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INFLUENCE OF THE ROLLER VELOCITY ON THE FLOW OF LUBRICATING OIL IN A ROLLING PISTON COMPRESSOR

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ABSTRACT

The oil flow through the minimal clearance in a rotating compressor is numerically investigated in presence of a moving piston. The full Navier-Stokes equation written in bicylindrical coordinates for laminar, incompressible and steady state conditions was employed to construct a three-dimensional model of the problem. Velocities and pressure results are presented as a function of the minimal clearance width and oil flow rate. Recirculation regions were observed and their influences on the oil leakage are explored.

INTRODUCTION

A small clearance between the piston and the cylinder in a rolling piston rotating compressor separates the suction and discharge chambers. During the compressor operation the clearance is filled with oil that serves as lubricant as well as sealant to prevent leaking of the refrigerating gas from the discharge to the suction chamber. The oil flow through this minimal clearance is driven by the pressure difference between the two chambers and by the rotation of the rolling piston.

The compressor volumetric efficiency is greatly affected by the refrigerant leaking through the minimal clearance. It is estimated that 70% of the internal losses of the refrigerating gas are due to leaking through the minimal clearance. Despite its importance, few works have deal with this problem. Costa et al. (1990) have made a literature review on the subject and introduced an experimentally validated one-dimensional model. Later, Ferreira et al. (1992) employed the full Navier-Stokes equation written in bicylindrical coordinates to construct a three-dimensional model of the flow through the minimal clearance. With their model a better agreement was obtained between experiment and computation for higher oil flow rates and larger clearances.

In the aforementioned models the piston rotation was ignored. However, in the real situation the rolling piston possesses a variable tangential relative velocity with respect to the cylinder. The purpose of the present paper is to incorporate the tangential velocity of the rolling piston in the three-dimensional model introduced by Ferreira et al. (1992). It is expected that the piston movement will significantly alter the flow pattern originating recirculation regions both at the entrance and exit of the clearance. In what follows, velocity and pressure profiles will be presented and the main features of the flow will be discussed.
PROBLEM FORMULATION

The geometry of the minimal clearance of the rolling piston compressor can be well described by a bicylindrical coordinate system (ψ, η and z), as shown in Fig. 1.

![Fig. 1 Geometry of the problem](image)

Indicated in the figure are the cylinder radius, \( R_c \), the piston radius, \( R_p \), and the \( ψ \) and \( η \) coordinates; at \( R_c, \ η=η_2 \), and at \( R_p, \ η=η_1 \). The \( z \) coordinate is along the direction perpendicular to the plane of the figure and is not shown. The high pressure region is separated from the low pressure region in one side by the blade and in the other side by the minimal clearance. According to the figure, the oil flows clockwise through the minimal clearance from the high pressure to the low pressure region while the rolling piston moves either in the clockwise or counter-clockwise direction depending on the position of the minimal clearance.

The oil flow is governed by the continuity and Navier-Stokes equations written in bicylindrical coordinates for incompressible, isothermal and steady state conditions, that is,

\[
1/h^2 \left[ \partial (phu) / \partial \psi + \partial (phv) / \partial \eta + \partial (ph^2 w) / \partial z \right] = 0
\]

\[
1/h^2 \left[ \partial (phuu) / \partial \psi + \partial (phvu) / \partial \eta + \partial (ph^2 wu) / \partial z \right] = -(1/h) \partial p / \partial \psi + (\mu / h^2) \left[ \partial^2 u / \partial \psi^2 + \partial^2 u / \partial \eta^2 + h^2 \partial^2 u / \partial z^2 \right] + \mu / h^2 \left[ (2/h)(\partial u / \partial \eta)(\partial v / \partial \eta) - (2/h)(\partial \psi / \partial \psi)(\partial v / \partial \psi) - (2/h)(\partial \psi / \partial \eta)(\partial v / \partial \eta) + (u / h)(\partial^2 \psi / \partial \eta^2 + \partial^2 \eta / \partial \eta^2) \right] - (p / h^2)[uv\partial h / \partial \eta - v^2 \partial h / \partial \psi]
\]
\[
1/h^2[\partial(\rho u \psi) / \partial \psi + \partial(\rho v \psi) / \partial \eta + \partial(\rho w \psi) / \partial z] = -(1/h)\partial p / \partial \psi + (\mu / h^2)[\partial^2 \psi / \partial \eta^2 + \partial^2 \psi / \partial \eta^2 + \\
+ h^2 \partial^2 \psi / \partial z^2] + \mu / h^2 [(2/h)(\partial \psi / \partial \eta)(\partial \sigma / \partial \eta) - (2/h)(\partial \psi / \partial \eta)(\partial \sigma / \partial \eta)] + \\
-(v/h)(\partial^2 \sigma / \partial \eta^2 + \partial^2 \sigma / \partial \eta^2) - (\rho / h^2)[uv \partial \psi / \partial \eta - u^2 \partial \psi / \partial \eta] 
\]

\[
1/h^2[\partial(\rho u \psi) / \partial \psi + \partial(\rho v \psi) / \partial \eta + \partial(\rho w \psi) / \partial z] = -\partial p / \partial z + \\
+(\mu / h^2)[\partial^2 w / \partial \psi^2 + \partial^2 w / \partial \eta^2 + h^2 \partial^2 w / \partial z^2] 
\]

where \(u, v\) and \(w\) are, respectively, the velocity components in the \(\psi, \eta\) and \(z\) directions, \(p\) is pressure, \(\rho\) is the oil density, \(\mu\) its viscosity, and \(h\) is the square root of the metric (Lami's coefficient), given by

\[
h = a / (\cosh \eta - \cos \psi) 
\]

in which \(a\) is the geometric parameter of the coordinate system (Gasche, 1992).

The boundary conditions associated to equations (1) to (4) are zero velocity at all solid walls, except at the rolling piston \((\eta = \eta_1)\) in which \(u = U\). A fictitious boundary condition is applied at the blade \((\psi = \beta\) and \(2\pi - \beta, \beta = \pi/36\) is half the blade thickness\)) to explore different mass flow rates through the minimal clearance. Therefore, at \(\psi = \beta\) and \(2\pi - \beta\), \(v = w = 0\) and \(\int udA = \bar{m} / \rho\), in which \(A\) is the flow cross-sectional area and \(\bar{m}\) is a prescribed mass flow rate. To cope the prescribed mass flow rate boundary condition with the \(u = U\) and \(u = 0\) conditions at \(\eta = \eta_1\) and \(\eta_2\), respectively, an iterative process was required. It should be noted that, in reality, suction and discharge valves are located in each side of the blade. However, since the main focus of the present work is to investigate the flow in the minimal clearance, the aforementioned fictitious boundary condition provides a suitable manner to explore leakage as a function of the oil flow rate, \(\bar{m}\).

**NUMERICAL METHODOLOGY**

The differential equations (1)-(4) and the associated boundary conditions were discretized using a finite volume methodology. Staggered meshes with respect to pressure were employed for each velocity component. The numerical integration of the convective-diffusive terms adopted the power law interpolation scheme according to Patankar (1980); all other terms were discretized using central difference. The coupled between pressure and velocity was accomplished via Patankar's SIMPLE algorithm, and the algebraic equations were solved through a combination between the Tri-Diagonal Matrix Algorithm and the Gauss Seidel methodology. A block correction strategy accelerated the convergence of the numerical solution. More details on the discretization as well as on other aspects of the numerical methodology and solution can be found in Gasche (1992).

The final mesh used to generate the results to be presented here has 2480 nodal points, being 31 points in the \(\psi\) direction, 20 points in the \(\eta\) direction and 4 points in the \(z\) direction. Finer meshes have been tested without showing greater improvement of the results.
RESULTS AND DISCUSSIONS

The influence of the tangential velocity of the rolling piston on the pressure profile along the minimal clearance is explored in Figs. 2 and 3 for a constant value of the oil flow rate, \( m \), and minimal clearance thickness, \( \delta_{\text{min}} \). Figure 2 and 3 refer, respectively, to \( \delta_{\text{min}}=80 \) \( \mu \)m and 10 \( \mu \)m, and for both figures, \( m=0.001 \) kg/s. Three piston velocities are covered, \( U=1 \), zero and -1 m/s; negative values occur when the prescribed velocity is contrary to the direction of the oil flow, that is, from the high to the low pressure region. The suction pressure, \( P_s \), was set equal to zero and the discharge pressure, \( P_d \), varied according to \( m \), \( \delta_{\text{min}} \) and \( U \). In all curves the same pattern is observed: pressure remains constant for most of the clearance and at the minimal clearance an abrupt pressure drop is observed due to the big change on the flow cross sectional area. As can be seen from the figures, the discharge pressure for a piston velocity contrary to the direction of the oil flow is considerably higher than that for both piston and oil rotating in the same direction. Furthermore, the dependence of \( P_d \) with \( U \) is stronger for higher values of \( \delta_{\text{min}} \), due to the increasing importance of the inertia terms for increasing values of \( \delta_{\text{min}} \).

![Fig. 2 Pressure profile along the clearance; \( \delta_{\text{min}}=80 \) \( \mu \)m.](image1)

![Fig. 3 Pressure profile along the clearance; \( \delta_{\text{min}}=10 \) \( \mu \)m.](image2)

Velocity profiles along the radial direction for different locations in the clearance are shown in Figs. 4 an 5. Figure 4 is for \( U=1 \) m/s and Fig. 5 for \( U=-1 \) m/s. For both figures the oil flow rate is from the left to the right, \( m=0.001 \) kg/s and \( \delta_{\text{min}}=80 \) \( \mu \)m. In preparing Figs. 4 and 5, instead of the \( \psi \) coordinate use was made of \( \theta \); the computation domain associated with the figures corresponds to \( 0 \leq \theta \leq 144^\circ \), and the location of the minimal clearance is 72°. Also, the radial location, \( r \), was normalized by the radial gap \( \delta(\theta) \). The first thing to be noted in the figures is the occurrence of recirculation zones as the oil approaches and exits the minimal clearance. This is so even when the piston velocity is in the same direction of the oil flow. The recirculation zone for oil and piston rotating in the same direction occurs due to the shear action associated with the piston velocity that drags more oil into the minimal clearance than that corresponding to the given flow rate. Similar phenomenon occurs when the oil exits the minimal clearance. For oil and piston rotating in opposite directions, the recirculation zones close to the piston surface are obvious.
To explore the influence of the piston tangential velocity on the discharge pressure, $p_d$, a dimensionless pressure was introduced,

$$ p_d^* = \frac{\left(p_d - p_s\right)p\delta_{\text{min}}^2}{\mu^2} \quad (6) $$

![Fig. 4](image1.png) Velocity profiles in the clearance; $U = 1$ m/s.

![Fig. 5](image2.png) Velocity profiles in the clearance; $U = -1$ m/s.

![Fig. 6](image3.png) Discharge pressure as a function of piston tangential velocity.

![Fig. 7](image4.png) Oil flow rate as a function of piston tangential velocity.
Values of $p_d^*$ as a function of the U having $\delta_{\text{min}}$ as curve parameter are shown in Fig. 6 for $\dot{m}=0.001$ kg/s. To explore such figures it should be kept in mind that the suction pressure, $p_s$, remains constant and equal to zero. The increase in $p_d$ for decreasing values of $\delta_{\text{min}}$ is due to the greater pressure drop associated to the smaller gap between piston and cylinder. The dependence of $p_d$ with U for fixed values of $\delta_{\text{min}}$ is also expected: since $\dot{m}$ remains constant, as U increases carrying more oil, $p_d$ decreases to compensate for that.

Of more practical interest is the dependence of $\dot{m}$ with U for fixed values of $\delta_{\text{min}}$ and $P_d$. To this extent, Fig. 7 was prepared. An interesting finding is the virtually small influence of the piston velocity on the oil flow rate for small values of $\delta_{\text{min}}$. This is so because for small gaps between piston and cylinder the momentum equation is governed by a balance between pressure and friction with inertia playing no role. As $\delta_{\text{min}}$ increases, the inertia term becomes more important and for a fixed pressure difference $(p_d-p_s)$, $\dot{m}$ increases with increasing values of U.

CONCLUSIONS

The present work is an extension of that proposed by Ferreira et al. (1992) in which the piston rotating velocity is incorporated in the model of the oil flow rate through the minimal clearance in a rolling piston compressor. The introduction of an additional mechanism to drive oil into the compressor clearance resulted in recirculating zones whose effect upon the oil flow rate increased in importance with increasing values of the gap between piston and cylinder. For small gaps between piston and cylinder, the momentum equation is governed by a balance between friction and pressure with inertia playing no role. In turn, the results for small values of $\delta_{\text{min}}$ were virtually insensitive to the piston velocity. For larger values of $\delta_{\text{min}}$ the oil flow rate was greatly affected by the piston velocity.

REFERENCES


