1990

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ABSTRACT

The leakage through the minimal clearance between roller and cylinder wall is the most significant mass loss of a rolling piston rotary compressor, and mostly controls its volumetric efficiency.

The models presented in the preceding literature are discussed.

The present work studies the leakage using bench tests to establish its flow pattern in actual compressor operating conditions. A stroboscope and video technique is used to record the flow pattern of the leakage.

The influence of the oil in the leakage of gas is shown.

A new model to cope with the flow behaviour is introduced.

INTRODUCTION

The leakage through the minimal clearance between roller and cylinder wall in a rolling piston compressor for household refrigeration is cited in the literature as the most influent leakage over the volumetric efficiency. It is estimated that it counts for around 70% of the total internal gas leakage. Several authors have presented models to predict that leakage.

Chu et al (1) and Pandeya and Soedel (2) modelled the minimal clearance leakage assuming isentropic flow through a nozzle and used a restrictive nozzle coefficient of around one third to set experimental and numerical results.

Reed and Hamilton (3) proposed that the leakage mass flow would lie between an isentropic flow model of refrigerant as a maximum limit and an incompressible flow model of oil as a minimum. They were the first ones to realize the influence of the oil in the gas leakage.

Yanagisawa and Shimizu (4) proposed that the leakage flow would be counted as only gas flow with a friction coefficient that would be given by the variable lenght of a constant diameter duct.

Lately Lee and Min (5) assumed that the refrigerant gas leaks as a mixture of oil and gas, as Reed (3) have proposed. They calculated the leakage by using the difference of solubility between high and low sides, and the Reynolds flow equation multiplied by a coefficient (= 0.22) admitted as a function of the shaft rotating speed i.e. the time period available for the absorbed gas to separate from the oil.

Although the prior models would give satisfactory results in terms of an overall compressor simulation, the leakage phenomenon deserves further investigation in order to propose a model nearer to reality. This is the object of the present work.

DYNAMIC TEST BENCH

In order to investigate the leakage phenomenon in the way it really occurs during compressor operation, the following test bench was design for flow visualization and video taping.

As shown in fig 1 and 2 the bench consists of a rotary compressor with the housing end open near the frame fixation. This set up allowed the operation of the pump, roller especially, to be seen through an acrilic cylinder. The lubricating oil pressure inside the pump was kept high using an oil separator hermetically connected with the discharge line and the shaft oil hole. The clearances between the moving parts were kept in the same order of the original compressor assembly. A stroboscope light was used to visualize the flow at a fixed crankshaft angle or with a small progressing velocity. So that one can set the lamp to the point of the passing minimal clearance of the roller.

FLOW VISUALIZATION

Analysing the dynamic test bench under operation, it can be observed that there is a strong formation of bubbles just after the minimal clearance line is transpassed by the flow. This bubble formation can be explained considering that the flow in the minimal clearance is formed basically by the two oil films which are adhered to the cylinder and roller walls, and that will stick together at the vicinity of the minimal clearance (fig.3). One has to have in mind that rolling piston rotaries are basically oil flooded compressors.

After the minimal clearance line is transpassed by the leakage a abrupt pressure drop in the oil flow will cause the liberation of the absorbed refrigerant gas in the form of bubbles.

STATIC TEST BENCH

From the described leakage behavior in the dynamic test bench, it appears that a proper model for the leakage through the minimal clearance would be that of a flow of oil through convergent - divergent plates with bubble formation.

In terms of the compressor volumetric efficiency the gas leakage can be determined by the difference of concentration of gas in the oil at the compression and suction volumes conditions during operation.

A question that remains is how the bubble formation will retard, partially blocking, the leakage flow.

To provide further insight into the leaking mechanism, it was built another test bench (fig. 4 and 5). A pumping set of rolling piston type was kept stationary and with all clearances hermetically tignt, except the minimal clearance. An acrilic cylinder could be mounted for visualization purpose. The pressure difference was kept constant by an auxiliary compressor, used in connection with two gas plus oil bottles at the upstream and dowstream of the cylinder. The mass flow rate was measured with a Bernoulli's mass flowmeter.

With the acrilic cylinder it was seen that the flow through the minimal clearance is of the same pattern as that show in the dynamic tests, i.e. solely oil flow in the upstream and bubbles formation plus oil flow in the downstream. This is a strong confirmation of the previously reported flow behaviour.

Using the mass flowmeter and reading the difference of oil levels in the up stream and downstream bottles for a given time, the mass flow rates could be obtained for a large number of pressure differences and upstream temperatures (oil viscosity).

PROPOSED MODEL FOR THE OIL FLOW

The flow geometry to be analyzed is shown in fig. 6 where one can identify R as the cylinder radius, Rp as the rolling piston radius, Ec as the rolling piston eccentricity with respect to the cylinder center (o) and r and θ as the radial circunferential independent variables for the fluid flow.

From fig. 6 one can easily obtain a as function of θ as shown in Eq. 1, where a is the radial distance with respect to the cylinder center, which identifies the solid boundary over the rolling piston. The other solid boundary for the flow is shown at r = R. $\theta=\pi/2$ identifies the location of the minimum clearance.

$$a = E_c \sin\theta + \sqrt{R_p^2 - E_c^2 * \cos^2\theta} \qquad (1)$$

The flow geometry is therefore a convergent-divergent passage which induced former researchers (1, 2) to consider it as being the flow through a convergent-divergent nozzle.

The flow is typically viscous and pressure driven, therefore the hypothesis of isentropic flow is not acceptable at all. Yanagisawa and Shimizu (4) recognized that the viscous effects were important in this type of flow and considered the flow through the minimum clearance as being the isentropic flow through a convergent nozzle and a viscous flow through a constant diameter duct.

In order to calculate the flow rate through the minimum clearance, it is necessary to establish the force balance in a fluid element, as shown in figure 7.

The velocity distribution of the fluid is calculated considering a steady, incompressible, constant properties 2-D flow. Inertial effects are neglected although curvature in the geometry is considered.

In θ - direction, the force balance on the fluid element is given by Eq. (2).

$$dF_1 - dF_2 + dF_3 - dF_4 = 0 \qquad (2)$$

or, for a unit depth element.

$$\left(p - \frac{\partial p}{\partial \theta}\frac{d\theta}{2} \right) dr - \left(p + \frac{\partial p}{\partial \theta}\frac{d\theta}{2} \right) dr + \left(\tau_{r\theta} + \frac{\partial \tau_{r\theta}}{\partial r}\frac{dr}{2} \right)\left(r + \frac{d r}{2} \right) d\theta \qquad (3)$$

$$- \left(\tau_{r\theta} - \frac{\partial \tau_{r\theta}}{\partial r}\frac{dr}{2} \right)\left(r - \frac{d r}{2} \right) d\theta = 0$$

Simplifying and grouping the similar terms, one gets

$$\frac{\partial}{\partial r}\left(r\tau_{r\theta} \right) = \frac{\partial p}{\partial \theta} = f(\theta) \qquad (4)$$

It is important to recognize that Eq.(4) is dependent on θ because the pressure gradient is variable along the flow.

Considering the shear stress $\tau_{r\theta}$ being dominantly generated by the velocity variation in r - direction one can write

$$\tau_{r\theta} = \mu \frac{\partial u}{\partial r} \qquad (5)$$

where μ is the fluid dynamic viscosity.

Substituting Eq.(5) into Eq.(4), a general solution which satisfies this equation is

$$u = \frac{1}{\mu}\frac{\partial p}{\partial \theta} r + f_1(\theta) \ln r + f_2(\theta) \qquad (6)$$

where $f_1(\theta)$ and $f_2(\theta)$ are determined by the following boundary conditions:

For r = a and r = R, u = o.

These conditions yields

$$f_1 = \frac{-(R - a)}{\mu \ln(R/a)} \frac{\partial p}{\partial \theta} \qquad (7a)$$

and

$$f_2 = -\frac{1}{\mu}\left(R - \frac{(R - a) \ln R}{\ln(R/a)} \right) \frac{\partial p}{\partial \theta} \qquad (7b)$$

Substituting Eqs. (7a) and (7b) into Eq. (6) and neglecting the centripetal effects one gets the velocity profile for each position along the flow, as shown in Eq. (8).

$$u = -\frac{1}{\mu}\left(\frac{dp}{d\theta} \right) R \left(1 - \frac{r}{R} - \frac{1-a/R}{\ln(a/R)} \ln(r/R) \right) \qquad (8)$$

By neglecting the velocity variation in z - direction, the volumic flow rate can be determined by Eq.(9) where H is the piston width.

$$q = \int_a^R u\, H\, dr \qquad (9)$$

After substituting Eq.(8) into Eq.(9) and integrating one gets the pressure gradient along the flow as a function of θ,

$$\frac{dp}{d\theta} = - 2\mu q/HR^2 \Big/ \left(1 - \frac{a^2}{R^2} + \frac{2\ (1 - a/R\ )^2}{\ln(a/R\ )} \right) \qquad (10)$$

in which a is given by Eq. (1).

The numerical integration of Eq.(10) yields the pressure profile along the flow, as q is constant.

Fig. 8 shows the calculated pressure distribution along the flow, using a fourth order Runge-Kutta scheme with an integration step of 5×10^{-4} rd and an initial condition of p_i = 11,85 bar for $\theta = \theta_i$. The choice of the initial angle of integration does not influence the pressure distribution for the flow of oil in the minimum clearance. This result was obtained for q = $5{,}0 \times 10^{-6}$ m^3/s and μ = 50 cP.

The pressure profile shows a very steep gradient around which is the location of the minimum clearance and a very small variation over $\pm 10^{\circ}$ in both directions. Beyond this region the pressure is practically constant. The region of pressure variation depends upon the geometrical dimmensions of the roller. This type of pressure profile for the flow of oil is compatible with the flow observations and the refrigerant bubbles growing process pointed before. When the flow of oil with dissolved refrigerant reaches the minimum clearance, the bubbles grow up very rapidly making the local temperature to decrease. This makes the oil viscosity to increase with a reduction in the overall leakage.

Velocity profiles, for the same case shown in Fig. 8, is presented in Fig.9. The velocity profiles are symmetric over the gap and show a strong dependance with θ . If only oil is present in the flow, and this is the only case analysed in this paper, there is also a symmetry in the velocity profile with respect to the location of the minimum clearance.

COMPARISON OF THE NUMERICAL AND EXPERIMENTAL RESULTS

Using the static test bench described before, several results for mass flow rate and total absolute pressure ratio (ps/pd) are presented in Fig.10, together with the numerical results. These results were obtained for δ= 18 μm and δ= 22 μm, where δ is the dimension of the minimum clerance, and keeping all the other parameters constant.

This type of flow is highly influenced by the dimension of the minimum clearance and the viscosity of the oil therefore the geometrical dimensions and the temperature measurements have to be controlled carefully. The uncertainty for the minimum clearance measurements has to be on the order of $\pm$1 μm and for the temperature measurements about $\pm$0.5° C to achieve a good agreement between numerical and experimental results.

FINAL COMMENTS

A novel model for the flow of oil through the minimal clearance in a rolling piston type compressor has been presented and validated.

The flow of oil is responsible for the leakage of refrigerant from the high pressure to the low pressure side of the compressor. Therefore, in order to adequately calculate the interaction of the refrigerant and the oil, the appropriate model has to be able to compute the pressure variation along the flow.

The visualization experiments indicated a very strong formation of bubbles after the minimum clearance due to a very steep pressure gradient. This expansion process reduces the temperature of the oil film, increases the viscosity of the oil and diminishes the flow rate and the refrigerant leakage.

Comparisons between experiment and prediction yielded strong support for the proposed model.

ACKNOWLEDGMENT

To our technician Sidnei K. Himpel whose ingenuous and skilled work achieved the experiment benches.

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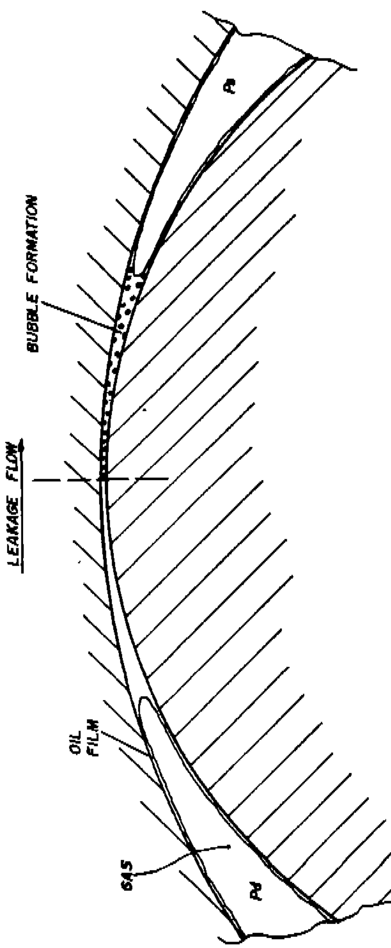

FIG. 3 — LEAKAGE FLOW PATTERN

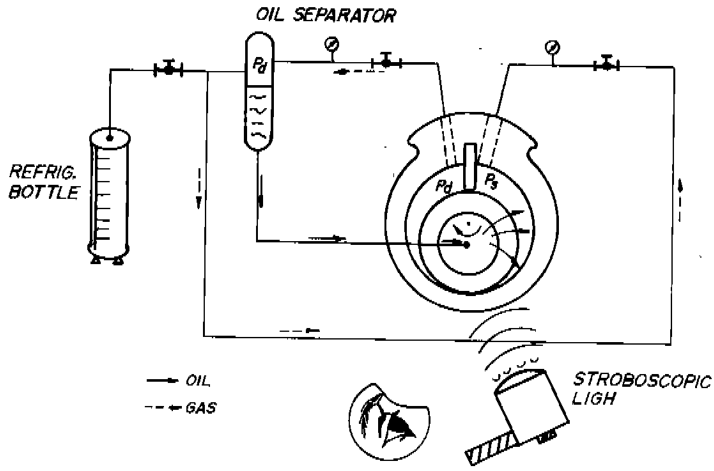

FIG I - DYNAMIC BENCH DIAGRAM

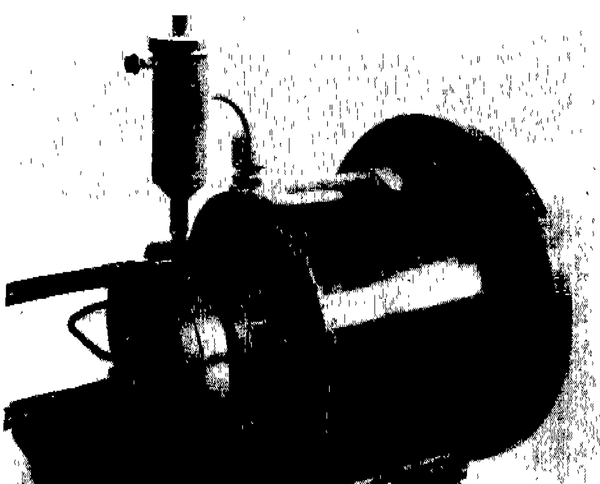

FIG. 2 - DYNAMIC BENCH COMPRESSOR

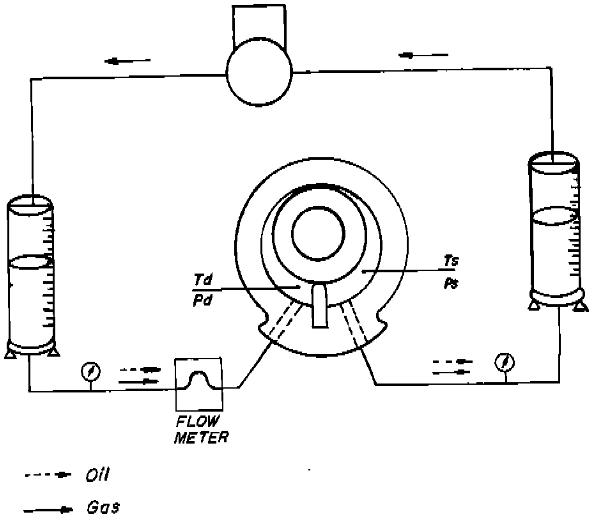

FIG. 4 - STATIC BENCH DIAGRAM

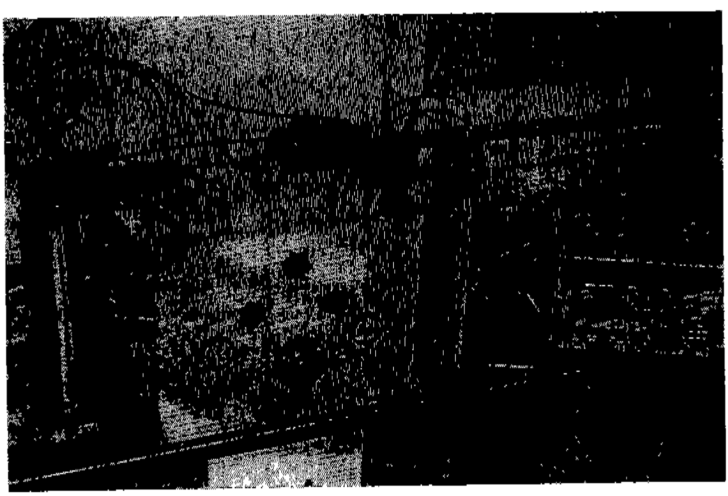

FIG.5 - STATIC BENCH

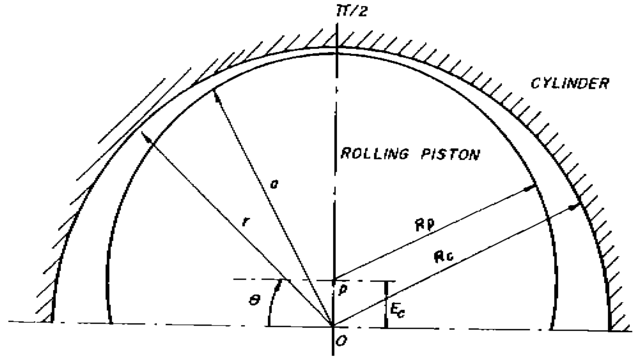

FIG. 6 — FLOW GEOMETRY THROUGHOUT THE MINIMUM CLEARANCE

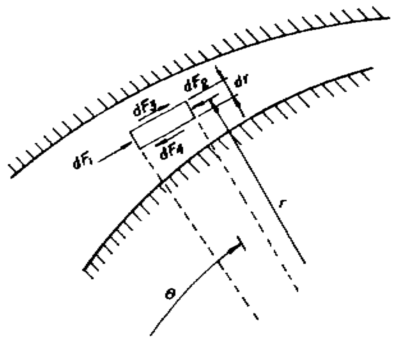

FIG. 7 — FORCES ON THE FLUID ELEMENT

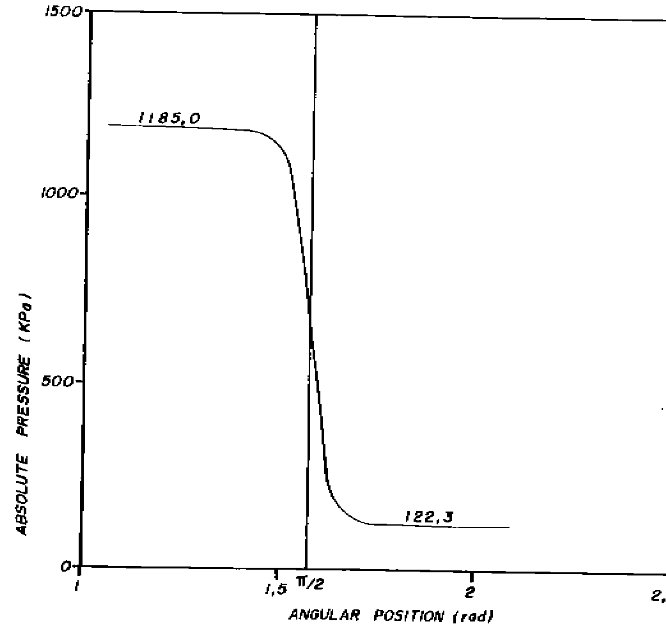

FIG 8 — PRESSURE DISTRIBUTION ALONG THE FLOW

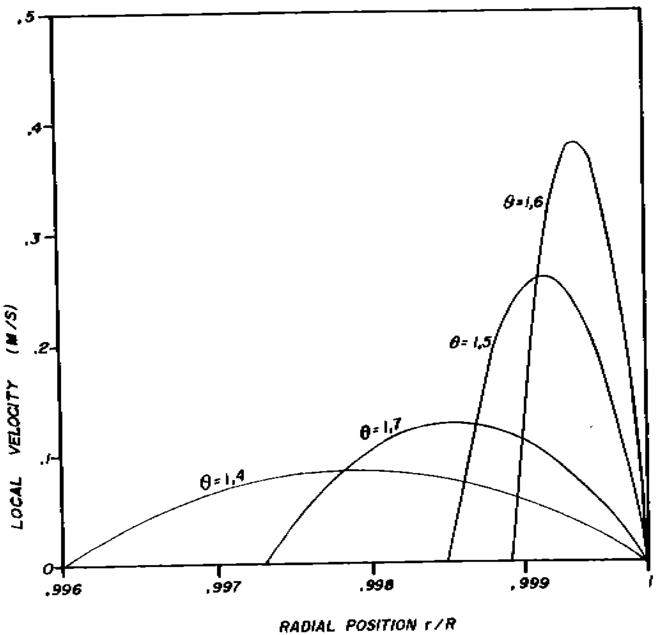

FIG. 9 – VELOCITY PROFILES ALONG THE FLOW

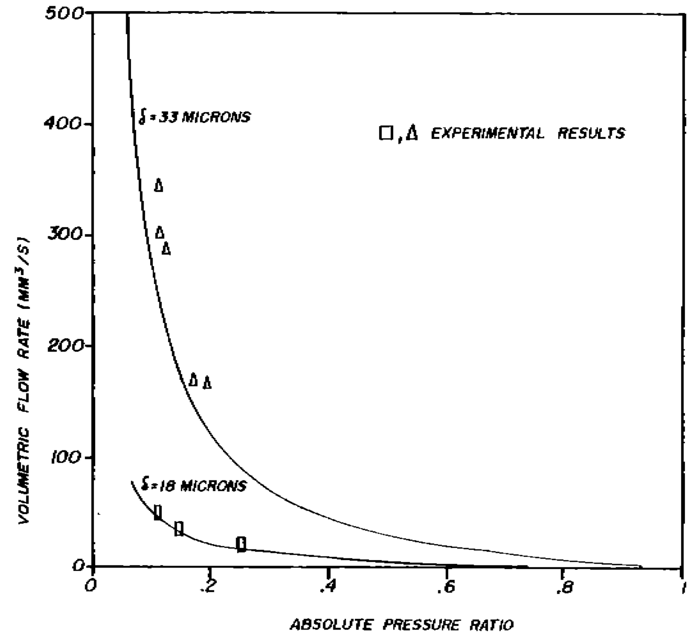

FIG. 10 – COMPARISON OF NUMERICAL AND EXPERIMENTAL RESULTS