One may conclude that the treatment of precession-rolling-motion-induced oil film force is significant for higher performance lubrication of the thrust slide bearing.

The oil film force due to the precession-rolling was about 85% of resultant oil film force at the thrust slide bearing with taper. This result indicates that the oil film forces due to the precession-rolling motion is dominant. In addition, the floating height of the oil film thickness for the tapered thrust plate is larger than for the flat thrust plate, indicating that the tapering of the housing bearing surface is effective to generate the larger oil film force. These floating amounts were also consistent with the tendency for seizure in actual thrust slide bearing.

6. CONCLUSION

The high-performance lubrication at the thrust slide bearing of a large-capacity scroll compressor under high load conditions of R 32 refrigerant was well verified, where the oil film forces due to both the sliding and precession-rolling motions were calculated for the elastically deformed orbiting scroll, simulated from the exact FEM analyses. The calculated results confirmed that the thrust slide bearing can be sufficiently floated by the oil film forces caused by both the sliding and precession-rolling motions, though it cannot be floated by only the conventional wedge oil film force. The calculated oil film thickness and tilting angle were appropriate to understand the tendency of higher lubrication of the actual thrust slide bearing.

The present study has made possible to design the larger-capacity scroll compressors which achieves smaller size and higher efficiency while ensuring high reliability of lubrication of the thrust slide bearing. The compressor capacity has been expanded from the conventional 12HP to 25HP, where the mass of the compressor was reduced to about 74% and the size to about 85%, compared with the 2 sets of 12HP compressor.

NOMENCLATURE

H, H_1, H_2	clearance gap (oil film thickness)	(m)
h, h_m	reduced oil film thickness	(-)
L	pad length	(m)
m	gap ratio	(-)
P, P_1, P_2	pressure	(Pa)
P_{max}	maximum oil film pressure	(Pa)
f_{oil}	reduced oil film force	(-)
F_{oil}	oil film force	(N)
p, p_1, p_2	reduced pressure	(-)
R	mean radius of curvature	(m)
r	radius of rolling	(m)
S_0	minimum oil film thickness	(m)
U	orbiting velocity	(m/s)
V_r	rolling velocity	(m/s)
V_s	sliding velocity	(m/s)
Z_0	coordinate	(-)
α	tilt angle	(deg)
η	oil viscosity	(Pa s)
θ	orbiting angle	(deg)

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ACKNOWLEDGEMENT

The authors would like to express their sincere gratitude to Daikin Industries for their financial support and nurturing a research environment. The authors are grateful to the staffs in Compressor Development & Engineering group for their cooperation in measuring the resistance to seizure of the actual compressor.