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Modeling of Gas Leakage through Compressor Valves

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ABSTRACT

Gas leakage in the incomplete sealing of valves may significantly affect the volumetric and isentropic efficiencies of compressors. This paper describes a model developed to predict gas leakage in reed type valves of small reciprocating compressors commonly adopted for household refrigeration. Computations are carried out for a wide range of pressure differences that prevail in the valve clearances during the complete compression cycle. Results for mass flow rate in the suction and discharge valves are used to analyze the influence of geometric parameters on the leakage process. Reed deflection into the valve port due to pressure load is seen to be an important parameter of the analysis.

1. INTRODUCTION

Refrigeration compressors adopt automatic valves, which open due to the pressure difference between the compression and suction/discharge chambers. The specification of such valves is one of the most important steps in the design of a high efficiency compressor. For instance, if there are defects in the sealing of a valve due to surface irregularity or misalignment, gas leakage will occur and significantly reduce the compressor performance. This paper reports a simulation model developed to predict gas leakage in compressor valves, including the deflection of the valve into the valve port brought about by pressure load. The lubricating oil in compressors acts also as a sealing element for valves. However, since very little information is available about the amount of oil that is actually present between the valve and its seat, the presence of oil has been neglected in the model.

There are several studies in the literature about the flow through reed-type valves considering the simplified model of laminar flow in radial diffuser with small gap between parallel disks. Livesey (1960) adopted the approximated integral method of Von Karman and the hypothesis of a parabolic velocity profile to obtain the pressure distribution throughout the radial diffuser for an incompressible laminar flow. Savage (1964) solved the momentum equation by perturbing the creeping flow solution with a power series expansion for the laminar flow, without the need of specifying a velocity profile, and the results were seen to be in better agreement with experimental data obtained by Moller (1963).

Fleming *et al.* (1984) solved the compressible flow in radial diffuser taking into account the effect of viscous friction and variation of flow cross section area. Ghila (1995) solved the incompressible flow in a radial nozzle by finite difference method. He showed that the previous theories are acceptable only for low Reynolds numbers and in most cases underestimate the wall shear stress. Sato *et al.* (2005) developed a methodology to linearize the equations governing the adiabatic compressible flow between parallel disks for both outward and inward flow. His approximation is valid when the effect of viscous friction is much smaller than that caused by area change. Elhaj *et al.* (2008) carried out a simulation of an air double stage reciprocating compressor and presented a mathematical model for gas leakage through valves based on isentropic compressible flow through an orifice.

The pressure difference between the compression and suction/discharge chambers is the driving force of gas leakage through valves. The model developed in the present work assumes a one-dimensional flow through the gap between the reed valve and its seat, considering the effects of viscous friction, compressibility and valve deflection into the port due to pressure load. The main interest is to quantify the effect of leakage on both the isentropic and volumetric efficiencies.

2. MATHEMATICAL MODEL

2.1 Leakage through valves

The conservation equations are solved for an ideal gas, with the assumptions of laminar, compressible, adiabatic, steady, one-dimensional flow in a duct with area change. Considering an infinitesimal element of length dr , width $2\pi r$ and height $\delta(r)$, as shown in Figure 1, it is possible to obtain the equations that describe the variations of flow properties due to friction and area change through a duct:

$$\frac{dp}{p} = \frac{\gamma M^2}{1-M^2} \left[\frac{dA}{A} - \frac{2C_f [1+M^2(\gamma-1)] dr}{D_h} \right] \quad (1)$$

$$\frac{d\rho}{\rho} = \frac{M^2}{1-M^2} \left[\frac{dA}{A} - \frac{2\gamma C_f dr}{D_h} \right] \quad (2)$$

$$\frac{dM}{M} = \frac{1 + \frac{(\gamma-1)}{2} M^2}{1-M^2} \left[\frac{dA}{A} - \frac{2C_f \gamma M^2 dr}{D_h} \right] \quad (3)$$

where p , ρ and M represent pressure, density and Mach number, respectively; γ is the ratio of specific heats; C_f is the wall friction coefficient and D_h is the hydraulic diameter.

Figure 1: Infinitesimal control volume.

According to Equations (1)-(3), flow properties vary due to the superposition of two effects: area change and viscous friction, represented by the first and second terms, respectively, inside the brackets on the right hand side of Equations (1)-(3). For subsonic flow ($M < 1$), viscous friction causes a reduction of pressure and density and an increase of Mach number. The same effects occur when the cross section area is reduced.

The deflection of the valve, $w(r)$, was estimated via a model for supported circular plate under uniform load, represented in Figure 2. Therefore, the local gap between the valve and seat, $\delta(r)$, is a function of both the valve deflection $w(r)$ and the reference clearance at the edge of the valve port, δ_e .

The solution of the deflection for the interval $r_o \leq r \leq r_d$ is given by:

$$w(r) = \frac{pr_o^3 (r - r_o)}{8D(1+\nu)} \quad (4)$$

where p represents the pressure difference acting on the valve, r_o is the radius of the orifice, r_d is the radius of the disc, ν is the Poisson's ratio and D is the flexural rigidity, which was expressed as being equivalent to that for beams:

$$D = \frac{Et^3}{12(1-\nu^2)} \quad (5)$$

with E being the modulus of elasticity and t the valve thickness.

Figure 2: Schematic of the valve deflection model.

When valve deflection is taken into account, the flow cross section area can be calculated as

$$A(r) = 2\pi r [w(r) + \delta_e] \quad (6)$$

In the case of non-deflected valve $w(r) = 0$. The following dimensionless coordinate transformation was adopted for convenience

$$r^* = \frac{r_d - r}{r_d - r_o} \quad (7)$$

If the flow properties are known, the mass flow rate can be calculated as follows:

$$\dot{m} = \rho(r^*) V(r^*) A(r^*) \quad (8)$$

2.2 Modeling the compression cycle

To evaluate the influence of valve gas leakage on the compressor efficiency, the leakage model was implemented into a simulation code originally developed by Ussyk (1984). The code accounts for the piston displacement as a function of time, thermodynamic process inside the cylinder, fluid flow through the valves, valve dynamics, gas pulsation inside mufflers, motor momentum-power-efficiency relationship, bearing simulation, thermal simulation and refrigerant thermodynamic and thermophysical properties. Several parameters are calculated along the compressor cycle, such as instantaneous pressure throughout the compressor, mass flow rate, valve dynamics, energy and mass losses, refrigerating capacity, energy consumption, etc.

The volumetric efficiency η_v , is defined as the ratio between the actual mass flow rate $\dot{m}$ and the ideal mass flow rate $\dot{m}_{th}$, as shown in Equation (9). The ideal mass flow rate would be obtained in the absence of cylinder clearance volume, gas leakages, flow restriction and backflow in valves, and gas superheating in the suction process.

$$\eta_v = \frac{\dot{m}}{\dot{m}_{th}} \quad (9)$$

The isentropic efficiency η_s is defined as the ratio between the specific compression work for an isentropic process w_s and the actual compression work w , necessary to compress the same amount of the refrigerant:

$$\eta_s = \frac{w_s}{w} \quad (10)$$

The model described in the previous section to quantify gas leakage through valves was implemented into the compressor simulation code as additional sources of volumetric and isentropic inefficiencies. Thus, by applying the principle of conservation of mass to the control volume shown in Figure 3, the rate of change of mass inside the cylinder can be obtained:

$$\left(\frac{dm}{dt} \right)_{CV} = (\dot{m}_s - \dot{m}_d - \dot{m}_{bs} + \dot{m}_{bd} - \dot{m}_{lpc} - \dot{m}_{ls} + \dot{m}_{ld}) \quad (11)$$

In Equation (11), $\dot{m}_s$ and $\dot{m}_d$ are the mass flow rates in the suction and discharge valves, respectively, whereas $\dot{m}_{bs}$ and $\dot{m}_{bd}$ represent eventual backflow in the same valves. The leakage of gas in the clearance between the piston and cylinder is represented by $\dot{m}_{lpc}$. Finally, the terms $\dot{m}_{ls}$ and $\dot{m}_{ld}$ represent gas leakages in the suction and discharge valves, respectively, which are evaluated following Equations (1) - (8).

The balance of energy for the cylinder associated with the mass flow rates indicated in Equation (11) can be written in the following manner:

$$\sum m h)_{CS} = (-\dot{m}_s h_{sc} + \dot{m}_d h_{dc} + \dot{m}_{bs} h_{cyl} - \dot{m}_{bd} h_{dc} + \dot{m}_{lpc} h_{cyl} + \dot{m}_{ls} h_{cyl} - \dot{m}_{ld} h_{dc}) \quad (12)$$

where the h_{sc} , h_{dc} and h_{cyl} are the enthalpy of the gas in the suction chamber, discharge chamber and cylinder, respectively.

Figure 3: Control volume for balances of mass and energy in the cylinder.

3. SOLUTION PROCEDURE

Manipulating the term dA/A , it is possible to rewrite in Equations (1)-(3) as first order ordinary differential equations, which can be solved with the fourth-order Runge-Kutta method. The required boundary conditions are the upstream stagnation pressure p_o and temperature T_o , and downstream outlet pressure p_{out} . Naturally, the proposed model is valid for suction and discharge valves. For the discharge valve, stagnation properties are evaluated in the discharge chamber and the gas flows through the valve clearance towards the valve port. On the other hand, the stagnation condition for the flow through the suction valve is based on the properties inside the cylinder.

The first step in the solution procedure is to estimate the Mach number at the entrance of the valve clearance. Following the hypothesis of isentropic flow, the stagnation properties remain constant up to the inlet region and the flow properties can be evaluated there based on the Mach number. This initial value for Mach number allows the evaluation of other flow properties along the valve clearance via the Runge-Kutta method. The outlet pressure obtained from the numerical solution is compared with the actual value within a given tolerance. If the condition is satisfied, the iterative method is considered to be converged; otherwise, another estimate for the Mach number at the inlet is adopted and the process is repeated until convergence is achieved. If the Mach number along the duct achieves the value 1 within a specified tolerance, then the choked flow condition is present in the nozzle. Equations (1) - (8) are solved for each crank angle along the compression cycle.

4. RESULTS AND DISCUSSION

The present section show results for gas leakage in the valves and the effect of valve deflection on the phenomenon. Moreover, the effect of leakage on volumetric and isentropic efficiencies is also analyzed for different values of valve clearance, δ_e . It should be mentioned that the crank angle $\omega t = 0^\circ$ represents the bottom dead center and, hence, the top dead center occurs at $\omega t = 180^\circ$.

4.1 Flow regime

A gas is assumed to behave as a continuum if the molecular mean free path, λ , is small compared to the flow characteristic length, L . Rarefaction effects become more important as the Knudsen number, $Kn (= \lambda/L)$ increases. With reference to the Knudsen number, a classification of different flow regimes can be given as follows: continuum flow ($Kn \leq 10^{-2}$); slip flow ($10^{-2} < Kn \leq 10^{-1}$); transitional flow ($10^{-1} < Kn < 10$); free-molecular flow ($Kn \geq 10$). The mean free path is affected by pressure and temperature in the valve clearance, which vary with the crank angle ωt . Results for Knudsen number in the discharge and suction valves along the compression cycle (Figure 4) show that slip flow is expected for clearances $\delta_e = 1 \mu\text{m}$. In spite of that, all simulations assumed a continuum flow regime.

Figure 5 shows the outlet Mach number as a function of the crank angle for both valves, but neglecting deflection. It is observed that the choked flow condition ($M = 1$) is not reached in the discharge valve for the clearance $\delta_e = 1 \mu\text{m}$, and the same happening for the suction valve. Results for Reynolds number in the outlet region (Figure 6) indicate that the maximum values of Reynolds number are approximately 1150 and 1400 for the discharge valve and suction valve, respectively, for the largest clearance. Therefore, the adopted hypothesis of laminar flow is adequate.

Figure 4: Knudsen number at the outlet for non-deflected valves: (a) discharge valve (b) suction valve.

Figure 5: Mach number at the outlet for non-deflected valves: (a) discharge valve (b) suction valve.

Figure 6: Reynolds number at the outlet for non-deflected valves: (a) discharge valve (b) suction valve

4.2 Valve gas leakage

Estimates of gas leakage in the suction and discharge valves are shown in Figure 7 as a function of crank angle ωt , assuming no valve deflection. Naturally, there is no meaning in establishing gas leakage when the valves are open and absence of leakage is indicated for such periods. As expected, the higher is the clearance the smaller is the flow restriction and, consequently, the higher is the leakage. It is also evident that the maximum leakage occurs at the bottom dead center for the discharge valve and at the top dead center for suction valve, because the maximum pressure difference is reached in such crank angle positions for each valve.

The influence of the clearance size on the compressor volumetric and isentropic efficiencies is depicted in Figure 8, again without valve deflection. The results show that the volumetric and isentropic efficiencies are reduced by 4.1% and 4.9%, respectively, for a clearance of 3 μm . The results also suggest that leakage in the discharge valve is more critical for the compressor performance.

Figure 9 was prepared to analyze the gas leakage in the discharge valve when deflection due to pressure load is taken into account. Regardless the clearance size ($\delta_e = 1\mu\text{m}$ and $3\mu\text{m}$), gas leakage is strongly increased as an outcome of the smaller flow restriction brought about by deflection. As a result, when deflection is considered for the discharge valve, the volumetric and isentropic efficiencies are seen to be reduced by 6.8% and 8.3%, respectively (Figure 10), for the case of $\delta_e = 3\mu\text{m}$. Hence, valve deflection must be considered in the analysis.

Figure 7: Leakage for non-deflected valves: (a) $\delta_e = 1 \mu\text{m}$; (b) $\delta_e = 3 \mu\text{m}$.

Figure 8: Influence of leakage on (a) volumetric efficiency and (b) isentropic efficiency, without valve deflection.

Figure 9: Leakage through the discharge valve with and without deflection: (a) $\delta_e = 1 \mu\text{m}$; (b) $\delta_e = 3 \mu\text{m}$.

Figure 10: Influence of valve deflection on (a) volumetric efficiency and (b) isentropic efficiency.

5. CONCLUSIONS

The present paper reported a model developed to estimate gas leakage through compressor valves, with and without its deflection into the valve port due to pressure load. It has been observed that leakage can significantly reduce the efficiency of compressors even for very small valve clearances. In fact, the effect of gas leakage on the compressor efficiency shown in this paper is significant enough to justify a more detailed analysis of the phenomenon with the inclusion of other aspects, such as slip flow regime, surface finishing and valve misalignment.

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