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SOME APPROACHES TOWARDS A HIGH EFFICIENT ROTARY COMPRESSOR

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ABSTRACT

The paper refers to some approaches towards the realization of a rolling piston type hermetic compressor with high efficiency for a room air conditioner and an unitary air conditioner.
There are several loss factors which affect a rotary compressor. Particularly in this paper, the losses which are caused in the cylinder of the rotary compressor have been investigated, experimentally in the main.
As a result, volumetric efficiency and compression efficiency have been greatly improved, and a high efficient rotary compressor has been realized.

INTRODUCTION

A rolling piston type rotary compressor, which has remarkable features that are small size, light weight and high efficiency, has been used widely for a room air conditioner answering the request of the society, that is energy saving, in these days.

So several kinds of approaches were studied for offering a rotary compressor with high efficiency. Some reports[1~4] of performance analysis for a rolling piston type rotary compressor were presented up to today. The factors of losses that cause the reduction of efficiency were analyzed experimentally or theoretically, and those are going to be almost clear.

But all these were the studies of a rotary compressor with small capacity for a small scale air conditioner, and that of a rotary compressor with relatively large capacity (for example 42,000 BTU class) has not yet been reported.

This paper deals with analysis and improvement of efficiency for a rolling piston type rotary compressor with large capacity (42,000 BTU class) by experiment and theory. We have analyzed losses on the basis of the P-V diagram obtained by the measurements of cylinder pressure and evaluated each efficiency of the compressor.

Authers have developed and introduced the discharge valve device of new type, which is a round valve, in a rotary compressor to obtain higher compression efficiency.
It is reported in the present paper that compression efficiency has been improved greatly comparing with a flat valve of old type.
Furthermore some approaches towards higher efficiency were studied. These are the reduction of discharge and suction passage losses.
As a result of this study, compression efficiency has increased from 69% to 86% comparing with a flat valve of common type.
Volumetric efficiency has been also improved and reached 98% as an additional effect.

EXPERIMENTAL APPARATUS AND METHOD

Fig. 1 shows a front cross sectional view of a test compressor used for the experiments.
The test compressor was a rolling piston type rotary compressor of nominal output 42,000 BTU.
Table 1 shows the main specifications of the test compressor.

The measurements of pressure in the cylinder and suction inlet were performed with small piezo type pressure transducers and a strain gage type pressure transducer. The temperatures of refrigerant gas and oil were measured with thermocouples.
A signal of crank angle was picked up and the motion of a crankshaft in the bearing was measured with gap sensors. Gas flow rate and consumption power were measured by a secondary refrigerant compressor calorimeter.
Table 2 shows operating conditions ASHRAE "T" of compressor.

Table 1 Main Specifications of Test Compressor

Power Source	200 V - 50/60 Hz - 3 Phase
Stroke Volume	56.9 cc/rev (3.47 cu.in/rev)
Nominal Output	42,000 BTU

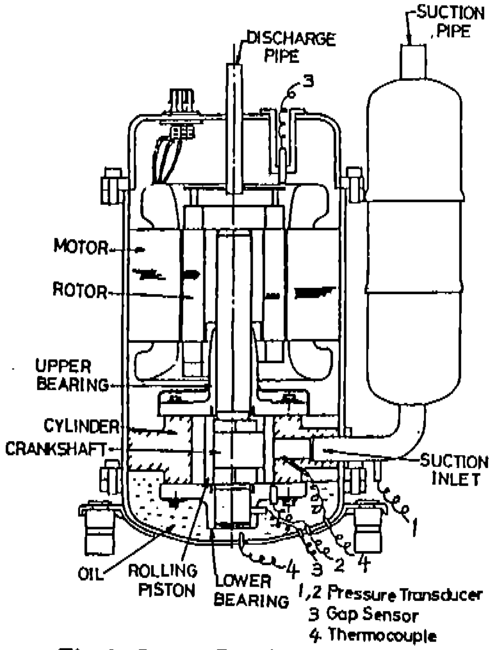


Fig.1 Cross Sectional View

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)

LOSS AND EFFICIENCY

Various losses are produced in the compressor. Classification of loss factors are shown in Table 3.

Table 3 Classification of Loss Factors

- Consumption power
  - Motor loss
    - Copper loss
    - Iron loss
  - Motor output
    - Mechanical loss
      - Rolling piston bearing loss
      - Bearing loss
      - Thrust loss
      - Vane tip loss
      - Vane side loss
    - Indicated work
      - Adiabatic work
      - Overshooting loss
      - Undershooting loss
      - Reexpansion loss
      - Heat loss (including leakage loss)

Each of efficiency is defined in this paper as follows.

Motor efficiency
$$\eta_m = \frac{\text{Motor output}}{\text{Consumtion power}} \quad (1)$$

Mechanical efficiency
$$\eta_{me} = \frac{\text{Indicated work}}{\text{Motor output}} \quad (2)$$

Compression efficiency
$$\eta_c = \frac{\text{Theoretical adiabatic work}}{\text{Indicated work}} \quad (3)$$

Compressor efficiency
$$\eta_{com} = \eta_m \cdot \eta_{me} \cdot \eta_c \quad (4)$$

Volumetric efficiency
$$\eta_v = \frac{\text{Actual gas flow}}{\text{Theoretical gas flow}} \quad (5)$$

These were obtained mainly by the experiments in this paper. On the other hand, theoretical analysis that estimated the improvement of efficiency was studied too, compared with the results of the experiments and the applicability of it was confirmed.

Motor Efficiency

It was obtained by motor test and calorimeter test.

Mechanical Loss

Total mechanical loss was obtained by the following equation. Each of mechanical loss was theoretically analyzed.

$$L_{me} = L_c \cdot \eta_m - L_i = L_m - L_i \quad (6)$$

Indicated Work

It was calculated on the basis of the P-V diagram (Fig.2) obtained by the experiments.

$$L_i = \oint P\,dV \quad (7)$$

Theoretical Adiabatic Work

It was calculated by the following equation.

$$L_{ad} = \frac{G_a \cdot \Delta i}{0.86} = \eta_v \frac{n}{n-1} P_s V_s \left\{ \left(\frac{P_d}{P_s} \right)^{\frac{n-1}{n}} - 1 \right\} \frac{N}{60} A \quad (8)$$

Overshooting Loss and Undershooting Loss

$$L_{dis} = \int_{V_{cl}}^{V_1} (P - P_d)\,dV \quad (9)$$

$$L_{suc} = \int_{V_{cl}}^{V_2} (P_s - P)\,dV \quad (10)$$

Overshooting loss and undershooting loss were experimentally obtained using the above equations. Those were theoretically also analyzed by the characteristic method for expectation of the improvement.

Reexpansion Loss

Reexpansion loss is the noneffective work to compress residual gas in clearance volume again.

It was calculated by the following equations.

$$\Delta P_s = P_s\left\{(V_s + V_{cl}\rho^{n})/(V_s+V_{cl})\right\}^{n} - P_s \qquad (11)$$

$$L_{Top} = \left\{\frac{n}{n-1}(P_s+\Delta P_s)V_s(\rho^{\frac{n-1}{n}}-1)+\Delta P_s\cdot V_s\right\}\frac{N}{60}A - L_{ad} \qquad (12)$$

Heat Loss

Heat loss is produced by heat transfer and leakage in the process of both suction and compression in the cylinder. It was experimentally obtained by the following equation.

$$L_h = L_i - (L_{ad}+L_{dis}+L_{suc}+L_{Top}) \qquad (13)$$

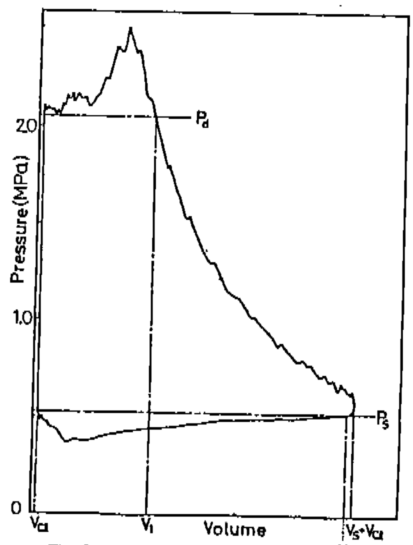


Fig. 2 P-V Diagram

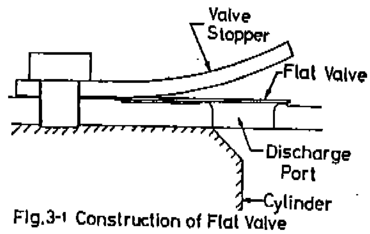


Fig.3-1 Construction of Flat Valve

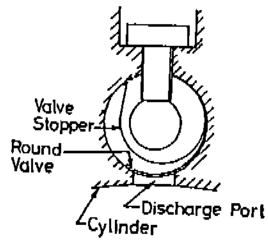


Fig.3-2 Construction of Round Valve

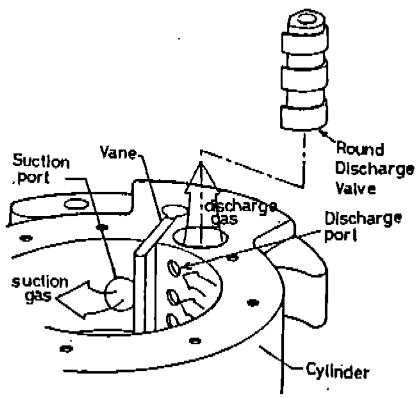


Fig. 4 Schematic View of Valve Setting

EXPERIMENTS

Comparison of a Round Valve with a Flat Valve

The discharge valve widely applied for a compressor is a flat valve. Fig. 3-1 shows a typical construction of the discharge valve device with a flat valve. There is usually one discharge port in this construction.

The discharge port diameter will have to be larger to be used in a compressor with large capacity, which has much gas flow rate.
But it will sometimes be impossible structurally to make a large discharge port corresponding to capacity. Even if the discharge port could be made large, clearance volume would increase as it. Compression efficiency and volumetric efficiency will decrease consequently. As valve thickness inevitably increases to ensure the reliability, overshooting loss due to valve delay increases.

Round Valve for Discharge

Authers have developed and introduced the discharge valve device of new type, that is a round valve, in a rolling piston type rotary compressor instead of a flat valve.

Fig. 3-2 shows a construction of the discharge valve device with a round valve. Fig. 4 shows how to be assembled in the cylinder.
The features of a round valve are as follows.

1. It is easy to make the discharge port area wide without large clearance volume.
2. Overshooting loss reduces to make gas flow smooth.
3. As a result of the above features, compression efficiency and volumetric efficiency are improved.

The features mentioned above are more remarkable under the conditions of much gas flow above all.

Results of Experiments

Fig. 5 shows the measured cylinder pressure comparing a round valve with a flat valve. A large difference between them is apparently found.

Each of efficiency was calculated using the equations (1)~(13) on the basis of the P-V diagram obtained by the measurements of cylinder pressure and crank angle.

Fig. 6 shows comparison of efficiency. Compression efficiency is improved as is evident from Fig. 6 . Fig. 7 shows the ratio of losses to consumption power of a flat valve in this measurements.

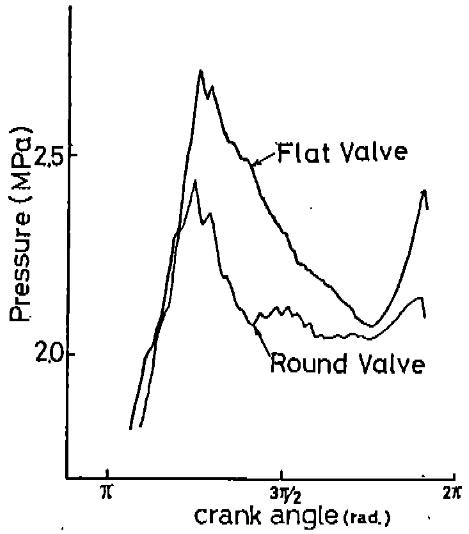


Fig.5 Measured Pressure

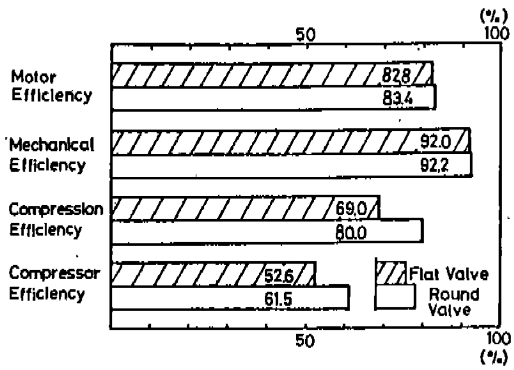


Fig.6 Comparison of Efficiency

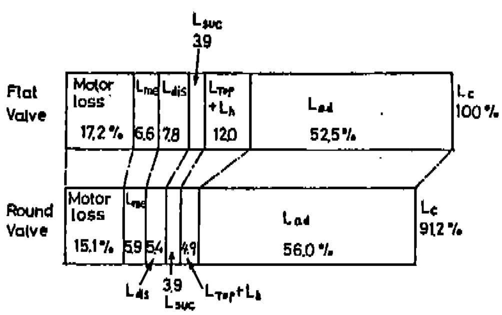


Fig.7 Ratio of Loss

FRICTION LOSS ANALYSIS

Total mechanical loss was experimentally obtained by the equation (6). Each of mechanical loss (friction loss) was theoretically analyzed, after the equilibrium equations of forces shown in Fig. 8 and Fig. 9 were solved.

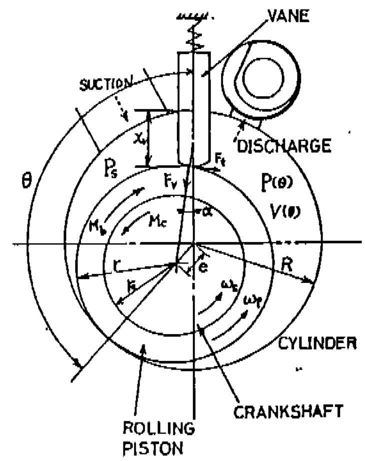


Fig.8 Schematic View and Moment acting on Rolling Piston

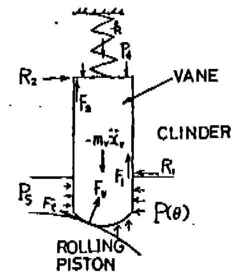


Fig.9 Forces acting on Vane

Loss at Vane Tip

Sliding velocity at vane tip and angular velocity of rolling piston were obtained by solving numerically the following equations.[5)(6)]

$$I_p\dot{\omega}_p = M_c - rF_t - M_b \tag{14}$$

$$\mathcal{V} = r\omega_p + e\omega_c\frac{\cos\theta}{\cos\alpha} \tag{15}$$

Friction loss at vane tip was obtained by the following equation.

$$L_v = \frac{A}{2\pi}\int_0^{2\pi} F_t \mathcal{V} d\theta \tag{16}$$

Loss at Vane Side

Friction loss at vane side was obtained by the following equation.

$$L_{vs} = \frac{A}{2\pi}\int_0^{2\pi}(F_1+F_2)\dot{x}_v d\theta \tag{17}$$

Loss at Bearing

Friction loss at bearing was obtained by solving the basic equation for journal bearing of finite length under fluctuating load. The basic construction of journal bearing is shown in Fig. 10.

$$\frac{1}{r_j^2}\frac{\partial}{\partial\theta_b}\left(h^3\frac{\partial P}{\partial\theta_b}\right) + h^3\frac{\partial^2 P}{\partial z^2} = 6\eta C_r\left[-\varepsilon\{\omega_j+\omega_b-2(\dot{\phi}+\dot{\psi})\sin\theta_b\} + 2\dot{\varepsilon}\cos\theta_b\right] \tag{18}$$

where, $C_r = r_b - r_j$, $h = C_r(1+\varepsilon\cos\theta_b)$

$\varepsilon = \frac{e_j}{C_r}$, $U = r_j\omega_j$

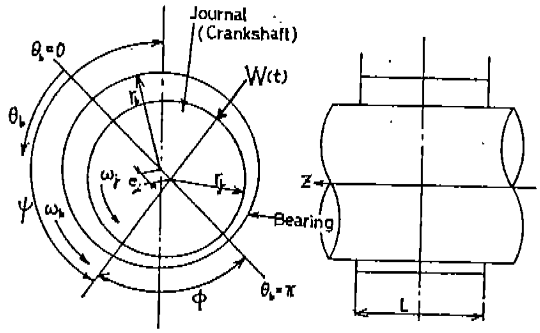


Fig.10 Model of Journal Bearing

$$F_j = \eta U L \frac{r_j}{C_r}\left\{\frac{\omega_j-\omega_b}{\omega_j}\frac{2\pi}{\sqrt{1-\varepsilon^2}} + \frac{\varepsilon}{2}\left(\frac{C_r}{r_j}\right)^2\frac{W}{\eta U L}\sin\phi + \left(\frac{C_r}{r_j}\right)\frac{2\pi}{\omega_j}\dot{\phi}\left(\frac{1}{\sqrt{1-\varepsilon^2}}-1\right)\right\} \tag{19}$$

$$L_b = \frac{A}{2\pi}\int_0^{2\pi} F_j U d\theta_b \tag{20}$$

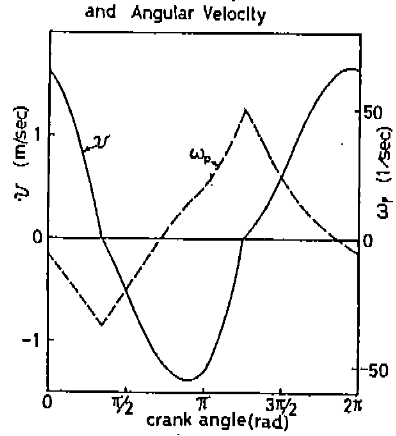
Loss at Thrust

Friction loss at thrust bearing was calculated considering weight of crankshaft and rotor, axial component of motor torque as load.

Results of Analysis

Sliding velocity at vane tip and angular velocity of rolling piston are shown as the results of computation in Fig. 11.
Fig. 12 shows friction loss at vane tip and vane side fluctuating to angular displacement, as the ratio to theoretical adiabatic work.
Theoretical total mechanical loss agreed approximately with experimental total mechanical loss obtained by the equation (6) as in Table 4.

Fig.11 Sliding Velocity and Angular Velocity



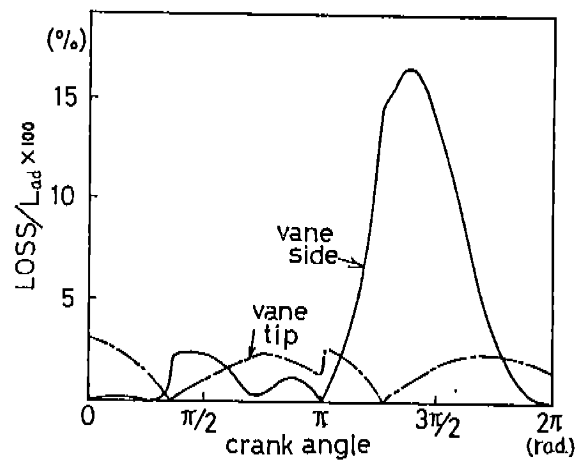


Fig.12 Friction Loss at Vane

Table 4 Mechanical Loss

loss ratio \ means	Experimental	Theoretical
$\frac{L_{me}}{L_{ad}} \times 100$ (%)	10.5	9.6

IMPROVEMENT OF EFFICIENCY

Compression efficiency was much improved with the round valve of new type as described before.
At next step, more improvement of efficiency was investigated. The decrease of overshooting and undershooting was expected theoretically at first, and lastly confirmed experimentally.

Theoretical Investigation

Theoretical investigations of overshooting and undershooting were made by the characteristic method[1] assuming unsteady one-dimensional isentropic flow.
Momentum equation is as follows.

$$\rho\frac{\partial u}{\partial t} + \rho u\frac{\partial u}{\partial x} + \frac{\partial P}{\partial x} = 0 \tag{21}$$

Equation combined the continuity equation and the energy equation is as follows.

$$\rho a^2\frac{\partial u}{\partial x} + \frac{\partial P}{\partial t} + u\frac{\partial P}{\partial x} = 0 \tag{22}$$

where , ρ = density of gas
u = velocity of gas
a = sound velocity of gas
t = time
x = axis along discharge port or suction inlet

The characteristic and compatibility equations corresponding to the coupled partial differential equations mentioned above are as follows.

$$\left(\frac{dx}{dt}\right)_{\pm} = u \pm a \qquad (23)$$

$$dP_{\pm} \pm \rho a du_{\pm} = 0 \qquad (24)$$

Overshooting loss and undershooting loss were computed by use of the equations mentioned above following the means of improvement as described bellow.

1. Discharge port diameter was changed to improve overshooting.
2. Suction inlet diameter was changed to improve undershooting.

Results of Investigations for Improvement

The results of measured and computed overshooting are shown in Fig. 13. It is seen from Fig. 13. that the computed overshooting approximately agrees with the measured.
Fig. 14 shows the measured and computed overshooting loss corresponding to the changed diameter of discharge port as the ratio to theoretical adiabatic work.
Fig. 15 shows the measured and computed undershooting loss corresponding to the changed diameter of suction inlet as the ratio to theoretical adiabatic work.

The computed loss shows a similar tendency to the measured. So theoretical investigations can be considered to be sufficient to estimate the improvement of efficiency in advance.

As a result that the discharge port diameter and the suction inlet diameter were increased, consumption power decreased remarkably and each efficiency was improved. Experimental results with the improved specifications, which are the discharge port diameter of 9 mm (0.354 in.) and the suction inlet diameter of 19.8 mm (0.78 in.), are shown in Fig. 16 and Fig. 17.
Fig. 16 shows the ratio of improved loss to consumption power of a flat valve.
Fig. 17 shows each of improved efficiency.

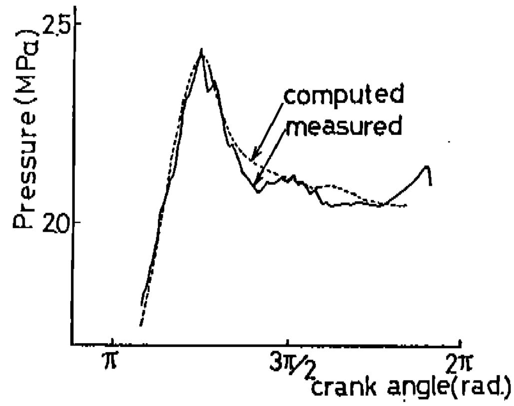


Fig.13 Computed Pressure

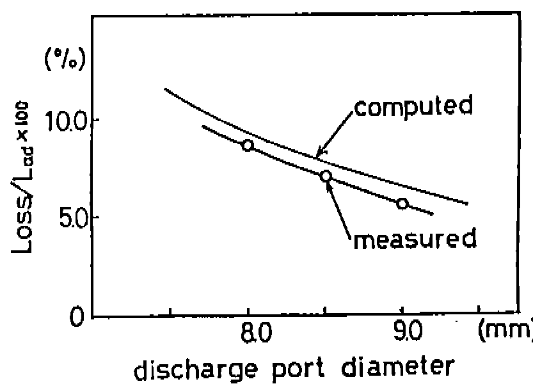


Fig.14 Overshooting Loss

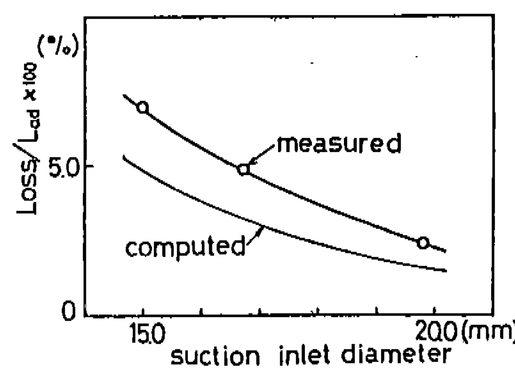


Fig.15 Undershooting Loss

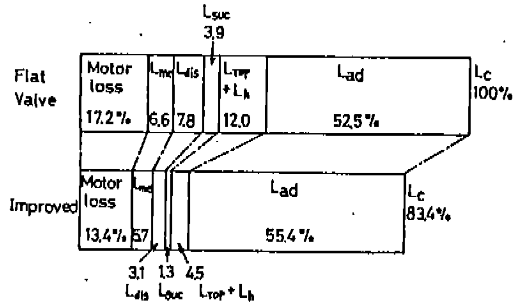


Fig.16 Ratio of Improved Loss

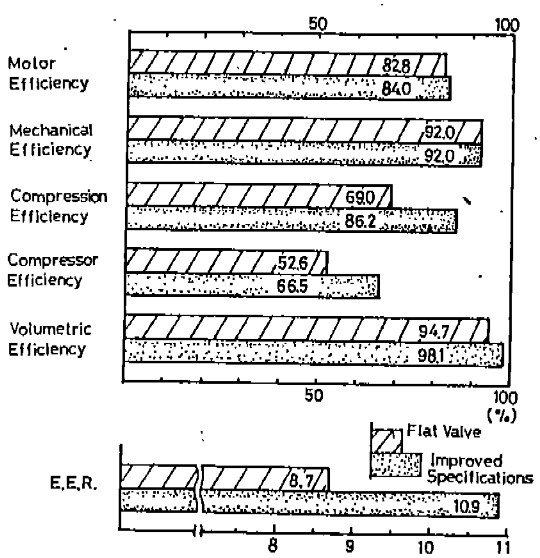


Fig.17 Comparison of Efficiency

CONCLUSION

Some approaches towards a high efficient rotary compressor of rolling piston type for a room air conditioner and an unitary air conditioner were studied in this paper.

Those are summarized as follows.

1. Efficiency of a rotary compressor with large capacity (42,000 BTU class) was analyzed, and improvement of it was studied experimentally and theoretically.

2. A round valve of new type for discharge which were developed and introduced in a rotary compressor instead of a flat valve improved each efficiency, compression efficiency above all.

3. Mechanical loss was theoretically analyzed, and it relatively agreed with the measured.

4. Theoretical analyses of overshooting loss and undershooting loss were carried out by the characteristic method.
   Those were sufficient to estimate improvement of efficiency in advance.

5. As a result of decreasing overshooting and undershooting, compression efficiency was improved to 86% in this investigation,and E.E.R reached 10.9 .

Improvement of motor was not investigated in this paper.
At the end, we would spare no efforts to study a higher efficient rotary compressor including improvement of motor in succession for the future.

NOMENCLATURE

η_m = motor efficiency

η_{me} = mechanical efficiency

η_c = compression efficiency

η_{com} = compressor efficiency

η_v = volumetric efficiency

L_c = consumption power

L_m = motor output

L_i = indicated work

L_{ad} = theoretical adiabatic work

L_{me} = mechanical loss

L_{dis} = overshooting loss

L_{suc} = undershooting loss

L_{Top} = reexpansion loss

L_h = heat loss

G_a = actual gas flow

G_{th} = theoretical gas flow

P = cylinder pressure

V = cylinder volume

Δi = increment of enthalpy

n = adiabatic exponent

P_s = suction pressure

P_d = discharge pressure

N = compressor speed (rpm)

A = 9.807×10^{-2} (W/(kg·cm/sec))

ΔP_s = increment of suction pressure

V_s = stroke volume

V_{cl} = clearance volume

R = cylinder radius

r = rolling piston radius

θ = crank angle

x_v = vane extension
$x_v = (R+r_v)-\{(r+r_v)\cos\alpha+e\cos\theta\}$

• = time differential

r_v = vane tip radius

α = offset angle of rolling piston center
$\alpha = \sin^{-1}(e/(r+r_v) \times \sin\theta)$

e = eccentricity of crank
e=R−r

R_1, R_2 = reaction force at vane side

F_1, F_2 = friction force at vane side
$F_1 = \mu_1 R_1$, $F_2 = \mu_2 R_2$

μ = coefficient of friction

k = coefficient of vane spring

m_v = mass of vane

I_p = moment of inertia of rolling piston

ω_p = angular velocity of rolling piston

M_b = friction moment at rolling face

M_c = friction moment at rolling piston bearing

F_v = reaction force at vane tip

F_t = friction force at vane tip
$F_t = \mu_v F_v$

ω_c = angular velocity of crankshaft

V = sliding velocity at vane tip

L_v = friction loss at vane tip

L_{vs} = friction loss at vane side

θ_b = coordinate of bearing angle

ϕ = attitude angle of bearing

ψ = directional angle of bearing load

ω_j = angular velocity of journal

ω_b = angular velocity of bearing

z = coordinate of bearing length

η = viscosity of lubricating oil

L_b = friction loss at bearing

p = oil pressure in bearing

R_b = bearing radius

r_j = journal radius

L = bearing length

e_j = eccentricity of journal

W(t) = bearing load

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