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ABSTRACT

The work presents a numerical methodology to simulate the dynamic behavior of reed type valves of reciprocating compressors used in refrigeration. A one-degree of freedom model is adopted for the reed motion and a finite volume method is employed to obtain the flow field through the valve. The valve dynamics and the time dependent flow field are coupled and solved simultaneously. As a first step, the flow through the valve is assumed laminar, incompressible and isothermal. Due to the inclined valve motion a three-dimensional numerical model was developed to take into account the variation with time of the computation domain. Through a coordinate transformation a moving coordinate system was obtained; this new system was able to expand and compress according to the valve displacement. Results for the force acting on the reed were obtained for a prescribed periodic velocity for the piston. The resulting flow field through the valve reveals a complex flow pattern, and that is explored in the paper.

INTRODUCTION

The adoption of automatic valves is a common practice in reciprocating compressors. These valves open and close depending on the pressure difference established by the piston reciprocating motion between the cylinder and the suction and discharge chambers. In designing the valve system one seeks fast response, large mass flow rate, low pressure drop when opened, and good backflow blockage when closed. To satisfy the high valve performance requirements necessary for a competitive compressor, a detailed understanding of the fluid flow through the valve as well as the dynamics of the valve is necessary.

Fig. 1 is a schematic representation of a compressor and a discharge valve that can be used to explain the principles of the valve operation (for the suction valve a similar situation is found). Basically, the net force resulting from the pressure difference between the cylinder and the discharge chamber brings about the reed displacement. This creates a gap between the reed and valve seat through which the gas is allowed to flow. As illustrated in Fig. 1, the fluid enters the valve through the feeding orifice flowing axially and then, due to the presence of the reed, it is forced to flow radially until the valve exit. The pressure difference between the orifice entrance and the valve exit, together with the displacement gap between reed and valve seat, govern the fluid flow throughout the valve. In turn, the flow dictates the pressure distribution on the reed and, consequently, the resultant force that governs its dynamics.

The analysis of the flow through reed type valves has received the attention of many researchers. A good review of works on the subject can be found in MacLaren (1972, 1982) and Prata & Ferreira (1990). One of the first attempts to model the coupling between the flow and the reed dynamics was carried out by Lopes & Prata (1997) for an isothermal, incompressible, laminar flow. Due to computational limitations, Lopes & Prata (1997) decided to approximate the reed geometry by a circular disc positioned concentric and parallel to the valve seat. The transient character of the flow was approximated through a

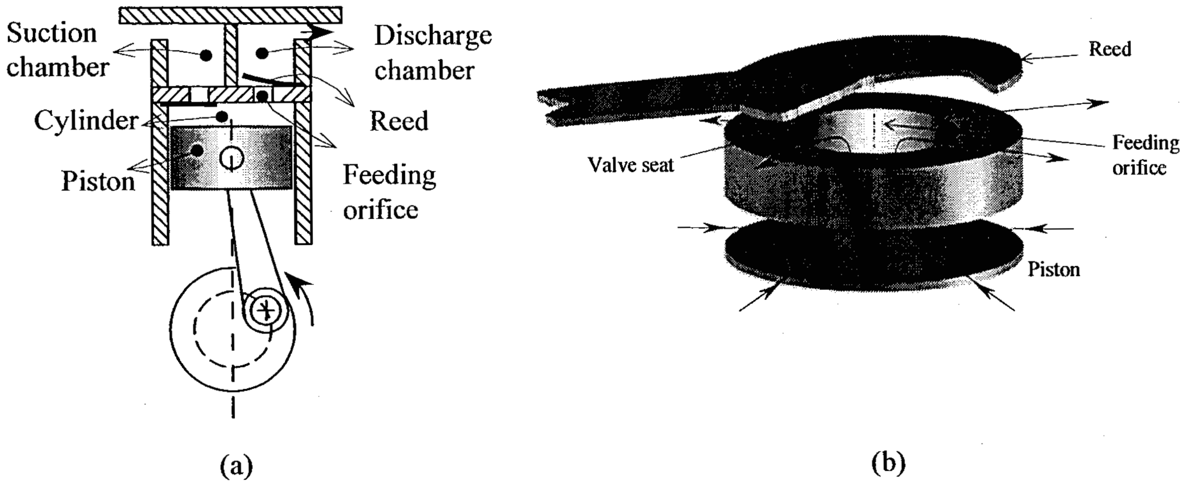


Fig. 1 - Schematic view of o compressor and a discharge valve.

periodic inlet flow condition at the feeding orifice entrance. A numerical solution for the flow field was obtained and from this the resultant force on the reed was evaluated. A one-dimensional dynamic model was then employed to solve for the reed motion.

More recently, Matos et al. (1999) have extended the work of Lopes & Prata (1997) by replacing the periodic inlet velocity condition at the feeding orifice with a pressure drop through the valve that varied with time. This boundary flow condition seems to be more realistic but there are some doubts as to whether it can applicable to practical valve designs. An alternative to circumvent this problem that is considered here is to include the cylinder region and the piston movement into the numerical model. As will be explained shortly, the solution domain for the cylinder region was limited to a radial position equal to the reed radius, as can be seen from Fig. 1b.

PHYSICAL MODEL

In the present work the reed is considered to be rigid and articulated at one of its extremities (Fig. 2). The reed dynamics is modeled by the following equation:

$$m\ddot{\alpha} + C\dot{\alpha} + K\alpha = \frac{F - F_0}{R} \tag{1}$$

where F_0 is a pre-load force on the reed, α is the instantaneous reed inclination angle, R is the distance between the point of application of force F and the articulation point shown in Fig. 2. The valve stiffness and damping coefficients, K and C, respectively, as well as the valve mass, are determined experimentally. In order to solve equation (1) for the reed inclination α, the force F due to the flow pressure distribution on the reed surface is evaluated according to

$$F = \int_0^{2\pi} \int_0^{D/2} p\, r\, dr\, d\theta \tag{2}$$

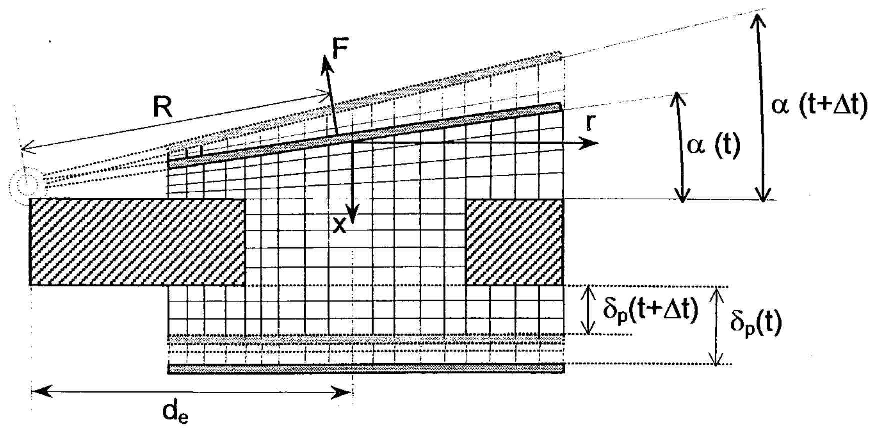


Fig. 2 – Schematic view of the computational domain.

where r and θ are the radial and circumferential coordinates, respectively, and D is the reed diameter. The instantaneous pressure distribution p in equation (2) is obtained from the numerical solution of the flow through the valve.

The point of application of the force due to the flow pressure distribution, R, is evaluated as

$$R = \frac{\int_0^{2\pi}\int_0^{D/2} p\, r\,(d_e + r\cos\theta)\sec\alpha\, dr\, d\theta}{\int_0^{2\pi}\int_0^{D/2} p\, r\, dr\, d\theta} \tag{3}$$

It should be noted that in equation (3) both force and its point of application vary with time since they are dependent on the flow. In the present work R is also considered to be the gravity mass center of the reed.

The flow illustrated in Fig. 1 was considered to be isothermal, incompressible and laminar. The more realistic situation of turbulent flow has been considered by Salinas-Casanova et al. (1999) for the case of inclined reed at a fixed displacement. The inclusion of turbulence and compressibility effects will be a next step in the development of the methodology presented herein.

Equations (1-3) combined with the conservation equations for mass and momentum (velocity components u, v e w along axial, radial and circumferential directions, respectively) fully describe the problem and are used to evaluate the seven unknowns (α, F, R, u, v and p). Boundary conditions required to solve such equations are discussed in the follow section.

NUMERICAL METHODOLOGY

A moving coordinate system as proposed by Lopes & Prata (1997) is adopted here to solve the governing equations in the physical domain that expands and contracts as the valve reed moves up and down, respectively. The main feature of this coordinate system is that it transforms the physical domain into a computational domain that remains unchanged regardless the reed and the piston motions.

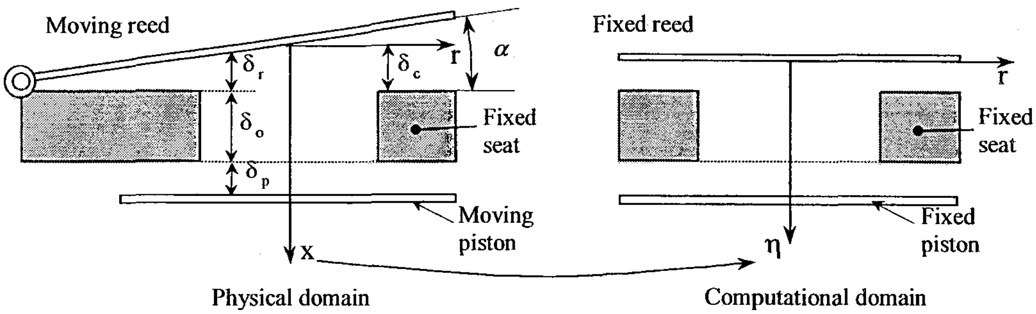


Fig. 3 - Physical and computational domains.

Accordingly, the axial coordinate, x, of the inertial system is replaced by a new axial coordinate, η, using the following transformation,

$$\eta = \frac{x(t) - x_r}{x_s(t) - x_r}; \qquad \eta = x; \qquad \eta = \frac{x(t) - x_h}{x_p(t) - x_h} \tag{4}$$

corresponding to the region between reed and seat, the feeding orifice and the cylinder, respectively. The subscripts "s", "r", "p" and "h" refer to seat, reed, piston and cylinder head. The physical and computational domains are shown in Fig. 3.

Transforming the continuity and momentum equations from the fixed coordinate system to the moving coordinate system yields,

$$\frac{1}{\delta}\frac{\partial}{\partial t}(\rho\delta\,\phi)+\frac{1}{\delta}\frac{\partial}{\partial \eta}(\rho\,\tilde{u}\,\phi)+\frac{1}{r}\frac{\partial}{\partial r}(\rho\, r\, v\,\phi)+\frac{1}{r}\frac{\partial}{\partial \theta}(\rho\, w\,\phi)=\frac{1}{\delta}\frac{\partial}{\partial \eta}\left(\frac{\Gamma^{\phi}}{\delta}\frac{\partial \phi}{\partial \eta}\right)+ \\ +\frac{1}{r}\frac{\partial}{\partial r}\left(r\Gamma^{\phi}\frac{\partial \phi}{\partial r}\right)+\frac{1}{r}\frac{\partial}{\partial \theta}\left(\frac{\Gamma^{\phi}}{r}\frac{\partial \phi}{\partial \theta}\right)+\hat{S}^{\phi} \tag{5}$$

where $\phi = 1$ and $S^{\phi} = \Gamma^{\phi} = 0$ for continuity, $\phi = u$, $\Gamma^{\phi} = \mu$ and $S^{\phi} = -(1/\delta)(\partial p/\partial \eta)$ for momentum conservation along the axial direction, $\phi = v$, $\Gamma^{\phi} = \mu$ and $S^{\phi} = -(\partial p/\partial r) - (2\mu/r^2)(\partial w/\partial \theta) - (\mu v/r^2) + (\rho w^2/r)$ for momentum conservation along the radial direction, $\phi = w$, $\Gamma^{\phi} = \mu$ and $S^{\phi} = -(1/r)(\partial p/\partial \theta) + (2\mu/r^2)(\partial v/\partial \theta) + (\rho v w/r) - (\mu v/r^2)$ for momentum conservation along the circumferential direction.

The transformed coordinate η varies from 0 to 1 in the region between the reed and the valve seat, from δ_r to $(\delta_r+\delta_o)$ in the feeding orifice and from 0 to 1 in the region between the piston and the cylinder head. In equation (5) the value for δ is equal to δ_r (= $[d_e + r\cos\theta]\tan\alpha$) in the region between the reed and the valve seat, to δ_o (= 1) in the feeding orifice and to δ_p (= clearance between the piston and the cylinder head) in the cylinder region. The axial velocity component u originally in equation (5) was replaced by $\tilde{u}$ which is the fluid axial velocity with respect to the moving coordinate, η. Once u is determined, $\tilde{u}$ can be obtained from

$$\tilde{u} = u - u_g \tag{6}$$

where u_g is the instantaneous velocity of the coordinate η given by,

$$u_g = \left(\frac{\partial x}{\partial t}\right)_{\eta,r} = \eta\frac{\partial \delta_r}{\partial t} = \eta\dot{\delta}_r \;; \qquad u_g = 0 \;; \qquad u_g = \left(\frac{\partial x}{\partial t}\right)_{\eta,r} = \eta\frac{\partial \delta_p}{\partial t} = \eta\dot{\delta}_p \tag{7}$$

according to the domain being considered; i.e. region between reed and seat, feeding orifice and cylinder, respectively. In equation (7) $\dot{\delta}_r$ is the instantaneous local velocity on the reed surface and $\dot{\delta}_p$ is the piston velocity.

The conservation equations for mass and momentum are solved using the finite volume discretization method on a staggered grid scheme. The system of algebraic equations that result from the integration of the governing differential equations over each control volume are solved using the Tridiagonal Matrix Algorithm (TDMA). Interpolation of unknown quantities at the control volume faces were obtained using the QUICK interpolation scheme (Hayase et al., 1992) which is considered to be a second order procedure. A segregate approach was employed to solve for the equations and the coupling between pressure and velocity was handled through the SIMPLEC algorithm (Versteeg & Malalasekera, 1995). A fully implicit approximation was adopted for the time interpolation.

At the solid walls all velocity components were taken equal to zero except at the surface of the valve reed and the surface of the piston where the axial velocity are $\dot{\delta}_r$ and $\dot{\delta}_p$, respectively. The velocity of the reed is obtained from equation (1) whereas the piston velocity is prescribed as

$$\delta_p = 0.05\, d\, (1 + 0.7 \text{ sen } \omega t) \quad , \tag{8}$$

where d is the feeding orifice diameter and $\omega = 2\pi f$.

At the symmetry plane (θ=0 and θ=π) symmetry boundary conditions were the natural choice, i.e. $w = \partial u/\partial\theta = \partial v/\partial\theta = 0$. To reduce computational cost it was decided to limit the extent of the solution domain in the cylinder region to a radial location equal to the reed radius. The instantaneous velocity condition needed at this boundary was derived from a simple integral relation for mass conservation applied to the remaining of the cylinder volume. A further hypothesis associated to this boundary was to assume the presence of no velocity components in the axial and circumferential directions.

At the exit boundary a condition for the radial velocity component was made possible by using the momentum and mass conservation with the outlet pressure set to the atmospheric condition. For the other two components, u and w, the conditions were $\partial u/\partial r = \partial w/\partial r = 0$.

RESULTS

Possamai et al. (1995) conducted an experimental and computational investigation of the incompressible laminar flow through the valve, considering the reed at different fixed inclinations and under a constant mass flow rate. The geometry adopted in their work is very similar to that in Fig.2., with the exception that they did not include the cylinder region in the solution domain. Here their numerical and experimental results are used to validate the present methodology. To this end a numerical simulation was performed for a constant mass flow rate and with a periodic flow condition, given by a prescribed reed motion

$$\alpha = 0.4\pi/180\,(1 + 0.3 \text{ sen } \omega t) \qquad (9)$$

with a low frequency condition (f = 0.01 Hz).

The mass flow rate is characterized by the Reynolds number, Re, based on the average velocity in the feeding orifice of diameter d (= 34.8 mm). Other geometric parameters are the reed diameter D (= 104.4 mm) and the feeding orifice length e (= 100 mm). Air was the operating fluid with $\mu = 1.86 \times 10^{-5}$ Pa.s and $\rho = 1.16$ kg/m^3.

The computational grid used in the simulation had a total of 20 x 58 x 7 (x, r, θ directions) volumes in the region between reed and valve seat, and 21 x 23 x 7 (x, r, θ directions) volumes in the feeding orifice region. Fig. 4 shows a comparison between results for pressure distribution on the reed surface according to the present methodology and to the work of Possamai et al. (1995) for a reed inclination α=0.4° and a gap δ_c = 0.0201d. As can be seen the agreement between both methodologies is satisfactory and, therefore, the present numerical model was considered validated.

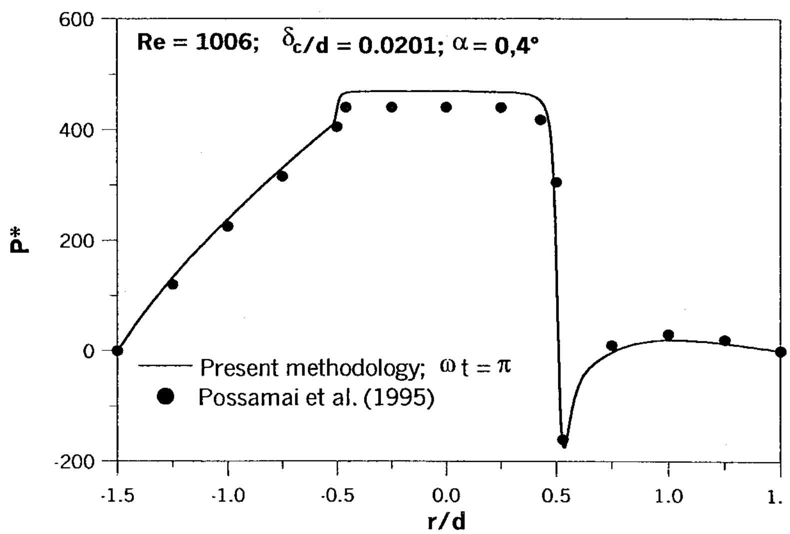


Fig. 4 – Pressure distribution on the reed surface according to different methodologies.

After validating the computational code, the work was directed to the solution of the flow with the piston moving in a periodic manner, according to equation (8) and with f = 60 Hz. For this case, the feeding orifice length e, the feeding orifice diameter d, the reed diameter D and the piston diameter D_p were equal to 0.525 mm, 3 mm, 4mm and 11.25 mm, respectively. The parameters in equation (1) were set to K = 200 N/m, C = 0.5 N s/m, m = 3.2 g e $F_0 = 0$. The reed displacement had its amplitude limited within the range $0.01125\text{ d} < \delta_c < 0.375\text{ d}$.

The solution domain shown in Fig. 2 had a computational grid with 19x68x7 (x, r, θ directions) volumes in the region between reed and seat, 15x29x7 (x, r, θ) volumes in the feeding orifice region, and 24x68x7 (x, r, θ directions) volumes in the cylinder region.

Fig. 5 shows results for force on the reed and the reed dynamics resulting for this flow condition. As can be seen, the force on the reed due to the pressure distribution experiences negative and positive values, which are directly linked to the piston motion. For parameters K and C adopted in equation (1), the reed gap does not change significantly along the whole cycle. This is the main reason why the flow field hardly affects the force on the reed, as one would expect if the reed displacement was large (see for instance Matos et. al., 1999). The inclusion of compressibility and turbulence effects into the present model will allow numerical simulations of much higher mass flow rates through the valve and, as a consequence, will provide the opportunity to investigate such range of valve gaps.

In Fig. 6 results for velocity vectors are shown for the plane of symmetry at two different positions of the cycle ($\omega t=1,34\pi$ and 2π). Fig. 6a represents the flow as the piston approaches top dead center. At this position, the flow is forced to exit the valve and two strong recirculating regions arise next to the feeding orifice wall. On the other hand, when $\omega t=2\pi$ (Fig. 6b) the piston is in its descending movement and the fluid suction is clearly shown; with the appearance of an equally complicated flow pattern.

The condition of incompressible flow employed here means that the suction and the discharge of the gas are in phase with the piston motion. This deficiency will be removed with the adoption of a compressible flow formulation.

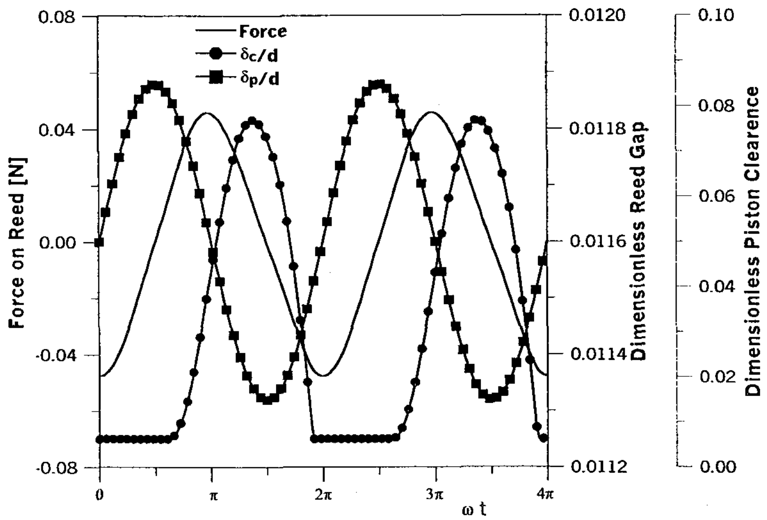
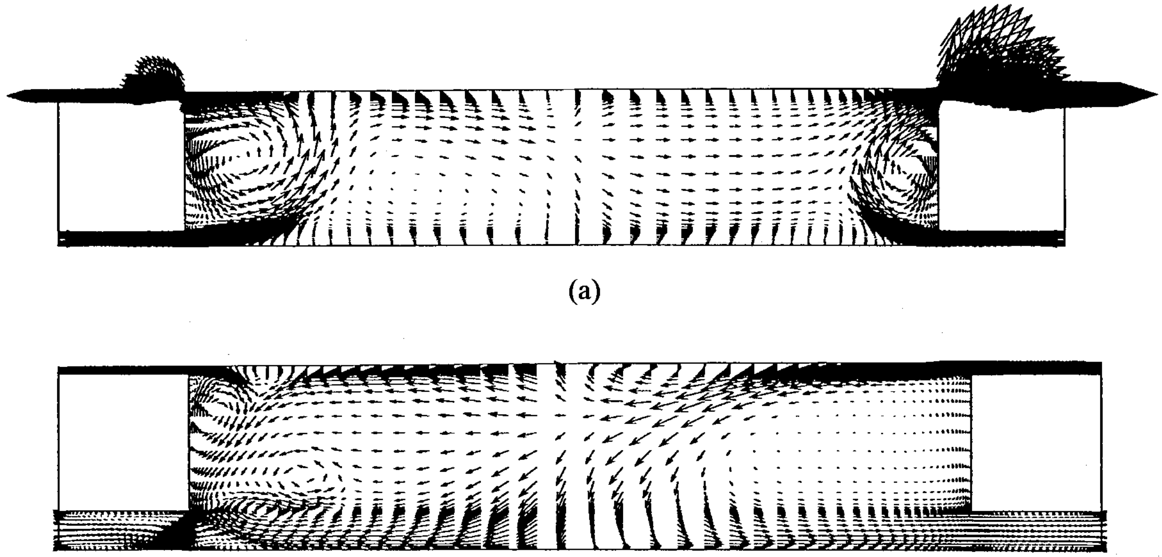


Fig. 5 - Gap between reed and seat and resultant force on the reed for a prescribed periodic piston motion.

CONCLUSIONS

The work presented a methodology to numerically solve the reed dynamics coupled with the flow through the valve of reciprocating compressors. The piston motion is included into the numerical model and originates the flow in the valve. The pressure distribution on the reed surface dictates the closing and the opening of the valve. The numerical solution indicates a complicated flow pattern in the feeding orifice. Although some basic features associated to both the reed dynamics and the flow through the valve are captured by the present methodology, the physical model has to be extended to take into account more realistic flow conditions, including effects of turbulence and compressibility.



(a)

(b)

Fig. 6 - Results for velocity vectors at the symmetry plane: (a) $\omega t=1,34\pi$; (b) $\omega t= 2\pi$.

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