2014

Mixed Lubrication Analysis of Vane Sliding Surface in Rotary Compressor Mechanisms

Yasutaka Ito
Corporate Research & Development Center, Toshiba Corporation, Japan, yasutaka.ito@toshiba.co.jp

Hitoshi Hattori
Corporate Research & Development Center, Toshiba Corporation, Japan, hit.hattori@toshiba.co.jp

Kazuhiko Miura
Toshiba Carrier Corporation, kazuhiko2.miura@toshiba.co.jp

Mixed lubrication analysis of vane sliding surface in rotary compressor mechanisms

- Influences of friction of vane sliding surface on lubricating condition between vane top and rolling piston -

Yasutaka Ito[1], Hitoshi Hattori[1] and Kazuhiko Miura[2]

[1] Corporate Research & Development Center, Toshiba Corporation
1, Komukai toshiba-cho, Saiwai-ku, Kawasaki 212-8582, Japan
Tel: +81-44-549-2380, Fax: +81-44-549-2383
E-mail: yasutaka.ito@toshiba.co.jp
E-mail: hit.hattori@toshiba.co.jp

[2] Toshiba Carrier Corporation
336, Tadehara, Fuji-shi, Shizuoka-ken 416-8521, Japan
Tel: +81-545-62-5591, Fax: +81-545-66-0305
E-mail: kazuhiko2.miura@toshiba.co.jp

ABSTRACT

To investigate the influences of friction of the vane sliding surface between the vane and vane slot on the lubricating conditions between the vane top and the rolling piston, we perform a numerical analysis of rolling piston and vane motion, considering mixed lubrication of the vane sliding surface between the vane and vane slot. Analysis results indicate that the risk of scuffing is high between the vane top and piston due to increased PV value near crank angle ψ = 180°, at which the lubrication is poor when the contact friction forces of the vane sliding surface increase. Moreover, the vane separated from the piston may collide with the piston at crank angle ψ = 180°, and excessive impact may occur between the vane top and the piston due to increased contact friction forces of the vane sliding surface.

1. INTRODUCTION

In the compression mechanisms of a rotary compressor for air conditioners, a vane and a rolling piston separate the suction and compression chambers. The rolling piston is driven by a crank eccentrically revolving on the shaft axis while rotating on its own axis. The vane undergoes reciprocating motion, and its top usually comes into contact with the rolling piston by the spring force and discharge pressure. The lubricating condition between the rolling piston and the vane top is critical because the contact pressure is high owing to line contact of the vane top. To prevent damage such as scuffing or abnormal wear, it is important to clarify the lubricating characteristics between the vane top and the rolling piston. Because the normal force acting on the piston at vane contact is greatly influenced by friction of the vane sliding surface between the vane and the vane slot, mixed lubrication analysis of the vane sliding surface between the vane and the vane slot is required to accurately obtain the normal force acting on the piston at vane contact.

To investigate the lubricating characteristics between the vane top and the rolling piston, we perform a numerical analysis of the motion of the rolling piston and the motion of the vane considering mixed lubrication of the vane sliding surface between the vane and vane slot. The cases of different friction coefficients of the vane sliding surface at solid contact between the vane and vane slot were calculated, and the results were compared. The results elucidate the influences of friction of the vane sliding surface between the vane and vane slot on the lubricating characteristics between the vane top and the rolling piston.

2. GOVERNING EQUATIONS

In the compression mechanism of air conditioner rotary compressors, the compression chamber is composed of a rolling piston, a vane, and a cylinder. The rolling piston, which is driven by the crank, rotates eccentrically, decreasing the volume of the compression chamber and increasing the pressure of the refrigerant.

This analysis solves, as a coupled problem the equation of vane motion and the equations of equilibrium of forces and moments, the equations of equilibrium of forces for the rolling piston, the rotational equation of motion of the rolling piston, the modified Reynolds equation, the elastic contact equation of the vane sliding surface between the vane and vane slot, and the Reynolds equation of the sliding section between the rolling piston and the crank area.

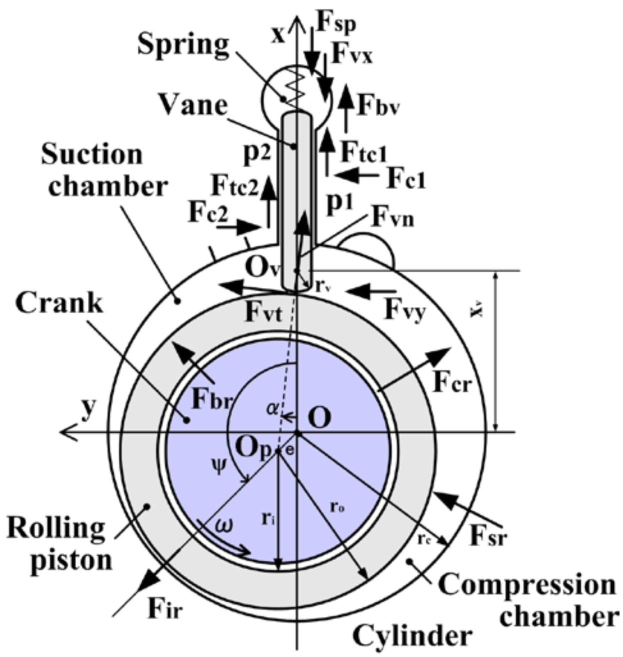


Figure 1: Static coordinate system with origin *O* at cylinder center

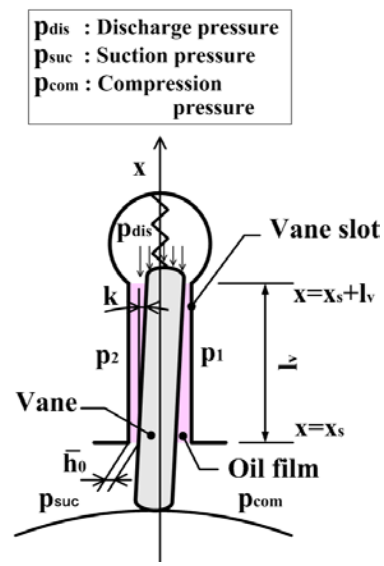


Figure 2: Coordinate system for calculation of friction forces of vane sliding surface

2.1 Motion of vane and rolling piston in the *x*- and *y*-directions

Figure 1 shows the static coordinate system with origin *O* at the cylinder center. As that figure shows, the equation of motion of the vane and the equations of equilibrium of forces and moments are expressed as follows[1, 2]:

$$m_v \ddot{x}_v = F_{vnx} + F_{vtx} + F_{to1} + F_{to2} + F_{tc1} + F_{tc2} + F_{bv} - F_{vx} - F_s \tag{1}$$

$$0 = F_{vny} + F_{vty} + F_{o1} - F_{o2} + F_{c1} - F_{c2} + F_{vy} \tag{2}$$

$$0 = M_{vn} + M_{vt} + M_{o1} + M_{o2} + M_{c1} + M_{c2} \tag{3}$$

The equation of motion of the rolling piston and the equations of equilibrium of forces are expressed as follows[3, 4]:

$$0 = F_{crx} + F_{srx} + F_{brx} - F_{irx} - F_{vnx} - F_{vtx} \tag{4}$$

$$0 = -F_{cry} + F_{sry} + F_{bry} + F_{iry} + F_{vny} - F_{vty} \tag{5}$$

Here, the subscripts *x* and *y* indicate the *x*- and *y*-direction components, respectively.

It is assumed that the solid contact region between the rolling piston and vane is in a boundary lubrication zone. Therefore, the friction force acting on the piston at vane contact F_{vt} is calculated by Coulomb's law as follows:

$$F_{vt} = \gamma_1 \mu_p F_{vn} \tag{6}$$

$$\gamma_1 = \mathrm{sgn}(V_{pv}) \tag{7}$$

$$V_{pv} = r_o \omega_p + (r_o + r_v)\dot{\alpha} \tag{8}$$

The equations and the explanations for calculation of the friction forces on the vane sliding surface are described as follows. Figure 2 shows the coordinate system for calculation of friction force on the vane sliding surface. The vane always inclines in the clockwise direction (Fig. 2) because of the effect of the compression pressure p_{com}[1]. Therefore, solid contact between the vane and vane slot occurs at the lower end on the suction side and at the upper end on the discharge side. Assuming that the vane sliding surface between the vane and vane slot is in a mixed lubrication zone, we perform a mixed lubrication analysis to calculate the friction forces on the vane sliding surface between the vane and vane slot. The influence of side leaks is negligibly small because the vane width is long enough as compared with the length of effectual oil film on the vane sliding surface near the lower end of the suction side and near the upper end of the discharge side between the vane and vane slot. Therefore, the vane sliding surface is treated as a surface having infinite width along the perpendicular of the plane in Fig. 2. The oil film reactive forces of the vane sliding surface on the discharge side and the suction side between the vane and vane slot are calculated by the modified Reynolds equations, as follows(5):

$$\frac{\partial}{\partial x}\left(\Phi_X \frac{\bar{h}^3}{\eta}\frac{\partial p}{\partial x}\right) = 6U\frac{\partial h_T}{\partial x} + 6U\sigma\frac{\partial \Phi_s}{\partial x} + 12\frac{\partial h_T}{\partial t} \tag{9}$$

The vane sliding surface is treated as a surface having longitudinal surface roughness. The direction parameter of the surface roughness χ used here is 3.0.

The local oil film thickness of vane sliding surface h_T is expressed as follows:

$$h_T = \frac{\bar{h}}{2}\left\{1 + erf\left(\frac{\bar{h}}{\sqrt{2}\sigma}\right)\right\} + \frac{\sigma}{\sqrt{2\pi}}\exp\left(-\frac{1}{2}\left(\frac{\bar{h}}{\sigma}\right)^2\right) \tag{10}$$

The average oil film thicknesses on the discharge side and the suction side between the vane and vane slot are

$$\bar{h}_1 = c_v - \bar{h}_0 - k(x - x_s) \tag{11}$$

$$\bar{h}_2 = \bar{h}_0 + k(x - x_s) \tag{12}$$

For calculating the contact forces of the vane sliding surface between the vane and vane slot, Patir and Cheng's approximate expression, based on Greenwood and Tripp's theory, is used [6, 7]:

$$p_c = \begin{cases} k_c E' \times 4.4086 \times 10^{-5}\left(4 - \dfrac{\bar{h}}{\sigma}\right)^{6.804} & (\bar{h} < 4\sigma) \\ 0 & (\bar{h} \geq 4\sigma) \end{cases} \tag{13}$$

The oil film reactive forces F_{o1}, F_{o2} and contact forces F_{c1}, F_{c2} are calculated by integration of oil film pressure p and of contact pressure p_c.

The contact friction forces on the vane sliding surface between the vane and vane slot F_{tc1}, F_{tc2} are expressed as follows:

$$F_{tc1} = \gamma_2 \mu_v w_c \int p_c(\bar{h}_1)dx \tag{14}$$

$$F_{tc2} = \gamma_2 \mu_v w_c \int p_c(\bar{h}_2)dx \tag{15}$$

$$\gamma_2 = \mathrm{sgn}(-\dot{x}_v) \tag{16}$$

The fluid friction forces on the vane sliding surface on the discharge side and on the suction side between vane and vane slot F_{to1}, F_{to2} are assumed to obey Newton's law of viscosity.

$$F_{to1} = w_c \int \eta \dot{x}_v / \bar{h}_1 dx \tag{17}$$

$$F_{to2} = w_c \int \eta \dot{x}_v / \bar{h}_2 dx \tag{18}$$

2.2 Motion of the rolling piston in the rotation direction

Figure 3 shows a moving coordinate system with origin O_p at the crank center. The moving coordinate system eccentrically rotates the circumference of the cylinder center. A moving coordinate system is used for the calculation of angular velocity of piston rotation ω_p. As shown in Fig. 3(a), the rotational equation of motion of a rolling piston about the piston center is expressed as follows[3, 8]:

$$I\dot{\omega}_p = M_r - M_s - r_o F_{vt} \tag{19}$$

Here, the viscous friction moment due to refrigerant gas between the piston and cylinder is negligibly small, and therefore not considered further in this analysis.

The friction moments M_s on both end surfaces of the rolling piston are expressed as follows[9]:

$$M_s = \frac{2\pi\eta\omega_p\left(r_o^{\,4} - r_i^{\,4}\right)}{c_s} \tag{20}$$

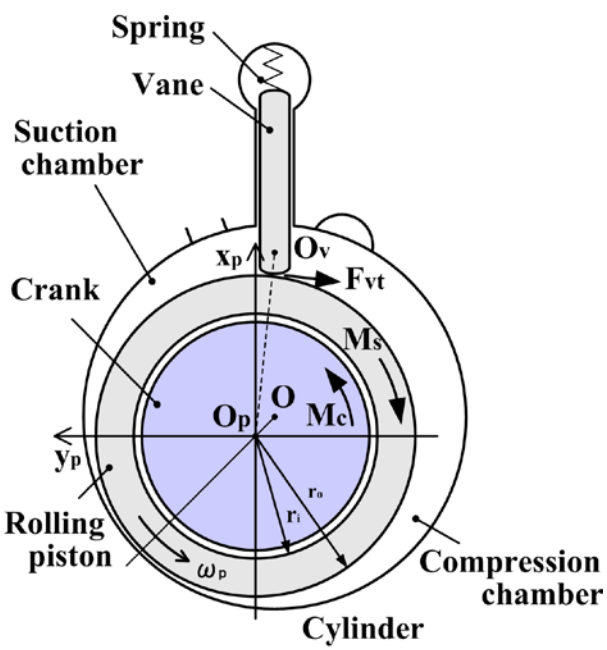
The equations for the calculation of the friction force and moment due to oil film viscosity between the piston and crank are explained as follows. Figure 3(b) shows an analysis model of oil film between the piston and crank. The direction and the value of the oil film reactive force between the piston and crank F_{cr} in Fig. 3(b) are obtained by Eqs. (4)–(5). The sliding section between the rolling piston and crank is regarded as a hydrodynamic journal bearing. The oil film pressure between the piston and crank is calculated by the Reynolds equation [8,10]. Because the width diameter ratio (w_s / (2*r_i), where w_s is the width of the crank) is less than 4.0, the Reynolds equation of finite width bearing is used, as follows:

$$\frac{\partial}{r_i^2\partial\theta}\left(\frac{h_r^{\,3}}{\eta}\frac{\partial p_r}{\partial\theta}\right) + \frac{\partial}{\partial z}\left(\frac{h_r^{\,3}}{\eta}\frac{\partial p_r}{\partial z}\right) = 6\omega_c\frac{\partial h_r}{\partial\theta} + 12\frac{\partial h_r}{\partial t} \tag{21}$$

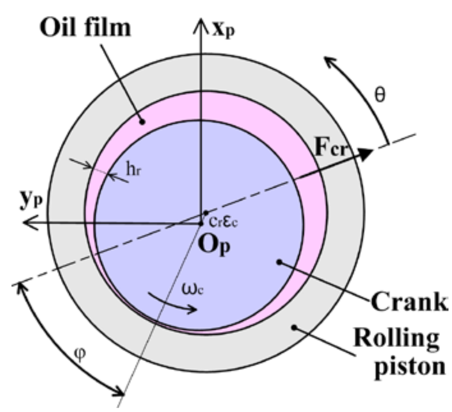
$$h_r = c_r \times \left(1 + \varepsilon_c \cos(\theta - \varphi)\right) \tag{22}$$

Using boundary conditions such that the ambient pressure at both ends equals the discharge pressure p_{dis}, oil film pressure between the piston and crank is calculated from Eq. (21). Although the oil film might break under negative pressure, the following calculation indicates that the oil film pressure is positive over the entire bearing area owing to the high discharge pressure. Thus, the frictions moment M_r due to oil film viscosity between piston and crank are calculated by integration over the whole bearing area, as follows:

$$M_r = \int\int\left(\frac{\eta\omega_c r_i}{h_r} + \frac{h_r}{2r_i}\frac{\partial p_r}{\partial\theta}\right) r_i^{\,2}\, d\theta dz \tag{23}$$



(a) Friction forces and moments acting on rolling piston



(b) Analysis model of oil film between piston and crank

Figure 3: Moving coordinate system with origin O_p at crank center

3. ANALYSIS PROCEDURE AND CONDITIONS

3.1 Analysis Procedure

Eqs. (1)–(5), (9), (13), (19), and (21) are solved as a coupled problem, and the average oil film thickness at the lower end of vane slot $\overline{h}_0$, the inclination of the vane k, the eccentricity of the crank with respect to piston center ε_c, the angle from vane centerline to minimum oil film position φ, the angular velocity of piston rotation ω_p, and the normal force acting on the piston at vane contact F_{vn} are numerically calculated by the Newton–Raphson method. The time differentials of $\overline{h}_0$ and k used in the calculation of the squeeze terms in the modified Reynolds equations, and the time differentials of ε_c and φ, used in the calculation of the squeeze terms in the Reynolds equations, are approximately calculated by the following equation:

$$\left.\begin{matrix} \dot{h}_0 \\ \dot{k} \\ \dot{\varepsilon}_c \\ \dot{\varphi} \end{matrix}\right\} = \frac{1}{\Delta t}\left\{\begin{matrix} \overline{h}_{0t+\Delta t} - \overline{h}_{0t} \\ k_{t+\Delta t} - k_t \\ \varepsilon_{ct+\Delta t} - \varepsilon_{ct} \\ \varphi_{t+\Delta t} - \varphi_t \end{matrix}\right\} \tag{24}$$

Moreover, the force variations and the moments acting on the rolling piston and vane are considered. Consequently, the coupled problem is solved recursively along the time axis.

3.2 Analysis conditions

The analysis conditions are shown in Table 1. The cylinder radius is 36.5 mm. The rolling piston outer radius is 29.7 mm. The crank eccentricity is 6.8 mm. The calculations were executed for the case of different friction coefficient of vane sliding surface at solid contact between the vane and vane slot μ_v. Moreover, the friction coefficient between the vane and piston μ_p is 0.12.

Figure 4 shows the relationship between the crank angle ψ and the compression pressure p_{com}, which was used as the analysis condition. It is assumed that the refrigerant discharge starts when the compression pressure p_{com} increases to 1.2-fold the discharge pressure p_{dis}. [9]

Table 1: Analysis conditions

Radius of cylinder, r_c [mm]	36.5
Outer radius of piston, r_o [mm]	29.7
Inner radius of piston, r_i [mm]	20.0
Eccentricity of crank, e [mm]	6.8
Oil viscosity, η [10^{-3} Pa・s]	2.8
Rotating Frequency, f_0 [Hz]	60
Discharge pressure / Suction pressure, p_{dis} / p_{suc} [MPa]	4.25/1.27 (Δp=2.98)
Friction coefficient at solid contact between vane and vane slot, μ_v	0.0, 0.10, 0.20, 0.30, 0.40
Friction coefficient between piston and vane, μ_p	0.12

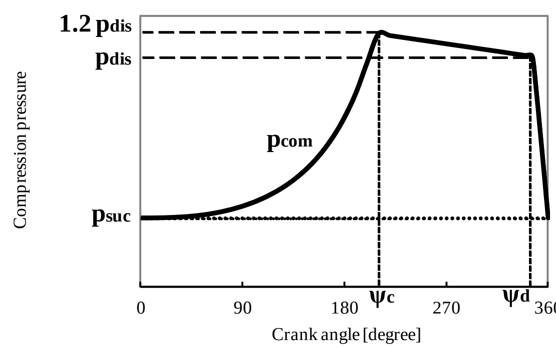
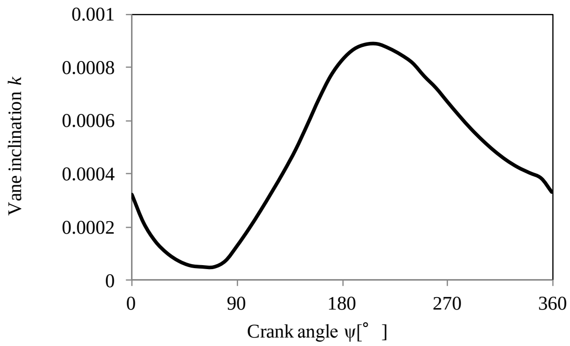


Figure 4: Pressure variation of refrigerant in compression chamber during one cicle

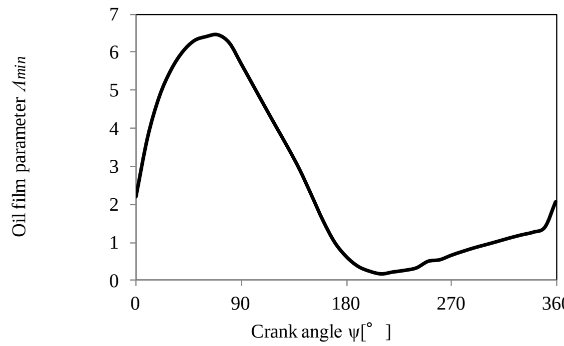
4. RESULTS AND DISCUSSION

4.1 Lubricating characteristics of vane sliding surface

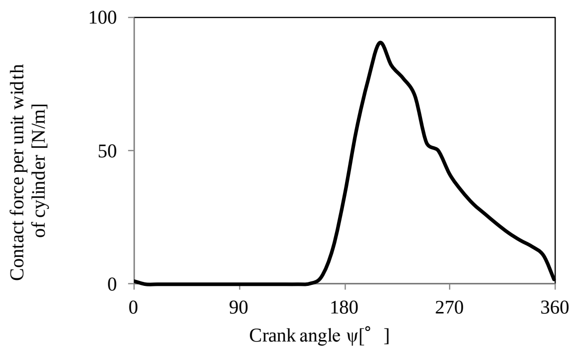
Figure 5 shows the analysis results when the discharge pressure p_{dis} is 4.25 MPa, the suction pressure p_{suc} is 1.27 MPa, the rotating frequency f_0 is 60 Hz, and the friction coefficient at solid contact between vane and vane slot μ_v is 0.1. The horizontal axis is the crank angle ψ. Figures 5(a) and (b) respectively show variations of the vane inclination k and the minimum oil film parameter of the vane sliding surface on the suction side between the vane and vane slot Λ_{min} (= h_{min}/σ) through one crank revolution. Figures 5(c) and (d) respectively show variations of the contact force of vane sliding surface per unit width of cylinder F_{c2} / w_c and oil film reactive force of vane sliding surface per unit width of cylinder F_{o2} / w_c on the suction side between the vane and vane slot through one crank revolution. Figures 5(e) and (f) respectively show variations of the contact friction force of vane sliding surface per unit width of cylinder between the vane and vane slot and the normal force acting on the piston at vane contact per unit width of cylinder F_{vn} / w_c through one crank revolution.



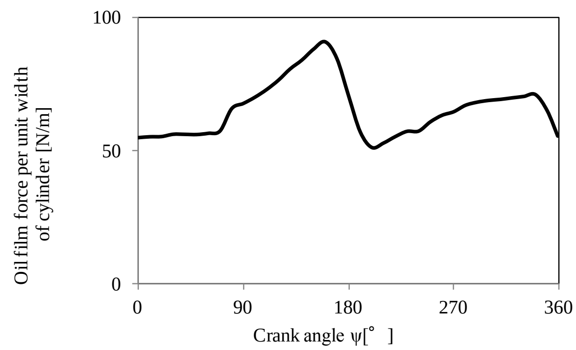
(a) Inclination of vane



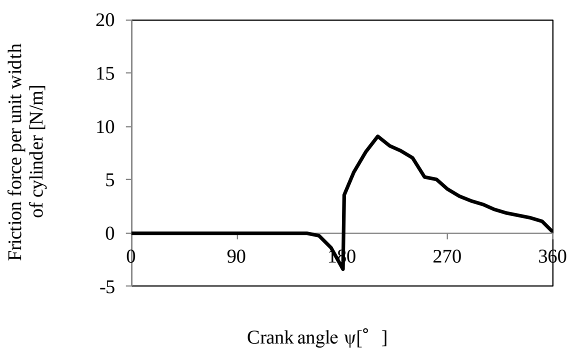
(b) Minimum oil film parameter of vane sliding surface on suction side between vane and vane slot



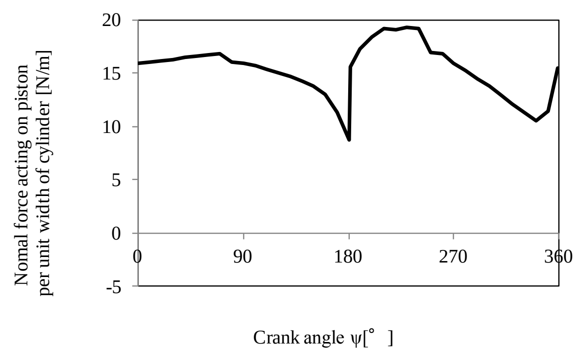
(c) Contact force of vane sliding surface on suction side between vane and vane slot



(d) Oil film reactive force of vane sliding surface on suction side between vane and vane slot



(e) Contact friction force of vane sliding surface

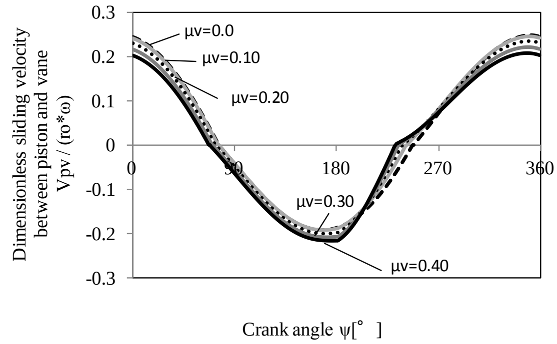


(f) Normal force acting on piston at vane contact

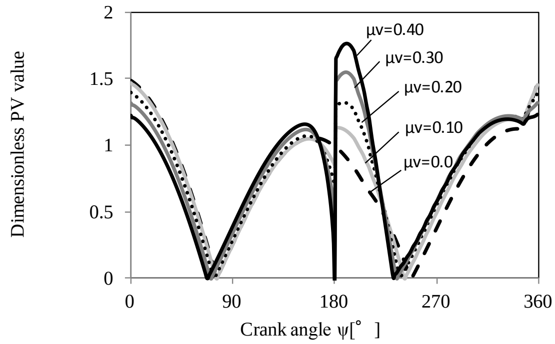
Figure 5: Lubrication characteristics of vane sliding surface
(p_{dis} / p_{suc} = 4.25 / 1.27 MPa (Δp = 2.98 MPa), f_0 = 60 Hz, μ_v = 0.10)

As shown in Fig. 5(a), the vane always inclines in the clockwise direction (Fig. 2) because the vane inclination k is always larger than 0°. Therefore, it can be seen from the vane geometry (Fig. 2) that $\overline{h}_0$ becomes the minimum oil film thickness h_{min} on the suction side between the vane and vane slot. As shown in Fig. 5(b), Λ_{min} (= h_{min}/σ) increases between ψ = 0° and ψ = 180° because the oil film pressure on the suction side is raised by the wedge film effect. Because the wedge film effect disappears when the vane motion reverses at ψ = 180°, Λ_{min} decreases from ψ = 180°. Therefore, the oil film reactive force on the suction side between the vane and vane slot decreases near ψ = 180°, as shown in Fig. 5(d), and the contact forces on the suction side between the vane and vane slot increases rapidly near ψ = 180° as shown in Fig. 5(c).

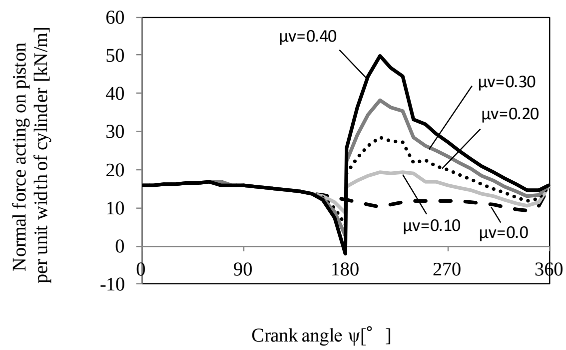
Under these analysis conditions, solid contact does not occur on the discharge side between the vane and vane slot. Therefore, the contact friction force of the vane sliding surface is based on the contact force on the suction side between the vane and vane slot. As shown in Fig. 5(e), it is found that the contact friction force of the vane sliding surface is negative from ψ = 0° to ψ = 180° and is positive from ψ = 180° to ψ = 360°. This is because the contact friction force of the vane sliding surface acts in opposite direction from vane movement. From ψ = 0° to ψ = 180°, the contact friction force of the vane sliding surface acts on the vane to separate the vane and the piston, and its value is maximum at ψ = 180°. Therefore, the normal force acting on the piston at vane contact is minimized at ψ = 180°, as shown in Fig. 5(f).



(a) Relative sliding velocity between vane and rolling piston



(b) PV value between vane top and rolling piston



(c) Normal force acting on piston at vane contact

Figure 6: Influences of friction of vane sliding surface (p_{dis} / p_{suc} = 4.25 / 1.27 MPa (Δp = 2.98 MPa), f_0 = 60 Hz)

4.2 Influences of friction of vane sliding surface between vane and vane slot

Figure 6 shows the analysis results when the friction coefficient of the vane sliding surface at solid contact between vane and vane slot μ_v is 0.0, 0.10, 0.20, 0.30, and 0.40.

Figure 6(a) shows the relative sliding velocity between the piston and the vane through one revolution of the crank. It is found that the sliding velocity decreases for all crank angles when the friction force of the vane sliding surface increases. This is because friction force at vane contact F_{vt}, which controls the rotating motion of the piston to prevent slip at the vane top, increases when the normal force acting on the piston at vane contact F_{vn} increases owing to an increase in μ_v.

Figure 6(b) shows variation of the PV value between the piston and vane in a dimensionless form. Here, the dimensionless PV value is defined as the ratio of the PV value to that of the μ_v = 0.0, crank angle ψ = 180° case. The PV value is calculated from F_{vn} and the relative sliding velocity V_{pv}. It can be seen that the PV value near ψ = 180° increases when the contact friction force of the vane sliding surface increases. This is because F_{vn} near ψ = 180° increases when μ_v increases. Near ψ = 180° is an area where the risk of scuffing is high owing to high PV value through one revolution of the crank. Consequently, it is found that the risk of scuffing between the piston and vane top increases when the contact friction force of the vane sliding surface increases.

Figure 6(c) shows variation of the normal force acting on a piston at vane contact per unit width of cylinder F_{vn} / w_c. The value of F_{vn} becomes negative near the crank angle ψ = 180° when the friction coefficient μ_v is 0.40. It is found that the vane separates from the piston at ψ = 180° because of the increase of the friction forces of the vane sliding surface. Because F_{vn} increases rapidly after ψ = 180°, the vane separated from the piston collides with the piston and an excessive impact occurs between the vane top and the piston. Thus, the risk of surface damage at the vane top becomes high. The analysis results reveal the necessity of decreasing the friction force of vane sliding surface to prevent separation of the vane and the piston.

5. CONCLUSION

The numerical analysis of the motion of the rolling piston and the motion of the vane considering mixed lubrication of the vane sliding surface has been performed to investigate the influences of the contact friction of the vane sliding surface on the lubricating condition between the vane top and rolling piston in a rotary compressor. The following results were obtained.

1. The risk of scuffing is high between the vane top and piston because of increased PV value near crank angle ψ=180°, at which the lubricating condition is severe when the contact friction forces of the vane sliding surface increases.

2. The vane separated from the piston collides with the piston and an excessive impact occurs between the vane top and the piston because of the increased the contact friction forces of the vane sliding surface. Thus, the risk of surface damage at the vane top becomes high.

3. Consequently, to maintain good vane top surface condition, it is important to guarantee and to maintain the sufficient lubrication of the vane sliding surface.

NOMENCLATURE

c_r	Radial clearance between piston and crank
c_s	Clearance between piston end and bearing end
c_v	Clearance between vane and vane slot
d_v	Vane height
E'	Equivalent elastic modulus
e	Crank eccentricity
F_{br}	Fluid friction force on piston end faces attributable to revolution
F_{bv}	Fluid friction force on vane end faces
F_{c1}, F_{c2}	Contact forces of vane sliding surface between vane and vane slot
F_{cr}	Oil film reactive force between piston and crank
F_{ir}	Centrifugal force attributable to revolution
F_{o1}, F_{o2}	Oil film reactive forces of vane sliding surface between vane and vane slot
F_s	Friction forces on both end surfaces of piston
F_{sp}	Spring force
F_{sr}	Gas pressure-difference force between compression chamber and suction chamber
F_{tc1}, F_{tc2}	Contact friction forces of vane sliding surface between vane and vane slot
F_{to1}, F_{to2}	Fluid friction forces of vane sliding surface between vane and vane slot
F_{vn}	Normal force acting on piston at vane contact
F_{vt}	Friction force acting on piston at vane contact

F_{vx} , F_{vy}	Forces acting on vane due to refrigerant pressure difference
f_0	Rotating frequency
$\bar{h}$	Average oil film thickness of vane sliding surface
$\bar{h}_0$	Average oil film thickness at lower end of vane slot on the suction side at $x = x_s$ in Fig. 6
$\bar{h}_1$, $\bar{h}_2$	Average oil film thicknesses on the discharge side and suction side between vane and vane slot
h_T	Local oil film thickness of vane sliding surface
h_{min}	Minimum average oil film thickness on the suction side between vane and vane slot
h_r	Oil film thickness between piston and crank
I	Mass moment of inertia of a piston
k	Vane inclination
k_c	Constant in force-compliance relationship
l_v	Vane slot length
M_{c1}, M_{c2}	Moments of contact force of vane sliding surface between vane and vane slot
M_{o1}, M_{o2}	Moments of oil film reactive forces of vane sliding surface between vane and vane slot
M_r	Friction moment due to oil film viscosity between piston and crank
M_s	Fluid friction moment on piston end faces due to revolution
M_{vn}	Moment of normal force acting on vane at piston contact
M_{vt}	Friction moment acting on vane at piston contact
m_v	Vane mass
O	Cylinder center
O_p	Crank center
O_v	Center of vane top arc
O-x,y	Static coordinates with origin at cylinder center
O_p-x_p,y_p	Moving coordinates with origin at crank center
p	Oil film pressure of vane sliding surface between vane and vane slot
p_1, p_2	Oil film pressure between vane and vane slot corresponding to discharge side and suction side
p_c	Contact pressure between vane and vane slot
p_{com}	Compression pressure
p_{dis}	Discharge pressure
p_{suc}	Suction pressure
p_r	Oil film pressure between piston and crank
r_c	Cylinder radius
r_i	Inner piston radius
r_o	Outer piston radius
r_v	Vane top radius
t	Time
U	Vane sliding velocity
V_{pv}	Relative sliding velocity between piston and vane
x_v	Displacement of vane along the x-direction
x_s	Length from cylinder center to lower end of vane slot along the x-direction
w_c	Cylinder width
α	Attitude angle of crank center with respect to center of vane top arc
γ_1	Constant with value –1, 0, or 1 (–1 when $V_{pv} < 0$, 0 when $V_{pv} = 0$, and 1 when $V_{pv} > 0$)
γ_2	Constant with value –1, 0, or 1 (–1 when $-x_v < 0$, 0 when $-x_v = 0$, and 1 when $-x_v > 0$)
ε_c	Piston eccentricity with respect to crank center
η	Oil viscosity

θ	Rotational angle of crank from crank centerline in the direction of F_{cr}
Λ_{min}	Minimum oil film parameter of vane sliding surface between vane and vane slot
μ_v	Friction coefficient of vane sliding surface at solid contact between vane and vane slot
μ_p	Friction coefficient between piston and vane
σ	Standard deviations of composite roughness
Φ_x	Pressure flow factor
Φ_s	Shear flow factor
φ	Angle from crank centerline in the direction of F_{cr} to minimum oil film position
ψ	Crank angle
ψ_c	Starting angle of refrigerant discharge
ψ_d	Angle of discharge valve
ω	Angular velocity of crank
ω_c	Relative angular velocity between crank and piston
ω_p	Angular velocity of piston rotation on the moving coordinate system

REFERENCES

[1] Ito, Y., Hattori, H., Miura, K., Hirayama, T., 2007, Mixed Lubrication Analysis of Vane Sliding Surface in Rotary Compressor Mechanisms, Tribology Online, vol. 2, no. 3: p. 73-77.

[2] Ito, Y., Hattori, H., Miura, K., 2009, Mixed Lubrication Analysis of Vane Sliding Surface in Rotary Compressor Mechanisms - Influences of Elastic Deformation at Surface End of Vane-Slot -, Tribology Online, vol. 4, no. 5: p. 96-102.

[3] Ito, Y., Hattori, H. and Miura, K., 2010, Numerical analysis for rotating motion of a rolling piston in rotary compressors, Proceeding of the 2010 International Compressor Engineering Conference at Purdue, CD-ROM: p. 1-8.

[4] Padhy, S. K., 1993, On the Dynamics of a Rotary Compressor: part1 – Mathematical modeling, Proc. of the A.S.M.E. Design Automation Conference., vol. 65, no. 1:p. 207-217.

[5] Patir, N., Cheng, H. S.., 1978, An Average Flow Model for Determining Effects of Three Dimensional Roughness on Partial Hydrodynamic Lubrication, Transaction of the ASME, Journal of Lubrication Technology., vol. 100, no. 1:p. 12-17.

[6] Greenwood, J. A., Tripp, J. H., 1970, The Contact of Two Nominally Flat Surfaces, Proceeding of the Institution of Mechanical Engineers., vol. 185, no. 48: p. 625-633.

[7] Patir, N., Cheng, H. S., 1978, Effect of surface roughness orientation on the central film thickness in E.H.D. contacts, Proceeding of the Institute of Mechanical Engineering Part I., vol. 185, no. 48, p. 15-21.

[8] Yoshimura, T., Ono K., Inagaki K., Kotsuka H., Korenaga A., 1999, Analysis of Lubricating Characteristics of Rotary Compressor for Domestic Refrigerators, Transaction of the ASME, Journal of Tribology, vol. 121, no. 7: p. 510-516.

[9] Tanaka, S., Nakahara, T., Kyogoku, K., 1996, Lubrication Characteristics of Refrigerator/Air Conditioning Rotary Compressor, The Tribologist (in Japanese), vol. 41, no. 3: p. 247-254.

[10] Kobayashi, H., 1993, Tribology of Rotor and Blade in Rotary Compressor, The Tribologist (in Japanese), vol. 38, no. 7, p. 599-604.