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EVALUATION OF THE LEAKAGE THROUGH THE CLEARANCE BETWEEN PISTON AND
CYLINDER IN HERMETIC COMPRESSORS

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ABSTRACT

Leakage through the clearance between piston and cylinder is analyzed. A simple mathematical model is used to calculate the total refrigerant leakage from the cylinder in reciprocating hermetic compressor. The calculation of the total mass considers both the flow of pure refrigerant and the flow of refrigerant dissolved in the oil.

An experimental device used to measure the refrigerant leakage is described. The measured results with this device were in much agreement with the theoretical predictions of leakage for different clearance values between piston and cylinder.

INTRODUCTION

Leakage through the clearance between piston and cylinder in reciprocating hermetic compressors diminishes performance by the reduction of capacity and by the power consumption of lost gas. The simple reduction of the clearance between piston and cylinder yields higher production costs yet it does not necessarily lead to a higher energy efficiency ratio as the viscous dissipation increases. This is a relevant topic to be analyzed when looking for ways to improve compressor efficiency.

Leakage of refrigerant fluid from inside of the cylinder can happen in two different ways: in one the effect of the increasing pressure inside the cylinder breaks the oil film, establishing a two-phase oil/refrigerant flow. This is called a direct leakage. In the other way, the refrigerant gets out of the cylinder dissolved in the lubricating oil when the piston descends. This is called an indirect leakage. The refrigerant separates from the oil inside the shell where the pressure is lower than in the clearance.

The phenomena of the mass flow through the clearance has been brought to the attention of many researchers especially in rotating compressors [1] - [4], where a higher number of leakage paths exists, when comparing it to reciprocating compressors.

The calculation of the mass loss in rotating compressors has been normally done [1] - [3] considering an isentropic flow of an ideal refrigerant fluid through a convergent - divergent nozzle, as the clearance length is small. The evaluation of the mass loss for reciprocating compressors has been done [5] only for pistons with rings, assuming an isothermal expansion, as the clearance length is also small.

The leakage through the clearance is governed by two different mechanisms: i) the difference between the pressure inside the cylinder and in the shell per unit of clearance length, and ii) the rate at which momentum is transmitted from the piston to the fluid inside the clearance between piston and cylinder. During the ascending movement of the piston, the fluid inside the clearance is lubricating oil. If oil viscosity is much higher than refrigerant viscosity the phenomena of piston momentum transmission overcomes the effect of pressure build-up inside the cylinder. Therefore, the oil seals the clearance reducing the leakage. During the descending movement of the piston, the existing oil inside the clearance contains dissolved refrigerant. This is brought to the shell by the effects of pressure difference and momentum transmission. During this stroke, there is a much higher probability for direct leakage to happen.

On the other hand, if lubricating oil viscosity is not much higher than refrigerant viscosity, or the clearance thickness is too great, pressure forces on the fluid

predominate over viscous forces and a direct leakage will occur during the piston ascending movement.

Therefore, it is important that the conditions in which the leakage occurs be determined a priori. The relation of pressure forces and viscous forces in Eq.(1) gives an idea of the leakage condition as presented in Table 1.

$$r_f = \frac{(p_o - p_L) \delta^2}{\mu L v_p} \quad (1)$$

where

- p_o - pressure inside the cylinder
- p_L - pressure inside the shell
- δ - radial clearance between piston and cylinder
- μ - viscosity of the fluid in the clearance
- L - clearance length
- v_p - characteristic velocity of the piston

Table 1 - Flow Condition in the Clearance

r_f	Movement of the Piston	
	Ascending	Descending
$\gg 1$	Oil/Refrigerant	Oil/Refrigerant
~ 1	Oil	Oil/Refrigerant
$\ll 1$	Oil	Oil

As the flow condition oil/refrigerant is more general than considering only the presence of oil in the clearance, the main objective of this paper will concentrate on the analysis of the flow of two fluids comparing theoretical and experimental results.

THEORETICAL ANALYSIS

The theoretical analysis of the problem was done on the flow model presented in Fig.1, considering the following assumptions:

- i) the phenomena is a quasi-steady flow;
- ii) the boundary conditions variations are instantaneously transmitted throughout all fluids;
- iii) the oil film has constant thickness;
- iv) the flow of both Newtonian fluids is incompressible;
- v) the end effects on the flow are disregarded;
- vi) inertia forces are negligible compared to viscous and pressure forces.

This flow condition depicted in Fig. 1 happens when during the compression stroke

the piston drags in a film of oil which completely fills all the clearance and, as the pressure builds up inside the cylinder, the refrigerant pulls back part of the oil film causing a gap and setting up the leakage. Part of the oil, due to its high viscosity, sticks to the piston reducing the area of direct leakage.

The balance of pressure and viscous forces for both fluids yields

$$-\frac{d}{dr}(r \tau_{rz}) = (p_o - p_L) \frac{r}{L} \quad (2)$$

Integration of Eq. (2) for both fluids subjected to the following specific boundary conditions:

$$r = k_1 R \quad \tau_{rz}^r = \tau_{rz}^o \quad (3a)$$

$$r = k_1 R \quad v_z^r = v_z^o \quad (3b)$$

$$r = k R \quad v_z^o = v_p \quad (3c)$$

$$r = R \quad v_z^r = 0 \quad (3d)$$

gives the velocity profile of the oil (v_z^o) and of the refrigerant (v_z^r):

$$v_z^o = v_p \left[1 + \frac{\ln(r/kR)}{\frac{\mu_o}{\mu_r} \ln k_1 + \ln\left(\frac{k}{k_1}\right)} \right] + \frac{p_o - p_L}{4 \mu_o L} R^2 \times \left[k^2 - \left(\frac{r}{R}\right)^2 - \ln\left(\frac{r}{kR}\right) \frac{\frac{\mu_o}{\mu_r} (1 - k_1^2) + (k_1^2 - k^2)}{\frac{\mu_o}{\mu_r} \ln k_1 + \ln\left(\frac{k}{k_1}\right)} \right] \quad (4)$$

$$v_z^r = v_p \left[\frac{\ln(r/R)}{\ln k_1 + \frac{\mu_r}{\mu_o} \ln\left(\frac{k}{k_1}\right)} \right] + \frac{p_o - p_L}{4 \mu_r L} R^2 \times \left[1 - \left(\frac{r}{R}\right)^2 - \ln\left(\frac{r}{R}\right) \frac{(1 - k_1^2) + \frac{\mu_r}{\mu_o} (k_1^2 - k^2)}{\ln k_1 + \frac{\mu_r}{\mu_o} \ln\left(\frac{k}{k_1}\right)} \right] \quad (5)$$

When only one fluid fills completely all the clearance, the velocity distribution can be determined from Eq. (4) or Eq. (5) with $k_1 = 1$

$$v_z^o = v_p \left[\frac{\ln(r/R)}{\ln k} \right] + \frac{p_o - p_L}{4 \mu_o L} R^2 \left[1 - \left(\frac{r}{R}\right)^2 - \frac{1 - k^2}{\ln k} \ln\left(\frac{r}{R}\right) \right] \quad (6)$$

The viscous dissipation power can be calculated by differentiating Eq. (4) with respect to r , and evaluating it at $r = kR$ and then multiplying the result by the velocity of the piston and the contact area.

$$\dot{P}_c = \frac{2 \pi L V_p^2}{\frac{\ln k_1}{\mu_r} + \frac{\ln \left(\frac{k_1}{k}\right)}{\mu_o}} - \frac{(\rho_o - \rho_L) \pi R^2 V_p}{2} \times \left[2k^2 + \frac{\frac{\mu_o}{\mu_r} (1 - k_1^2) + (k_1^2 - k^2)}{\frac{\mu_o}{\mu_r} \ln k_1 + \ln \left(\frac{k}{k_1}\right)} \right] \quad (7)$$

The refrigerant mass flow rate by direct leakage can be determined by integrating Eq. (5) in its domain ($k_1 \leq r \leq 1$), giving:

$$\dot{m}_o = \rho_r \frac{\pi R^2}{2} \left[(1 + 2k_1^2 \ln k_1 - k_1^2) \cdot A + (1 - k_1^2) \cdot B \right] \quad (8)$$

where

$$A = \frac{\frac{\rho_o - \rho_L}{4L \mu_r} R^2 \cdot \left[\frac{(1 - k_1^2) + \frac{\mu_r}{\mu_o} (k_1^2 - k^2)}{\ln k_1 + \frac{\mu_r}{\mu_o} \ln \left(\frac{k}{k_1}\right)} \right] - V_p}{\ln k_1 + \frac{\mu_r}{\mu_o} \ln \left(\frac{k}{k_1}\right)} \quad (9a)$$

$$B = \frac{\rho_o - \rho_L}{4L \mu_r} R^2 \quad (9b)$$

ρ_r - specific density of refrigerant in the conditions of the cylinder. The indirect leakage of refrigerant can be determined by Eq. (10)

$$\dot{m}_R = \rho_o (W'_p - W'_s) \frac{\pi R^2}{2} \left\{ C \cdot \left[k^2 + 2k_1^2 \ln \left(\frac{k_1}{k}\right) - k_1^2 \right] + (k_1^2 - k^2) \cdot \left[2V_p - D (k_1^2 - k^2) \right] \right\} \quad (10)$$

where

$$C = \frac{V_p - \frac{\rho_o - \rho_L}{4L \mu_o} R^2 \left[\frac{\mu_o}{\mu_r} (1 - k_1^2) + (k_1^2 - k^2) \right]}{\frac{\mu_o}{\mu_r} \ln k_1 + \ln \left(\frac{k}{k_1}\right)} \quad (11a)$$

$$D = \frac{\rho_o - \rho_L}{4L \mu_o} R^2 \quad (11b)$$

ρ_o - Specific density of oil at the working temperature.

W'_p - Ratio of the mass of liquid refrigerant contained in the oil and the mass of oil in the conditions inside the cylinder.

W'_s - Ratio of the mass of liquid refrigerant contained in the oil and the mass of oil in the conditions of the shell.

W'_p and W'_s are calculated^[6] using Eqs. (12) and (13) for the mixture oil/R-12.

$$\log_{10} P = 9.9972 - 0.558 W^{-1/2} - \frac{(1177.67 - 98.753 W^{-1/2})}{T} - \left[0.002338 (W - 0.6)^2 - 0.000075 \right] (T - 273.16) \quad (12)$$

and

$$W' = \frac{W}{1 - W} = \frac{m \text{ REFRIGERANT}}{m \text{ OIL}} \quad (13)$$

p in [Pa] and T in [K]

As V_p , L , P_0 and ρ_r vary for each position of the piston, the equations which allow an evaluation of the instantaneous leakage and the viscous dissipation power are solved for each crankangle. After one machine cycle the average value of leakage and power are calculated.

EXPERIMENTAL AND NUMERICAL RESULTS

The calculation of the total leakage is only possible if the thickness of the oil film ($k_1 - k$) R is known and if the flow condition has been determined. In order to get more insight about the phenomena a leakage measuring device was built as shown in Figs. 2 and 3.

The device has the following components: two tanks of 20 liters each, one level indicator, valves, pipes and fittings. It is necessary to convert the compressor to direct suction, in order to use the shell as a recipient for the leakage.

From the shell, the leakage is conducted to tank 1 lowering the water level. The water from tank 1 goes to tank 2 and part of the refrigerant in tank 2 goes to the suction of the compressor. Therefore the pressure inside the tanks is kept constant during the measuring period.

During the warm-up period of the compressor, a sufficient amount of refrigerant is left inside the tanks, in direct contact with the water, in order to avoid any possible dissolution of the refrigerant during the tests. During this period valves 1 and 2 are closed and the by-pass valve is opened.

During the measuring period, valves 1 and 2 are opened and the by-pass valve is closed.

The working condition of the compressor used to get the experimental results were -30; 55; 32°C and the compressor had the following characteristics:

Displacement - 5.52 cm^3
 Mean diameter of the piston - $21.011 \times 10^{-3} \text{ m}$
 Stroke length - 3540 rpm
 Viscosity of the lubricating oil at 120°C - $38.8 \times 10^{-4} \text{ kg/m.s}$

The results gotten from the calorimeter are shown in Table 2 for different values of the clearance.

In the piston/cylinder sets used for the tests, the form error was kept below $1.0 \mu\text{m}$.

The calculation of the instantaneous leakage was done using Eqs. (8), (10) and (12). The total leakage was determined by the sum of the direct and indirect leakage.

Fig. 4 shows the comparison between numerical and experimental results for different radial clearances. As it can be seen the agreement of these results is quite good. The oil film thickness which provides the agreement between numerical and experimental results decreases as the radial clearance increases.

It is important to mention that form errors in the piston/cylinder set yield completely different results, as the oil film will break only in the regions of greater clearances, thus creating preferable paths.

CONCLUSIONS

The main points which should be emphasized in this work are:

- i) The relation of pressure and viscous forces is important to determine the flow condition in the clearance.
- ii) When there is no break in the oil film, the only leakage of refrigerant is that dissolved in the oil.
- iv) The device used to directly measure the total leakage has been described and it produces accurate and repeatable data.
- v) The experimental results well agree with the numerical calculation based on the mathematical model described here, especially for small clearance values.

Table 2 - Results from Leakage Tests

Characteristic Unit	Mean actual radial clearance ⁽¹⁾ [μm]					
	2.75	5.60	6.92	9.08	11.25	12.00
Capacity [kcal/h]	103	79	89	89	86	83
Power consumption [W]	123	102	110	109	113	115
E E R [BTU/Wh]	3.32	3.07	3.21	3.24	3.02	2.86
Total Leakage Mass [kg/h]	0.017	0.048	0.069	0.115	0.29	0.443
Pumped Mass [kg/h]	2.91	2.25	2.53	2.53	2.45	2.36

(1) Clearance was corrected for differential dilatation at 120°C

REFERENCES

[1] Reed, W.; Hamilton, J. F. - "Internal Leakage in Sliding Vane Rotary Compressors". Proc. 1980 Purdue Compressor Tech. Conf., 112 - 124.

[2] Pandeya, P.N.; Soedel W. - Rolling Piston Type Rotary Compressors with Special Attention to Friction and Leakage". Proc. 1978 Purdue Compressor Tech. Conf., 209 - 218.

[3] Chu, I.; Shiga, T.; Ishijima, K.; Sakaino, M. - "Analysis of the Rolling - Piston Type Rotary Compressor". Proc. 1978 Purdue Compressor Tech. Conf., 219 - 225.

[4] Hirano, T.; Sakitani, K. - "Finite Element Method Analysis of Leakage Flow in the Narrow Clearance Between the Rotor and Side Plates of a Sliding Vane Rotary Compressor". Proc. 1982 Purdue Compressor Tech. Conf., 305 - 312.

[5] Jacobs, J. J. - "Analytic and Experimental Techniques for Evaluating Compressor Performance Losses". Proc. 1976 Purdue Compressor Tech Conf., 116 - 123.

[6] Bambach, G. - "Das Verhalten von Mineralöl - F12 - Gemischen in Kältemaschinen". Abhandlungen des DKV Nr. 9, CF Muller Verlag, Karlsruhe, 1955.

W	- ratio of the mass of liquid refrigerant and the mass of oil in the conditions inside the cylinder
W_s	- ratio of the mass of liquid refrigerant and the mass of oil in the conditions inside the shell
μ	- absolute viscosity
μ_r	- absolute viscosity of the refrigerant
μ_o	- absolute viscosity of the oil
δ	- radial clearance between piston and cylinder
ρ_o	- specific density of oil at the working conditions
ρ_r	- specific density of refrigerant at the cylinder conditions
τ_{rz}	- shear stress of the fluid in the clearance

NOMENCLATURE

A,B,C,D	- function specified in the text
k	- ratio of diameters of piston and cylinder
k_l	- ratio of diameters of oil film and cylinder
L	- instantaneous length of the clearance
\dot{m}_D	- refrigerant mass flow rate by direct leakage
\dot{m}_R	- refrigerant mass flow rate by indirect leakage
P_O	- pressure inside the cylinder
P_L	- pressure inside the shell
\dot{P}_C	- viscous dissipation power
r	- radial independent variable
r_f	- ratio of pressure and viscous forces
R	- cylinder radius
V_P	- instantaneous velocity of the piston
V_z^o	- velocity profile of the oil in the clearance
V_z^r	- velocity profile of the refrigerant in the clearance
W	- ratio of the mass of liquid refrigerant and the sum of liquid

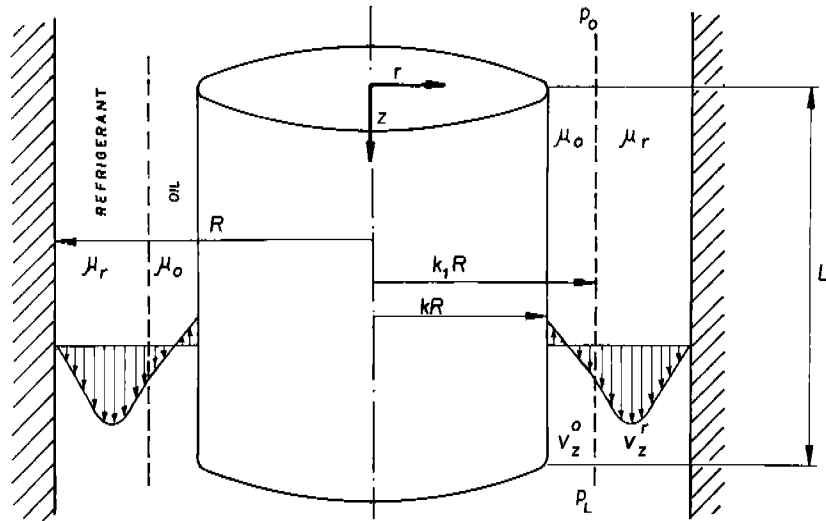


Fig. 1 - Geometry of the flow problem

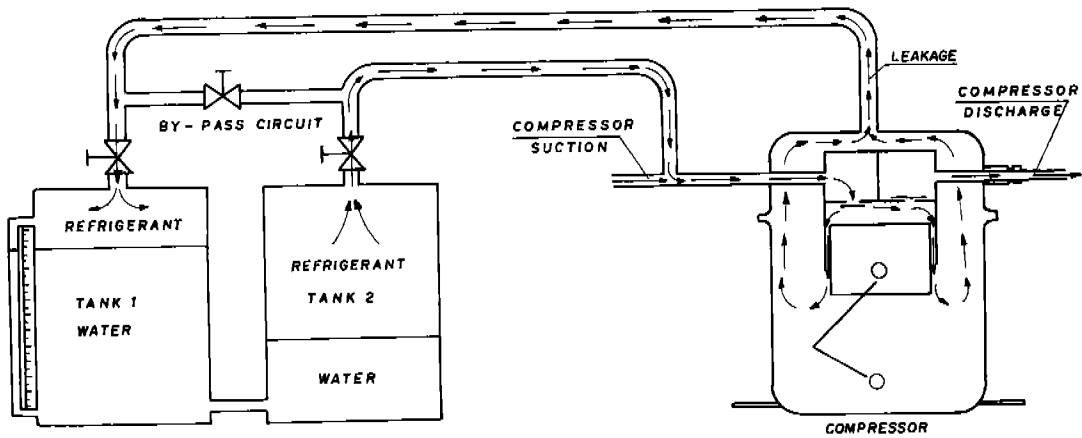


Fig. 2 - Leakage measurement device

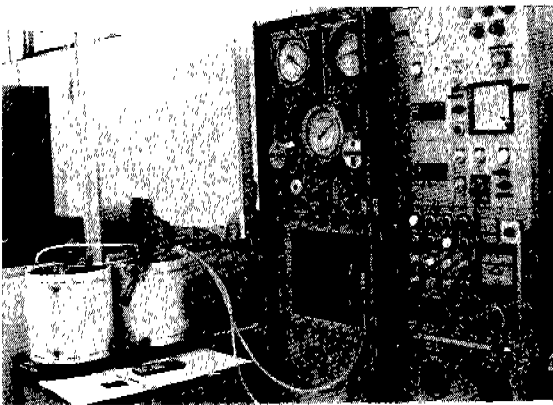


Fig. 3 - General view of experimental set up

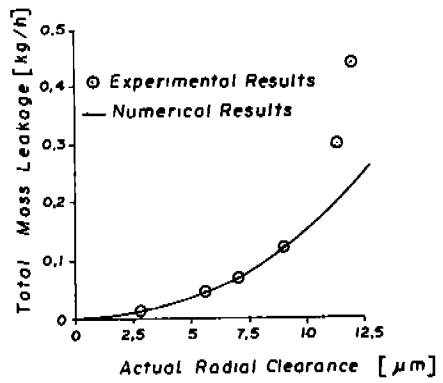


Fig. 4 - Comparison of numerical and experimental results.