Analysis of the Dynamic Characteristics of a Variable Speed Compressor

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ANALYSIS OF THE DYNAMIC CHARACTERISTICS OF A VARIABLE SPEED COMPRESSOR

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ABSTRACT

A dynamic simulation model of a rolling piston type rotary compressor is developed to predict the dynamic characteristics of a variable speed compressor. The model is based on the principles of conservation, real gas equations, kinematics of the crankshaft and roller, mass flow loss due to leakage, and heat transfer. For the computer simulation of the compressor, the experimental data were first obtained with the motor performance tests at various operating frequencies. Using the model developed, the compression performance, friction loss, mass flow loss and heat transfer loss etc. were estimated as a function of the crankshaft speed in a variable speed compressor.

Since the transient state of the compressor strongly depends on the system, the transient model developed for a variable speed compressor is combined with a transient system simulation program to get transient variations of the compression process in the system.

INTRODUTION

The variable speed compressor is designed to change the motor speed compared to a constant speed compressor. The efficiency of a constant speed motor is dependent on the load of the motor. However, the efficiency of variable speed motor is changed with not only the load of the motor but also the driving frequency. Therefore, the efficiency of the compressor with a variable speed motor in the compression and suction process will be continuously changed with driving frequency and load.

This study is concerned with developing a dynamic program to predict the effect of the frequency on compressor performance, friction loss, mass flow loss, and heat transfer loss etc.

Several computational models for the compressor are based on the polytropic process. In this study, the model is developed with conservation equations and a detailed simulation model of heat transfer, friction and mass losses.

The present study characterizes like as follows.

1) The model in this study is a dynamic analysis model based on the principles of the mass and energy conservation equations. Therefore, the variations of temperature and pressure of the refrigerant can be calculated with the rotation of the crankshaft.
2) The refrigerant is considered as a real gas rather than an ideal gas.
3) All of the efficiencies of the compressor are calculated or measured. There is no assumption about the efficiencies.
4) With analysis of the dynamic equation for the moving part, it is possible to calculate accurate friction loss although there is angular velocity variation in the crankshaft and roller.
5) This model considered variation of the heat transfer coefficient and area with rotation of crankshaft.
6) The variable motor efficiency was measured experimentally. Therefore, the error of the motor efficiency is reduced and it is possible to predict the motor efficiency with the variation of driving frequency.

MATHEMATICAL MODELLING

The rolling piston type rotary compressor in this model is widely used for controlling the speed of the compressor due to the simple mechanism. The calculations are performed by increasing the angle of the crankshaft with small time intervals. The refrigerant temperature, pressure, heat transfer, and mass loss can be calculated by analyses of the compression and suction process in the cylinder at the same time. The mechanical loss, indicated work, volumetric efficiency etc. also can be calculated at the end of the one cycle. In order to more accurately calculate the losses in the compression process, dynamic analysis of the compressor is necessary. In this model, the dynamic equations of the mechanical part of the compressor are used to predict accurate friction loss with calculated velocity of the moving parts.

Governing Equations

The refrigerant in the cylinder is being sucked into the suction chamber simultaneously with compressing in the compression chamber. The rate of change of mass in the control volume is given by equation (1) and temperature variation equation (2) that is obtained by energy conservation equation and real gas equation.

$$\frac{dm}{dt}=\sum \dot{m}_{in}-\sum \dot{m}_{out} \tag{1}$$

$$\frac{dT}{dt}=\left(\frac{dQ_c}{dt}+\sum\frac{dm_i(i_i-i_c)}{dt}-\sum\frac{dm_o(i_o-i_c)}{dt}-m_c\left[\left(\frac{\partial i}{\partial v}\right)_T-v\left(\frac{\partial P}{\partial v}\right)_T\frac{dv}{dt}\right]\right)\Bigg/\left(m\left[\frac{\partial i}{\partial T}\right]_v-v\left(\frac{\partial P}{\partial T}\right)_v\right) \tag{2}$$

The rate of change of pressure, which is a function of mass, temperature and specific volume is calculated from equation (3).

$$\frac{dP}{dt}=P\left(\frac{1}{m_c}\frac{dm_c}{dt}+\frac{1}{T}\frac{dT}{dt}-\frac{1}{V}\frac{dV}{dt}\right) \tag{3}$$

The refrigerant flow rate discharged through the compressor discharge line from the compressor can be achieved by conservation equation of mass and energy.

$$\dot{m}_{cano}=\dot{m}_{in}+\dot{m}_{lv}+\frac{M_l}{v_v}\frac{dv_v}{dt}+\frac{M_v}{v_v}\frac{dv_l}{dt}+\frac{v_v}{v_l}\cdot\dot{m}_{lv} \tag{4}$$

Where, $\dot{m}_{lv}$ is rate of evaporating refrigerant from liquid to vapor at saturation state i.e.

$$\dot{m}_{lv}=\left(\dot{m}_{in}(i_{in}-i_v)+\frac{dQ_{cyl}}{dt}+\frac{dQ_{mot}}{dt}-\frac{dQ_{can}}{dt}-M_v\frac{di_v}{dt}-M_l\frac{di_l}{dt}+V\frac{dP}{dt}\right)\Bigg/(i_v-i_l) \tag{5}$$

Dynamic Equation of Crank Shaft

Because the angular velocity of the motor is changed with motor and load torque, angular velocity of the crankshaft with revolution is changed with angle of crankshaft. If the motor torque and load torque are assumed constant with time, the angular velocity of the crank shaft obtained by equation (6)[Matsuoka and Nagamoto(1988)].

$$I_s\frac{dt}{d\omega}=T_M-T_L \tag{6}$$

Where, I_s is inertia moment of crankshaft, T_M is motor torque.

The load acting on the compressor motor is the same as the total moment exerted on the crankshaft[Yanagisawa, et. al(1984) and Mitsunaga, et. al.(1987)] as compressor start. The load torque is obtained by following equation.

$$T_L=M_f+M_j+M_t \tag{7}$$

Moment acting on the crankshaft by the roller, M_f The forces acting on crankshaft by the roller are composed of 1) radial direction force (F_{vn}) on center of the roller imposed by vane, 2) friction force $(f_v F_{vn})$ caused by aforementioned force, 3) force by pressure difference between compression and suction chamber and 4) centrifugal force from orbital motion of roller [Imaich, et. al.(1982)]. Among the forces, the force by the pressure difference is the highest force acting on the crankshaft. Table 1 shows the force exerted on the crankshaft.

$$M_f=e\cdot F_p\cdot\sin\left(\frac{\alpha+\theta}{2}\right)-e\cdot F_{vn}\cdot\sin(\alpha+\theta)+e\cdot f_{vn}\cdot\cos(\alpha+\theta) \tag{8}$$

Moment of thrust bearing, M_t

$$Mt=\pi\mu_t\frac{\omega}{2\delta}\left(2R_j^4-R_s^4\right) \tag{9}$$

Table 1 Forces exerted on the crankshaft by the roller

Force of vane acting on center of the roller	$F_{vn}=\left[-l_s(F_d+F_k+F_m)+f_{vs}F_h(l_s+x+t\cdot f_{vs})\right]/ \begin{bmatrix}-(\cos\alpha+f_v\sin\alpha)(l_s+2f_{vs}R_v\sin\alpha)+f_{vs}(\sin\alpha-f_v\cos\alpha)\cdot\\ \left[2x+l_s+t\cdot f_{vs}-2R_v(1-\cos\alpha)\right]\end{bmatrix}$
Force by pressure difference between the suction and discharge chamber	$F_p=2R_r h\sin[(\theta+\alpha)/2][P_d-P_s]$

Moment of journal bearing, M_j The journal bearing consists of a main bearing and sub bearing sustained by the spring of the vane. Friction force caused by the weight of the rotor and crankshaft is applied to the journal bearing as a moment[Itami, et. al.(1982)].

$$M_j=\frac{0.66\mu_j hR_s^3\omega}{\delta} \tag{10}$$

Dynamic Equation of Roller

The roller rotates centering on its center regardless of rotation of crankshaft. The motion of the roller is determined only by the moment that is acting on the roller. The dynamic equation of the roller can be determined by the moment equation to the center of it as in table 2. The friction between roller and cylinder is negligible compared to other terms.

$$\frac{d\omega}{dt}=2g\left[M_k-2M_r-f_vR_rF_{vn}\right]/\left[\gamma\pi l_r\left(R_r^4-R_e^4\right)\right] \tag{11}$$

Table 2 Moment acting on the roller

Moment by friction force between crank fin and inner surface of the roller	$M_k=2\pi\mu hR_e^3\omega_e/\delta$
Moment by friction force between roller and cylinder head	$M_r=\pi\mu_o\left(R_r^4-R_e^4\right)\omega_r/\delta$

Mass Flow Losses

Suction gas heating, oil circulation, re-expansion at residual volume and back flow and leakage through the discharge valve affect mass flow losses in the compressor. The mass flow loss by suction gas heating can be neglected in the low-pressure type rotary compressor due to the refrigerant being directly sucked to the cylinder from the accumulator or evaporator. The effect of the oil is also neglected in this study. During the compression process, the mass flow loss comes out through contact surface between the roller and cylinder, the vane and the roller and roller and cylinder head etc. The calculation of mass losses are based on the following assumptions: 1) The mass losses only came from pressure difference, 2) The effect of Coutte flow due to the velocity difference of the stationary part and the moving part is neglected, 3) The type of path is assumed as a converge-diverge nozzle, 4) the refrigerant losses through small paths are critical flow. The flow area in equation (12) is calculated at each angle of rotation of the crankshaft.

$$\dot{m}_{loss}=P_uA\sqrt{\frac{2k}{(k-1)RT_u}\left[r_c^{2/k}-r_c^{k+1/k}\right]} \tag{12}$$

Heat Transfer

In this study, the refrigerant discharged from the cylinder is accumulated at the compressor can. The heat transfer to the refrigerant in the can is accompanied by the motor heat(Q_{mot}), heat transfer from the cylinder (Q_{cyl}) and heat transfer from the can(Q_{can}). The motor heat and heat transfer from the cylinder can be easily caculated by motor efficiency and heat transfer equation. The heat transfer from the cylinder comes out through the cylinder, roller, cylinder head, suction line and vane. The heat transfer is calculated by the following equation (13) with coresponding heat transfer area[Yanagisawa, et.al.(1983)].

$$Q_{cyl}=\frac{h}{\omega_s}\int_{\beta}^{\beta+2\pi}f(\theta)\cdot d\theta \tag{13}$$

Where, h is heat transfer coefficient, ω_s is angular velocity of the crankshaft, and $f(\theta)$ is function of heat transfer and temperature difference.

RESULTS AND DISCUSSION

Experimental Results of Motor Performances

The motor dynamometer test was conducted to get the motor efficiency. The motor efficiency, motor torque and speed variation with variation of load are measured at each driving frequencies.

Motor slip is the ratio of actual rotational frequency to driving frequency that is applied to the motor. If the slip ratio is small, it means the driving frequency is similar to motor rotational frequency. Figure 1 shows rotational frequency-efficiency diagram at 60Hz. The efficiency increased with slip ratio until slip ratio became approximately 0.04(3456rpm). After the efficiency reached the maximum efficiency, the efficiency decreased with slip ratio. If the compressor using this motor as its driving device is designed as the load match with this rotational frequency(3456rpm) at 60Hz, the compressor efficiency should be maximized. The motor torque is linearly proportional to slip. In addition, the motor output that is a function of torque and rotational frequency shows the same trend as the motor torque.

As the experimental results show, if the driving frequency is changed from 30Hz to 60Hz, the efficiency at 25kg·cm motor torque is increased from 67% to 80% approximately. Nevertheless, as the driving frequency increases over 60Hz, the efficiency of the motor maintains a constant value up to the 110Hz driving frequency. Therefore, the motor efficiency is lower at low frequency region than high frequency region. Moreover, if the driving frequency reaches the rating frequency (60Hz), there is no big difference of the efficiency at higher frequencies over the rating point.

Compression Characteristics

The analyses are conducted by varying the crankshaft angle through one complete suction and compression process. In this analysis, the operating condition for the compressor is 7.2°C evaporating temperature and 54.4°C condensing temperature. Figure 2 shows a pressure variation with crankshaft angle. The suction process was finished at 360° and the compression process starts at 397.75° due to the location of the suction port(37.75°). If the angle became over 300°, the pressure was a little higher than the evaporating pressure due to the re-expansion of the compression process(673°). Although the pressure drop during the suction process is small, the suction losses are not small compare to discharge losses in the compression process due to the prolonged suction process.

The load and motor torque are shown in figure 3, which is calculated by the dynamic equation of the roller. The difference of the load torque and motor torque makes an angular velocity variation as in figure 4. From the angular velocity variation, the friction loss in the compressor can be correctly calculated. The slip ratio shown in the figure 4 is in inverse proportion to the angular velocity of the crankshaft. The motor efficiency can be calculated at specified angular velocity using the obtained slip ratio. Because the angular velocity and friction losses have been connected with each other, these should be calculated by iteration method. Figure 5 shows efficiencies in the compressor obtained by dynamic equations. Due to the slip ratio of the crankshaft at this operating condition being close to 0.095, the motor efficiency is very low.

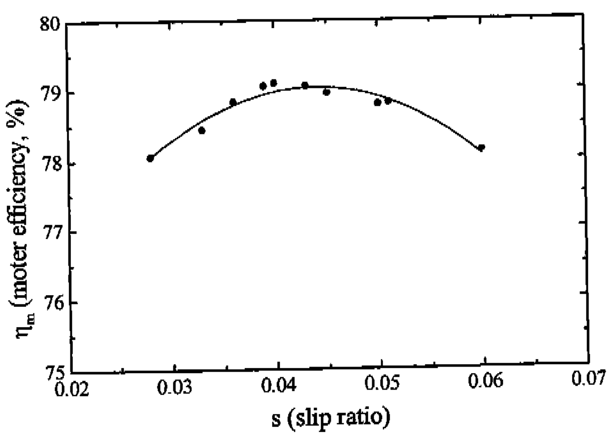

Figure 1 Variation of motor efficiency with slip ratio at rated frequency(60Hz)

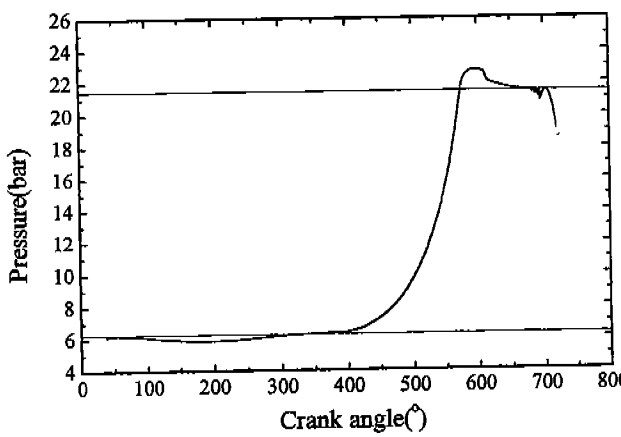

Figure 2 Variation of refrigerant pressure in cylinder with angle of crankshaft

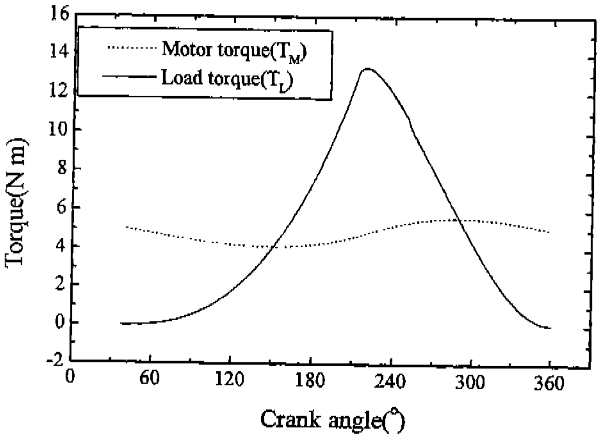

Figure 3 Variation of load torque and motor torque with angle of crankshaft

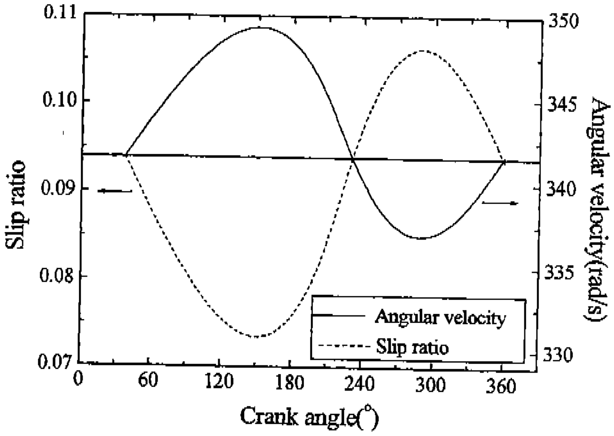

Figure 4 Variation of slip ratio and angular velocity with angle of crankshaft

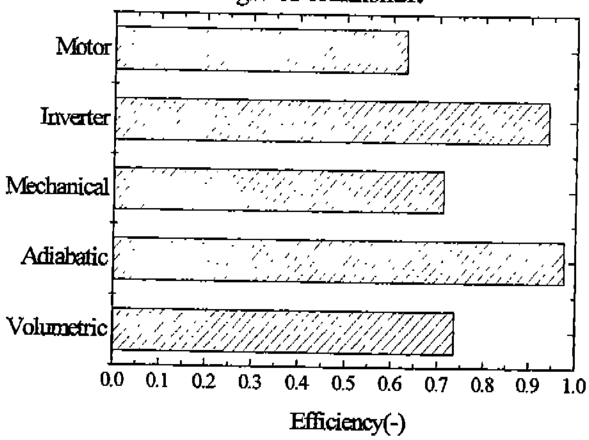

Figure 5 Comparison of various efficiencies of a variable speed compressor at 60Hz

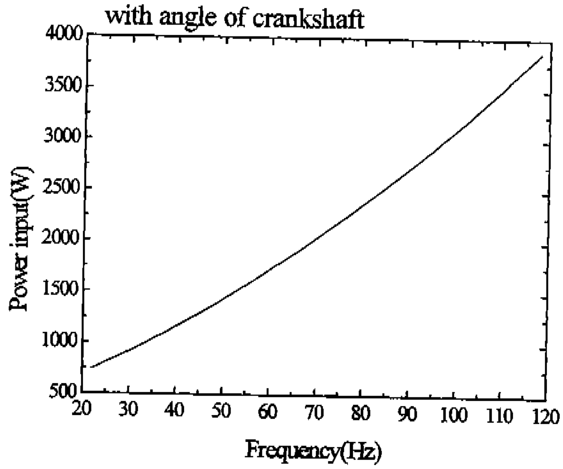

Figure 6 Variation of power input of compresor with driving frequency

The variation of the compressor performance with driving frequency is also calculated. The results shown in figure 6 indicate that the power input of a variable compressor is linearly proportional to the frequency. As the results of the calculation shows, the volumetric efficiency decreases slowly when the frequency deviates from the rated condition. The indicated efficiency is almost constant for low frequency; however, it decreases due to the increase of the suction and discharge loss as the frequency increases. The mechanical efficiency is inversely proportional to the angular velocity of the crankshaft, it decreases as the frequency of the compressor increases.

Startup Characteristics of Compressor in conjunction with System Simulation Program

Most of the studies on compressors are focused on the compressor performance at steady state. The dynamic simulation program that is developed in this study is applied to predict transient variation of the compression process during startup period. Because the transient phenomena depend on the system, the dynamic compressor model is combined with a transient system simulation program. The refrigerant pressure, work done by refrigerant and mass of the refrigerant in the cylinder with angle of crankshaft are shown in figure 7 to 9. In these figures, the variation of the parameters with time is depending only on system performance. The pressure variation in figure 7 shows that the discharge loss at the early stage of start-up is significantly large due to the discharge valve opening so early(450°) and lasting for a long period(700 °). As time goes on, the discharge valve opens at 570 ° angle of crankshaft and becomes the steady state value.

The work of the refrigerant during suction and discharge processes are shown in figure 8 as the energy term. Negative means work done by refrigerant and positive is the work acting on the refrigerant. Since in the early state of the compressor start-up the suction pressure is higher than steady state, the work done by

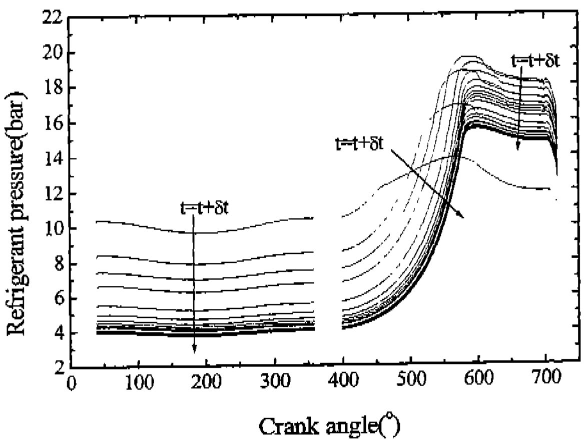

Figure 7 Variation of refrigerant pressure in cylinder with angle of crankshaft from starting of compressor

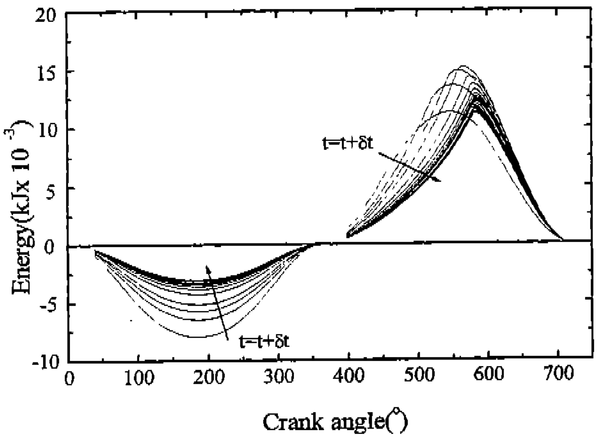

Figure 8 Variation of energy transfer in cylinder with angle of crankshaft from starting of compressor

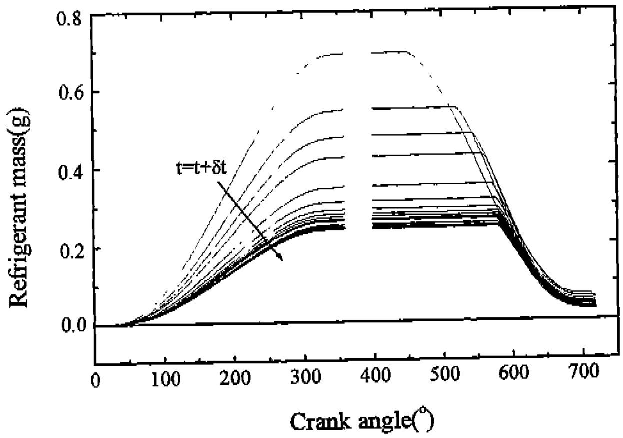

Figure 9 Variation of refrigerant mass in cylinder with angle of crankshaft

refrigerant to compressor is also higher. The refrigerant temperatures in the suction and discharge chambers are strongly dependent on the system pressure variation.

The refrigerant mass during the suction and discharge process varies as in figure 9. If the compressor is started, the mass of refrigerant in the cylinder is large due to the large suction loss, which makes a large pressure difference between the evaporating and suction chamber pressures.

CONCLUSIONS

A dynamic simulation model for the variable speed compressor is presented. The model is based on the physical principles of conservation of mass and energy including the dynamic equation of the roller and crankshaft. In addition, the model considered most of the parameters that affect the compressor performance such as the leakage, friction, heat transfer and re-expansion of compressed refrigerant etc. Using this model, the compressor efficiency and energy losses can be predicted at various compressor-operating frequencies. It is also possible to predict the variation of compressor characteristics with time at start-up of the system.

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