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# Numerical Analysis for Rotating Motion of a Rolling Piston in Rotary Compressors

## - Effective Factors for Characteristics of Rotating Motion of a Rolling Piston -

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### ABSTRACT

In this study, in order to investigate the characteristics of rotating motion of a rolling piston in a rotary compressor, a numerical analysis for the rotating motion of the rolling piston has been performed. The analysis takes into consideration four kinds of friction forces that act on the rolling piston. The analysis results reveal that the friction force between the side wall of the rolling piston and the vane top has the largest influence. And it can be seen that the rotating motion is also influenced by the piston's own inertia. It is clarified that the relative sliding velocity between the piston and the vane can be reduced by reducing the weight of rolling piston.

### 1. INTRODUCTION

In the compression mechanisms of a rotary compressor for air conditioners, the rolling piston driven by the crank revolves with the eccentric rotation, and also rotates on its own axis around the crank shaft. A vane is provided to separate the suction chamber and the compression chamber. The vane which undergoes reciprocating motion is in contact with the side wall of the rolling piston at its top. Generally, the lubricating condition between the rolling piston and the vane top is very severe because the contact pressure becomes very high owing to the line contact of the vane top. Therefore, for the prevention of scuffing, abnormal wear and so on, it is important to clarify the characteristics of the rotating motion and to optimize the lubricating condition between the rolling piston and the vane top.

Various friction forces influence the rotating motion of the rolling piston. They are the friction force between the side wall of the rolling piston and the vane top, the friction forces on both end surfaces of the piston, the friction force due to the oil film viscosity between the piston and the crank, and the friction force due to the refrigerant viscosity between the piston and the cylinder.

Taking into consideration the four kinds of friction forces, the numerical analysis for the rotating motion of the rolling piston has been performed in order to investigate the characteristics of the rotating motion of the rolling piston<sup>(1-5)</sup>. In this paper, the governing equations and the procedures of the analysis are described. And the analysis results obtained by changing the parameters, namely, the rotating frequency, the pressure difference between the discharge and the suction pressures, and the weight of the rolling piston, are discussed.

### 2. GOVERNING EQUATIONS

#### 2.1 Analysis model

Figure 1 shows a brief drawing of the compression mechanism in a rotary compressor. The compression chamber is composed of the rolling piston, vane and cylinder. The rolling piston driven by the crank rotates eccentrically. Owing to this eccentric rotation, the volume of the compression chamber decreases, and the pressure

of refrigerant becomes high. The vane is provided to separate the suction chamber and the compression chamber. The vane is in contact with the side wall of the rolling piston at its top owing to the spring force and the discharge pressure  $p_{dis}$  acting on the tail-end of the vane. And the vane that undergoes reciprocating motion is driven by the eccentric rotation of the rolling piston.

As shown in Fig.1, the friction forces acting on the rolling piston are the friction force between the side wall of the rolling piston and the vane top, the friction forces on both end surfaces of the piston, the friction force due to the oil film viscosity between the piston and the crank, and the friction force due to the refrigerant viscosity between the piston and the cylinder. The lubrication condition between the piston and the vane is considered to be boundary lubrication. Moreover, the lubrication condition between the piston and the crank is considered to be hydrodynamic lubrication, as shown in Fig.2. And the sliding section between the rolling piston and the crank is regarded as a hydrodynamic journal bearing<sup>(6)</sup>. Since it is considered that the solid contact is not generated on end surfaces of the piston or between the piston and the cylinder, the viscous friction of oil film and refrigerant is taken into consideration in calculation of the friction forces on end surfaces of the piston and between the piston and the cylinder.

## 2.2 Equation of motion of rolling piston

Figure 1 shows the external friction forces acting on the rolling piston. The equation of motion of the rolling piston is expressed as follows<sup>(1-3)</sup>:

$$I\dot{\omega}_p = r_i F_c - M_s - r_o F_{vt} - r_o F_r \quad (1)$$

where,  $I$  is mass moment of inertia of piston,  $\omega_p$  is angular velocity of piston rotation,  $F_c$  is friction force due to the oil film viscosity between piston and crank,  $M_s$  is friction moments on both end surfaces of piston,  $F_{vt}$  is friction force between side wall of rolling piston and vane top,  $F_r$  is friction force due to refrigerant viscosity between piston and cylinder,  $r_i$  is inner radius of piston, and  $r_o$  is outer radius of piston.

## 2.3 Friction force between side wall of rolling piston and vane top

As described in chapter 2.1, it is assumed that the lubrication condition between the piston and the vane is boundary lubrication. Therefore, the frictional force between the side wall of the rolling piston and the vane top  $F_{vt}$  is calculated according to Coulomb's law of friction.  $F_{vt}$  is expressed as follows:

$$F_{vt} = \gamma \mu F_{vn} \quad (2)$$

$$\gamma = \text{sgn}(V_{pv}) \quad (3)$$

$$V_{pv} = r_o \omega_p + (r_o + r_v) \dot{a} \quad (4)$$

where,  $F_{vn}$  is normal force between side wall of rolling piston and vane top,  $\mu$  is friction coefficient between piston and vane,  $\gamma$  is a constant which is value of -1, 0 or 1 ( $\gamma$  is -1 when  $V_{pv} < 0$ ,  $\gamma$  is 0 when  $V_{pv} = 0$ ,  $\gamma = 1$  when  $V_{pv} > 0$ ),  $a$  is attitude angle of piston center with respect to vane center, and  $V_{pv}$  is relative sliding velocity between piston and vane. ( $V_{pv}$  is relative velocity between piston and vane, and is a component of sliding velocity except a component of rolling velocity.)

$F_{vn}$  includes the friction forces between the vane and vane-slot, the spring force and the force due to the discharge pressure  $p_{dis}$  acting on the tail-end of the vane. The value of  $F_{vn}$  is obtained from the mixed lubrication analysis of vane sliding surface<sup>(4-5)</sup>.

## 2.4 Friction force due to oil film viscosity between piston and crank

The sliding section between the rolling piston and crank is regarded as a hydrodynamic journal bearing. Figure 2 shows the analysis model of rolling piston and crank lubrication. The oil film pressure between piston and crank  $p$  is calculated by the Reynolds equations as follows<sup>(1)</sup>:

$$\frac{\partial}{r_i^2 \partial \theta} \left( \frac{h^3}{\eta} \frac{\partial p}{\partial \theta} \right) + \frac{\partial}{\partial z} \left( \frac{h^3}{\eta} \frac{\partial p}{\partial z} \right) = 6\omega_c \frac{\partial h}{\partial \theta} + 12 \frac{\partial h}{\partial t} \quad (5)$$

$$h = c \times (1 + \varepsilon_c \cos(\theta - \varphi)) \quad (6)$$

where,  $h$  is oil film thickness between piston and crank,  $p$  is oil film pressure between piston and crank,  $\eta$  is viscosity of oil,  $\theta$  is rotational angle of crank from vane centerline,  $\varphi$  is angle from vane center line to minimum oil film position,  $\omega_c (= \omega - \omega_p)$ ,  $\omega$  is angular velocity of crank) is relative angular velocity between crank and piston,  $c$  is radial clearance between piston and crank, and  $\varepsilon_c$  is eccentricity of crank with respect to piston center.

And the friction force due to the oil film viscosity between piston and crank  $F_c$  is obtained as follows.

$$F_c = \iint \left( \frac{\eta r_i \omega_c}{h} - \frac{h}{2r_i} \frac{\partial p}{\partial \theta} \right) r_i d\theta dz \quad (7)$$

#### 2.4 Friction forces on both end surfaces of piston

The friction forces and moments on both end surfaces of piston are expressed as follows <sup>(6)</sup>:

$$F_s = \frac{8\pi\eta\omega_p(r_o^3 - r_i^3)}{3c_s} \quad (8)$$

$$M_s = \frac{2\pi\eta\omega_p(r_o^4 - r_i^4)}{c_s} \quad (9)$$

where,  $F_s$  is friction forces on both end surfaces of piston, and  $c_s$  is clearance between piston end and bearing end.

#### 2.5 Friction force due to refrigerant viscosity between piston and cylinder

The friction force due to refrigerant viscosity between the piston and the cylinder is calculated by the friction equation of Reynolds as follows <sup>(7)</sup>:

$$F_r = 2.4677\eta_g r_o \omega w_c \sqrt{\frac{2r_e}{c_p}} \quad (10)$$

where,  $\eta_g$  is viscosity of refrigerant,  $w_c$  is width of cylinder,  $r_e$  is equivalent curvature radius of  $r_o$  and  $r_c$  ( $r_c$  is inner radius of cylinder), and  $c_p$  is clearance between piston and cylinder.

### 3. ANALYSIS PROCEDURE AND CONDITIONS

#### 3.1 Analysis Procedure

In the rotary compressor, the external force and the moment acting on piston are varied owing to the crank's eccentric rotation. Therefore, Eqs. 2, 7, 8, 10 are solved for every crank angle  $\psi$ , and each friction force acting on piston is obtained. The values of such friction forces are assigned to Eq. 1, and the angular velocity of piston rotation  $\omega_p$  is calculated. This calculation is performed through one revolution of the crank, and the variations of  $\omega_p$  through one revolution of the crank are obtained.

#### 3.2 Analysis conditions

The analysis conditions are shown in Table 1. The radius of cylinder is 19.5mm. The outer radius of rolling piston is 15.9mm. The eccentricity of crank is 3.6mm. The calculations were executed for the case of different pressure difference, different rotating frequency and different weight of rolling piston. The difference of weight of rolling piston is expressed with the difference of dimensionless density of rolling piston  $\rho/\rho_r$  ( $\rho$  is the density of rolling piston,  $\rho_r$  is the density of cast iron). Figure 3 shows the variation of refrigerant pressure in the compression chamber through one revolution of the crank, obtained by using the analysis condition. In this calculation, the effect of over-compression is considered.

### 4. RESULTS AND DISCUSSION

#### 4.1 Analysis results

Figure 4 shows the analysis results when the discharge pressure  $p_{dis}$  is 2.54MPa, the suction pressure  $p_{suc}$  is 0.87MPa and the rotating frequency  $f_0$  is 60Hz. The horizontal axis is the crank angle  $\psi$ . Figure 4(a) and (b) shows the variations of angular velocity of piston rotation  $\omega_p$  and the relative sliding velocity  $V_{pv}$  between piston and vane, through one revolution of the crank respectively. Figures 4(c) and (d) show the variations of four kinds of friction forces acting on rolling piston through one revolution of the crank.

As shown in Fig.4(b), the value of  $V_{pv}$  is positive from the crank angle  $\psi=0^\circ$  to  $\psi=40^\circ$  and from  $\psi=210^\circ$  to  $\psi=360^\circ$ . It means that the side wall of rolling piston moves in the direction in which the crank shaft rotates at the vane top. The value of  $V_{pv}$  is negative from  $\psi=40^\circ$  to  $\psi=210^\circ$  as shown in Fig.4(b). It means that the side wall of rolling piston moves contrary to the direction in which the crank shaft rotates at the vane top. Therefore, it is found

that the direction in which friction force between the side wall of rolling piston and the vane top  $F_{vt}$  acts on the rolling piston turns over at  $\psi=40^\circ$  and  $\psi=210^\circ$ . As a result, the value of  $F_{vt}$  becomes negative from positive at  $\psi=40^\circ$  and becomes positive from negative at  $\psi=210^\circ$  as shown in Fig.4(c). The peaks of  $\omega_p$  occur at  $\psi=40^\circ$  and  $\psi=210^\circ$  as shown in Fig.4(a). This is because the direction in which  $F_{vt}$  acts on the rolling piston turns over at  $\psi=40^\circ$  and  $\psi=210^\circ$ . Moreover, as shown in Fig.4(c) and (d), it can be seen that  $F_{vt}$  is larger than other friction forces. As a result, it is found that the influence of  $F_{vt}$  on the rotating motion of rolling piston is the largest among the four kinds of forces acting on the rolling piston.

#### 4.2 Effects of pressure difference

Figure 5 shows the analysis results of the cases when the pressure difference  $\Delta p$ (= discharge pressure  $p_{dis}$  - suction pressure  $p_{suc}$ ) is 1.17MPa, 1.67MPa, 2.17MPa, 2.67MPa, 3.17MPa. Then, the rotating frequency  $f_0$  is 60Hz. Figure 5(a) and (b) shows the variations of angular velocity of piston rotation  $\omega_p$  and the relative sliding velocity  $V_{pv}$  between piston and vane, through one revolution of the crank respectively. Figure 5(c) shows the variation of absolute value of the friction force between side wall of rolling piston and vane top  $F_{vt}$  through one revolution of the crank. As shown in Fig. 5(a), the absolute value of  $\omega_p$  increases when  $\Delta p$  increases. As shown in Fig. 5(b), the absolute value of  $V_{pv}$  decreases when  $\Delta p$  increases. This is because  $F_{vt}$  which controls the rotating motion of piston to stop the slip of the piston at the vane top increases when  $\Delta p$  increases as shown in Fig. 5(c). Therefore, when the pressure difference is high ( $\Delta p = 3.17\text{MPa}$ ), the sliding and rolling condition between the piston and the vane top is in pure rolling condition ( $V_{pv}=0$ ) through one revolution of the crank.

#### 4.3 Effects of rotating frequency

Figure 6 shows the analysis results of the cases when the rotating frequency  $f_0$  is 30Hz, 45Hz, 60Hz, 75Hz and 90Hz. Then, the discharge pressure  $p_{dis}$  is 2.54MPa, and the suction pressure  $p_{suc}$  is 0.87MPa. Figures 6(a) and (b) shows the variations of angular velocity of piston rotation  $\omega_p$  and the relative sliding velocity  $V_{pv}$  between piston and vane, through one revolution of the crank respectively. Figure 6(c) shows the variation of friction force due to the oil film viscosity between piston and crank  $F_c$  through one revolution of the crank. As shown in Fig. 6(a), the average of  $\omega_p$  through one revolution of the crank increases when  $f_0$  increases. This is because  $F_c$  acting on the piston in the direction in which the crank shaft rotates increases when  $f_0$  increases as shown in Fig. 6(c). As shown in Fig. 6(b), the absolute value of  $V_{pv}$  increases when  $f_0$  increases. This is because the inertia force of piston rotation increase by increasing  $\omega_p$ , and the effect of the friction force between the side wall of rolling piston and the vane top  $F_{vt}$ , which restrains the slip of piston at the vane top, decreases.

#### 4.4 Effects of weight of rolling piston

Figure 7 shows the analysis results of the cases when the dimensionless density of rolling piston  $\rho/\rho_r$  is 0.5, 0.75, 1.0 and 1.25. Then, the discharge pressure  $p_{dis}$  is 2.54MPa, the suction pressure  $p_{suc}$  is 0.87MPa, the rotating frequency  $f_0$  is 60Hz. Figure 7(a) shows the variation of angular velocity of piston rotation  $\omega_p$  through one revolution of the crank. Figure 7(b) shows the variation of relative sliding velocity between piston and vane  $V_{pv}$  through one revolution of the crank. As shown in Fig.7(a) and 7(b), the absolute value of  $\omega_p$  increases and the absolute value of  $V_{pv}$  decreases when the weight of rolling piston decreases. This is because the effects of friction forces acting on the rolling piston increase relatively owing to the reduction of the mass moment of inertia of piston  $I$ , with the increase in the effect of  $F_{vt}$ , which restrains the slip of piston at the vane top, being especially marked. As a result, it is clarified that  $V_{pv}$  can be reduced by reducing the weight of rolling piston. For high-rotating-frequency region, the reduction of  $V_{pv}$  contributes to the reduction of PV value between the piston and the vane top, and it contributes to the prevention of scuffing between the piston and the vane top and the improvement of reliability in rotary compressors.

## 5. CONCLUSION

The numerical analysis for the rotating motion of rolling piston has been performed in order to investigate the characteristics of rotating motion of rolling piston in a rotary compressor. The following results have been obtained.

1. The influences of the friction force between the side wall of rolling piston and the vane top on the rotating motion of rolling piston is the largest among the four kinds of forces acting on rolling piston.

2. The friction force between the side wall of rolling piston and the vane top  $F_{vt}$  controls the rotating motion of rolling piston to stop the slip of the piston at the vane top. Therefore, when the pressure difference is high, because the absolute value of  $F_{vt}$  is high, the sliding and rolling condition between the piston and the vane is in pure rolling condition through one revolution of the crank.
3. The average of the angular velocity of piston rotation through one revolution of the crank increases when the rotating frequency increases. This is because friction force due to the oil film viscosity between piston and crank increases.
4. The relative sliding velocity between piston and vane  $V_{pv}$  can be reduced by reducing the weight of rolling piston. For high-rotating-frequency region, the reduction of  $V_{pv}$  contributes to the reduction of PV value between the piston and the vane top, and it contributes the prevention of scuffing between the piston and the vane top and the improvement of reliability in rotary compressors.

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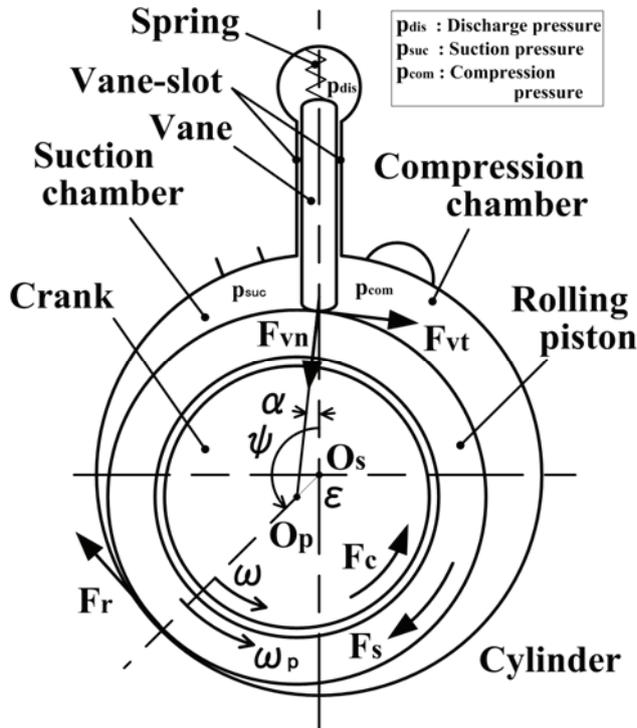


Figure1. Brief drawing of compression mechanism in rotary compressor

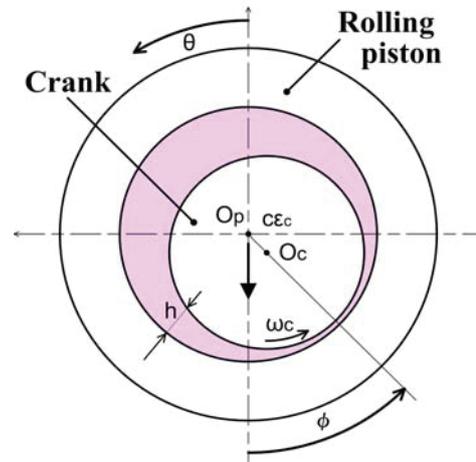


Figure2. Analysis model of rolling piston and crank lubrication

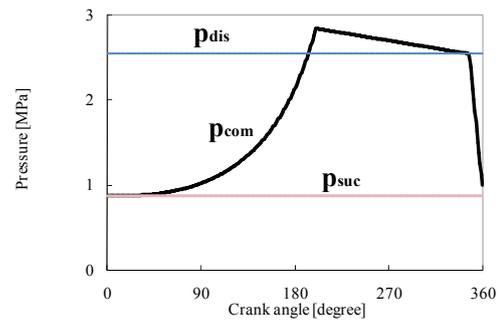
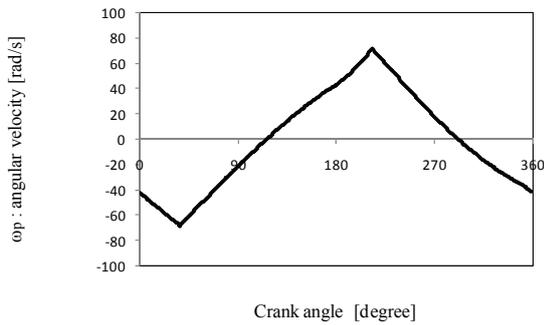


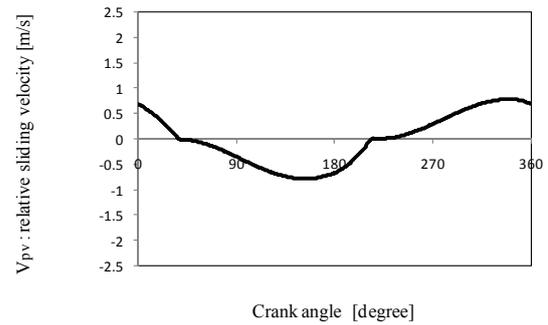
Figure3. Pressure of refrigerant in compression chamber

Table 1 Analysis conditions

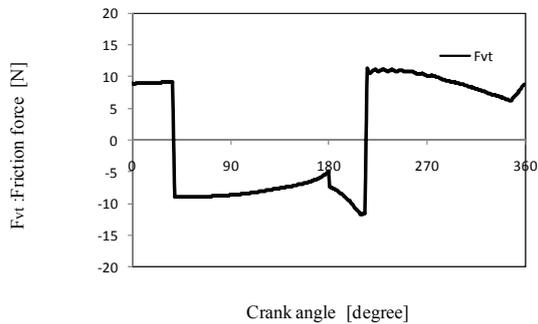
Radius of cylinder [mm]	19.5
Outer radius of piston [mm]	15.9
Inner radius of piston [mm]	10.6
Eccentricity of crank, $\epsilon$ [mm]	3.6
Oil viscosity, $\eta$ [ $10^{-3}$ Pa · s]	2.8
Rotating Frequency, $f_0$ [Hz]	30, 45, 60, 75, 90
Discharge pressure / Suction pressure, $p_{dis} / p_{suc}$ [MPa]	2.04/0.87 ( $\Delta p=1.17$ )
	2.54/0.87 ( $\Delta p=1.67$ )
	3.04/0.87 ( $\Delta p=2.17$ )
	3.54/0.87 ( $\Delta p=2.67$ )
	4.04/0.87 ( $\Delta p=3.17$ )
Dimensionless density of rolling piston, $\rho/\rho_r$	0.5, 0.75, 1.0, 1.25



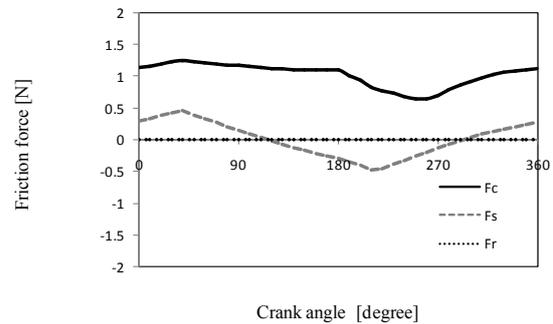
(a) Angular velocity of piston rotation



(b) Relative sliding velocity between piston and vane

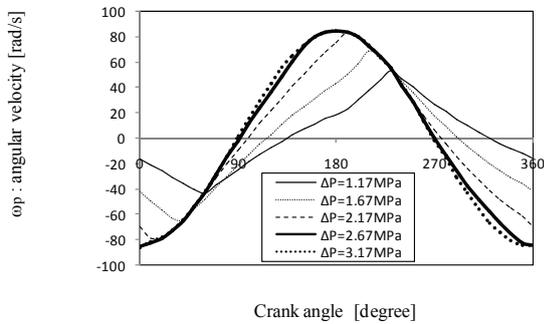


(c) Friction force between the side wall of rolling piston and the vane top

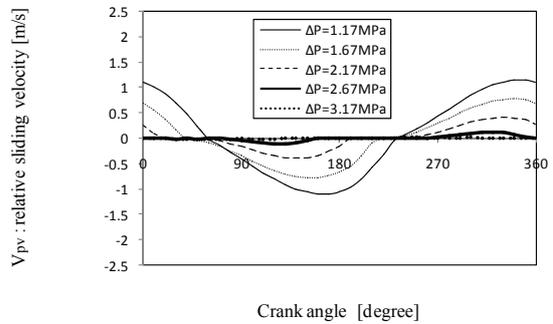


(d) Friction forces which act on rolling piston

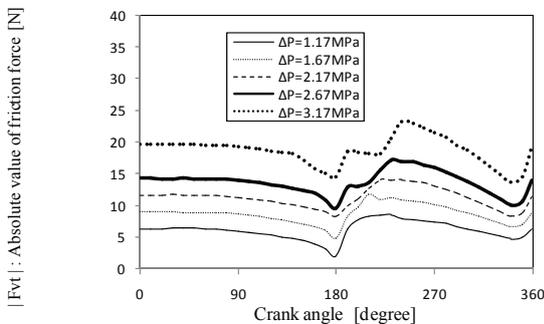
Figure4. Analysis results ( $p_{dis}/p_{suc}=2.54/0.87\text{MPa}$  ( $\Delta p=1.67\text{MPa}$ ),  $f_0=60\text{Hz}$ )



(a) Angular velocity of piston rotation

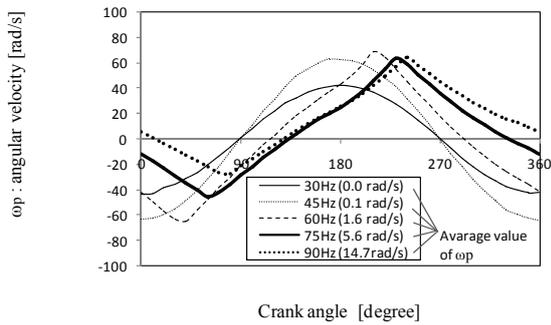


(b) Relative sliding velocity between piston and vane

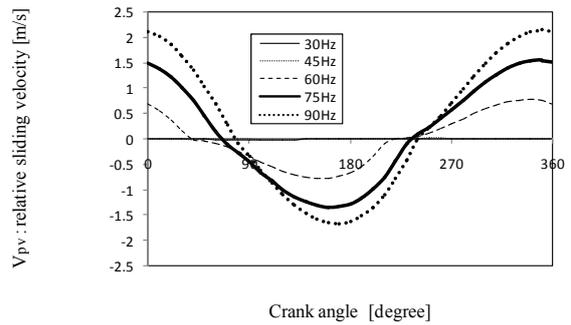


(c) Absolute value of friction force between the side wall of rolling piston and the vane top

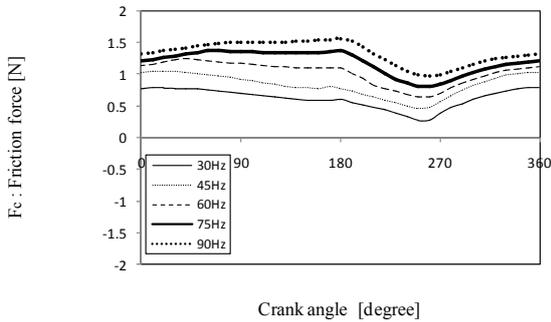
Figure5. Effects of pressure deference ( $f_0=60\text{Hz}$ )



(a) Angular velocity of piston rotation

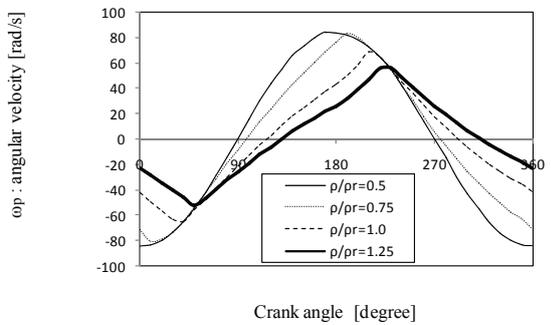


(b) Relative sliding velocity between piston and vane

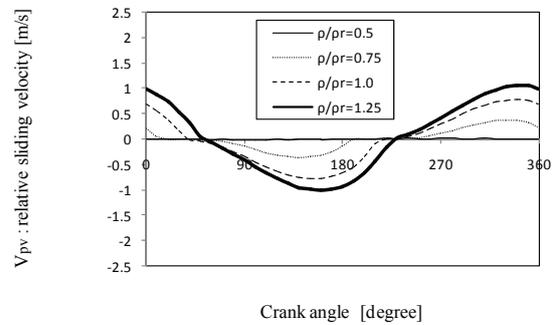


(c) Friction force due to the oil film viscosity between piston and crank

Figure6. Effects of rotating frequency ( $p_{dis}/p_{suc} = 2.54/0.87\text{MPa}$  ( $\Delta p = 1.67\text{MPa}$ ))



(a) Angular velocity of piston rotation



(b) Relative sliding velocity between piston and vane

Figure7. Effects of weight of rolling piston ( $p_{dis}/p_{suc} = 2.54/0.87\text{MPa}$  ( $\Delta p = 1.67\text{MPa}$ ),  $f_0 = 60\text{Hz}$ )