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THE STUDY OF DUAL CYLINDER ROTARY COMPRESSOR

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ABSTRACT

Rolling piston type rotary compressor has been widely used nowadays because of the features of high efficiency, compact size and light weight. On the other side rotary compressor with large capacity, for example more than 5 HP , has a tendency to be difficult in the application to an equipment on account of vibration characteristics.
The paper refers to the basic study of dual cylinder rotary compressor which has also the excellent feature of low vibration. Dynamic characteristic analysis, efficiency improvement, and efficiency evaluation were studied theoretically and experimentally as compared with single cylinder rotary compressor of conventional type. The results of this study suggest much applicability of dual cylinder rotary compressor to the refrigerating system with large capacity.

INTRODUCTION

Rolling piston type rotary compressor has been widely used for refigerator, room air conditioner and unitary nowadays, particularly rapidly these several years. Because it has the attractive features which are, as well known, high efficiency, compact size, light weight and so on. And these meet the requirement of the times, that is the saving of energy. But on the other hand of the remarkable features mentioned above, it has a tendency to be difficult in the application to an equipment on account of vibration characteristics as it increases in capacity.

We made the basic study of dual cylinder rotary compressor which has low vibration besides the original features of rolling piston type rotary compressor. The results of the study of the model in 5 P class are reported in this paper. Dynamic characteristic analysis, the investigation of efficiency improvement and the evaluation of each efficiency were carried out theoretically and experimentally. On the other hand single cylinder rotary compressor with the same capacity was also examined for the purpose of comparison.

Dynamic characteristics was analyzed by solving numerically the dynamic equations of parts in motion, these are crankshaft, vane and rolling piston. The theory of finite length bearing was applied to upper bearing, lower bearing and eccentric bearing under the condition of fluid lubrication. The proper coefficients of friction were used at vane tip, vane side and thrust bearing under the condition of boundary lubrication. In efficiency improvement, the investigation based on authors' report at preceding Conference, the most suitable path of suction for dual cylinder rotary compressor and so on were examined one by one. In efficiency analysis, .each efficiency was evaluated using the result of electric motor test and P-V diagram obtained by measurement of cylinder pressure.

As a result of this study, dynamic characteristics of dual cylinder rotary compressor has been made clear in comparison with that of single cylinder type and compression efficiency has reached about 86% by the investigation of efficiency improvement. The results reported in this paper suggest much applicability of dual cylinder rotary compressor with high efficiency and low vibration to the large capacity class.

EXPERIMENTAL MODEL OF DUAL CYLINDER ROTARY COMPRESSOR

Fig.1 shows the fundamental construction of dual cylinder rotary compressor. This is the subject of experimental and theoretical study in this paper. The model of this study has nominal output of 3.75 kw (5 P class). The main specifications are described in Table 1.

Table 1   Main specifications
of Experimental Model

Power Source	200 V – 50/60 Hz – 3 Phase
Stroke Volume	80.6 cc/rev (4.92 cu.in/rev)
Nominal Output	60,000 BTU

The mechanism of dual cylinder rotary compressor can be explained simply as follows. Partition plate is held between upper cylinder and lower cylinder. Indivisual mechanism in cylinder is the same as conventional rolling piston type rotary compressor. Crankshaft has two eccentric cams with phase difference of 180 degrees. Gas compression of two times per a revolution of crankshaft is performed. The above mentioned points are the fundamental difference between dual cylinder type and single cylinder type.

EXPERIMENTAL APPARATUS AND METHOD

The measurements of pressure in cylinder and suction passage were performed with small piezo type pressure transducers and a strain gage type pressure transducer. A signal of crank angle was picked up with gap sensor. The temperatures of refrigerant gas and oil were measured with thermocouples. Gas flow rate and consumption power were measured by operating actually the experimental model installed in a secondary refrigerant compressor calorimeter. All theoretical and experimental studies in this paper were made under operating conditions ASHRAE "T" of compressor as shown in Table 2.

Table 2 Operating Conditions
(ASHRAE "T")

Power source	200 V - 60 Hz
Condensing Temp.	54.4° C (130° F)
Evaporating Temp.	7.2° C (45° F)
Return Gas Temp.	35.0° C (95° F)
Liquid Temp. entering Exp. Valve	46.1° C (115° F)
Ambient Temp.	35.0° C (95° F)

DYNAMIC CHARACTERISTICS ANALYSIS

Dynamic analysis was performed in order to make dynamic characteristics clear in comparison with single cylinder rotary compressor. It was necessary for dynamic analysis to obtain beforehand gas compression torque and electric motor output torque acting on crankshaft. In order to get gas compression torque, cylinder pressure corresponding to crank angle was measured by operating the experimental model. Fig.2 shows the measured cylinder pressure of compression plenum and suction plenum. Regarding motor torque, the characteristics of electric motor corresponding to slip was obtained by motor test. This reult could be made use of in efficiency evaluation mentioned later. The characteristic curves of electric motor are shown in Fig.3. Friction torque caused at bearings and reaction torque by constraint force at vane tip also act on crankshaft besides the torques mentioned before.

Motion equation of crankshaft is as follows.

$$\mathrm{I}c \cdot \ddot{\theta} = \mathrm{T_m} - (\mathrm{T_g} + \mathrm{T_v} + \mathrm{T_f}) \tag{1}$$

Motor output torque Tm shown in Fig.3 was used as a function of angular velocity $\dot{\theta}$.

$$\mathrm{T_m} = \mathrm{T_m}(\dot{\theta}) \tag{2}$$

Gas compression torque Tg can be expressed as following equations by use of measured cylinder pressure Pc and Ps.

$$\mathrm{T_g} = \sum_i \mathrm{Wg}_i \cdot e \cdot \sin((\alpha_i + \theta_i)/2) \qquad i = 1, 2 \tag{3}$$

$$\mathrm{Wg}_i = (\mathrm{Pc}_i - \mathrm{Ps}_i) \cdot 2\, r_p \cdot \sin((\alpha_i + \theta_i)/2) \cdot \mathrm{H} \tag{4}$$

$$\theta_1 = \theta \quad , \quad \theta_2 = \theta + \pi/2 \tag{5}$$

Subscripts 1 and 2 in the above equations mean respectively upper cylinder and lower cylinder.

Fig.4 and Fig.5 show the schematic view of forces and moments acting on rolling piston and vane. Fv, Fs and Fd in Fig.4 and Fig.5 are constraint forces acting on vane tip and sides. These values can be obtained by solving the equilibrium equations and motion equation of rolling piston discribed as (6) considering the effect of angular velocity $\dot{\theta}$ and angular acceleration $\ddot{\theta}$ of crankshaft.

$$\mathrm{I}p \cdot \dot{\omega}p = \mathrm{Mc} - r_p \cdot \mu_v \cdot \mathrm{Fv} - \mathrm{Mb} \tag{6}$$

The above mentioned torque Tv is expressed using reaction force Fv as follows.

$$\mathrm{Tv} = - \sum_i \mathrm{Fv}_i \cdot e \cdot \sin(\alpha_i + \theta_i) \qquad i = 1, 2 \tag{7}$$

As crankshaft at upper, lower and eccentric bearing can be considered as being under the condition of fluid lubrication, journal bearing model as shown in Fig.6 can be introduced. Friction force can be obtained by solving the basic equation (8) for journal bearing of finite length under fluctuating load. In this solution the average angular velocity of crankshaft was used for simplicity.

$$\frac{1}{r_j^2} \frac{\partial}{\partial \theta_b} \left(h_j^3 \frac{\partial p}{\partial \theta_b} \right) + h_j^3 \frac{\partial^2 p}{\partial z^2} = 6\eta \cdot \mathrm{Cr} [-\varepsilon \{ \omega_j + \omega_b -2(\dot{\phi} + \dot{\psi})\sin\theta_b\} \\ + 2\dot{\varepsilon} \cdot \cos\theta_b] \tag{8}$$

$$\mathrm{Ff} = \eta\, \omega_j r_j^2 \frac{\mathrm{L}}{\mathrm{Cr}} \left\{ \frac{\omega_j - \omega_b}{\omega_j} \frac{2\pi}{\sqrt{1-\varepsilon^2}} + \frac{\varepsilon}{2} \left(\frac{\mathrm{Cr}}{r_j}\right)^2 \frac{\mathrm{W} \cdot \sin\phi}{\eta\, r_j\, \omega_j \mathrm{L}} + \frac{\mathrm{Cr}}{r_j} \cdot \frac{2\pi}{\omega_j} \dot{\phi} \left(\frac{1}{\sqrt{1-\varepsilon^2}} - 1\right) \right\} \tag{9}$$

Total friction torque Tf is expressed as follows.

$$Tf = \sum_i Ff_i \cdot r_i + Tthrust \qquad\qquad i=1,2,3 \qquad\qquad (10)$$

Subscripts 1,2 and 3 in the above equation mean respectively upper,eccentric and lower bearing.

Results Of Analysis And Experiment

All equations mentioned above were solved simultaneously and calculations were continued till solutions converged. In this numerical calculation, the coefficients of friction under the condition of boundary lubrication were determined so that calculated shaft power agreed with that of the experimental model operated actually. Furthermore dynamic analysis of single cylinder rotary compressor was also performed by the same method.

Fig.7 shows fluctuating angular velocity of crankshaft. The measured values are also plotted in the figure. The calculated values agree approximately with the measured. It can be seen from Fig.7 that the amplitude of fluctuation of dual cylinder rotary compressor is about one-fourth of that of single cylinder type.

Fig.8 shows the calculated torque curves.
The above mentioned fluctuation of angular velocity and vibration characteristics are greatly affected by the amplitude of fluctuating torque.
The measured power spectra of vibration are shown in Fig.9. There is much difference in power spectrum of first order which significantly affects vibration characteristics.

Fig.10 shows the polar plot of upper or lower bearing load. Fig.11 shows the predicted locus of crankshaft motion in the bearing. The fluctuation of those in dual cylinder rotary compressor is relatively slight in comparison with that of single cylinder rotary compressor.

SOME INVESTIGATIONS OF EFFICIENCY IMPROVEMENT

The fundamental specification and efficiency improvement were examined mainly experimentally. Some investigations are introduced in the paper.

High efficient round valve was used as discharge valve. Authors already introduced the effect of it in comparison with a conventional type of flat valve at preceding Conference. The schematic construction of round valve is shown in Fig.12.

Regarding the structure of suction passage, branch-
ing path as shown in Fig.13 and two paths as shown in
in Fig.14 were examined by experiments. The results are
shown in Fig.15.
It seems to be considered that decrease in refrigerating
capacity of the compressor with branching suction path
was caused by interference of suction gas. The method
of two paths improved a little E.E.R. as a result.

The effect of length and diameter of inlet pipe was
also studied as another investigation. The results are
shown in Fig.16. The values calculated by the method of
characteristics agree qualitatively with the experimen-
tal values. The most suitable inlet pipe was investi-
gated from the results mentioned above.

As for another examination, for example,deformation
analysis of bearing, partition plate etc were carried
out by FEM and the appropriate shape was attempted.

EFFICIENCY ANALYSIS

Each efficiency was evaluated in the experimental
model which adopted the specifications investigated be-
fore. P-V diagram obtained by measurement of cylinder
pressure and motor efficiency by motor test were used in
this analysis. Each efficiency is defined in this paper
as follows.

Motor
efficiency $\quad \eta_m = \dfrac{\text{Motor output}}{\text{Consumption power}} \qquad (11)$

Mechanical
efficiency $\quad \eta_{me} = \dfrac{\text{Indicated work}}{\text{Motor output}} \qquad (12)$

Compression
efficiency $\quad \eta_c = \dfrac{\text{Theoretical adiabatic work}}{\text{Indicated work}} \qquad (13)$

Compressor
efficiency $\quad \eta_{comp} = \eta_m \cdot \eta_{me} \cdot \eta_c \qquad (14)$

Volumetric
efficiency $\quad \eta_v = \dfrac{\text{Actual gas flow}}{\text{Theoretical gas flow}} \qquad (15)$

Each of analyzed efficiency is shown in Fig.17 com-
paring with that of single cylinder rotary compressor.
The difference seen in compression efficiency and volu-
metric efficiency is caused by the increase of overcom-
pression and oil quantity circulating in a calorimeter.
The growth of discharge pulsation in single cylinder
rotary compressor can be considered as the reason of
these phenomena mentioned above.

As for mechanical loss, the result of the above dynamic analysis agreed.as described in Table 3,approximately with the value obtained using following equation based on experiment.

$$Lme = Lm - Li \qquad (16)$$

Table 3   Mechanical Loss

Means Loss Ratio	Experimental	Theoretical
$\dfrac{Lme}{Lad} \times 100\,(\%)$	9.9	9.1

CONCLUSION

The basic study of dual cylinder rotary compressor in 5 HP class was performed by theory and experiment as compared with single cylinder rotary compressor of conventional type.

Those are summarized as follows.
1) Dynamic characteristics of dual cylinder rotary compressor of nominal output 3.75 kw was made clear by solving numerically motion equations and operating the experimental model actually.
   Particularly the amplitude of fluctuating angular velocity of crankshaft and torque was about one-fourth of that of single cylinder type.
   As a result, vibration was very low.

2) The fundamental structure and efficiency improvement were examined.
   As suction passage, the method of two paths was suitable to dual cylinder rotary compressor in particular.

3) Each of efficiency was evaluated mainly by experiment. In this evaluation, compression efficiency and volumetric efficiency were superior to those of single cylinder rotary compressor.
   Particularly compression efficiency was about 86 % and E.E.R. reached 11.1 .

It seems to indicate that these results reported above mean much possibility of dual cylinder rotary compressor with high efficiency and low vibration in large capacity class.

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NOMENCLATURE

Ic = motion of inertia of crankshaft and rotor
θ = crank angle
$\cdot$ = time differential
Tm = motor output torque
Tg = gas compression torque
Tv = reaction torque
Tf = friction torque
e = eccentricity of crank
α = offset angle of rolling piston center
r_p = rolling piston radius
H = rolling piston height
Pc = pressure measured in conpression plenum
Ps = pressure measured in suction plenum
Fv = reaction force at vane tip
Ff = friction force at bearing
r = crankshaft radius
T_{thrust} = friction torque at thrust bearing
Ip = moment of inertia of rolling piston
ω_p = angular velocity of rolling piston
μ = coefficient of friction
Mc = friction moment at rolling piston bearing
Mb = friction moment at rolling piston face
θ_b = coordinate of bearing angle
z = coordinate of bearing length
ϕ = attitude angle of bearing
ψ = direction angle of bearing load
ω_j = angular velocity of journal
ω_b = angular velocity of bearing
r_j = journal radius
Cr = radius clearance
ε = eccentricity ratio of journal
L = bearing length
η = viscosity of lubricating oil
W = bearing load
Lm = motor output
Li = indicated work
Lad = theoretical adiabatic work
Lme = mechanical loss

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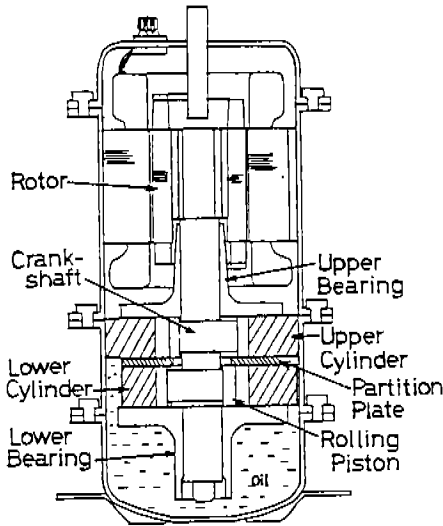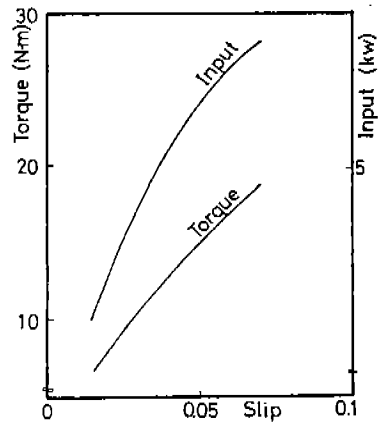
Fig.1 Sectional View of Experimental Model
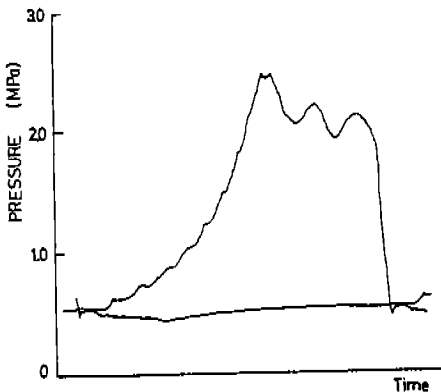
Fig.2 Measured Pressure

Fig.3 Motor Characteristics

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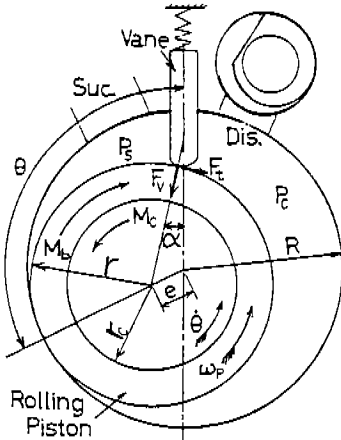
Fig.4　Schematic View and Moment acting on Rolling Piston
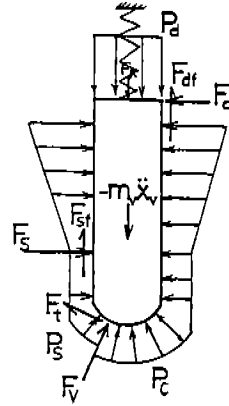
Fig.5　Forces acting on Vane
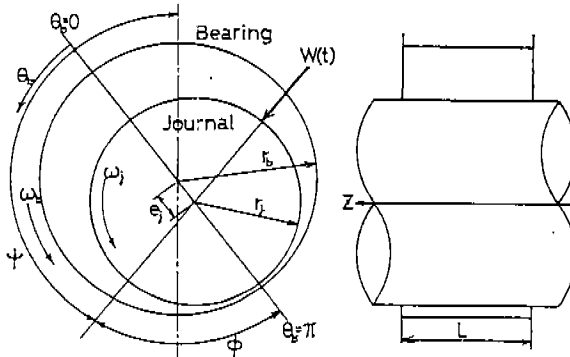
Fig.6　Model of Journal Bearing
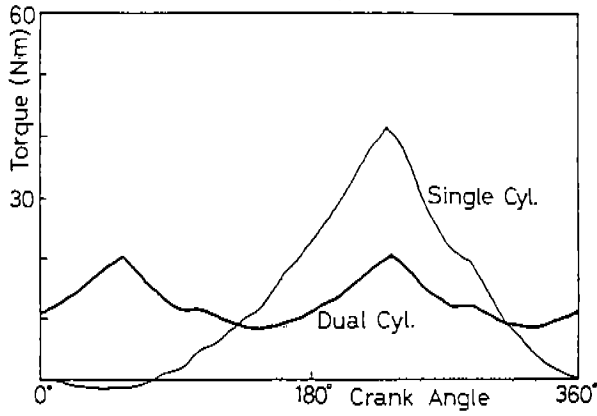
Fig. 8   Calculated Torque
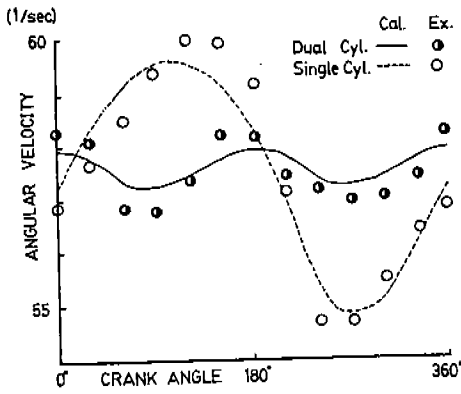
Fig.7   Angular Velocity
of Crankshaft
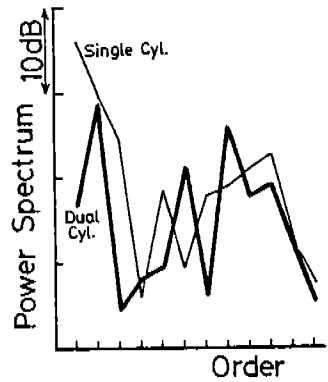
Fig.9   Power Spectra
of Vibration

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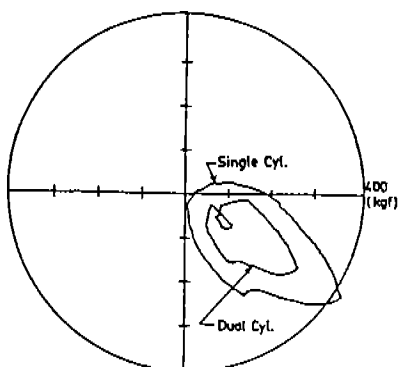
Fig.10   Polar Plot of Bearing Load
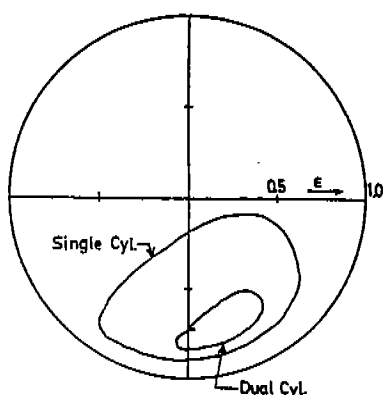
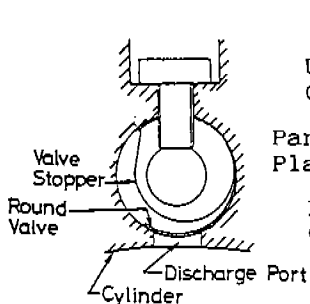Fig.11   Predicted Locus of Crankshaft Motion
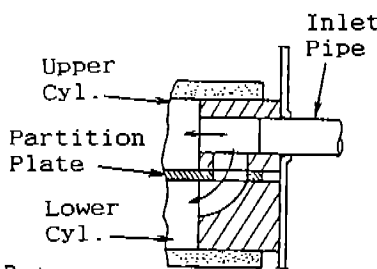

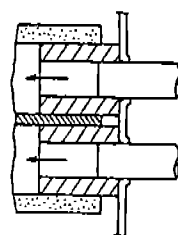Fig.12   Construction of Round Valve
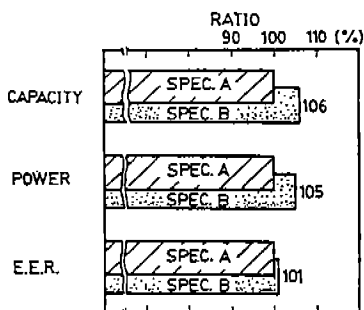

Fig.13   Branching (Spec. A)

Fig.14   Tow Paths (Spec. B)

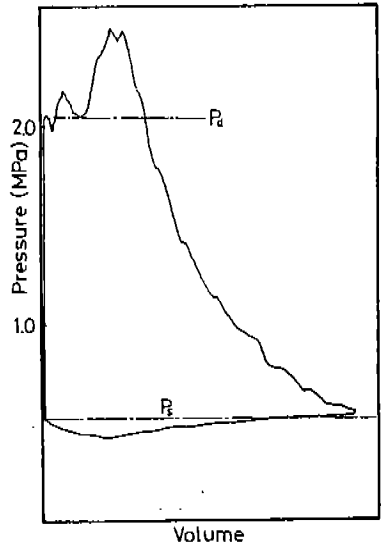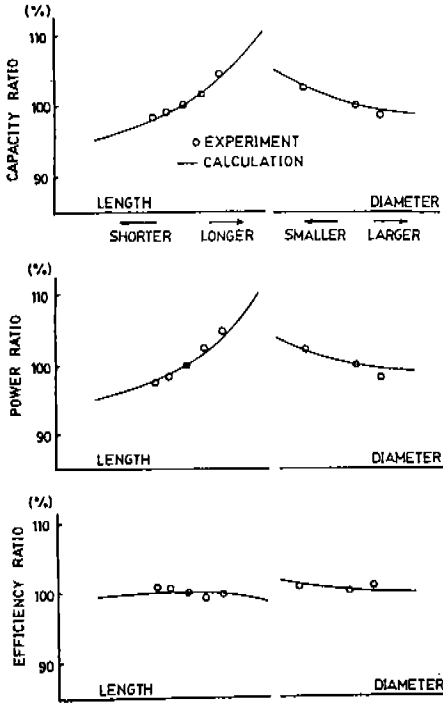Fig.15   Comparison between Spec. A and Spec. B

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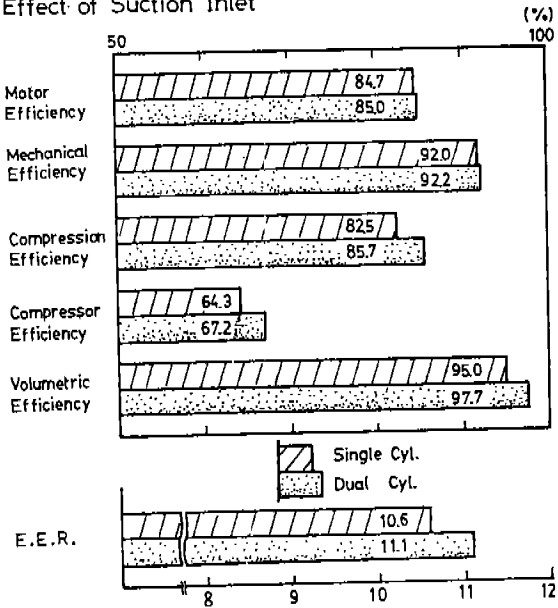Fig.16 Effect of Suction Inlet

Fig17 P-V Diagram

Fig.18 Efficiency Evaluation

STUDY ON ACTUAL PROFILE SURFACE AND ENGAGING CLEARANCE OF SCREW COMPRESSOR ROTORS

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ABSTRACT

A simple and effective method of obtaining the mathematical description of actual profile surface of screw rotors is put forward in this paper by assuming that the actual surface is composed of a theoretical surface and a surface difference. The surface difference is obtained through mathematical processing by computer to the screw line error, profile line error and graduation error determined by a special measurement apparatus. According to the engaging principle, we can calculate the actual engaging clearance between two mating screw rotors.

NOMENCLATURE

A centre distance between male rotor and female rotor

ΔA centre distance error

i transmission ratio

p characteristic parameter of screw line

Δp characteristic parameter error

T screw line leader

ΔT screw line leader error

t profile line parameter

τ screw line parameter

τ_z wraple angle of rotor

X, Y, Z profile line or surface coordinates