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The Estimation of Compressor Performance Using A Theoretical Analysis of the Gas Flow through the Muffler Combined with Valve Motion

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ABSTRACT

This paper describes the features of a calculation, developed to optimize the design of a gas flow path and valves, and shows the comparison between its results and experimental data for maximized cooling capacity and efficiency in reciprocating compressors.

The theoretical approach is studied and analyzed considering the interactions of compressor valve motions, pressure changes in the cylinder and the gas flow through the suction and discharge path, including the suction muffler. The valve motion and pressure changes in the cylinder are analyzed by the equation of motion and the equation of state through the Runge-Kutta method. The gas flow through the suction and discharge path, including the suction muffler, are calculated by using transfer matrix methods (4-pole relations) based on the assumption of one dimensional isentropic compressible fluid flow. The calculation accuracy was improved by using a correction coefficient, which is determined by comparing the theoretical and experimental transmission loss in various simple mufflers.

It is concluded that this theoretical analysis is very effective to evaluate the gas flow between compressor inlet and outlet. This is based on the fact that experimental results show the same pattern as theoretical analysis for valve motions, pressure changes in the cylinder, cooling capacity, efficiency and transmission loss of a suction muffler. The analysis technique described in this paper provides a method to optimize valve and muffler design in reciprocating compressors that maximizes cooling capacity and efficiency.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Av</td>
<td>area of valve port</td>
</tr>
<tr>
<td>A, B</td>
<td>four pole</td>
</tr>
<tr>
<td>C, D</td>
<td>damped ratio of valve</td>
</tr>
<tr>
<td>c_v</td>
<td>spring constant of valve</td>
</tr>
<tr>
<td>m_v</td>
<td>equivalent mass of valve</td>
</tr>
<tr>
<td>M_in</td>
<td>mass flow rate through suction valve</td>
</tr>
<tr>
<td>M_out</td>
<td>mass flow rate through discharge valve</td>
</tr>
<tr>
<td>P_c</td>
<td>pressure in cylinder</td>
</tr>
<tr>
<td>P_in</td>
<td>pressure at suction valve upstream</td>
</tr>
<tr>
<td>P_out</td>
<td>pressure at discharge valve</td>
</tr>
<tr>
<td>P_1, P_2</td>
<td>amplitude of pressure oscillation</td>
</tr>
<tr>
<td>t</td>
<td>time</td>
</tr>
<tr>
<td>U_1, U_2</td>
<td>amplitude of volume velocity</td>
</tr>
<tr>
<td>V_c</td>
<td>cylinder volume</td>
</tr>
<tr>
<td>x_v</td>
<td>displacement of valve</td>
</tr>
<tr>
<td>x_0</td>
<td>initial displacement of valve</td>
</tr>
<tr>
<td>\kappa</td>
<td>specific heat ratio</td>
</tr>
<tr>
<td>\rho_c</td>
<td>density in cylinder</td>
</tr>
<tr>
<td>\rho_in</td>
<td>density at suction valve upstream</td>
</tr>
<tr>
<td>\eta</td>
<td>effective force area</td>
</tr>
</tbody>
</table>
INTRODUCTION

Higher efficiency and lower noise levels have been demanded for household refrigerator compressors in recent years. Some of the factors affecting efficiency and noise are the suction and discharge valves and the gas flow path, including the suction muffler. For this reason, studies have been made on estimating gas flow characteristics from the compressor inlet, including the suction muffler, to the outlet pipe and the associated pressure changes. In addition, a variety of methods for estimating transmission loss in suction mufflers have been discussed for reducing operating noises. However, the effect of the suction muffler on the cooling capacity and efficiency has not been accurately estimated yet.

This paper analyzes suction and discharge valve motions, pressure changes in the cylinder, and the gas flow path, including the suction muffler, considering the interactions. In addition, the paper describes the development of a theoretical analysis program that can estimate cooling capacity, efficiency, and transmission loss of a suction muffler simultaneously and the results verified by experiments. An advantage using this method is estimation of sound characteristics.

CONSTRUCTION OF A RECIPROCATING COMPRESSOR

The cross-section of a conventional reciprocating compressor is shown in Fig. 1, and the basic construction of the suction and discharge paths in Fig. 2. The piston and cylinder mechanism is located in the lower part of the hermetic casing while the motor is placed in the upper part. As the piston of this compressor moves toward bottom dead center (suction stroke), the pressure in the cylinder reduces. The suction valve opens and refrigerant in the hermetic casing is guided into the cylinder through the suction muffler, cylinder head, valve plate and suction valve. When the piston moves toward top dead center (compression stroke), the refrigerant in the cylinder is compressed. The discharge valve then opens and the refrigerant flows out to the refrigeration system. This paper makes a theoretical estimation by modeling the construction of the suction and discharge system of the compressor described above.

METHOD OF THEORETICAL ANALYSIS

The theoretical analysis consists of two major parts. The first part is for estimating the suction and discharge valve motions and the pressure changes in the cylinder. The second part estimates the transmission loss of the suction muffler and pressure oscillation in each part. These two parts are then solved simultaneously for obtaining cooling capacity and efficiency.

The method of analysis on each part is described below.

(I) Suction/discharge valve motions and pressure changes in the cylinder

The fundamental equations used for analyzing the suction/discharge valve motions and pressure changes in the cylinder are shown below.

Equation of motion for the valve

\[ m_v \frac{d^2 x_v}{dt^2} + c_v \frac{dx_v}{dt} + k_v x_v = -k_v x_0 + \eta A_v \left( P_{in} - P_c \right) \]

\[ m_v \frac{d^2 x_v}{dt^2} + c_v \frac{dx_v}{dt} + k_v x_v = -k_v x_0 + \eta A_v \left( P_c - P_{out} \right) \]
Equation of pressure change in the cylinder
\[
\frac{dP_c}{dt} = \frac{\kappa}{V_c} \left( -P_c \frac{dV_c}{dt} + \frac{P_{in}}{\rho_{in}}M_{in} - \frac{P_{out}}{\rho_c}M_{out} \right)
\]  
(2)

The equations (1) and (2) are analyzed by using the Runge-Kutta method and the pressure changes in the cylinder and suction/discharge valve motions are calculated.

(II) Transmission loss of suction muffler and pressure oscillation in each part

The fundamental equation of the transfer matrix method used for analyzing the transmission loss of the suction muffler and the pressure oscillation in the suction valve upstream and the discharge valve downstream is shown below.

\[
\begin{pmatrix} P_1 \\ U_1 \end{pmatrix} = \begin{pmatrix} A & B \\ C & D \end{pmatrix} \begin{pmatrix} P_2 \\ U_2 \end{pmatrix}
\]  
(3)

Fig. 3 shows the model of the suction muffler used for the analysis. As shown in Fig. 3, the suction and discharge system is divided into pipe elements, chamber, resonator etc. Four pole of each element is calculated, and then the pressure oscillation and volume velocity are calculated for each point of an element. The transmission loss of the suction muffler is calculated using the ratio volume velocity at inlet and outlet. The calculation is made by assuming the flow to be one-dimensional isentropic compressible fluid flow.

The error between theoretical and experimental transmission loss occurred because of many assumptions described above. The calculation accuracy was improved by using a correction coefficient, which is determined by comparing the theoretical and experimental transmission loss in various simple mufflers.

(III) Cooling performance

Cooling capacity and efficiency are calculated by following procedure. The suction valve upstream pressure and the discharge valve downstream pressure are set as initial values. The change of volume velocity flowing into and out of the cylinder is calculated by using method (I). The change of volume velocity is then transformed to obtain a Fourier series for each frequency. The Fourier series is given as a boundary condition of volume velocity at the point of the suction valve upstream or the discharge valve downstream used in method (II). The amplitude of pressure oscillation and volume velocity at the frequency is calculated on each element. The calculated data for all frequency components are finally synthesized. The calculation is repeated until the difference between the obtained suction valve upstream pressure and the discharge valve downstream pressure and the corresponding initial value get smaller than the convergence judgment error. The cooling capacity or volumetric efficiency is calculated from the obtained gas volume velocity into the cylinder.

In an actual compressor, the intake gas temperature increases by heat generation, thereby changing gas density. The gas temperature in this analysis, however, is assumed constant in the suction path. The COP was calculated by cooling capacity / (Friction loss + Motor loss + Indicated work). The sum of the friction loss and motor loss was assumed to change linearly with frequency.
METHOD OF EXPERIMENT

(I) Suction and discharge valve motions, and pressure changes in each part

In order to verify the result of the theoretical analysis, the pressure changes in each part and the motion of the suction valve were measured. The measurement points are shown in Fig. 1 and Fig. 2. The pressure change in the cylinder (P1), at the suction valve upstream (P2), and at the discharge valve downstream (P3) were measured by installing quartz-type piezoelectric pressure sensor in each part. The displacement of the suction valve was measured by installing a displacement sensor installed on the valve plate (D1). A slit disc was mounted on the crankshaft to detect the rotating angle by sensing the slit with a displacement sensor (D2).

(II) Transmission loss of the suction muffler

In order to verify the transmission loss of the suction muffler, acoustic characteristics using a speaker was measured in air and were converted to an equivalent frequency in refrigerant by using the difference in sound velocity.

(III) Cooling performance

Test #1
By operating a compressor (cylinder volume 7.7 cm³) that equipped with measuring instruments on a calorimeter, cooling performance, along with the pressure oscillation and suction valve motion, was measured with the sensors described above. The suction muffler shown in Fig. 3 was used. The operating conditions of the calorimeter were evaporation temperature -30°C, condensation temperature 40°C, power frequency 45~70Hz (5Hz increment) and hermetic casing temperature 65°C.

Test #2
In addition, for the purpose of verifying the difference in cooling performance by suction muffler model used for theoretical analysis, two different models of suction muffler were fabricated and installed in another compressor for measuring cooling performance. The two muffler models used for the analysis and experiment are shown in Fig. 4. The operating conditions of the calorimeter for the second setup were evaporation temperature -23.3°C, condensation temperature 54.4°C, power frequency 50~65Hz (5Hz increment) and hermetic casing temperature 65°C.

Both of the tests were made with R134a refrigerant and ester oil (VG10).

RESULTS AND DISCUSSION

(I) Suction/discharge valve motions and pressure changes in the cylinder

Suction valve displacement, and pressure in the cylinder at 60Hz operation are shown in Fig. 5 and Fig. 6, respectively. Both figures compare the experimental data and calculated values. Fig. 5 shows good matching of suction valve displacement in the number of opening and in qualitative motion. Also, the importance of predicting the number suction valve openings for estimating volumetric efficiency has been known by previous studies[3]. Fig. 6 indicates good correlation between the calculated and experimental pressure changes in the cylinder both in motion and quantity. In particular, the pressure changes below the suction pressure (Ps) and above the discharge pressure (Pd) that respectively cause suction and discharge loss indicates good correlation.

(II) Transmission loss of suction muffler and pressure oscillation in each part

Shown in Fig. 7 are the results of experimental and calculated transmission loss of the suction muffler model shown in Fig. 3. This figure indicates close agreement of resonance frequency and loss level under approximately 2800Hz, the frequency range in which one-
dimensional theory applies. It is possible that transmission loss of suction muffler in an actual compressor is estimated with acceptable error by using the correction coefficient determined in various simple mufflers.

The suction valve upstream pressure and discharge valve downstream pressure, and experimental and calculated values on both pressures are shown in Fig. 8 and Fig. 9, respectively. Both Fig. 8 and Fig. 9 show qualitative agreement in two sets of data. Since the suction valve upstream pressure and discharge valve downstream pressure are known to affect the motion of respective suction and discharge valve, agreement of the two sets of data is essential for estimating efficiency.

(III) Cooling performance

The results of the experimental and calculated volumetric efficiency are shown in Fig. 10. This figure indicates qualitative agreement of experimental and calculated values, volumetric efficiency of 55Hz operation is lower while that of 65Hz is relative higher. Volumetric efficiency characteristics by operating frequency is heavily depended on the timing of open/close operation of the suction valve and the number of such operations[3]. For accurate estimation of the valve motion, simultaneous solution with a flow analysis as described in this report is essential.

Shown in Fig. 11 are the comparison results of the experimental and calculated COP. As seen in Fig. 11, the frequency at maximum COP in both experimental and calculated results agrees to a reasonable extent.

Shown in Fig. 12 is the experimental and calculated comparison of the volumetric efficiency characteristics of the two different suction muffler models. The experimental conditions and models are test #2. As indicated in Fig. 12, the calculated values can predict an greater difference in volumetric efficiency at 55Hz operation due to the difference in dimensions of suction mufflers.

CONCLUSIONS

Simultaneous analysis of suction/discharge valve motions, pressure changes in the cylinder, and gas flow path, including the suction muffler, considering the interactions, enabled estimation of suction/discharge valve motions, pressure changes in the cylinder, transmission loss of the suction muffler, pressure oscillation in each part, and cooling capacity. By using this theoretical analysis, simultaneous estimation of the effect of the suction muffler dimensions or suction/discharge valve dimensions on the cooling capacity and efficiency, along with estimation of transmission loss of the suction muffler has become possible.

REFERENCES

2. W.Soedel, “Mechanics, simulation and design of compressor valves, gas passage and pulsation mufflers”, Short Course Notes, 1992
Fig. 1 Schematic diagram of reciprocating compressor

Fig. 2 Suction and discharge path

Fig. 3 Modeling way of suction muffler

Fig. 4 Suction muffler model
Fig. 5 Suction valve motion (D1, test #1)

Fig. 6 Cylinder pressure (P1, test #1)

Fig. 7 Transmission loss of suction muffler shown in fig.3
Fig. 8 Suction valve upstream pressure (test #1)

Fig. 9 Discharge valve downstream pressure (test #1)

Fig. 10 Volumetric efficiency (test #1)

Fig. 11 COP (test #1)

Fig. 12 Comparison of suction muffler (test #2)