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INVESTIGATION OF PRESSURE PULSATION IN SUCTION PIPE ON ROTARY COMPRESSOR

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ABSTRACT

The compression performance of a rolling piston type rotary compressor was analyzed with attention to the pressure pulsation in the suction pipe.

In a displacement compressor, the pressure pulsation comes from intermittent suction. The authors analyzed and experimented with the super-charge effect of pulsation actively to increase the suction volume.

The mathematical model was constructed for the elements, and it was solved by the method of characteristics. The pressure pulsation measurements were performed under various pipe specifications and rotating speeds to verify the model. Then, the mechanism of suction line pressure pulsation and the influence of the suction specifications were discussed. It was confirmed that the best pipe length and diameter could result in a 10% increase of the charge for the rolling piston type rotary compressor.

INTRODUCTION

It is well known that a displacement compressor sucks and discharges intermittently, thereby generating pressure pulsations in the piping system. In regard to the pressure pulsation in the suction line of a reciprocating and rotary compressor, several studies are reported with attention to reducing the vibration and the suction passage loss.(1),(2) The pressure pulsation in the suction line may give rise to a charging effect, enabling the volumetric efficiency to be improved.(3)

In the present study, with attention focused on the influence of the pressure pulsation on a super-charge effect and compressor efficiency of the rolling piston type rotary compressor, the influences of suction pipe specifications were examined in detail, both theoretically and experimentally.

A mathematical model was constructed for the suction line elements, assuming that the pressure and the density in an accumulator were constant, and solved by the method of characteristics.

The suction volume and the suction chamber pressure are measured with air and R-22 respectively to verify the mathematical model. The mass flow rate and the indicated work were calculated under various pipe specifications and rotating speeds. And the mechanism of suction line pressure pulsation and the influence of the suction specifications were discussed.

As a results of this study, it was confirmed that the best pipe length and diameter could result in a 10% increase of the charge for the rolling piston type rotary compressor.

MATHEMATICAL MODEL

A schematic illustration of the suction line and cylinder suction chamber is presented in Fig.1. A mathematical model was constructed for the suction pipe and the suction chamber. The sketch of the simplified suction line is shown in Fig.2.

The suction chamber is assumed to be the pipe with the same diameter as the suction pipe. The reciprocating piston is assumed instead of the rolling piston in this model, and it is moved according to the rule that the volume velocity is equal to the rolling type. Consequently, the piston is moved to the left in Fig.2 between $\theta=0$ and $\theta=\theta s$, where the suction volume is decreased. The following assumptions are made also,

(1) The working fluid is an ideal gas.
(2) There is a one-dimensional, homentropic, unsteady flow, without wall friction.
(3) There is no heat exchange with the external environment.
(4) Quantities of state in an accumulator are constant.
(5) There is no pressure loss at the end of the pipe.

The continuity equation, conservation of moment, and the energy equation are as follows,

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

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

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

With the wave characteristics in x-t field, the equations (1) through (3) are transformed as follows,

$$dP - a^2 \cdot d\rho = 0 \qquad \cdots\cdots(4)$$

$$dP + \rho a \cdot du = 0 \qquad \cdots\cdots(5)$$

$$dP - \rho a \cdot du = 0 \qquad \cdots\cdots(6)$$

P, ρ, u are obtained by integrating these equations along to the characteristics lines.

The quasi-steady flow and following equations are assumed at the end of the pipe.(4)

(1) the flow from accumulator to pipe

$$P/\rho^{\kappa} = Pac/\rho ac^{\kappa} \qquad \cdots\cdots(7)$$

$$\frac{\kappa}{\kappa-1} \cdot \frac{P}{\rho} + \frac{u^2}{2} = \frac{\kappa}{\kappa-1} \cdot \frac{Pac}{\rho ac} \qquad \cdots\cdots(8)$$

(2) the flow from pipe to accumulator

$$P = Pac \qquad \cdots\cdots(9)$$

EXPERIMENTAL APPARATUS AND METHOD

Fig.3 is a schematic view showing the experimental apparatus.

The test compressor was a commercial 2.5 Kw class rolling piston type rotary compressor modified for experimental use. The measurement of pressure in the cylinder suction chamber was performed with a small piezo type pressure transducer. A signal of crank angle was picked up with a gap sensor. The temperatures of refrigerant gas were measured with thermocouples.

The gas flow rate and the consumption power were measured by actually operating the experimental model installed in a secondary refrigerant compressor calorimeter. To examine the super-charge effect in detail, the measurement of the gas flow rate was performed with a floating flow meter by operating the test compressor with air.

RESULTS OF ANALYSIS AND EXPERIMENT

The experimental pressure diagrams and calculated pressure diagrams with respect to the suction pipe length and rotating speeds are shown in Fig.4 and Fig.5. Fig.6 shows the air flow rate with respect to the suction pipe length. The measured values are also plotted in the figure. The air flow rate becomes maximal at approximately Ls=Lsr. Lsr is defined by (10) as the pipe length of gas column resonance,

$$Lsr = \frac{15a}{N} \quad \cdots\cdots (10)$$

The experimental results agreed well with the results of calculations both qualitatively and quantitatively, showing that the mathematical model is capable of representing phenomena with sufficient reproducibility.

The velocity and pressure pulsations in a standard-specification suction pipe are shown in Fig.7 and Fig.8, respectivly. [0],[4] in the figures mean respectively the end point of the pipe and the suction port. The velocity fluctuation at [4] is almost the same as the volume velocity in the suction chamber. The amplitude of velocity and pressure pulsation become maximal at the end of the pipe, and at the suction port, respectively. The velocity and pressure in another suction pipe are shown in Fig.9 and Fig.10, respectively. The pipe length Ls is 800(mm), and approximately equal to the resonant pipe length. The maximal amplitudes of velocity and pressure on Ls=800(mm) are about two times larger than those on Ls=260(mm). It can be seen from Fig.4 through Fig.10 that the amplitude of velocity and pressure fluctuation are greatly affected by Ls, and that the suction port pressure is minimal in the neighborhood of $\theta=180°$ at Ls=Lsr.

The compression power is given by the following equation,

$$W = 2\pi g \cdot \frac{N}{60} \int_0^{2\pi} [\ Pc(\theta) - Psc(\theta)\ ]\ H \cdot La(\theta) \cdot Lb(\theta)\ d\theta \quad \cdots\cdots (11)$$

$La(\theta)$ and $Lb(\theta)$ become maximal at $\theta=180°$, so that it is preferred that Psc is large in the neighborhood of $\theta=180°$ to reduce the compression power. However, as seen from Fig.5 and Fig.6, Ps becomes a bottom pressure at that crank angle, when Ls approximately equals Lsr; so that, the compression power also increases.

The refrigerating capacity ratio, Rs, the compression power ratio, Rp, and the compressor efficiency ratio, $R\eta$, with respect to the suction pipe length are shown in Fig.11, Fig.12, and Fig.13, respectively. The measured values are also plotted in the figures. Rs, Rp, $R\eta$ are defined in this paper as follows.

$$Rs = G/Gst \quad \cdots\cdots (12)$$

$$Rp = W/Wst \quad \cdots\cdots (13)$$

$$R\eta = \eta/\eta st \quad \cdots\cdots (14)$$

Subscript st in the above equations means the value under the standard suction pipe specifications, i.e., Ls=260 (mm), Ds=16.7 (mm). The calculated values agree approximately with the measured. It can be seen from Fig.11 that the refrigerating capacity increases more than 10% at the most suitable pipe length,compared with that of the standard pipe length. The compression power also increases similarly to the refrigerating capacity, due to the above reason, as shown in Fig.12. As a result of these, very little increase appears in the compressor efficiency, as shown in Fig.13.

The calculated charging ratio (Q/Q_0) with respect to the suction pipe diameter is shown in Fig.14. The maximal amplitude of the charging ratio is greatly affected by the suction pipe diameter, and increases with a reducing Ds.

CONCLUSIONS

The following conclusions are obtained,

1. A comparatively simple mathematical model could be constructed for the pressure pulsation in the suction line of the rolling piston type rotary compressor.
2. The affects of the suction pipe specifications on the refrigerating capacity, and on the compressor efficiency were derived from results of theoretical and experimental studies.
3. The most suitable length and diameter could result in a more than 10% increase of the refrigerating capacity.
4. The compressor efficiency increased only slightly.

NOMENCLATURE

Psc: pressure in suction chamber
Pc : pressure in compression chamber
Ls : suction pipe length
Lsr: pipe length of gas column resonance
La : arm length (see Fig.1)
Lb : distance between two contact point of the rolling piston (see Fig.1)
Ds : suction pipe diameter
H : rolling piston height
G : refrigerating capacity
W : compression power
Q : flow rate
N : compressor speed
Rs : refrigerating capacity ratio
Rp : compression power ratio
$R\eta$: compressor efficiency ratio

g : gravity acceleration
a : acoustic velocity
u : fluid velocity
x : distance
t : time
η : compressor efficiency
θ : crank angle
θs: crank angle where suction process starts
ρ : fluid density
κ : specific heat ratio

subscripts
st: standard suction pipe specification
ac: accumulator

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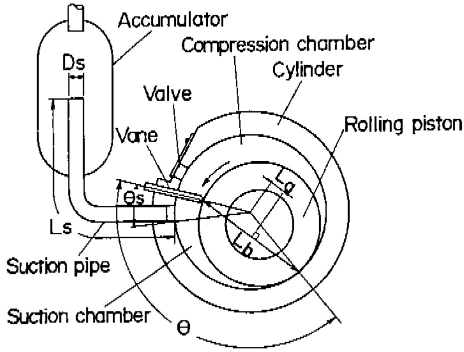


Fig. 1 Cylinder and Suction Line

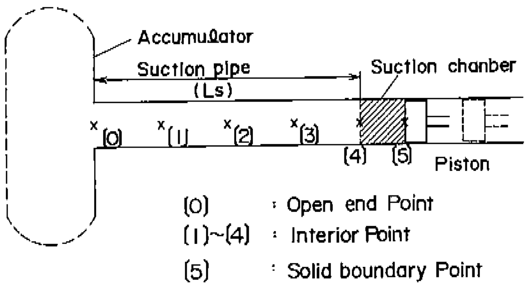
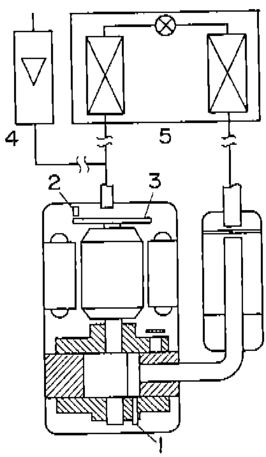


Fig. 2 Simplified Suction Line



1: pressure transducer
2: gap sensor
3: disc
4: float type flow meter (for Air)
5: secondary refrigerant compressor calorimeter

Fig. 3 Experimental Apparatus

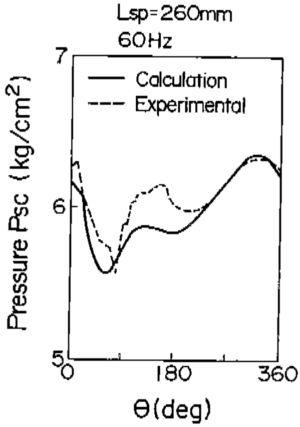


Fig. 4 Pressure in Suction Chamber (1)

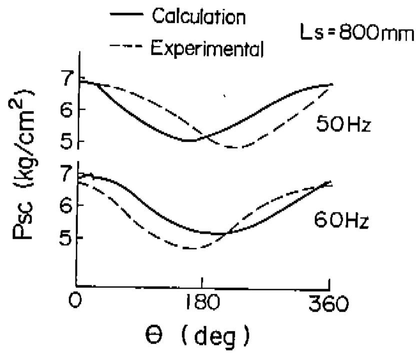


Fig. 5 Pressure in Suction Chamber (2)

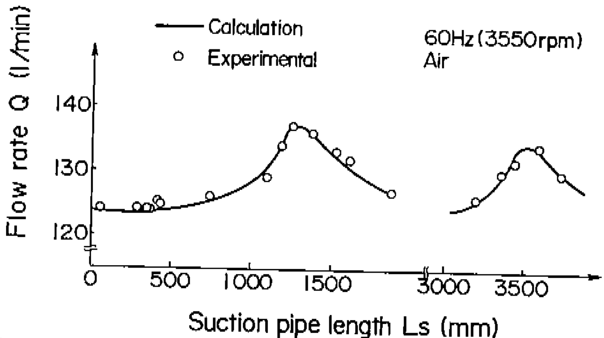


Fig. 6 Influence of Suction Pipe Length on Flow Rate

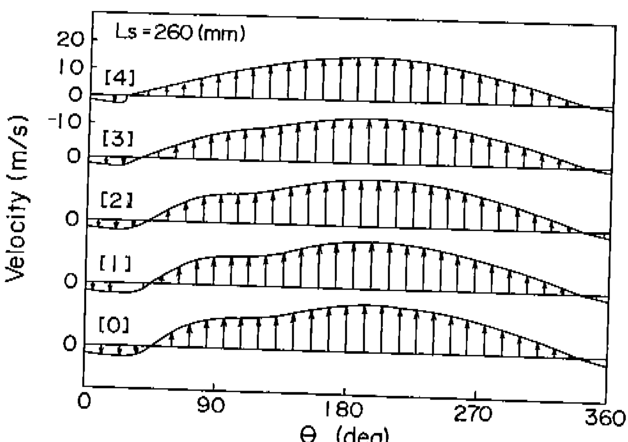


Fig. 7 Calculated Velocity in Suction Pipe (1)

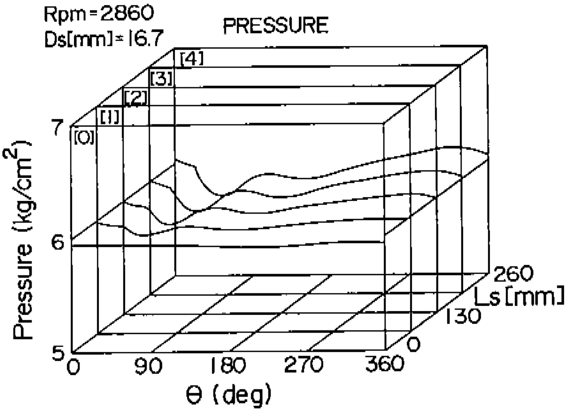


Fig. 8 Calculated Pressure in Suction Pipe (1)

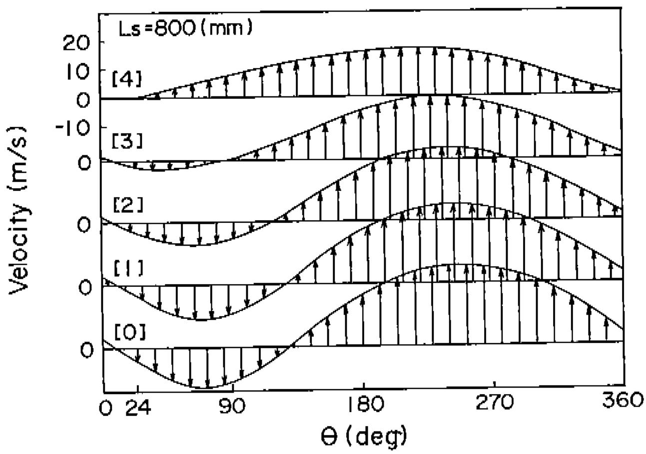


Fig. 9 Calculated Velocity in Suction Pipe (2)

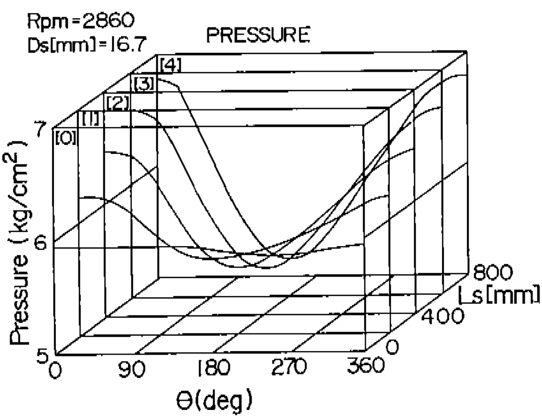


Fig. 10 Calculated Pressure in Suction Pipe (2)

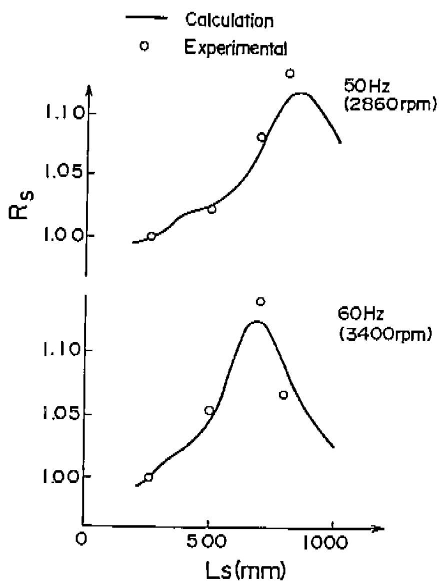


Fig. 11 Refrigerating Capacity Ratio

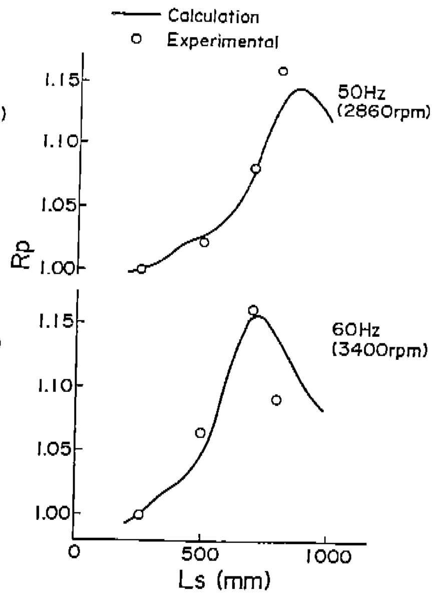


Fig. 12 Compression Power Ratio

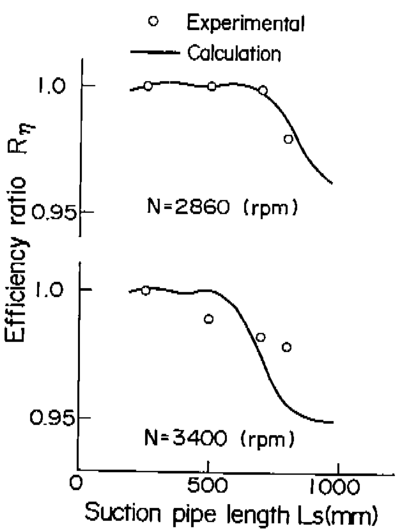


Fig. 13 Compressor Efficiency Ratio

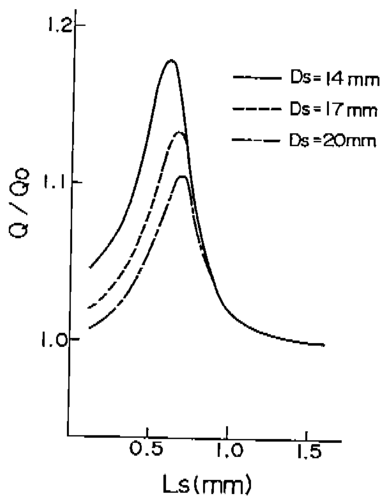


Fig. 14 Influence of Pipe Diameter on Charging Ratio