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Analysis of The Rolling-Piston Type Rotary Compressor

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ANALYSIS OF THE POLLING-PISTON TYPE
ROTARY COMPRESSOR

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ABSTRACT

We have being tried to evaluate exactly performance of hermetic rolling-piston type rotary compressor using for domestic room air conditioner. The performance is evaluated concerning with volumetric efficiency and consumption power. Our evaluating approach was founded expelimental measurement and logical calculation. The volumetric efficiency in this paper consists of leakage, suction gas heating and top clearance. The leakage is most characteristic factor which effects to volumetric efficiency and consumption power is this rotary type compressor. The consumption power in this paper contains of gas compression work, mechanical loss, and driving motor loss. The rolling-piston type rotary compressor in this evaluation is shown in Fig 1, modified for simplification.

DESCRIPTION OF EVALUATION

(1) Volumetric Efficiency

Concerning factors are leakage of inside, suction gas heating and top clearance. The analysis for the leakage of inside are very important in this rotary type compressor. These leakages through the cylinder inside clearances are separated into two types according to leakage substances which are refrigerant gas leakage and lubricant oil leakage, and they make the differential effects.

Leakage : The leakage channels are determined in Fig 2. We assume that gas leakage is adiabatic change and maximum flow velocity which occures in minimum sectional area in each leakage channel. Also, we assume that oil leakage is noncompressive viscous fluid.

Radial Leakage : W_R , is caused through at clearance area, $Ar(\theta)$, which is between cylindrical walls on cylinder inside and rolling-piston outside. This is assumed on gas leakage. This leakage is caused by differential pressure between compressing chamber pressure, $Pd(\theta)$, and suction pressure, Ps. They are formularized follows.

$$W_R = \psi_R \int_{\theta s}^{\theta c} Ar(\theta) \sqrt{2g \frac{n}{n-1} \cdot \frac{Pd(\theta)}{v(\theta)} \left\{ \left(\frac{Ps}{Pd(\theta)}\right)^{\frac{2}{n}} - \left(\frac{Ps}{Pd(\theta)}\right)^{\frac{n+1}{n}} \right.} \, d\theta$$

$$+ \psi_R \int_{\theta c}^{\theta d} Ar(\theta) \sqrt{g \cdot n \cdot \frac{Pd(\theta)}{v(\theta)} \left(\frac{2}{n+1} \right)^{\frac{n+1}{n-1}}} \, d\theta \quad -(1)$$

Functions are shown in Fig 2 or latter paragraph.

Vane side leakage : W_v, is caused by clearance area, $Av(\theta)$, between vane both ends and the walls to cover cylinder ends. This is assumed on gas leakage too.

Formulare is following :

$$W_v = \psi_v \int_{\theta s}^{\theta c} Av(\theta) \sqrt{2g \frac{n}{n-1} \frac{Pd(\theta)}{v(\theta)} \left\{ \left(\frac{Ps}{Pd(\theta)}\right)^{\frac{2}{n}} - \left(\frac{Ps}{Pd(\theta)}\right)^{\frac{n+1}{n}} \right.} \, d\theta$$

$$+ \psi_v \int_{\theta c}^{\theta d} Av(\theta) \sqrt{g \cdot n \cdot \frac{Pd(\theta)}{v(\theta)} \left(\frac{2}{n+1} \right)^{\frac{n+1}{n-1}}} \, d\theta \quad -(2)$$

Functions are shown in Fig 2 or latter paragraph. We got the coefficient of flow that was ψ_R / ψ_v =0.3~0.5 with exprimental measurement used actual compressor. This result explains that radial leakage is non-adiabatic change caused by the geometric shape of compressor chamber to have large heat transfer.

Vane side leakages : W_{vvs}, W_{vvd}, W_{vss}, and W_{vsd} are Caused by differential pressure between vane fack pressure, Pd, and cylinder chamber pressure, Ps or $Pd(\theta)$. These leakages are assumed on lubricant oil leakage. We do not consider W_{vvd} and W_{vvs} in this paper, since they are negligible. W_{vss} and W_{vsd} occur in the earliest stage of compression. These are reduced by the slant of vane in vane slot with gas pressure.

$$W_{vss} = \int_0^{2\pi} H\left(\frac{(P_d - P_s)(\delta_{vs}(\theta))^3}{12\,\mu\,\ell(\theta)} + \frac{U(\theta)\,\delta_{vs}(\theta)}{2}\right) d\theta \quad —(3)$$

$$W_{vsd} = \int_0^{\theta d} H\left(\frac{(P_d - P_d(\theta))(\delta_{vsMAX} - \delta_{vs}(\theta))^3}{12\,\mu\,\ell(\theta)} + \frac{U(\theta)(\delta_{vsMAX} - \delta_{vs}(\theta))}{2}\right) d\theta \quad (4)$$

These are not negligible in large size of δ_{vs}MAX. Be care, if the level of lubricant oil is not to reach the back of vane, gas leakage occurs and it much reduce volumetric efficiency.

Piston side leakage : W_{ss} and W_{sd} , are caused by the clearances between the rolling-piston both ends and the walls to cover cylinder ends. These are assumed the lubricant oil leakages. If the oil do not supply to the piston inside enough to seal the clearances, these change into gas leakage which much reduses volumetric efficiency. This is the most important for the rotary type compressor.

Formuleres are following :

$$W_{ss} = \frac{\alpha_{ps}\,\gamma\,(\overline{\delta_{ps}})^3 (P_d - P_s) \times R_{PI}}{12\,\mu\,(R_{po} - R_{PI})} \int_0^{2\pi} d\theta \quad —(5)$$

$$W_{sd} = \frac{\alpha_{ps}\,\gamma\,(\overline{\delta_{ps}})^3\; R_{PI}}{12\,\mu\,(R_{po} - R_{PI})} \int_0^{\theta d} (P_d - P_d(\theta))\, d\theta \quad —(6)$$

Where

$$\overline{\delta_{ps}} = \frac{\delta_{ps1} + \delta_{ps2}}{2},$$

$\alpha_{ps} = 2 : \delta_{ps1} = \delta_{ps2}$, $\alpha_{ps} = 2.67 : 2\delta_{ps1} = \delta_{ps2}$ or $\delta_{ps1} = 2\delta_{ps2}$ etc.

Gas Heating : Volumetric efficiency is reduced by increasing specific volume that brings about heat transfer in suction accumlator, cylinder inside walls, and by hot oil leakages which are W_{ss} , W_{vus}, and W_{vss}. These hot oil leakages influence a about half to total heating effect. These are not only characteristic, but also importance in this type of rotary, as described in paragraph of the leakage. Evaluated value of heat transfer in this paper used the formulare that was assumed by rectangular tube with steady flow, since it is very difficult to analyze exact modeling of heat transfer with walls in compressor. The above intention is to oftain effect of various dimensions of compression chamber and gas flow. The heated gas temperature, T_2 , and increased specific volume, ν_2 , are following:

$$T_2 = \frac{G_o \cdot C_p \cdot T_s + W_L \cdot C_{oil} \cdot T_{oil} + \alpha_c A_c (T_c - T_{s2}) + \alpha_a A_a (T_m - T_s)}{G_o \cdot C_p + W_L \cdot C_{oil}} \quad —(7)$$

$$\nu_2 = \frac{273 + T_2}{273 + T_s}\, \nu_s$$

$$W_L = W_{vss} + W_{ss} + W_{vus}$$

Top clearance volume : This effect reducing suction volume is assumed by re-expansion all of top clearance volume, however, there is some offset in actual result and it may explain that re-expansion gas, having been closed a suction port, is flowing out of the all through it.

(2) Comsumption Power

Eraluating comsumption power in this paper consists of gas compression work, mechanical loss, and driving motor loss.

Gas compression work : We estimate the gas compression work which consists of polytropic compression work, over shooting loss in discharge stage, under shooting loss in suction stage, gas leakage loss, heating loss by oil leakage, and compression loss of top clearance volume. These estimations are based on measured pressure curve in actual compressor and calculated the volumetric efficiency in above.

Polytropic compression work is assumed in following, and including loss by suction gas heating.

$$L_{pol} = \nu_2 \cdot \eta_v \cdot \frac{n}{n-1} P_s \cdot G_o \left\{ \left(\frac{P_d}{P_s}\right)^{\frac{n-1}{n}} - 1 \right\} \quad —(8)$$

Over shooting loss in discharge stage is

$$L_{os} \propto \int_{\theta d}^{2\pi} (P_d(\theta) - P_d) \cdot A_p(\theta) \cdot E(\theta)\, d\theta \quad —(9)$$

where

$P_d(\theta)$ given by measurement.

The leakage oil in discharge gas increases the over shooting loss as increasing specific weight of gas through a discharge port and valve.

Under shooting loss : It is fortunate that having no suction valve under shooting in suction stage is reduced. This charactor bring rolling-piston tyoe rotary compressor into high performance in condition of higher suction pressuer.

$$L_{us} \propto \int_0^{2\pi} (P_s - P_s(\theta)) \cdot A_p(\theta) \cdot E(\theta)\, d\theta \quad —(10)$$

$P_s(\theta)$ given by measurement.

Gas leakage loss is evaluated with integration of gas compression work of the leakage from θ_s to θ_d about W_R and W_V .

$$L_{LEAK} \propto \int_{\theta s}^{\theta d} W_V(\theta) \cdot i_{pol}(\theta)\, d\theta + \int_{\theta s}^{\theta d} W_R(\theta) \cdot i_{pol}(\theta)\, d\theta \quad —(11)$$

Heating loss of oil leakages is caused by W_{sd} , W_{vsd}, and W_{vud} which are evaluated in above. We assume it in following:

$$L_{HL} = G \cdot C_{oil} (W_{vud} + W_{sd} + W_{vsd})(T_{oil} - T_{s2}) \quad —(12)$$

Top clearane loss : We assume that all of compressing work in top clearance volume is non-recovery work. This charactor increases comsumption power at high compression ratio. And this performance has to be compared with that of reciprecating compressor and this performance is very characteristic for rotary compressor. This is recovery work for reciprocating compressor.

$$L_{Top} = \varepsilon \left(\frac{P_d}{P_s}\right)^{\frac{1}{n}} \cdot \frac{v_s}{v_0} G_i \cdot P_s \frac{n}{n-1} \left\{ \left(\frac{P_d}{P_s}\right)^{\frac{n-1}{n}} - 1 \right\} \quad -(13)$$

Mechanical Loss

We estimated the mechanical loss at journal fearings of cylinder both ends and rolling-piston inside, friction loss between the outer diameter of rolling-piston and the vane, and between the vane side and vane slot of the cylinder.

Journal bearings : These are evaluated as finite length journal bearings using mean force of compression pressure.

Rolling-piston-vane : This is characteristic loss in rolling-piston type compressor. Evaluating this loss, the motion of rolling-piston must be decided. We used results of the digital simulation by Okada (*1) and the exprimental measurement in actual compressor. We evaluated this loss using assumed coefficient of friction, so that we could not measure the coefficient. This motion is important not only increasing loss, but also increasing wear on the rolling-piston and the vane.

Vane slot : We evaluate the vane slot loss caused by friction loss. We assumed the forces and the coefficients of friction, since these could not measure. The vane slot loss is calculated by the digital simulation as described in the rolling-piston-vane loss, since this friction influences motion of rolling-piston.

Motor Loss

The motor loss consists with resistance of a stator winding conductor and a rotor conductor, and eddy courrent in magnet core. They calculated by motor design program and checked by measuring. The motor loss influence large part of the total compressor performance.

CONCLUSION

We could explain detal performance of the rolling-piston type compressor. It is important to reduce the leakages and the rotation of a rolling-piston,and these brings the compressor to high efficiency and reliability as explained. We estimated the performance in vary conditions and got the results as same for practical use as measured in that. This is shown in Table-3.

These results are very useful to design a new compressor. We did not use a large integrated simulation program by reason of a complication for handling. The separated programs for each factor are often useful in practical use. In conclusion we want that meny analys s for this rotary type compressor are presented by the many investigator.

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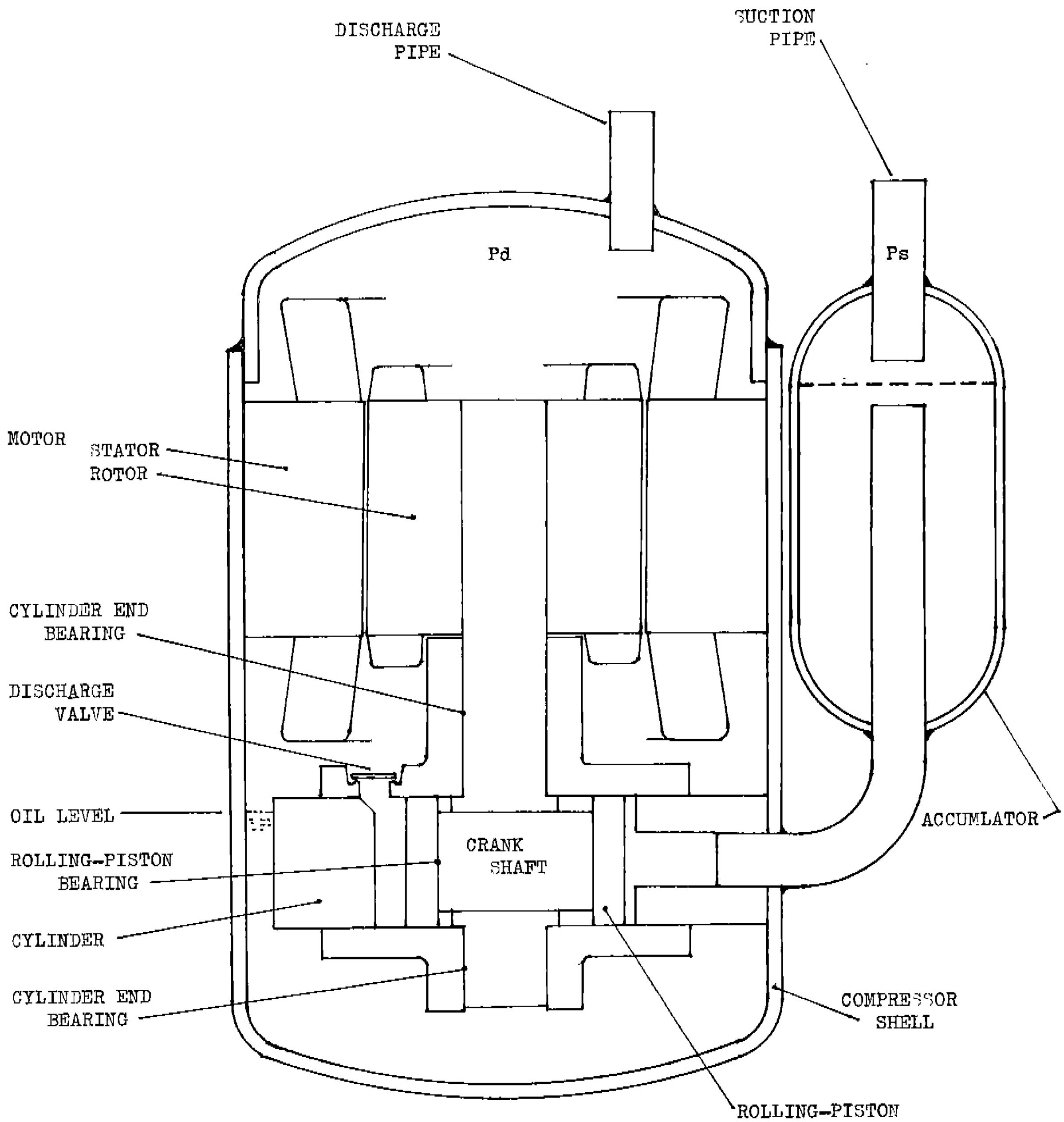
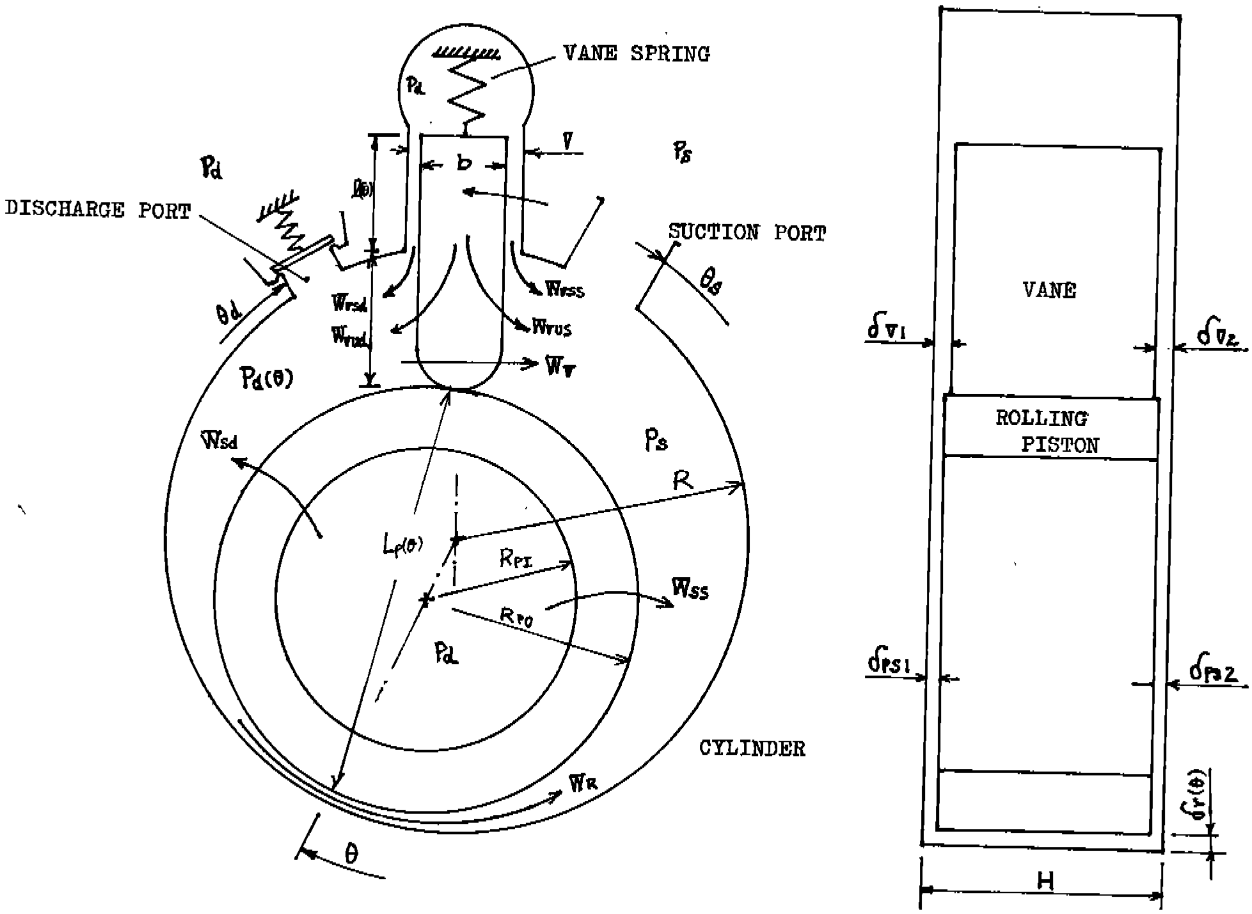


ROLLING-PISTON TYPE ROTARY COMPRESSOR

Fig -1



THE LEAKAGE CHANNELS

Fig - 2

NOTATION

Aa: Coefficient of heat transfer at accumlator.

Ac: coefficient of heat transfer at cylinder

$Ar(\theta)$: $\delta r(\theta) \cdot H$

$Av(\theta)$: $lv(\theta)(\delta_{v1} + \delta_{v2})$.

$Ap(\theta)$: $Lp(\theta) \cdot H$.

C oil: specific heat of oil.

Cp : specific heat at constant pressure.

$E(\theta)$: $R(1-K)\sin\{\frac{\pi}{2} - \frac{1}{2}\sin^{-1}(\frac{l_v(\theta)}{K R}\sin\theta)\}$

G_o : $V_o/V_s = (V_{st} \cdot N \cdot 60)/v_s$

G : actual gas flow

$i_{pol}(\theta)$: $\frac{n}{n-1} v_2 \cdot P_s \cdot \{(\frac{P_d}{P_s})^{\frac{n-1}{n}} - 1\}$.

K : R_{po}/R

$Lp(\theta)$: $2KR\sin\{\frac{\pi}{2} - \frac{1}{2}\sin^{-1}(\frac{l_v(\theta)}{K \cdot R}\sin\theta)\}$

$l_v(\theta)$: $R\{(1-K)\cos\theta + \{(1-K)^2\cos^2\theta + 2K - 1\}^{1/2}\}$

N : r.p.m.

n : polytropic exponent.

$Pd(\theta)$: compressing pressure. $P_d(\theta) = \left(\frac{\pi R^2(1-K^2)}{S_d(\theta)}\right)^n$

$$Sd(\theta) = \frac{R^2}{2}[(1-K^2)\theta - \frac{1}{2}(1-K)\sin 2\theta - (1-K)\sin\theta \cdot (K^2 - (1-K)^2\sin^2\theta)^{1/2} - K^2\sin^{-1}\frac{(1-K)\sin\theta}{K}]$$

Tc : temperature of cylinder walls.

Tm : temperature of accumlator.

Toil: temperature of oil.

Ts : compressor inlet gas temperature.

Ts_2 : sution port gas temperature.

$U(\theta)$: vane speed.

$v(\theta)$: $(\frac{P_d(\theta)}{P_s})^{-\frac{1}{n}} \cdot v_2$

Vst : stroke volume $= \pi R^2(1-K^2) \cdot H$

θ_c : angle at $P_d(\theta) = P_s/(\frac{2}{n+1})^{\frac{n}{n-1}}$

δ_{v2max}: $V - b$

$\delta_{\overline{PS}}$: $\frac{\delta_{PS1} + \delta_{PS2}}{2}$

ε : V_{Top}/V_{st}.

Testing condition for table -1 & 2

Ps :	4.92 Kg/cm^2G	(72. 4 psi)
(As evapolating temp.	5°C	(41°F)
Ts :	15 C	(59°F)
Pd :	19.7 Kg/cm^2G	(289.7 psi)
(As condensing temp.	52°C	(125.6°F)

100 Volt 60 Hz 3400 r.p.m

compressor out put 3/5 Hp

refrigerant R - 22

Volumetric efficiency as evaluated

$$\eta_v = \frac{G}{G_o} = \left[1-\varepsilon\left\{\left(\frac{Pd}{Ps}\right)^{\frac{1}{n}}-1\right\}\right]\frac{Vs}{V_2} - \frac{W}{G_o} - \frac{Wv}{G_o}$$

Total heating loss $(1-\frac{Vs}{V_2})$	5.3 %
Top clearance loss $((\frac{Pd}{Ps})^{\frac{1}{n}}-1)\frac{Vs}{V_2}$	3.3 %
Radial leak. loss $\frac{W_R}{G_o}$	3.6 %
Vane side leak. loss $\frac{Wv}{G_o}$	2.1 %
Total loss	14.3 %
Volumetric efficiency	85.7 %

Table - 1

Comsumptinn power

(Each evaluated value ratios of compressor input)

Polytropic work Lpol	56.9 %
Over shooting loss Los	4.5 %
Under shooting loss Lus	2.0 %
Radial leak.loss $W_R \cdot l$pol	1.8 %
Vane side leak. loss $W_v \cdot l$pol	0.9 %
Heating loss Lhl	4.3 %
Top clearance loss Ltop	3.1 %
Journal bearings loss	2.4 %
Piston-vane loss	1.1 %
Vane slot loss	1.0 %
Motor conductors loss	16.6 %
eddy courrent loss	5.4 %
TOTAL	100. %

TAble - 2

Condensing temp.	40°C (104°F)	40°C (104°F)	52°C (125.6°F)	60°C (140°F)	60°C (140°F)
Evaporating temp.	0°C (32°F)	10°C (50°F)	5°C (41°F)	0°C (32°F)	10°C (50°F)
Measured value	87.6 %	92.5 %	87.3 %	80.9 %	85.8 %
Evaluated value	86.7 %	90.4 %	85.7 %	80.3 %	85.7 %

The Comparison of Volumetric Efficiency between Calculated Values and Measured in Various Condition.

Table - 3