2002

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ABSTRACT

The technique of the modeling and analysis procedures of reciprocating compressor is described which predict the basic performance of the compressor, compute the pressure pulsations in cylinder and suction line and simulate valve motions. The polytropic process assumption was used for modeling cylinder thermodynamic process and four-pole transfer matrix for gas pulsations and spring-mass model for valve dynamics. The commercial software ANSYS and SYSNOISE were also used for numerical analysis of valve properties and impedance of suction line which can be used for a boundary condition. And comparing the simulated data such as energy efficiency ratio, cylinder pressure, suction pressure, etc to the experimental data, we confirmed the feasibility of developed technique.

NOMENCLATURE

k : Specific heat ratio

ρ : density

R : Gas constant

c : Speed of sound

INTRODUCTION

The developments of the reciprocating compressors for high efficiency and low noise had been performed constantly, since this machine was invented. Nowadays some particular methods were required to upgrade the performance. Among them, one of the most potential means is a simulation tool which can interpret the states of compressor in more details with high accuracy.

There have been numerous researchers in these areas. W. Soedel et al[1] simulated the compressor with cylinder process and gas pulsations separately and Wambsgans[2] developed the basis of detailed simulation model without gas pulsations. Brablik[3] was the first who coupled the cylinder process and gas pulsations in the compressor line. And Elson[4] et el. employed Fourier transformation technique to connect the cylinder process and pressure pulsations in compressor line. This approach was used in many situations and modified numerously[5]. In modeling gas pulsations, W. Soedel et al[1,4,6] adapted the linearized acoustic theory because typical components in gas manifold are small compared to the smallest wave length of general interest.

In this study, we simulated the compressor with coupling the cylinder process and pressure pulsation using the Fourier transformation tool and the commercial FEM code for calculating the pressure pulsation because the linear acoustic model does not valid any more in manifold with complex geometry and the frequency we are interested in is raised, so the small geometry and components was considered to the simulations for higher accuracy.

STRUCTURE OF ANALYSIS

Modeling Compression Parts

A Refrigerant flow passage from suction pipe to discharge pipe through the compressor is composed of many elementary component such as shell cavity, suction muffler, suction port/valve, cylinder, discharge port/valve, discharge plenum, discharge silencer and loop pipe in sequence. Following these passages, temperature and pressure of refrigerant was raised to a discharge state from a suction state. These states of refrigerant are predetermined by ASHRAE conditions.

Kinematics[7]

The compression part is composed of crank arm, connecting rod, piston and cylinder. And the volume of the cylinder is expressed by the rotation angle of crank.

$$V_c = V_{dead} + \frac{\pi D_e{}^2}{4}\left(\sqrt{(R_1+R_2)^2 - e^2} + R_1 Cos(\theta) - R_2 Cos(\beta)\right), \qquad \beta = Tan^{-1}\left(\frac{R_1 Sin(\theta)+e}{\sqrt{R_2{}^2 - (R_1 Sin(\theta)+e)^2}}\right) \tag{1}$$

where V_c is volume of cylinder, R_1 is eccentricity and R_2 is con-rod pitch. The piston is offset by the amount of e from the center line of the crank.

Thermodynamics

Assuming ideal gas, one dimensional flow and polytropic process, the pressure in cylinder is given as follows;

$$P_c = P_s\left(\frac{m(t)}{\rho_s V(t)}\right)^n \tag{2}$$

where P_c is the pressure of cylinder, m is refrigerant mass in cylinder and P_s is pressure of suction port. Refrigerant mass in cylinder is conserved during compression and expansion process.

Valve Flows and Valve Dynamics[2,8]

The flow through valve port can be modeled by a simple orifice flow assuming one-dimensional isentropic process, steady flow and stagnation upstream condition. The mass flow rate through the suction port is expressed as follows.

$$\dot{m}_s = A_{sv}P_s\left[\frac{2k}{(k-1)RT_s}\right]^{1/2}\left\{f(r_s) - \left(\frac{T_s}{T}\right)^{1/2} r_s f\left(\frac{1}{r_s}\right)\right\}$$

$$f(r) = \begin{cases} \left(r_c{}^{2/k} - r_c{}^{\frac{k+1}{k}}\right)^{1/2}, & 0 < r \le r_c \\ \left[(r)^{2/k} - (r)^{\frac{k+1}{k}}\right]^{1/2}, & r_c \le r \le 1 \\ 0, & r > 1 \end{cases} \qquad r_c = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \tag{3}$$

where m_s is mass flow rate through the suction port, A_{sv} is equivalent effective flow area, r is the ratio of upstream and downstream pressure, k is specific heat ratio, T_s is suction temperature and T is cylinder temperature.

Mass flow rate is controlled mainly by the ratio of upstream and downstream pressure to an extent that is $r<r_c$ and in $r>r_c$ region, the flow is choked, so choked equation is used. The above equation can also be used for reversal flow conditions.

Effective flow area which is equivalent to the port area of orifice model is given by

$$A_v = \frac{(K_0 A_0)}{\sqrt{1+\left(\frac{K_0 A_0}{K_0 A_0}\right)^2}} \tag{4}$$

where k_0 , k_1 is contraction and expansion loss coefficient.

Suction valve and discharge valve is modeled as one-dimensional equivalent mass-spring system and the principal force bring about valve motion is the pressure difference across the valve.

$$\ddot{y} + \frac{C_{eq}}{M_{eq}}\dot{y} + \frac{K_{eq}}{M_{eq}} y = \frac{(P_s - P_c)A_F}{M_{eq}} \tag{5}$$

where y is valve lift, A_f is effective force area of the valve which can be acquired from simple force equilibrium, M_{eq} is equivalent mass of the valve, K_{eq} is equivalent stiffness of the valve, C_{eq} is damping coefficient. The commercial software ANSYS was used to calculate the effective stiffness and first resonant frequency of the valve, in which FEM elastic shell element was used with exact three-dimensional geometry.

$$A_F = (K_V A_V)^2\left(\frac{A_P}{{A_r}^2} + \frac{1}{A_0}\right) \tag{6}$$

where A_v is effective flow area and k_v is loss coefficient of the port.

Cylinder Process[9]

Cylinder process is consist of compression, discharge, expansion and suction according to the piston positions or crank angles. Compression and expansion process can be simulated by polytropic equation (eq.2) assuming isentropic process that has no heat transfer with cylinder and the other parts and the mass inside the cylinder is conserved. In discharge and suction process, refrigerant goes in and out of the cylinder through the opened valve. The mass flow rate through this suction and discharge valve port can be calculated by coupling valve flow equation and valve dynamic equation. These two equations are initial value problems with three differential equations(eq.3), which can be solved numerically by the Runge-Kutta method.

$$F = \begin{cases} \dfrac{dy}{dt} = z \\ \dfrac{dz}{dt} = \dfrac{(P_c - P_d)}{M_{eq}} A_F - \dfrac{C_{eq}}{M_{eq}} z - \dfrac{K_{eq}}{M_{eq}} y \\ \dfrac{dm}{dt} = \dot{m}_s - \dot{m}_d \end{cases} \qquad \vec{v} = \begin{cases} m \\ y \\ z \end{cases} \tag{7}$$

Three unknown variables m, y, z which indicate mass in cylinder, valve lift and valve velocity respectively are integrated with time until suction/discharge valve was closed. At every time step, the internal state of the cylinder can be deduced from the equations as mentioned so far.

Modeling Suction Parts

The main driving force of valve motion is pressure difference between the cylinder and suction part. So the pressure of the suction part influences the valve opening and closing time and also governs the valve motion. This suction pressure varies periodically with time, which is called as pressure pulsation. To simulate this time varying variable, four-pole transfer matrix model was introduced.

The pressure spectrum at one position can be acquired by production with impedance and flow rate spectrum in frequency domain. And also pressure variation can be calculated by inverse Fourier transformation of the pressure spectrum. For computing the pressure acting on the valve, we need the pressure which position is at the exit of suction muffler.

In this study we used the commercial software SYSNOISE for accurately calculating the impedance with full geometric considerations. The finite element method was used for solving acoustic pressure variation. And the following Four-pole method applied for muffler cavity and shell cavity respectively.

$$\begin{bmatrix} Q_0 \\ P_0 \end{bmatrix} = \begin{bmatrix} A_M & B_M \\ C_M & D_M \end{bmatrix} \begin{bmatrix} Q_1 \\ P_1 \end{bmatrix} \qquad A_M = \left.\frac{Q_0}{Q_1}\right|_{P_1=0},\ B_M = \left.\frac{Q_0}{P_1}\right|_{Q_1=0},\ C_M = \left.\frac{P_0}{Q_1}\right|_{P_1=0},\ D_M = \left.\frac{P_0}{P_1}\right|_{Q_1=0} \tag{8}$$

$$\begin{bmatrix} Q_1 \\ P_1 \end{bmatrix} = \begin{bmatrix} A_S & B_S \\ C_S & D_S \end{bmatrix} \begin{bmatrix} Q_2 \\ P_2 \end{bmatrix} \qquad A_S = \left.\frac{Q_1}{Q_2}\right|_{P_2=0},\ B_S = \left.\frac{Q_1}{P_2}\right|_{Q_2=0},\ C_S = \left.\frac{P_1}{Q_2}\right|_{P_2=0},\ D_S = \left.\frac{P_1}{P_2}\right|_{Q_2=0} \tag{9}$$

Where $\boldsymbol{Q}$, $\boldsymbol{P}$ is complex variables which represent for the spectrum of flow rate and pressure respectively. The Four-pole of total suction parts come from the product of four-pole of muffler cavity and shell cavity.

$$\begin{bmatrix} Q_0 \\ P_0 \end{bmatrix} = \begin{bmatrix} A_M & B_M \\ C_M & D_M \end{bmatrix} \begin{bmatrix} A_S & B_S \\ C_S & D_S \end{bmatrix} \begin{bmatrix} Q_2 \\ P_2 \end{bmatrix} = \begin{bmatrix} A_T & B_T \\ C_T & D_T \end{bmatrix} \begin{bmatrix} Q_2 \\ P_2 \end{bmatrix} \tag{10}$$

Using the anechoic boundary condition (eq. 11) at the suction pipe, the pressure and flow rate spectrum are calculated as follow

$$Q_2 = \frac{A_2}{c\rho} P_2 \tag{11}$$

$$Z = \frac{P_0}{Q_0} = \frac{C_T \dfrac{A_2}{c\rho} + D_T}{A_T \dfrac{A_2}{c\rho} + B_T} \tag{12}$$

Eq.12 indicates that the impedance is a function of geometry and properties of the medium

Analysis Procedures

The pulsation pressures in the suction lines are periodic function which fundamental period is a reciprocal of operating frequency, and also the flow rates through the valves are. So the flow rates can be represented by a Fourier series and transformed to flow rates spectrum. The product of this spectrum and previously calculated impedance will make the pressure spectrum, where the spectrums use complex form.

$$\mathbf{P} = \mathbf{ZQ} \tag{13}$$

At the beginning, the flow rates were integrated by cylinder process with constant initial suction pressure. And the compensated pulsation pressure would be acquired by summation previous pressure and newly acquired pressure fluctuations which are calculated from the inverse transform of the pressure spectrum. With this revised pressure pulsations in suction part which varies with time, more developed flow rates are calculated. These procedures are iterated until the pressure pulsations, mass flow rate and other variables are converged within specific limit.

RESULTS AND DISCUSSION

With this developed technique, we analyzed four kinds of compressors and performed the experiments to make comparison and verify the feasibilities. Those four compressors are classified by two valves with thickness *0.204* and *0.150* and two mufflers which are different each other. One muffler has one cavity with pipes and the other muffler has two cavities with pipes, which is called pipe resonator.

All the conditions and parameters used in analysis are coincident with which of experiments. And the experiments were performed in calorimeter which accuracy is about *99.5%* for energy efficiency ratio. The pressures in cylinder and in suction parts were measured by piezoelectric pressure sensors and the crank angle was indicated by the gap sensor by checking the bottom dead center of the piston.

The energy efficiency ratio result of the analysis and experiment were compared in Fig.1. They have strong correlations with correlation coefficient *0.96*. The energy efficiency ratio can be calculated by the mass flow rate divided by the work output. In more detail, the pressure variations with time inside the cylinder was also compared with the measured data in Fig.2. The analysis can accurately depict the slightly changing cylinder pressure in even suction process. And also the pressure pulsations in suction line have a good agreement with the acquired experimental results (Fig. 3) and even also pressure spectrum characteristics were predicted precisely within *3khz* and all of those results were the basis of the accurate performance prediction.(Fig. 4)

CONCLUSIONS

The technique to predict performance of reciprocating compressor was developed which estimate the energy efficiency ratio, pressures in cylinder and suction line, valve motions, etc. By applying this technique to four different compressors, confirmed the feasibilities. The pressures in cylinder and suction pressure pulsation have good agreements with the experimental data, and even though the spectrums of pressures coincided well. The commercial FEM codes, ANSYS and SYSNOISE was introduced to valve dynamics and suction line impedances with full three-dimensional geometric considerations, so attained *96%* performance prediction accuracy.

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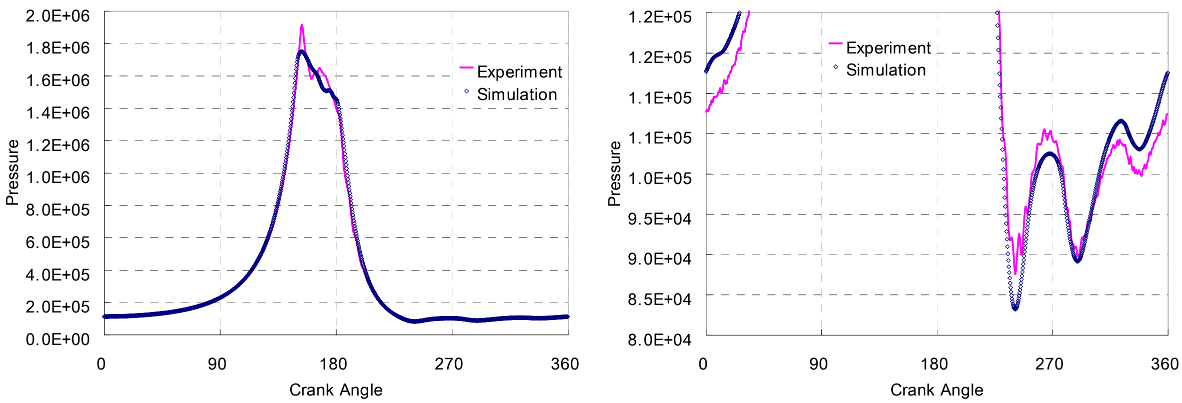

(a) Overall Range (b) Low Pressure Detail

Fig.2 Pressure in Cylinder

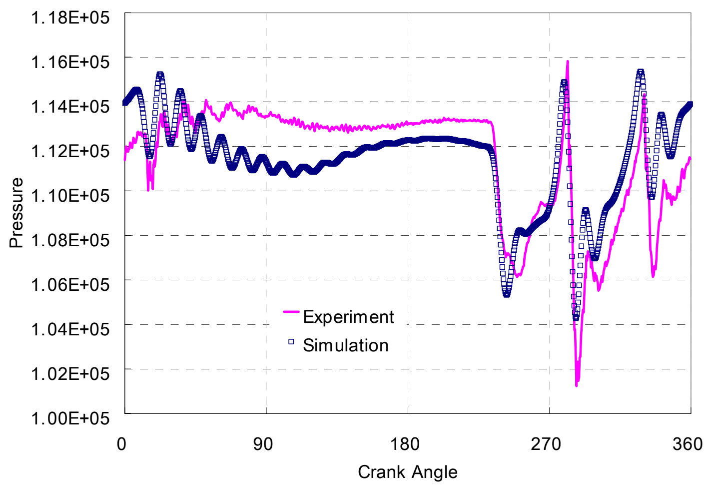

Fig.3 Suction Pressure Pulsation

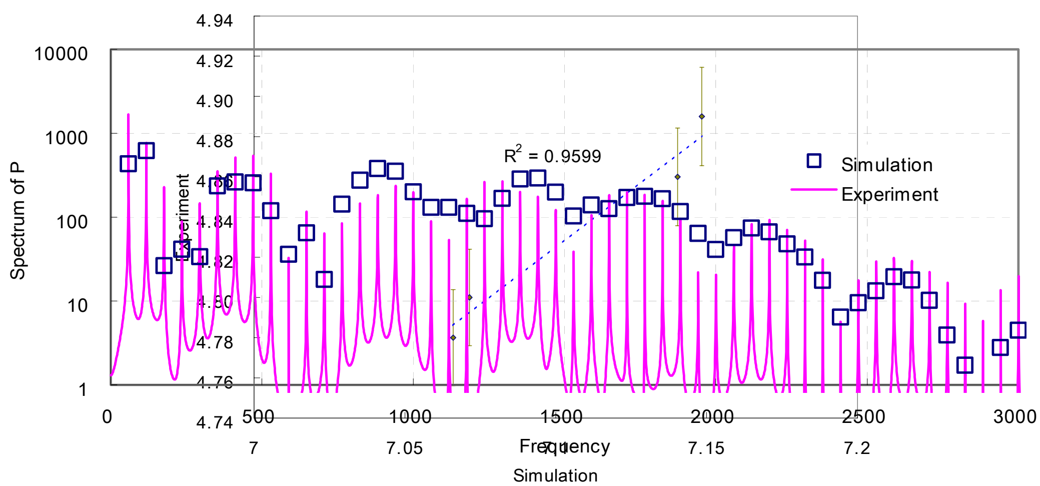

Fig.4 Spectrum of Suction Pressure

Figure 1: Comparison of measured refrigerator efficiencies.