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Keiko Anami
anami@osakac.ac.jp

Ryosuke Okamoto
mm19a005@oecu.jp

Masaru Tanaka
masaru.tanaka@daikin.co.jp

Kenichi Sata
kenichi.sata@daikin.co.jp

Hideki Matsuura
hideki.matsuura@daikin.co.jp

See next page for additional authors

New Understanding of Higher Oil Film Pressure between Scroll Wraps of Scroll Compressors Due to Rolling and Sliding

Keiko ANAMI[1]*, Ryosuke OKAMOTO[1],
Masaru TANAKA[2], Kenichi SATA[2], Hideki MATSUURA[2],
Yukiko MAEJIMA[2], Kosuke NISHIMURA[2], Kanetaka MIYAZAWA[2]

[1] Osaka Electro-Communication University, Department of Mechanical Engineering
Neyagawa, Osaka, Japan
anami@osakac.ac.jp

[2] Daikin Industries, Ltd., Technology and Innovation Center
Settsu, Osaka, Japan

* Corresponding author

ABSTRACT

The generation of relatively high oil film pressure in the small radial gap between two scroll wraps in scroll compressors has long been hypothesized. However, neither the precise physical mechanism of this high oil film pressure generation nor the significant characteristics of the oil film pressure has been clearly identified. This study investigates the generation of high oil film pressure between two scroll wraps in compressors, caused by inscribed rolling and sliding motions. Theoretical analysis, considering oil viscosity effects, yields dimensionless expressions for oil film pressure and resultant force. Experimental validation using a simplified equivalent model confirms theoretical findings. This research sheds light on the mechanisms behind oil film pressure generation and its implications for compressor performance.

1. INTRODUCTION

In a scroll compressor, oil injected into the refrigerant gas is drawn into the narrow space between the scroll wraps. Therefore, a relatively high oil film pressure is thought to be generated in this area. However, the characteristics and generation mechanism of the oil film pressure remain unclear owing to several factors. Fukuta et al. (2014) and Hiwata et al. (2010) estimated the oil film force between the wraps based on the load measurement of the main bearing. The indirect measurement method makes it difficult to accurately obtain the oil film pressure between the wraps, and the mechanism to generate oil film force could not be addressed.

Scroll contact points may be designed with a small gap. In addition, this small gap can be filled with sufficient oil via the injected oil into the suction gas for oil sealing. In such a case, a large oil film pressure can be easily generated by the viscous effect of the oil alone when a small orbiting scroll rolls and slides against a larger diameter fixed scroll. However, neither the precise physical mechanism of this high oil-film pressure generation nor the significant characteristics of the oil-film pressure has been clearly identified.

This study presents a theoretical analysis for the oil film pressure generated when scrolls inscribe rolling and sliding. In the analysis, the oil-film pressure and resultant oil film force, closely related to oil viscosity, are derived in simplified form of the dimensionless expressions. The validity of the analysis was verified by model experiments. Finally, the oil film pressure expected to occur near the scroll contact was calculated for a medium-capacity scroll compressor.

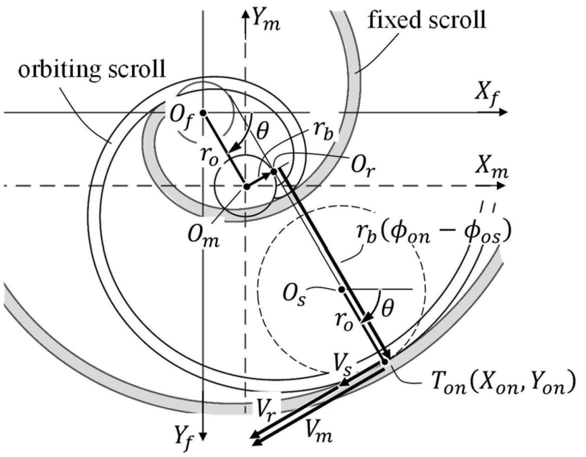

Figure 1: Rolling velocity V_r and sliding velocity V_s at the contact point between scroll wraps

2. ROLLING AND SLIDING AT SCROLL CONTACTS

Figure 1 shows the enlarged view of engagements of scroll wraps. An orbiting scroll moves clockwise relative to fixed scroll. The contact points on both scrolls appear on the common tangent of the base circle of both scrolls. The base circle radius is denoted by r_b, the orbiting radius is represented by r_o, and the clockwise orbiting angle is represented by θ.

The trajectory circle of the contact point T_{on} (n =0, 1, 2, here n =1) is illustrated by dashed line. Then, the circumferential velocity of the point T_{on}, represented as in the short arrow on the right as shown in Figure 1, indicates the sliding speed of orbiting scroll V_s. The radius is r_o of course.

Meanwhile, the movement speed V_m of the contact point T_{on}, is depicted by the long arrow. The movement speed V_m consistently surpasses the sliding speed V_s, with the difference being the rolling speed V_r, indicated by the relatively long arrow on the left.

Therefore, it can be understood that the contact point T_{on} inscribes to the fixed scroll with the sliding speed V_s and the rolling speed V_r. Since the rolling radius $r_b(\phi_{on} - \phi_{os})$ from O_r to T_{on}exceeds the orbiting radius r_o, the rolling speed V_r is considerably larger than the sliding speed r_o, in the outer circumference of the scroll.

3. THEORETICAL ANALYSIS OF INSCRIBED ROLLING AND SLIDING OIL FILM PRESSURE BETWEEN SCROLL WRAPS

German Walter Büche (1934a, 1934b) first reported a theoretical analysis of the rolling oil film pressure generated between two circumscribed cylinders. The theoretical analysis method can also be applied to examine the oil film pressure when scroll wraps inscribed rolling and sliding.

Figure 2 shows the analytical model of the oil film pressure generated within the gap. The curvature radii of the inside of the fixed scroll and the outside of the orbiting scroll, are denoted by r_1 and r_2, respectively. The gap between the two scrolls is represented by S_0. In the scroll compressor, the orbiting scroll moves at a speed of $(V_s + V_r)$ to the fixed scroll. On the other hand, in this analysis, both the fixed and orbiting scroll move counterclockwise at rolling velocity of V_r. In addition, the orbiting scroll moves in a counterclockwise at a sliding velocity of V_s. Therefore, the contact point keeps as stationary state at the origin O of the $X - Y$ coordinate. The origin O is the midpoint of the minimum gap.

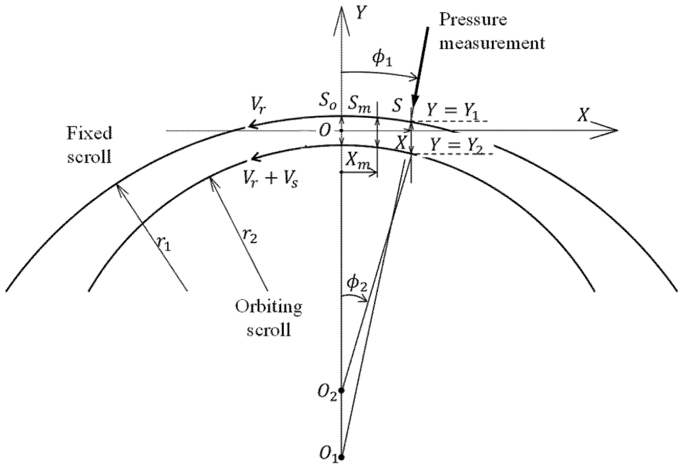

Figure 2: Analytical model of the fixed scroll with radius of r_1 and the orbiting scroll with radius of r_2, undergoing relative inscribed rolling V_r and sliding V_s motions

Since the oil film pressure increases only near the minimum gap, the analysis assumes that it is necessary to consider only the vicinity of the minimum gap ($X \ll r_1, r_2$). The oil film thickness can be given by using the mean radius of curvature R, which is derived by:

$$\frac{1}{R} = \frac{1}{2}\left(\frac{1}{r_2} - \frac{1}{r_1}\right) \tag{1}$$

The Navier-Stokes Equation, described in Lamb (1932), for the velocity in the oil film can be simplifies to:

$$\frac{dP}{dX} = \eta \frac{d^2 u}{dY^2} \tag{2}$$

u and η represent the velocity and viscosity of the oil, respectively. Where, it is assumed that the viscosity terms in the X and Z directions are small enough to be negligible. This assumption applies to the extremely thin oil film between the scroll wraps.

The Equation (2) can be solved with an assumption that the pressure P reaches the maximum at $X = +X_m$ near the minimum gap (see Figure 2), as in the following expression:

$$p = \frac{2kx(1 + k^2 x_m^2)}{(1 + k^2 x^2)^2} - \frac{kx(1 - 3k^2 x_m^2)}{1 + k^2 x^2} + (1 - 3k^2 x_m^2)\left(\frac{\pi}{2} - \tan^{-1} kx\right) \tag{3}$$

which represents the dimensionless oil film pressure defined by

$$p = \frac{P}{\frac{3}{4}\eta(2V_r + V_s)\sqrt{\frac{R}{S_0^3}}} \tag{4}$$

In the theoretical derivation, the following boundary condition was imposed, assuming that the generated oil film pressure is significantly larger than the compressed gas pressure, and thus, the effect of the compressed gas pressure can be ignored:

$$P = 0 \;\; at \;\; X = -X_m \tag{5}$$

The dimensionless coordinates x, x_m and the dimensionless variable k, introduced in Expression (3), are defined as in the following expressions:

$$x = \frac{X}{r_1}\ ;\ x_m = \frac{X_m}{r_1}\ ;\ k = \frac{r_1}{\sqrt{S_0 R}} \tag{6}$$

Here, assume the second boundary condition that the oil film pressure becomes zero at infinite distances, as in the following expression:

$$P = 0\ \ at\ \ X = +\infty \tag{7}$$

As a result, the maximum of the dimensionless oil film pressure, p_{max}, is reduced to a constant value of 1.014 at the dimensionless position $kx_m(= X_m/\sqrt{S_0 R}) = 0.47513$. Of quite significance is that the maximum of dimensional oil film pressure, P_{max}, can be reduced to the following expression:

$$P_{max} = 0.7605\eta(2V_r + V_s)\sqrt{\frac{R}{S_0^3}} \tag{8}$$

which is proportional to the oil viscosity η, the sum of the rolling and sliding velocities $(2V_r + V_s)$, 1/2 power of the mean radius of curvature R and inversely proportional to 3/2 power of the minimum gap S_0.

4. MODEL EXPERIMENT

Figure 3 shows the experimental equivalent model that reproduces the vicinity of the contact point of scroll wraps. This model replaces the actual component with an equivalent arc for simplicity. A small cylinder represents an orbiting scroll, inscribed along the inner wall of a fixed large cylinder, simulating a fixed scroll. The small cylinder was fixed on an orbiting thrust plate, which is forced to orbited by crankshaft and Oldham ring. In line with our theoretical analysis, which assumes that the effect of compressed gas pressure is negligible, the experimental model incorporates a sufficiently large release volume on the left half of the large cylinder, to minimize volumetric pressure changes.

Table 1 shows the specifications of the model. The radius r_1 of the fixed large cylinder is set to 56 mm. The small cylinder radius r_2 is set to 48.7 mm, which is fixed on an orbiting plate with the orbiting radius r_o of 3 mm. The small cylinder is fixed on the orbiting plate, thus inscribing on the larger cylinder. The mean radius of curvature, R, for the fixed large cylinder and the inscribed small cylinder is 747.2 mm. The gap between the small and larger cylinders was set to 20 μm under static conditions by using feeler gauge.

This model experiments were carried out under atmospheric pressure conditions within a sufficient oil chamber. To measure the oil film pressure generated, a semiconductor pressure transducer is embedded at the center of the height of fixed large cylindrical side. A thermocouple also embedded adjacent to this point to monitor the oil temperature.

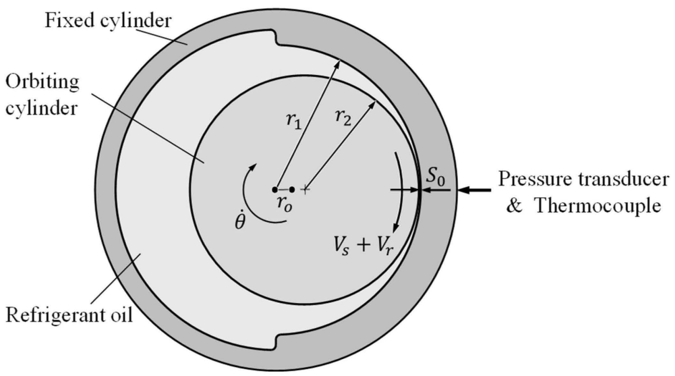

Figure 3: Equivalent model for experiment

Table 1: Major specifications of simplified equivalent physical model tests

Parameter	**Value**
Gas	Air
Oil	POE, VG68
Average oil temperature, T_{oil}	26.3 (℃)
Viscosity of oil, η (26.3℃)	0.135 (Pa·s)
Radius of orbiting cylinder, r_2	48.7 (mm)
Radius of fixed cylinder, r_1	56 (mm)
Orbiting radius, r_o	3 (mm)
Cylinder gap, S_0	20～150 (μm)
Rotation speed, N	341.4～1599 (min^{-1})

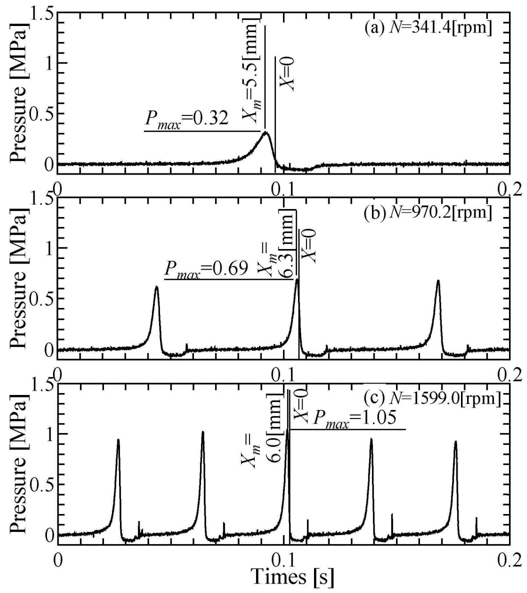

Figure 4: Measured oil film surface pressure variation with orbiting speed N from 341.4 min^{-1} to 1599 min^{-1}.

The oil film pressure was measured when the orbiting speed of the small cylinder was varied from 341.4 min^{-1} to 1599 min^{-1}. This adjustment altered the sliding speed V_s of the small gap from 0.11 m/s to 0.50 m/s. Then, the rolling speed V_r varies from 1.74 m/s to 8.15 m/s, over 10 times higher than the sliding speed.

The measured oil temperature during the experiment indicated with a mean value of approximately 26.3ºC. The oil viscosity coefficient η was addressed to be 0.135 Pa･s, calculated from the Walther's equation (JIS K2283(2000)). The viscosity in our experiment is considerably higher than that in the actual compressor under operation.

Figure 4 shows the oil film pressure measured when the small cylinder was orbited with a speed of 341.4 min^{-1}, 970.2 min^{-1} and 1599.0 min^{-1}, respectively. Each rotation of the crankshaft generates one pressure peak, regardless of the rotation speed. The higher the rotating speed, the steeper and larger these peaks become. The pressure reaches its maximum just before the pressure measurement point (location at X= 0). The maximum pressure appears at X_m=5.5 mm to 6.3 mm. The maximum value of the measured oil film pressure P_{max}, ranging from 0.32 MPa to 1.05 MPa as the orbiting speed increases.

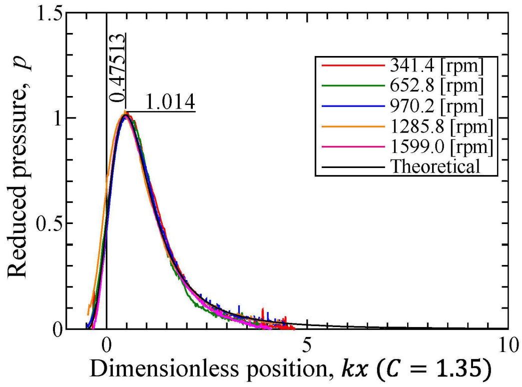

Figure 5: Reduced pressure p on dimensionless position kx, compared with theoretical one

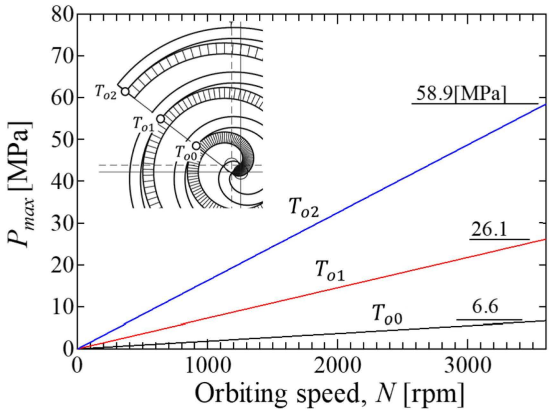

Figure 6: Calculated oil film pressure for a medium capacity scroll compressor

The minimum gap S_0 was calculated by an inverse calculation method, assuming that the maximum of non-dimensional pressure p_{max} aligns with the theoretical value of 1.014. The calculated gap values S_0 was relatively larger, ranging from 104 μm to 142 μm.

Figure 5 shows the non-dimensional oil film pressure, using the inverse calculation values of gaps. The horizontal axis represents th e dimensionless position kx. All the experimental results conducted at different orbiting speeds agree well with the theoretical analysis. Where, the correction factor C, multiplying the dimensionless horizontal axis kx was introduced, and was set to 1.35. The assumption used to determine the oil film thickness has an effect on the mean curvature of radius. Therefore, a correction factor C was introduced to rectify this. Careful consideration is needed regarding this correction factor.

By integrating the oil film pressure, the resultant oil film force F_{oil} can be calculated. The resultant oil film force F_{oil} reached sufficiently larger than the centrifugal force acting on the rotating part of the compressor.

5. OIL FILM FORCE FOR MEDIUM-CAPACITY SCROLL COMPRESSORS

The oil film pressure expected to occur near the scroll contact was examined assuming a medium-capacity scroll compressor. It was supposed that an asymmetric scroll with a base circle radius of 2.5 mm, an orbiting radius of 3.85 mm and a scroll height of 30 mm, operates with a suction pressure of 1 MPa and a discharge pressure of 3.4 MPa. The maximum oil film pressure was calculated with the contact points T_{o0}, T_{o1}, and T_{o2} which have mean radius of curvature R = 130.6 mm, 519.2 mm, and 1164.0 mm, respectively.

The minimum gap S_0 was set to 10 μm, while the orbiting speed was varied up to 3600 min^{-1}. Figure 6 shows the calculated the maximum oil film pressure P_{max}. The maximum oil film pressure increases with increasing the orbiting speed, and as the contacts move toward the outer circumference. The maximum oil film pressure P_{max} reaches 58.9 MPa at the outermost circumference with 3600 min^{-1}. In this case, the sum of the resultant oil film force of all of contact point will reach 12.9 kN, which greatly exceeds the centrifugal force of 0.6 kN.

6. CONCLUSIONS

This study theoretically demonstrated the significant oil film pressure due to oil viscosity, induced by the rolling and sliding motions of the orbiting scroll wrap, inscribed to the fixed scroll wrap. The oil film pressure was derived in a dimensionless form, where the maximum value was addressed to be a constant value of 1.014.

The oil film pressure generated in the orbiting speed up to 1600 min^{-1} was measured and compared with the theoretical analysis results. The measured oil film pressure was found to align closely with the theoretical analysis results. However, a correction coefficient was introduced to rectify the error occurs in the coordinates. Further improvements of correction coefficient are to be expected.

Finally, the oil film pressure was estimated for a medium-capacity scroll compressor. The large oil film force is generated for the specified compressor, which potentially causing deformation of the scroll wraps. In practical compressor, it can be thought that the gap between the scroll wraps takes an appropriate value such that the gas force, the resultant friction force and the resultant oil film force are balanced.

NOMENCLATURE

B	scroll height	(m)
C	empirical correction coefficient	
k	non-dimensional variable	
P, P_{max}	oil film pressure	(Pa)
p, p_{max}	non-dimensional oil film pressure	
R	mean radius of curvature	(m)
r_1, r_2	radius of curvature	(m)
r_b	base circle radius	(m)
r_o	orbiting radius	(m)
S, S_0, S_m	oil film thickness	(m)
u	velocity of oil	(m·s^{-1})
V_s, V_r	sliding and rolling speed	(m·s^{-1})
X_m, x_m	position of maximum pressure	

Greek symbols

η	viscosity of oil	(Pa·s)
θ	orbiting angle	(rad)
ϕ_{on}	expansion angle of involute	(rad)
ϕ_{os}	beginning angle of involute	(rad)

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