1984

Analysis of Suction Passage Loss in a Rotary Compressor

H. Hirano

ANALYSIS OF SUCTION PASSAGE LOSS IN A ROTARY COMPRESSOR

Hideo Hirano, Chief Engineer
Matsushita Electric Industrial Co., Ltd.
2275-3 Noji-cho, Kusatsu City, Shiga Pref. 525, Japan

ABSTRACT

The compression efficiency of a rolling piston type rotary compressor for room air-conditioners was analyzed.

Suction passage loss is one of the detracting factors in compression efficiency and we studied its main cause, i.e. the pressure pulsation generated in the suction line, and clarified the relation between suction line specification and suction passage loss. A method of evaluating the suction passage loss was also derived. First, the pressure pulsations in various suction line elements and cylinder suction chamber were measured to experimentally estimate the relation of the suction line specification with the pressure pulsation and suction passage loss.

Then, a mathematical model was constructed for each suction line element and its validity was tested. And the relation of the specification with the pressure pulsation and suction passage loss was theoretically estimated. Lastly, a discussion was made with reference to speed control.

INTRODUCTION

Several reports are available on the analysis of rotary compressor performance.(1),(2) As these studies have shown, the compression efficiency of this type of compressor is low as compared with its mechanical efficiency and it is an important subject of study to develop a method for improving the compression efficiency.

In the suction, compression and discharge processes of a rolling piston type rotary compressor, the following power consumptions occur to affect the compression efficiency.

1) Suction Passage Loss

2) Discharge Passage Loss

3) Heating Loss during Suction Process

4) Heating Loss during Compression Process

5) Clearance Volume Loss

In regard to the suction process, it has been reported that suction pipe length may give rise to a charging effect, enabling the volumetric efficiency to be improved.(3) Moreover, though it is related to the suction process of a reciprocating type compressor, some references have been made to the relation between pressure pulsation and suction passage loss, indicating that the consumption power increases due to resonance of the gas in the suction pipe.(4),(5),(6)

In the present study, with attention paid to the pressure pulsation occuring in the suction process of the rolling piston type rotary compressor, the influences of the suction line specification were examined in detail and an evaluation method was derived.

First, the pressure pulsations in the accumulator and suction pipe constituting the compressor suction line were accurately measured, whereby it was experimentally clarified that pressure pulsations in the suction line and cylinder suction chamber vary considerably with the specifications, influencing the suction passage loss.

Then, a mathematical model was constructed for each suction line element and the validity of the model was tested and confirmed. Thereafter, the mechanism of suction line pressure pulsation was theoretically estimated and the influence of the specification of each element was theoretically clarified.

The above evaluation method was also applied to the rotary compressor of speed control type so as to clarify the factors which influence the compression efficiency.

EXPERIMENTAL APPARATUS

Fig. 1 is a schematic view showing the experimental apparatus.

The test compressor was a commercial 900-watt class rolling piston type rotary compressor modified for experimental use. With regard to the pressures in the accumulator, suction pipe and cylinder suction chamber, the pressure pulsations were measured with piezoelectric pressure transducers (Type 601A Kistler) and as to the accumulator and suction pipe, the mean pressures were measured with Bourdon tubes. Regarding the crank angle of the test compressor, the T.D.C. (top dead center) and 10 deg. crank angles were optically determined by means of LED and phototransitor.

Test conditions were set with a secondary refrigerant compressor calorimeter. Based on these analog data, P-θ and P-V diagrams were constructed using an A-D converter and a computer and the suction passage loss defined by (1) was calculated.

$$LPs=\int_{Vcl}^{Vst+Vcl}(Pcs-Ps)dVcs\frac{N}{60}Ac \qquad (1)$$

MATHEMATICAL MODEL

A schematic illustration of the suction line and cylinder suction chamber is presented in Fig. 2. Mathematical models were constructed for the three suction line elements and the boundaries.[7],[8],[9]

1) Cylinder suction chamber

2) Suction pipe

3) Accumulator

4) Pipe ends

The refregerant is assumed to be an ideal gas.

Cylinder suction chamber

The following assumptions are made.

1) an adiabatic process, without heat exchange with the cylinder wall.

2) No wave action.

3) Top clearance volume disregarded.

4) Suction process starts at θs.

The pressure and weight are given by (2) and (3).

$$\frac{dPcs}{dt}=\chi\cdot Pcs\left(\frac{1}{Gcs}\frac{dGcs}{dt}-\frac{1}{Vcs}\frac{dVcs}{dt}\right) \qquad (2)$$

$$\frac{dGcs}{dt}=Ssp\cdot\rho spe\cdot g\cdot Uspe \qquad (3)$$

Suction pipe

The following assumptions are made.

1) Homentropic flow.

2) A one-dimensional unsteady flow, with wall friction.

3) No heat exchange with the external environment.

The continuity equation, conservation of momentum, and first law of thermodynamics hold.

$$\frac{\partial Psp}{\partial t}+\rho sp\frac{\partial Usp}{\partial x}+Usp\frac{\partial\rho sp}{\partial x}=0 \qquad (4)$$

$$\frac{\partial Usp}{\partial t}+Usp\frac{\partial Usp}{\partial x}+\frac{1}{\rho sp}\frac{\partial Psp}{\partial x}+\frac{2f}{Ds}Usp^2\frac{Usp}{|Usp|}=0 \qquad (5)$$

$$\frac{\partial}{\partial t}\left\{\rho sp\left(\frac{g}{A}Cv\cdot Tsp+\frac{Usp^2}{2}\right)\right\}+\frac{\partial}{\partial x}\left\{\rho spUsp\left(\frac{g}{A}Cv\cdot Tsp+\frac{Psp}{\rho sp}+\frac{Usp^2}{2}\right)\right\}=0 \qquad (6)$$

$$\frac{g}{A}Cv\cdot Tsp=\frac{a sp^2}{\chi(\chi-1)} \qquad (7)$$

With the wave characteristics in x-t field, the equations (4) through (6) are transformed as follows.

Direction conditions

$$\frac{dx}{dt}=Usp\pm a sp \qquad (8)$$

Compatibility conditions

$$\frac{dPsp}{dt}\pm\rho sp\cdot a sp\frac{dUsp}{dt}-(\chi-1)\frac{2\cdot f\cdot\rho sp\cdot Usp^3}{Ds}\frac{Usp}{|Usp|}\pm\frac{2\cdot f}{Ds}\rho sp\cdot a sp\cdot Usp^2\frac{Usp}{|Usp|}=0 \qquad (9)$$

Accumulator

The following assumptions are made.

1) An adibatic change, without heat exchange with the external environment.

2) No wave action.

As in the case of the cylinder suction chamber, the pressure and weight are given (10) and (11).

$$\frac{dPac}{dt}=\chi\ Pac\frac{1}{Gac}\frac{dGac}{dt} \qquad (10)$$

$$\frac{dGac}{dt}=Scp\cdot\rho cpe\cdot g\cdot Ucpe-Ssp\cdot\rho sps\cdot g\cdot Usps \qquad (11)$$

Pipe ends

An isoentropic flow is assumed.

EVALUATION OF THE MATHEMATICAL MODEL

The experimental pressure diagrams and calculated pressure diagrams with respect to the suction line specifications of Fig. 3 are shown in Figs. 4 and 5, respectively.

With regard to the ordinary differential equations concerning the cylinder suction chamber and accumulator, the method of Runge-Kutta-Gill was employed. As for the suction pipe and connecting pipe, computations were made by the method of characteristics. The friction factor was calculated for mean fluid velocity by use of Blasius' equation. These computations could be sufficiently converged in the fourth cycle.

The experimental results were in good agreement with the results of calculations both qualitatively and quantitatively, showing that the mathematical model is capable of representing phenomena with sufficient reproducibility.

RESULTS

The results of the study of the suction line including a type (I) accumulator are set forth below.

Suction pipe

The pressure pulsation and LPs with respect to pipe length Ls are shown in Figs. 6 and 7. The volume velocity in the cylinder suction chamber is maximal in the neighborhood of $\theta = 180°$ and, therefore, LPs becomes maximal for the pipe length giving a bottom pressure at $\theta = 180°$.

The crank angle θsr at the moment of return of the pressure pulsation generated in the cylinder suction chamber at suction start is given by the following equation.

$$\theta sr = \theta s + \frac{12 \cdot Ls \cdot N}{a s} \qquad (12)$$

The relation between θsr and the bottom pressure position is shown in Fig. 8. A good agreement is found, indicating that the pressure in the cylinder suction chamber drops till return of the pressure wave generated at the start of suction.

At $\theta sr > 180°$, the bottom pressure rose and LPs decreased but these were due to the influence of reflected wave and an increased inflow fluid velocity at suction start. The above results suggest that $\theta sr < 90°$ is preferred.

The pressure pulsation for pipe inner diameter Ds is shown in Fig. 9 and the suction passage loss corresponding to the Reynolds' number of mean flow velocity is shown in Fig. 10. While a large variation in magnitude of bottom pressure rather than in phase is apparent, both variations are due to a decreased flow velocity as a consequence of wall friction. The decreased inflow velocity of reflected pressure wave caused a delay in the recovery of pressure, thus causing a further increase of LPs.

It is clear that this is particularly pronounced at $Re > 2 \times 10^5$.

Accumulator

The pressure pulsation relative to volume Vac is shown in Fig. 11, and the suction passage loss in Fig. 12. While there is a change in pressure drop from suction start to θsr, this is because the outflow velocity at the start of suction is reduced as the volume is decreased. There is no major change in LPs but an influence on waveform is apparent at $(Vac/Vsp) < 10$.

Compressor speed

An example of application to compressor speed control is shown in Fig. 13. Within the scope of the present experimental study, the influence of θsr was the greatest. It is conceivable that the influence of Reynolds' number and volume appears, depending on compressor speed.

CONCLUSIONS

The foregoing observations lead us to the following conclusions.

1) A comparatively simple mathematical model for simulating the pressure pulsation in the suction line could be constructed.

2) A method for evaluating the suction passage loss was derived from results of experimental and theoretical studies.

3) In compressor speed control, the influence of θsr was found to be great.

NOMENCLATURE

a : acoustic velocity
as : acoustic velocity at Ps and Ts
A : inverse of mechanical equivalent of heat
Ac : coefficient of conversion
Cv : specific heat at constant volume
Ds : suction pipe diameter
f : friction factor
g : gravity acceleration
G : weight
Ls : suction pipe length
LPs: suction passage loss
N : compressor speed
P : pressure
Ps : suction pressure
Rc : cylinder radius
Re : Reynolds' number
Rp : piston radius
S : cross-sectional area
T : temperature
Ts : suction temperature
U : fluid velocity
V : volume
Vcl: top clearance volume
Vst: stroke volume
t : time
x : distance

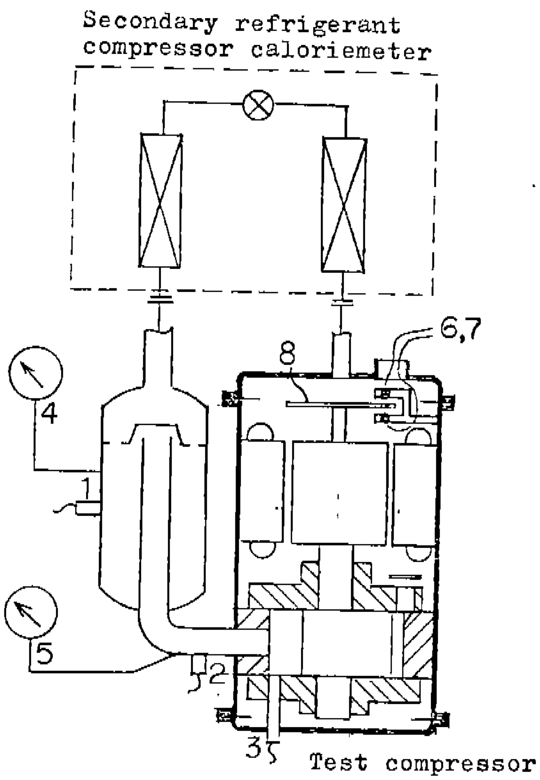
θ : crank angle
θbp: crank angle at bottom of pressure pulsation in suction chamber
θs : crank angle where suction process starts
θsr: crank angle where pressure wave returns
κ : specific heat ratio
ρ : fluid density

Subscripts

ac : accumulator
cp : connecting pipe
cpe: pipe end in downstream
cs : suction chamber
sp : suction pipe
spe: pipe end in downstream
sps: pipe end in upstream

REFERENCES

1. Chu, I. et al., "Analysis of the rolling-piston type rotary compressor", Proc. Purdue Comp. Tech. Conf., 1978.
2. Wakabayashi, H. et al., "Analysis of performance in a rotary compressor", Proc. Purdue Comp. Tech. Conf., 1982.
3. Yoshida, Y. et al., "Study of pressure pulsation in the suction line of a rotary compressor", Prepr. of Jpn. Soc. Mech. Eng., 1981, No. 815-1.
4. Bannister, F. K., "Induction ramming of small high-speed compressor", Proc. I. Mech. Engr., 1959, Vol. 173.
5. MacLaren, J. F. T. et al., "A model of a single stage reciprocating gas compressor accounting for flow pulsations", Proc. Purdue Comp. Tech. Conf., 1974.
6. Brablic, J., "Computer simulation of the working process in the cylinder of reciprocating compressor with piping system", Proc. Purdue Comp. Tech. Conf., 1974.
7. Benson, R. S. et al., "A numerical solution of unsteady flow problems", Int. J. Mech. Sci., 1964, Vol. 6.
8. Benson, R. S. et al., "Some further analysis of rociprocating compressor systems", Proc. Purdue Comp. Tech. Conf., 1974.
9. Wright, E. J., "Computer simulation of engine gas dynamic processes: a design package", S.A.E., Automotive Engineering Congress, 1971.



1~3 : pressure transducer
4,5 : Bourdon tube
6 : LED
7 : photo-transiter
8 : disc

Fig. 1 Experimental apparatus

Electric sources	100V-60HZ
Evaporating temp	7°C
Condensing temp	55°C
Suction temp	18°C

Refrigerant R-22

Table. 1 Test conditions

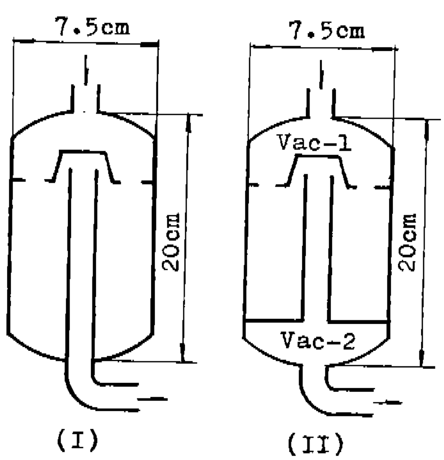
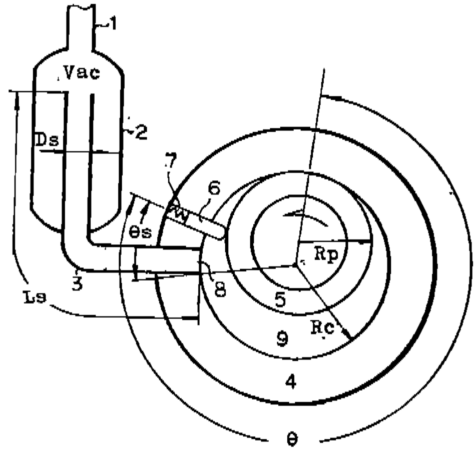


Fig. 3 Accumulators



1: connecting pipe
2: accumulator
3: suction pipe
4: cylinder
5: rolling piston
6: sliding vane
7: spring
8: suction port
9: suction chamber

Fig. 2 Cylinder and suction line

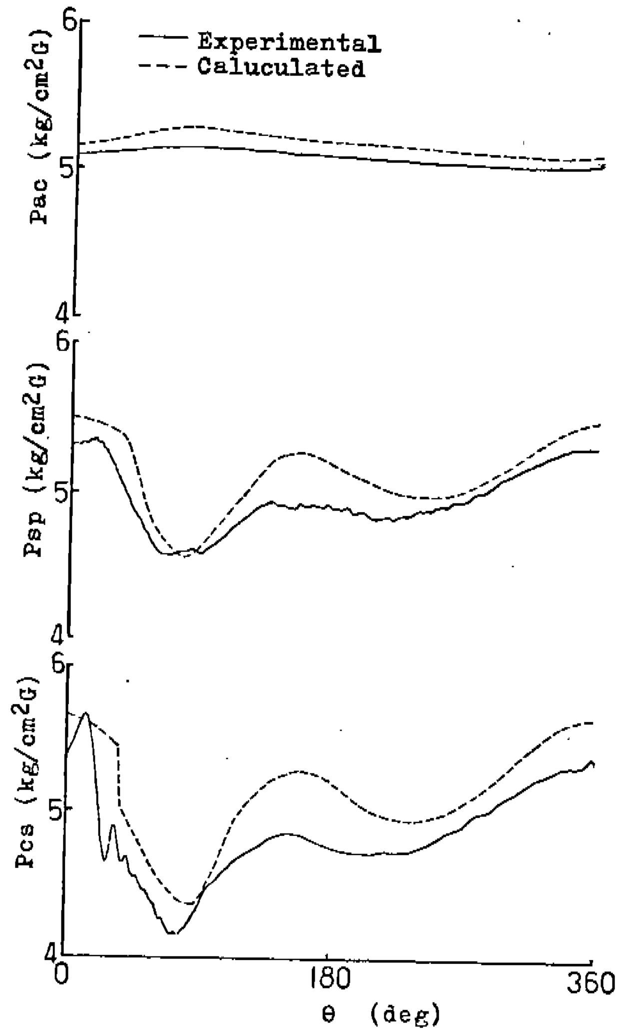
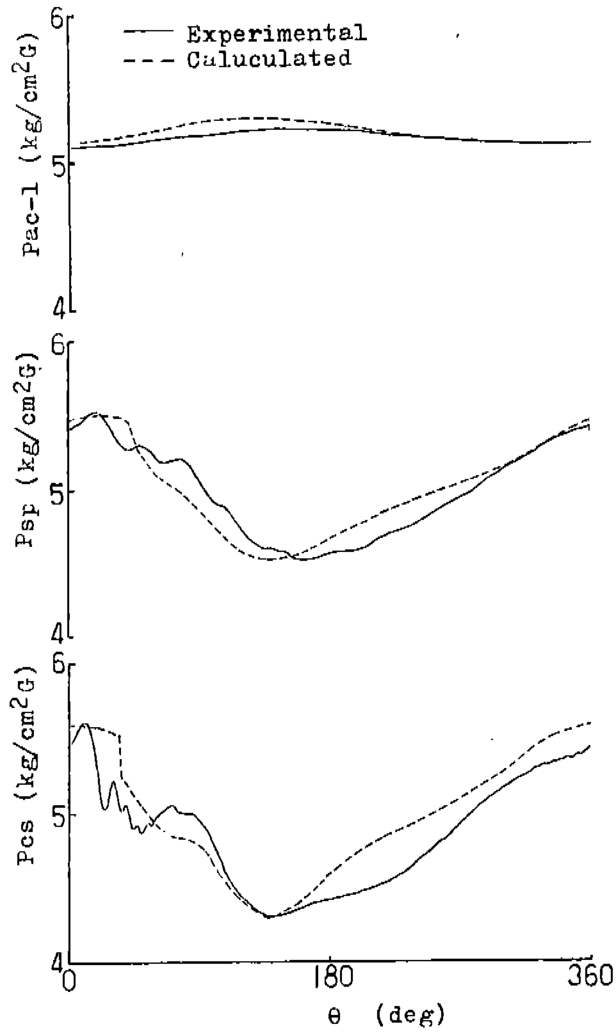
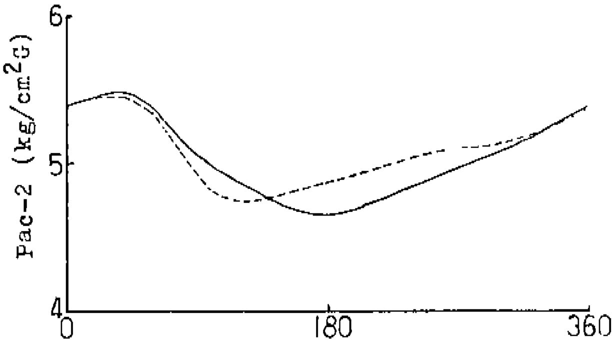


Fig. 4 Comparison of calculated and experimental results (type I)





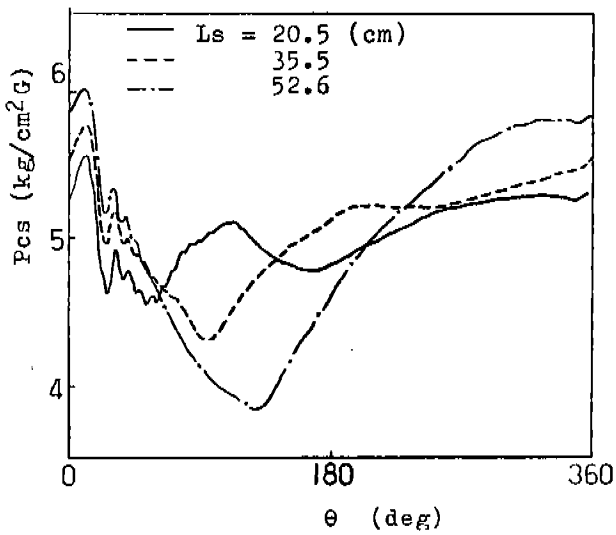


Fig. 6 Influence of suction pipe length to pressure pulsation in suction chamber

Fig. 5 Comparison of calculated and experimental results (type II)

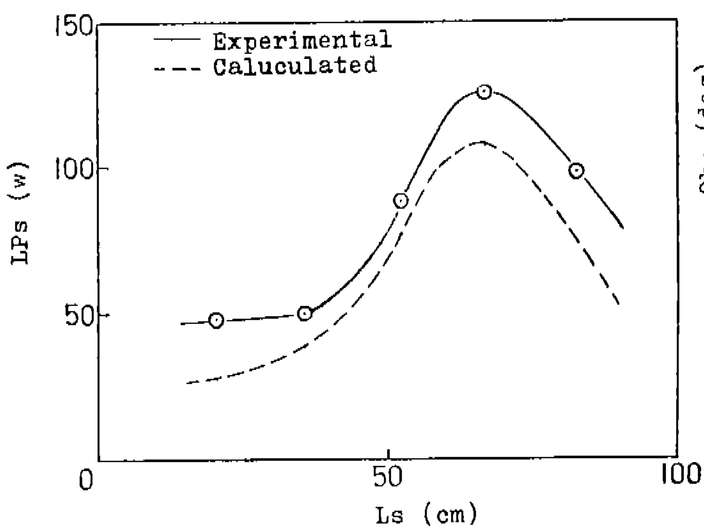


Fig. 7 Ls - LPs

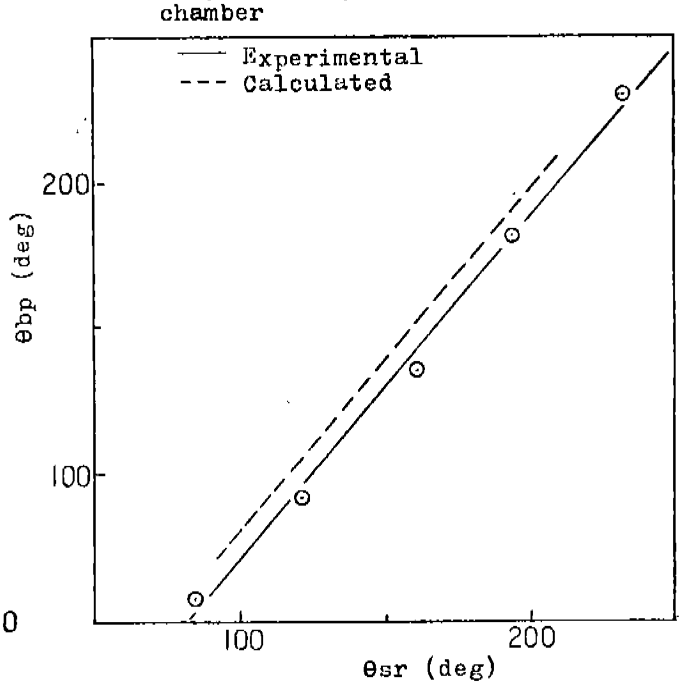


Fig. 8 θsr - θbp

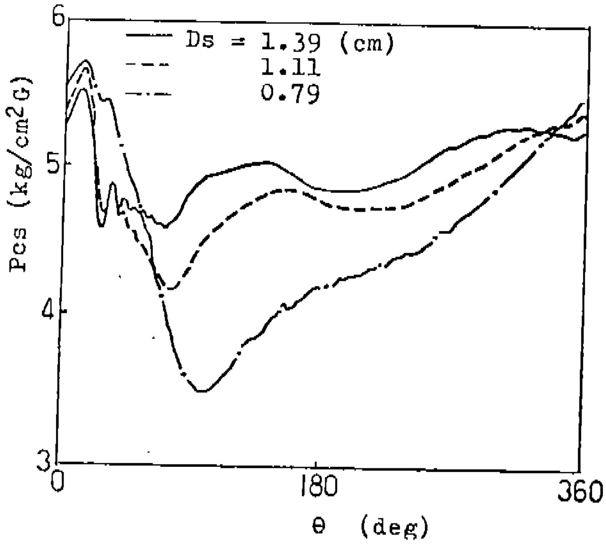


Fig. 9 Influence of suction pipe diameter to pressure pulsation in suction chamber

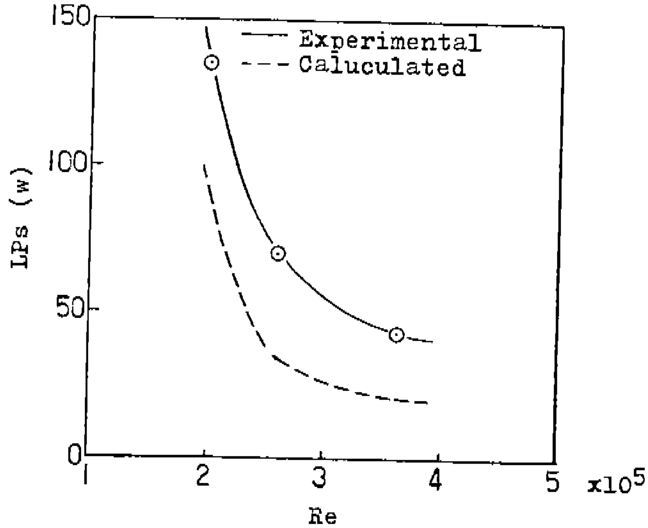


Fig.10 Ds - LPs

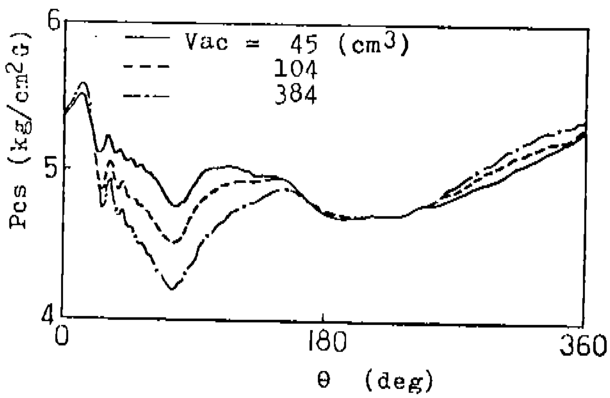


Fig.11 Influence of accumulator volume to pressure pulsation in suction chamber

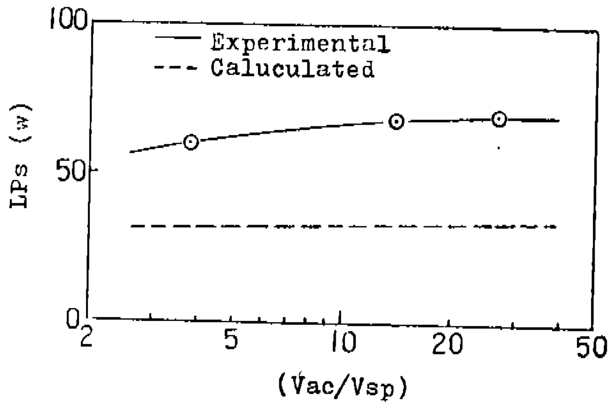


Fig.12 (Vac/Vsp) - LPs

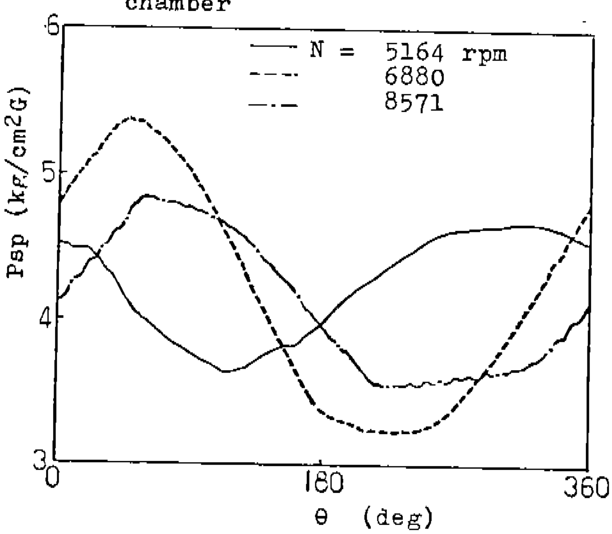


Fig.13 Influence of compressor speed to pressure pulsation in suction pipe