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H. Akashi  
*Matsushita Refrigeration Company Ltd.*

A. Yagi  
*Matsushita Refrigeration Company Ltd.*

S. Sugimoto  
*Matsushita Refrigeration Company Ltd.*

T. Yoshimura  
*Matsushita Refrigeration Company Ltd.*

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Influence of Pressure Wave in a Suction Path on Performances in Reciprocating Compressors

Hironari Akashi¹, Akio Yagi¹, Syuhei Sugimoto¹, Takao Yoshimura²
1 Compressor Development Office, Refrigeration Business R&D Center, Refrigeration Business Group, Matsushita Refrigeration Company
6-4-3 Tsujido Motomachi, Fujisawa, Kanagawa 251-8520, JAPAN
Phone 81-(0)466-30-1150; Fax 81-(0)466-30-1176; E-mail yagi@mrc.mei.co.jp
2 Compressor Division, Refrigeration Business Group, Matsushita Refrigeration Company

Abstract

This paper intends to clarify the influence of the supercharging effect on reciprocating compressor performance both experimentally and theoretically. The suction valve motion, the pressure fluctuation in the cylinder and in the suction pipe, and their effects on the compressor performance are researched experimentally. Furthermore, the theoretical analysis is performed by simultaneously calculating the pressure fluctuation and the suction valve displacement.

It is concluded that the compressor performance is great influenced by the pressure wave, which is formed in the suction pipe by the movement of suction valve. Consequently, it is important to synchronize the returning time of pressure wave to the cylinder inlet and the opening time of suction valve for higher supercharging effect.

Nomenclatures

\begin{align*}
a & \text{ velocity of sound} \\
A_s & \text{ cross sectional area of suction port} \\
C_d & \text{ damped ratio of suction valve} \\
f_0 & \text{ natural frequency of suction valve} \\
h & \text{ displacement of suction valve} \\
h_0 & \text{ initial displacement of suction valve} \\
m & \text{ mass of suction valve} \\
P & \text{ gas pressure} \\
R & \text{ universal gas constant} \\
s & \text{ spring constant of suction valve} \\
T & \text{ gas temperature} \\
u & \text{ gas velocity} \\
x & \text{ position on the suction pipe} \\
\Delta t & \text{ time interval} \\
\Delta x & \text{ distance step} \\
\eta & \text{ effective force ratio} \\
\kappa & \text{ specific heat ratio} \\
\rho & \text{ gas density} \\
\end{align*}

Subscripts

\begin{align*}
e & \text{ suction pipe end at suction valve} \\
v & \text{ inside in cylinder} \\
\end{align*}

Introduction

The demand for cheaper and smaller compressors with high efficiency and low noise characteristics for domestic refrigerators has been increasing in recent years. As one method of
achieving this purpose, supercharging to force the refrigerant into the cylinder by means of pressure waves generated in the suction pipe has been studied. The supercharging effect has been theoretically and experimentally confirmed to be effective in rolling piston rotary compressors [1] without a suction valve, and experimentally in reciprocating compressors [2]. In reciprocating compressors, however, the influence on the supercharging effect, due to the interaction between the suction valve behavior and the pressure wave in the suction pipe has not been clarified until now.

Experimental investigation and theoretical analysis of the supercharging effect on reciprocating compressor performance by examining the suction path consisting of a suction pipe and suction valve were made.

**Experimental and Theoretical Methods**

1. **Construction of a conventional reciprocating compressor**
   A cross-sectional view of a conventional reciprocating compressor is shown in Fig. 1, and the general construction of a suction path in Fig. 2, respectively. A piston-cylinder mechanism is placed at the upper part of the hermetic casing and a motor at the lower part. When the piston moves to the bottom dead center (suction stroke), the pressure in the cylinder falls. The suction valve opens at this time and the refrigerant in the hermetic casing is sucked into the cylinder through the suction muffler, cylinder head and valve plate in turns. When the piston moves to the top dead center (compression stroke), the refrigerant in the cylinder is compressed. Then, the discharge valve opens and the refrigerant is sent out to the refrigeration system.

2. **Experimental method**
   For the purpose of clarifying the supercharging effect, only a suction pipe was installed in place of a conventional suction muffler as shown in Fig. 3. Although the majority of the noise reducing function normally provided by the suction muffler is lost, this setup enables better understanding of the pressure fluctuations and supercharging effect.

   The pressure fluctuation in the suction pipe and the behavior of the suction valve were measured, on the assumption that they significantly influence the supercharging effect. It was assumed that the behavior of the suction valve, especially the valve displacement and open-close status, would influence the supercharging effect when a pressure wave forces refrigerant into the cylinder.

   In order to understand the process of pressure wave generation, transmission and reflection, and the pressure fluctuation in the cylinder, quartz-type piezoelectric pressure sensors were installed at the suction valve entrance (P2), at the center of the suction pipe (P3), and in the cylinder (P1). A displacement sensor (G1) was also installed on the valve plate to monitor the behavior of the suction valve. In addition, a slotted disk was attached to the crankshaft to measure the rotation angle of the crankshaft by detecting the slit with another displacement sensor (G2).

   The compressor used for this study had a displacement volume of 5.1 cm$^3$, to which was attached a suction pipe of length 250 mm and internal diameter 4.75 mm.

   Refrigeration performance was measured using a calorimeter under three revolution frequencies
of 50, 55 and 60 Hz, respectively. At the same time, the pressure fluctuations and suction valve behavior were measured. The operating conditions of the calorimeter were HFC-134a, ester oil (VG22), evaporation temperature -30 °C, condensation temperature 40 °C, and hermetic casing temperature 76 °C.

(3) Theoretical analysis

Theoretical analysis was carried out by simultaneously analyzing the flow in the suction pipe and suction valve behavior. The analysis model is illustrated in Fig. 4. The fundamental equations used in the analysis [1,3] were as follows.

Fundamental equations of flow

Equation of continuity

$$\frac{\partial \rho}{\partial t} + u \frac{\partial \rho}{\partial x} + \rho \frac{\partial u}{\partial x} = 0$$  (1)

Equation of motion

$$\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + \frac{1}{\rho} \frac{\partial P}{\partial x} = 0$$  (2)

Equation of energy

$$\left( \frac{\partial P}{\partial t} + u \frac{\partial P}{\partial x} \right) + a^2 \left( \frac{\partial \rho}{\partial t} + u \frac{\partial \rho}{\partial x} \right) = 0$$  (3)

Where  \( a^2 = \kappa RT \)  (4)

Equation of motion for the suction valve

$$\frac{\partial^2 h}{\partial t^2} + 2f_0C_{d} \frac{dh}{dt} + f_0^2 h_0 + f_0^2 h - \frac{A_1 \eta}{m}(P_e - P_v) = 0$$  (5)

Where  \( f_0 = s/m \)  (6)

The equations (1), (2) and (3) were solved by using the method of characteristics [1] considering the reflection of the pressure wave at the end of the suction pipe (where the pipe opens to the inside of the hermetic casing) in order to analyze the flow in the suction pipe and the pressure wave itself. The cross-sectional area of the suction pipe was constant, and the flow was assumed to be one-dimensional isentropic compressible. It was further assumed that the compressible gas was an ideal gas and thus was not influenced by fluid viscosity or thermal conduction.
With the method of characteristics, the inside of the suction pipe was divided into equal intervals $\Delta x$ in axial direction. The physical values at each node in a time interval of $\Delta t$ was calculated. The characteristic curve $C_+$ represents the locus of a forward wave and the curve $C_-$ represents that of a backward wave. They can be expressed as $C_0$: $dx=udt$, $C_+$: $dx=(u+a)dt$, $C_-: dx=(u-a)dt$. The values at node 4 in Fig. 4, $P$ (gas pressure), $\rho$ (gas density), and $u$ (gas velocity), are obtained from the value of each characteristic curve on the $x$-$t$ plane before the time interval of $\Delta t$, that is, the values of $P$, $\rho$, and $u$ at point 1, 2 and 3, which values calculated from the node 5, 6 and 7 by interpolation.

The behavior of the suction valve was obtained from the equation (5) and the equation of the gas flow through the valve, the equation of relationship between the cylinder internal pressure and volumetric flow rate, and the equations for each parameter such as density at the suction pipe end, by the Newton's method. The spring constant and natural frequency of the suction valve were assumed to be constant regardless of valve displacement.

Calculations were conducted by the following procedure. First, the initial values of the condition in the suction pipe and cylinder were assumed. Second, while forward time in steps of $\Delta t$, the pressure at each part, suction valve displacement and etc. was calculated for whole rotation of the crankshaft. At last, by using the calculation results after completing whole rotation of the crankshaft, the initial values are reassumed for calculating the subsequent whole rotation. These procedures were repeated until the difference of $P$, $\rho$, and $u$ values at angle between $0^\circ$ and $360^\circ$ reduced to less than the convergence judgment error.

**Results**

(1) **Experimental results and discussion**

The measured result of volumetric efficiency and COP of a conventional compressor and the tested compressor under each revolution frequency is shown in Fig. 5. It is shown that the volumetric efficiency increases by supercharging at 55 Hz and 60 Hz, but there is no improvement at 50 Hz.

The results of measurement of pressure fluctuations in the cylinder (P1 part) and suction pipe (P2 and P3 parts) and the suction valve behavior for crankshaft rotation angle at 50 Hz and 60 Hz are shown in Fig. 6. Both at 50 Hz and 60 Hz, there is a phase difference of the pressure fluctuation between at the entrance of the suction pipe and at the center part. The phase difference is equivalent to the time taken for the refrigerant gas to advance from the sensor P2 to P3 at the velocity of sound.

The relationship between pressure wave generation, transmission, reflection and suction valve behavior during supercharging is revealed in Fig. 6 (60 Hz).

As the suction valve opens, an expansion wave or lower pressure is generated at the entrance of the suction valve in the suction pipe (point A). The expansion wave travels approximately at the velocity of sound in the suction pipe, reflects at the suction pipe inlet and travels back in the reverse direction as a compression wave at higher pressure. The first time when the compression wave returns about when the suction valve closes (point A'). The refrigerant is pushed into the cylinder at
this time by the returned compression wave. The reflection of the compression wave reopens the suction valve before it closes (point B). The second opening of the suction valve generates an expansion wave again at the suction valve entrance in the pipe. This expansion wave, as the first expansion wave, travels in the suction pipe, reflects at the suction pipe inlet, and travels back in the reverse direction as a compression wave about when the suction valve closes, which is the second time (point B'). Similarly, this phenomenon continues until the suction valve completely closes (point C'). As that, the expansion wave and compression wave or the pressure wave travels back and forth in the suction pipe without being damped even after the suction valve is being closed and affects the following suction strokes.

This effect is also observed at 55 Hz. At 50 Hz, however, the third time when the pressure wave returns is during the early duration of the compression stroke (at a crank angle of approximate 200° (C')). This results in the pressure inside the suction pipe higher than that in the cylinder. Then the suction valve opens for the fourth time. This pressure wave pushes refrigerant into the cylinder, the cylinder internal pressure briefly increases at that time. However, the internal pressure quickly increases due to the compression stroke. It exceeds the pressure cylinder in the suction pipe soon. At this time, the refrigerant in the cylinder would flow back into the suction pipe. This phenomenon would be a cause of degradation in volumetric efficiency at 50 Hz.

In a word, the supercharging effect is closely related to revolutionary speed, suction pipe length and the suction valve behavior. It is necessary to make the suction pipe length match the opening timing of the suction valve to obtain a high supercharging effect.

(2) Theoretical results and discussion

Fig. 7 is a comparison between the experimental results and calculating ones of the suction valve displacement, cylinder internal pressure, and suction valve entrance pressure in the suction pipe at 50Hz. The results of the suction valve displacement and pressure fluctuation are quite accordant with each other, both qualitatively and quantitatively. The calculation results confirm that the suction valve opens 4 times at 50 Hz, whereas 3 times at 60 Hz.

The result of calculation of intake mass flow rate for one revolution of the crankshaft at 50 Hz and 60 Hz is shown in Fig. 8. A reverse flow occurs when the suction valve is opened at a crank angle more than 210°. It is shown that the reverse flow at 50 Hz is greater than that 60 Hz because of late closing of the suction valve. The calculated results for volumetric efficiency are shown in Table. 1. It is confirmed that the volumetric efficiency at 50 Hz is lower than that either at 55 Hz or at 60 Hz. This is a reason why the experimental result of volumetric efficiency is lower at 50 Hz in Fig. 5.

Theoretically, it is also clear that both the suction pipe length and the suction valve timing affect the supercharging effect. In addition, the method of analysis with the equations of one-dimensional isentropic compressible fluid flow and the suction valve motion was effective to predict supercharged compressor characteristics with large pressure fluctuations in the suction pipe.

Conclusions

The supercharging effect in a reciprocating compressor was examined both experimentally and
theoretically. The return timing of the pressure wave in the suction pipe and the open-close timing of the suction valve were confirmed to be a pair of important factors in the supercharging effect. In order to obtain a significant supercharging effect, reverse flow of refrigerant from the cylinder into the suction pipe needs to be prevented by ensuring that the higher pressure wave than that in the cylinder does not return when the suction valve is closed during the initial period of the compression stroke.

In addition, the method of analysis to simultaneously solve the equations of one-dimensional isentropic compressible fluid flow and the suction valve motion was confirmed to be effective to predict suction valve displacement, pressure fluctuation in the suction pipe or cylinder, and compressor performance.

Future work should examine the supercharging effect in the suction path with a muffler.

References

Fig. 1 Conventional reciprocating compressor

- $C_0$: Path line
- $C_L$: Left-running Mach line
- $C_R$: Right-running Mach line

Distance at suction pipe $x$

Fig. 2 Suction path

- $P_1$: Cylinder
- $P_2$: Suction valve entrance
- $P_3$: Center of suction pipe
- $G_1$: Suction valve displacement
- $G_2$: Shaft angle

Fig. 3 Experimental setup diagram

Fig. 4 The method of characteristics

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Fig. 6 Pressure pulsation in suction path and suction valve behavior

Fig. 7 A comparison between experimental results and calculating results

Table 1: Volumetric efficiency [%]

<table>
<thead>
<tr>
<th>Frequency [Hz]</th>
<th>50</th>
<th>55</th>
<th>60</th>
</tr>
</thead>
<tbody>
<tr>
<td>Experiment</td>
<td>60.5</td>
<td>66.4</td>
<td>64.7</td>
</tr>
<tr>
<td>Calculated</td>
<td>73.7</td>
<td>76.2</td>
<td>76.7</td>
</tr>
</tbody>
</table>

Fig. 8 Calculated intake mass flow rate