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ABSTRACT

In this paper, numerical and experimental analysis of rolling piston type rotary compressor for refrigerator is performed. In order to analyze rotary compressor, computer programing is developed to predict the behavior of compressor's moving parts and the leakage phenomena through clearances between parts such as roller, cylinder, shaft, main-bearing, sub-bearing and vane. Dynamics of valve motion is also considered. Optimal dimensions of the compressor are calculated using this program and a prototype is made according to the predictions.

Several experiments and numerical predictions reveal characteristics of rotary compressor. The results show that there are certain ranges of clearances which maximize the performance and gas leakage through clearances acts an important role in compressor refrigerating capacity.

NOMENCLATURE

B	Vane thickness
C_{rc}	Roller and cylinder clearance
C_{rb}	Roller and bearings "
C_{re}	Roller and eccentric shaft "
C_{sb}	Shaft and bearing "
C_{vb}	Vane and bearing "
C_{vc}	Vane and cylinder slot "
D	Cylinder depth
E	Shaft eccentricity
F_A	Normal force on contact point of roller and cylinder wall
F_{Af}	Friction force on contact point of roller and cylinder wall
F_P	Pressure force
F_V	Normal force on vane tip
F_{Vf}	Friction force on vane tip
H	Cylinder height
h_o	Minimum film thickness
I_r	Moment of inertia of roller
k	Spring constant
κ	Isentropic coefficient
L_{rb}	Coefficient for $\dot{m}_{rbd}$, $\dot{m}_{rbs}$
L_{rc}	Coefficient for $\dot{m}_{rc}$
L_{vb}	Coefficient for $\dot{m}_{vb}$
L_{vc}	Coefficient for $\dot{m}_{vcd}$, $\dot{m}_{vcs}$
$m(\theta)$	Mass of R-12 in cylinder
$\dot{m}_v$	Mass discharge rate via valve
$\dot{m}_{rbd}$	Mass leakage rate througt C_{rb}
$\dot{m}_{rbs}$	"
$\dot{m}_{rc}$	Mass leakage rate through C_{rc}
$\dot{m}_{vcd}$	Mass leakage rate through C_{cv}
$\dot{m}_{vcs}$	"
M_r	Mass of roller
M_v	Mass of vane
n	Polytropic coefficient
N_r	Angular velocity of roller
$P(\theta)$	Pressure in cylinder volume
P_d	Discharge pressure
P_s	Suction pressure
Q	Refrigrating capacity
R_c	Inner radius of cylinder
R_e	Radius of eccentric shaft
R_r	Outer radius of roller
R_s	Radius of shaft
T_M	Motor torque
T_r	Frictional torque on roller
T_s	" on eccentric shaft
$V(\theta)$	Cylinder volume
W_r	Force on roller
W_s	Force on shaft
X	Vane displacement
y	Valve lift height
$\mathcal{E}$	Solubility difference of R-12
$\rho(\theta)$	Density of R-12 in cylinder
ρ_{oil}	Density of lubricating oil
ρ_s	Density of suction gas
$\O_{FP}$	Acting angle of F_P
$\O_r$	Acting angle of W_r
$\O_s$	Acting angle of W_s
θ	Angular displacement of shaft
η	Viscosity of lubricating oil
μ	Friction coefficient at vane tip

INTRODUCTION

Rolling piston type rotary compressors are widely used for refrigerator and air conditioner because of its high efficiency and compactness in size. The performance of a compressor depends on the refrigerating capacity and required power to drive it. There are many factors that influence the performance such as over compression loss, friction loss, over heating and gas leakage through clearances. And several investions about compressor performance are presented by means of analytical and experimental analysis. But there are few experimental data about the effect of clearances on compressor performance, even though theoretical analysises are reported by many authors.[2,4] About roller and bearing clearanes, references [5] and [6] present its effect on performance.

This paper places special emphasis on the effect of gas leakage through clearances between parts so that engineers can predict the compressor's performance before production. Theoretical analysis is done using computer simulation. In this analysis, we consider each part as a free body. Calculating forces acting on the parts, dynamic behavior of the parts and required power to drive shaft can be conjectured. For bearings, we assumed it as infinite one dimensional journal bearing and valve system is modeled as a simple single degree of freedom.

Gas in the cylinder volume is pressurized as the volume is reduced. The pressure difference between suction and discharge chambers causes gas and lubricating oil to leak through the clearances and discharge via valve system. To evaluate the effect of clearances, two kinds of model equations are tried, one is nozzle flow model and the other is laminar poiseuille flow model.

Experiments meausring input power and refrigerating capacity were performed for various size of clearances with a prototype compressor mounted on caloriemeter stand. These results show that the leakage pattern is laminar poiseuille flow rather than nozzle flow.

THEORETICAL ANALYSIS

Theoretical analysis dealing with dynamic motions of the parts and gas leakage phenomena is presented here to predict the performance of rotary compressor. In this approach, each part of compressor is considered as a free body and force balance equations are solved by numerical method.

Pressure in the Cylinder Volume and Refrigerating Capacity

By considering continuity of mass, refrigerating capacity and properities of the gas in the cylinder volume can be obtained as follows:

$$Q = \frac{\Delta h \cdot \dot{\theta} \cdot 3600}{2 \cdot \pi} \cdot \int \{\dot{m}_v - (\dot{m}_{rbs} + \dot{m}_{rbd} + \dot{m}_{vcs} + \dot{m}_{vcd})\} \cdot dt \tag{1}$$

$$\frac{dm(\theta)}{dt} = -(\dot{m}_{rc} + \dot{m}_{vb} - \dot{m}_{rbd} - \dot{m}_{vcd}) - \dot{m}_v \qquad \text{I.C}: \ m(0) = \rho_s \cdot V(0) \tag{2}$$

$$\rho(\theta) = \frac{m(\theta)}{V(\theta)} \tag{3}$$

$$P(\theta) = P(\theta - \delta\theta) \cdot \left\{\frac{\rho(\theta)}{\rho(\theta - \delta\theta)}\right\}^n \tag{4}$$

Dynamics of Roller

Equations of motion of the roller for the free body diagram Fig.1 are given as Eq.5,6 and 7, where laminar viscous friction force and torque are applied for upper and lower sides of the roller.

$$0 = -F_P \cdot \cos\phi_{FP} + W_r \cdot \cos\phi_r + M_r \cdot E \cdot \dot{\theta}^2 - F_{Af} - F_V \cdot \cos(\theta+\alpha) + F_{Vf} \cdot \sin(\theta+\alpha) \tag{5}$$

$$0 = F_P \cdot \sin\phi_{FP} - W_r \cdot \sin\phi_r + F_{rs} + F_{Af} - F_V \cdot \sin(\theta+\alpha) - F_{Vf} \tag{6}$$

$$I_r \cdot \dot{N}_r = F_{Af} \cdot R_r - T_r - T_{rs} - F_{Vf} \cdot R_r \tag{7}$$

$$F_P = \sqrt{2} \cdot \{P(\theta) - P_s\} \cdot \overline{AB} \cdot H$$

$$\phi_{FP} = -\operatorname{atan}\left(\overline{AB} / \sqrt{4 \cdot R_r^2 - \overline{AB}}\right)$$

$$F_{rs} = \frac{4 \cdot \pi \cdot \eta \cdot \dot{\theta}}{C_{rb}} \cdot E \cdot (R_r^2 - R_e^2)$$

$$T_{rs} = \frac{2\cdot\pi\cdot\eta\cdot N_r}{C_{rb}}\cdot(R_r^4 - R_e^4)$$

Dynamics of Shaft

Referring to Fig.2, the motion of shaft is governed by Eq.8,9 and Eq.10. The rotational speed of the shaft is nearly constant.

$$0 = M_s\cdot E\cdot\dot{\theta}^2\cdot\cos\theta - W_r\cdot\cos(\theta+\phi_r) - W_s\cdot\cos\phi_s \tag{8}$$

$$0 = M_s\cdot E\cdot\dot{\theta}^2\cdot\sin\theta - W_r\cdot\sin(\theta+\phi_r) - W_s\cdot\sin\phi_s - F_{ss}\cdot\cos\theta \tag{9}$$

$$0 = T_M - W_r\cdot E\cdot\sin\phi_r - T_r - T_s - T_{ss} - F_{ss}\cdot E \tag{10}$$

$$F_{ss} = \frac{4\cdot\pi\cdot\eta\cdot\dot{\theta}}{C_{eb}}\cdot E\cdot R_e^2$$

$$T_{ss} = \frac{2\cdot\pi\cdot\eta\cdot\dot{\theta}}{C_{eb}}\cdot(R_e^4 - R_s^4)$$

Dynamics of Vane

To get the forces acting on the roller, it is necessary to know normal and tangential forces acting on the point B' at Fig.3, and the resulting equations are

$$F_v = -k\cdot(X-X_o) - M_v\cdot\ddot{X} + 4\cdot B\cdot\left(\frac{H}{C_{vs}} + \frac{D}{C_{vb}}\right)\cdot X + D\cdot H\cdot\left(P_d - \frac{P(\theta) - P_s}{2}\right) - H\cdot\overline{BB'}\cdot(P(\theta) - P_s) \tag{11}$$

$$F_{vf} = -\mu\cdot F_v\cdot\frac{V_{bt}}{|V_{bt}|} \tag{12}$$

$$X = R - E\cdot\cos\theta - \sqrt{R_r^2 - E^2\cdot\sin^2\theta}$$

$$\overline{BB'} = \frac{R_v\cdot E\cdot\sin\theta}{R_r + R_v}$$

$$V_{bt} = R_r\cdot\left[\frac{E}{R_r}\cdot\dot{\theta}\cdot\cos\theta\cdot\left\{\sqrt{1-\left(\frac{E}{R_r}\right)^2\cdot\sin^2\theta} + \left(\frac{E}{R_r}\right)^2\cdot\frac{\sin^2\theta}{\sqrt{1-\left(\frac{E}{R_r}\right)^2\sin^2\theta}}\right\} - N_r\right]$$

Contact Point A of Roller and Cylinder

We think it as a rolling contact with slip, and solving Reynolds' equation Eq.13 for one-dimensional rolling contact gives Eq.14 and 15. Detail procedure reaching to the results is shown in reference [1].

$$\frac{dP}{dx} = 6\cdot U\cdot\eta\cdot\frac{h - \bar{h}}{h^3} \tag{13}$$

$$F_A = \left(2 - \frac{8}{\pi}\cdot P^*\right)\cdot U\cdot\eta\cdot\frac{R_{com}}{h_o}\cdot H \tag{14}$$

$$F_{Af} = \frac{3\cdot\pi}{4}\cdot\eta\cdot H\sqrt{\frac{2\cdot R_{com}}{h_o}}\cdot\left\{-V_{rc} - \left(\frac{2}{3} - \frac{8}{3}\cdot\pi\cdot P^*\right)\cdot U\right\} \tag{15}$$

$$P^* = \frac{h_o^2}{\sqrt{2\cdot R_{com}\cdot h_o}\cdot 6\cdot U\cdot\eta}\cdot(P(\theta) - P_s)$$

$$R_{com} = \frac{R_r\cdot R_c}{R_c + R_c}$$

$$U = 2\cdot R\cdot\dot{\theta} - V_{rc}$$

$$V_{rc} = E\cdot\dot{\theta} - R_r\cdot N_r$$

Journal Bearing

There are two journal bearings in rotary compressor, one is roller and eccentric shaft and the other is bearing(sub and main) and shaft. It is cosidered as one dimensional infinitely long journal bearing with incompressible fluid. By solving Reynolds equation[7], eccentric ratio and attitute angle are discribed as Eq.17 and 18. Petroff's law is used to evaluate the frictional torque.

$$\frac{d}{R\cdot d\psi}(h^3\cdot\frac{dP}{R\cdot d\psi}) = 6\cdot\eta\cdot\Omega\cdot\frac{dh}{d\psi} + 12\cdot\eta\cdot\frac{dh}{dt} \tag{16}$$

$$\frac{2\cdot d(\psi+\emptyset)}{\Omega\cdot dt} = 1 - \frac{a\cdot W^*\cdot \sin\psi}{6\cdot\pi\cdot\varepsilon\cdot b} \tag{17}$$

$$\frac{2\cdot d\varepsilon}{\Omega\cdot dt} = \frac{W^*\cdot\cos\psi\cdot(1-\varepsilon^2)^{\frac{3}{2}}}{3\cdot\pi\cdot b} - \frac{2\cdot\varepsilon\cdot(1-\varepsilon^2)^{\frac{3}{2}}\cdot\sqrt{1+\{(\frac{1-\varepsilon^2}{1+\varepsilon^2/2})^2\cdot 1\}\cdot\cos^2\psi\cdot W^*}}{3\cdot\pi^2\cdot b^2} \tag{18}$$

$$W^* = \frac{W}{\eta\cdot\Omega\cdot R\cdot H\cdot(R/C)^2}$$

$$a = (2+\varepsilon^2)\cdot(1-\varepsilon^2)^{\frac{1}{2}}$$

$$b = \frac{c+3}{c+1.5} \quad \text{when} \quad \frac{2\cdot d\varepsilon}{\Omega\cdot dt} > 0$$

$$\frac{c}{c+1.5} \quad \text{when} \quad \frac{2\cdot d\varepsilon}{\Omega\cdot dt} < 0$$

$$c = (1-\varepsilon^2)^{\frac{3}{2}}\cdot\{(\frac{\Omega\cdot dt - 2\cdot d(\psi+\phi)}{2\cdot d\varepsilon})^2 + \frac{1}{\varepsilon^2}\}^{\frac{1}{2}}$$

$$T = 2\cdot\pi\cdot\eta\cdot\frac{\Omega\cdot R^3\cdot H}{C} \tag{19}$$

Valve System

Valve system is modeled as simple one degree of freedom[3]. With incompressible nozzle flow assumption, the mass flow flow rate through valve is calculated from Eq.21.

$$\frac{d^2y}{dt^2} + 2\cdot\zeta\cdot w_n\cdot\frac{dy}{dt} + w_n^2\cdot y = F(y) \tag{20}$$

$$\dot{m}_v(\theta) = A_v\cdot P(\theta)\cdot\sqrt{\frac{2\cdot k}{k-1}\cdot\rho(\theta)\cdot P(\theta)\cdot\{(\frac{P_d}{P(\theta)})^{\frac{k}{2}} - (\frac{P_d}{P(\theta)})^{\frac{k+1}{k}}\}} \tag{21}$$

Leakage

Gas leakage through the clearances as shown in Fig.4 is unavoidable and it is important to know how the gas leaks. In this section, Gas leakage model is tried to predict the phenomena. An assumption that refrigerant gas leaks as mixture of lubricating oil and refrigerant gas R-12 is considered to be resonable. Calculating oil flow rate and the difference of solubility between high and low pressure sides, we can predict the gas leakage rate.

Refrigerant gas leakage through roller and cylinder wall is driven from Reynolds' equation Eq.13, and reads as fellow.

$$\dot{m}_{rc} = L_{rc}\cdot Æ(P(\theta),P_s)\cdot\frac{\rho_{oil}\cdot H\cdot h_o}{2}\cdot(\frac{4\cdot U}{3} + \frac{P^*}{3}) \tag{22}$$

Gas leakage through roller sides and vane sides take place when the lubricating oil flows as laminar flow.

$$\dot{m}_{rbs} = L_{rb}\cdot Æ(P_d,P_s)\cdot\frac{\rho_{oil}}{12\cdot\eta}\cdot(\frac{C_{rb}}{2})^3\cdot\frac{R_r\cdot\theta}{R_r-R_e}\cdot(P_d-P_s)\cdot 2 \tag{23}$$

$$\dot{m}_{rbd} = L_{rb}\cdot Æ(P_d,P(\theta))\cdot\frac{\rho_{oil}}{12\cdot\eta}\cdot(\frac{C_{rb}}{2})^3\cdot\frac{R_r\cdot(2\cdot\pi-\theta)}{R_r-R_e}\cdot(P_d-P(\theta))\cdot 2 \tag{24}$$

$$\dot{m}_{vcs} = L_{vc}\cdot Æ(P_d,P_s)\cdot\frac{\rho_{oil}}{12\cdot\eta}\cdot(\frac{H\cdot C_{vc}^3}{D\cdot 2} + \frac{B\cdot C_{vb}^3}{D}\cdot 2)\cdot(P_d-P_s) \tag{25}$$

$$\dot{m}_{vcd} = L_{vc}\cdot Æ(P_d,P(\theta))\cdot\frac{\rho_{oil}\cdot H\cdot C_{vc}^3}{12\cdot\eta\cdot D\cdot 2}\cdot(P_d-P(\theta)) \tag{26}$$

On the contrary to the above equations, incompressible nozzle flow model is used to evaluate the leakage between vane and bearings because it is thought that there is no enough oil to fill the gap and its width is relatively small when compared with the other parts.

$$\dot{m}_{vb} = L_{vb}\cdot C_{vb}\cdot X\cdot P(\theta)\cdot\sqrt{\frac{2\cdot k}{k-1}\cdot\rho(\theta)\cdot P(\theta)\cdot\{(\frac{P_s}{P(\theta)})^{\frac{k}{2}} - (\frac{P_s}{P(\theta)})^{\frac{k+1}{k}}\}} \tag{27}$$

EXPERIMENT

A prototype compressor with flanged case was made for experimental purpose. Various sizes of clearances can be obtained by replacing parts which were machined precisely. Thirty two rollers, two shafts, twelve vanes, five main-bearing and five sub-bearing are used in this experiment. In assembling cylinder and main-bearing, we used electric micrometer to control concentricity within 2μm. It was attached to calorimetric test stand to measure refrigerating capacity and power input at ASHRAE condition shown below:

Condensing Temperature	:	54.4 °C (130 °F)
Evaporating Temperature	:	-23.3 °C (-10 °F)
Return Gas Temperature	:	32.2 °C (90 °F)
Ambient Temperature	:	32.2 °C (90 °F)
Sub-cooling Temperature	:	32.2 °C (90 °F)

The case temperature of the compressor was set to be 80 °C by forced cooling which may be usual case temperature when attached to refrigerator. Also several two phase induction motor were made to drive the mechanical parts. It has about 68% of efficiency at normal running condition and designed for 220 volt and 60 Hz.

RESULTS AND DISCUSSION

Fig.5,6,7,8,9 and Fig.10 show the results. Mark o represents experimental result and solid line represents predicted result. Values of input power, capacity and clearances in these figures are nondimensionalized. The predicted capacity curves are obtained from Eq.1 with coefficients L_{rc}, L_{rb}, L_{vc} and L_{vb} which are 0.22, 2.0, 2.0 and 0.95 respectively in leakage model equations.

From Fig.5, we know that refrigerating capacity and power input decrease as the clearance C_{rc} increases. The clearances related to the leakage througt roller and cylinder wall, $\dot{m}_{rc}$, are C_{rc}, C_{re} and C_{sb}. But the effect of C_{re} and C_{sb} on capacity is relatively small when compared with the effect of C_{rc}. This effect is due to the eccentric ratios of the journal bearings, which vary according to the size of clearances and influence the film thickness between roller and cylinder wall.

The coefficient L_{rc} is a kind of time constant and can be thought to be a portion in which refrigerant gas R-12 dissolves into and separates from lubricating oil. The rotating speed of the shaft is so fast that there is no enough time for gas to fully dissolve into and separate from the lubricating oil. This coefficient is thought to be function of rotational speed of the shaft and 0.22 in this prototype.

Leakage through clearance C_{rb} and C_{vc} acts as the most important role in compressor performance. Gas leaks from compressor case chamber to cylinder voulme. The leakage to suction side reduces refrigerating capacity, and the leakage to discharge side increases input power. It is supposed that the oil in the chamber contains much formy refrigerant gas because moving parts such as vane and eccentric shaft whirl the oil. So when the oil flows through this clearances, the refrigerant gas R-12 leaks not only in a state of solution but also in foam mixed with oil. Taking into account of this phenomena, we set the coefficient L_{rb} and L_{vc} as two. this model is in good agreement with experimental result.

For the leakage $\dot{m}_{vb}$ through clearance, C_{vb}, between vane and bearings, nozzle flow model with nozzle coefficient of 0.95 gives a good result. This clearance also has an effect on the leackage $\dot{m}_{cvs}$, which is included in Eq.25.

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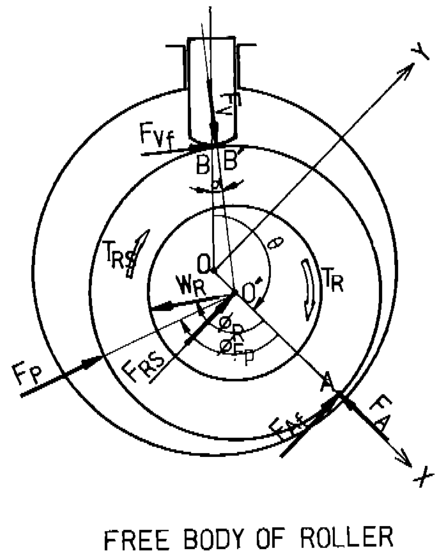

Fig.1 Free body diagram of Roller

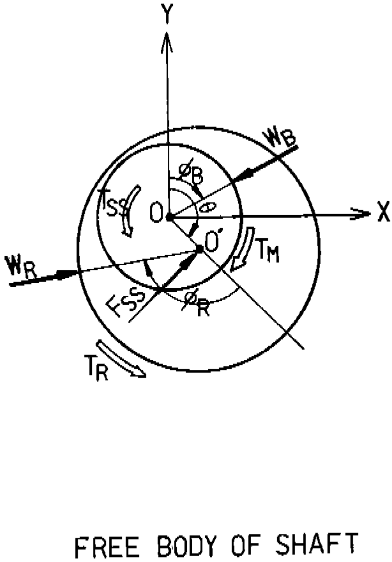

Fig.2 Free body diagram of shaft

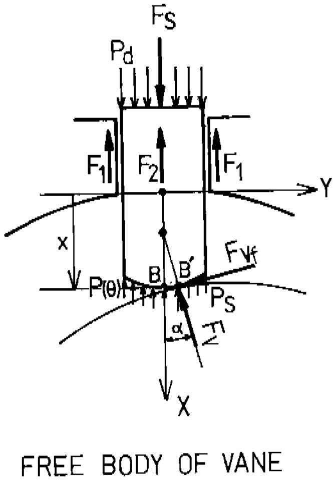

Fig.3 Free body diagram of vane

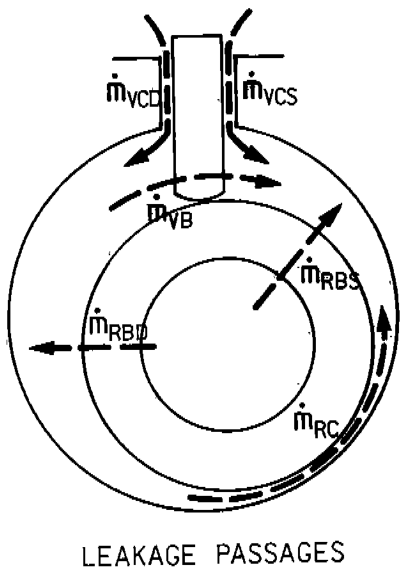

Fig.4 Leakage passages of rotary compressor

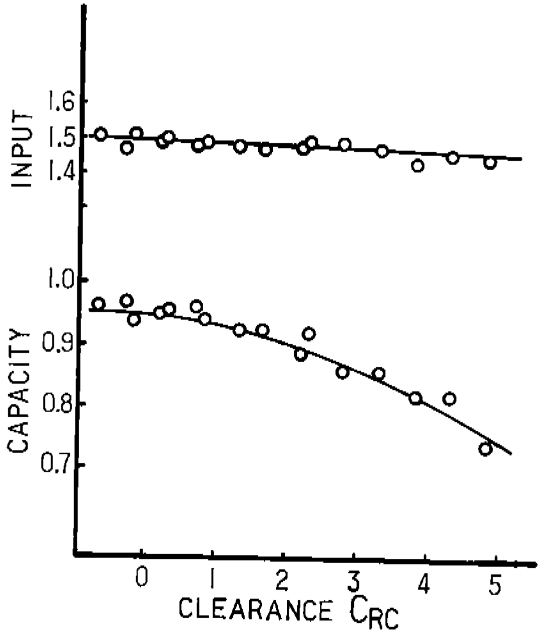

Fig.5 Effect of clearance between roller and cylinder wall

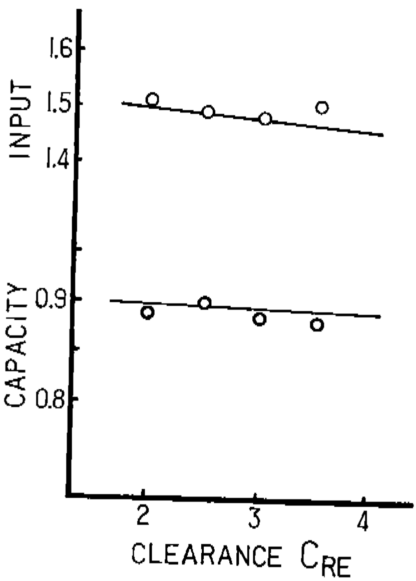

Fig.6 Effect of clearance between roller and eccentric shaft

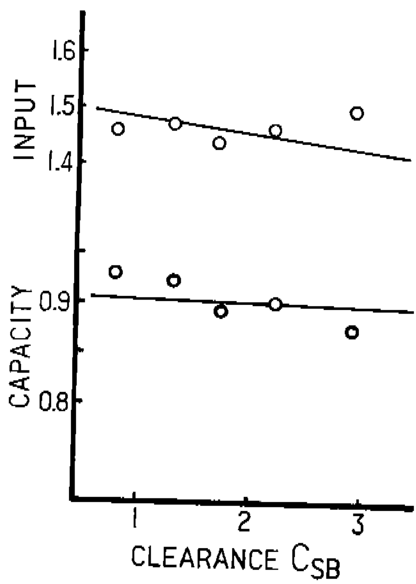

Fig.7 Effect of clearance between shaft and bearing

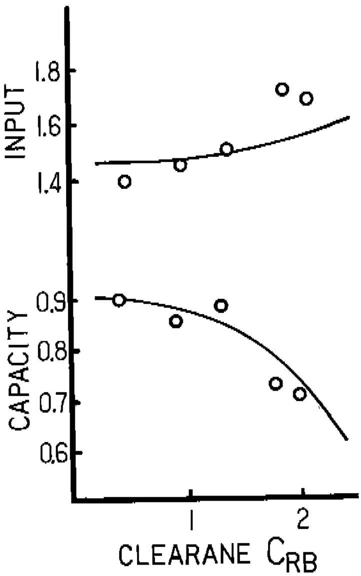

Fig.8 Effect of clearance between roller and bearings

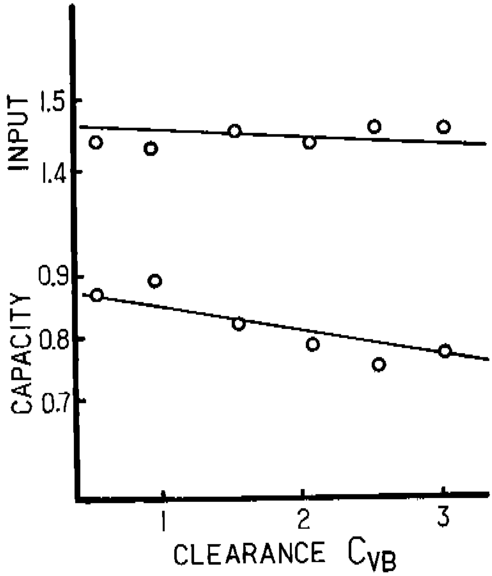

Fig.9 Effect of clearance between vane and bearings

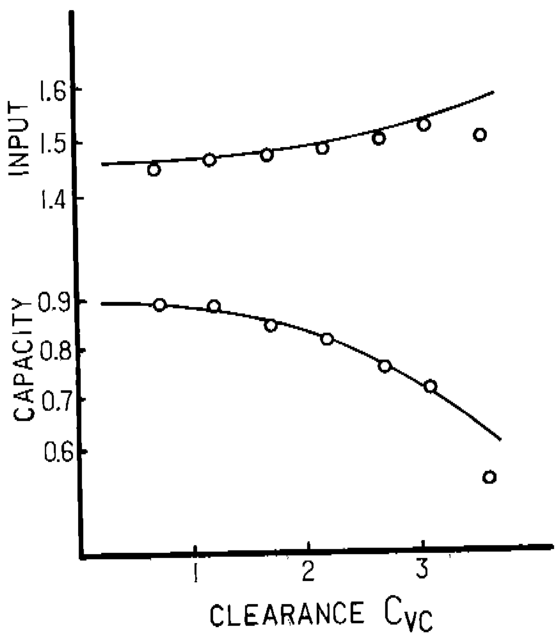

Fig.10 Effect of clearance between vane and cylinder slot