2006

A Simplified Numerical Methodology for the In-Cylinder Flow at the Top-Center Crank Position

F. A. Ribas
Embraco

Carlos J. Deschamps
Federal University of Santa Catarina

A SIMPLIFIED NUMERICAL METHODOLOGY FOR THE IN-CYLINDER FLOW AT THE TOP-CENTER CRANK POSITION

F.A. RIBAS JR[1], C.J. DESCHAMPS[2]

[1]EMBRACO S.A., Product and Process Technology Management,
Joinville, SC, Brazil
Fernando_A_Ribas@embraco.com.br

[2]Federal University of Santa Catarina, Department of Mechanical Engineering,
Florianopolis, SC, Brazil
deschamps@polo.ufsc.br

ABSTRACT

A one-dimensional numerical procedure was developed to predict the compressible in-cylinder flow at the top center (TC) crank position during the opening of the discharge valve. The differential governing equations are discretized via the finite volume methodology in a computational grid that expands and contracts according to the piston motion. The ideal gas hypothesis is adopted and the coupling between pressure and velocity fields is achieved using the SIMPLEC algorithm. The procedure is combined with an integral formulation, which is activated when the piston is far from the TC. The new methodology is seen to reproduce some physical features indicated by experimental data that cannot be captured by integral approaches, such as the pressure drop along the cylinder head originating from acceleration and friction effects.

1. INTRODUCTION

When the piston approaches the top center (TC) crank position and the discharge valve opens, a flow is originated in the cylinder towards the valve, as illustrated in Fig. 1, associated with the development of pressure waves in the clearance between the piston and cylinder head. If the cylinder diameter is large and/or the clearance between the piston and cylinder head is narrow, there will be considerable velocity and pressure variations due to viscous friction forces and gas inertia.

The gas inertia implies that the full mass flow condition through the valve will take some time to be established. Similarly, flow deceleration during the closing of the valve may result in mass flow rates greater than predicted by steady state theory. This phenomenon is expected to be more pronounced for large-bore and high-speed compressors.

Machu (1998) offered an interesting discussion on the effect of waves and unsteady flow inside the cylinder with some estimates using the method of characteristics. His study indicated that the gas inertia implies that the cylinder pressure is different to the pressure upstream of the discharge valve, such that its dynamics does not necessarily follow that of the cylinder pressure.

Aigner *et al.* (2005) presented an analysis of a double acting, 2-cylinder, air-reciprocating compressor, based on two- and three-dimensional flow simulations performed using a commercial code. The authors concluded that the one-dimensional model developed in their work was sufficient to predict accurately not only the gas flow and pressure distribution inside the cylinder but also the impact velocity of the valve plate and valve losses.

Rovaris and Deschamps (2005) developed a methodology to simulate reciprocating compressors, with particular attention to the discharge valve dynamics. In order to reduce the computational time, integral and differential formulations were combined. The compressible turbulent flow through the discharge valve was predicted via large-eddy simulation (LES). The model was seen to capture several important flow features found in compressors, such as pressure overshooting in the cylinder, recirculating flow regions and backflow through the discharge valve.

Matos *et al.* (2006) presented a differential methodology to simulate the whole operating cycle of a small refrigeration reciprocating compressor, including the compressible turbulent flow inside the cylinder. Additionally, the time dependent flow field through the discharge valve and its dynamics were coupled and solved simultaneously. Results for the cylinder pressure field showed the presence of a steep pressure drop in the cylinder clearance when the piston approaches the top dead center and the discharge valve is open.

Most compressor simulation methodologies are based on integral formulations, usually evaluating the compression process through the use of polytropic exponents or the first law of thermodynamics. Although the flow through the discharge valve is taken into account in the conservation equations, the flow inside the cylinder is not solved and the cylinder pressure is assumed to be homogeneous throughout the compression cycle.

Figure 2 shows a comparison between results for the indicator diagram obtained with an integral simulation model and a pressure transducer mounted on the cylinder wall. Pressure and volume values were normalized by the suction pressure line condition and the piston swept volume, respectively. There was a good agreement between the results for most of the compression cycle, indicating that the integral model is physically consistent. However, on examining the enlarged view of the pressure variation during the discharge process (Fig. 2b), the departure of the numerical predictions from the experimental data becomes clear. Such a discrepancy may be explained by the fact that the integral model is unable to account for any pressure variation along the clearance as registered by the transducer.

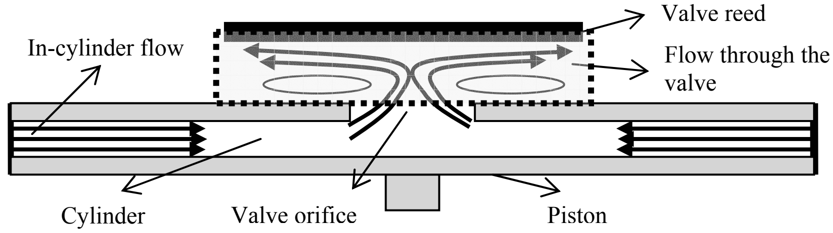


Figure 1: Schematic diagram of the flow geometry.

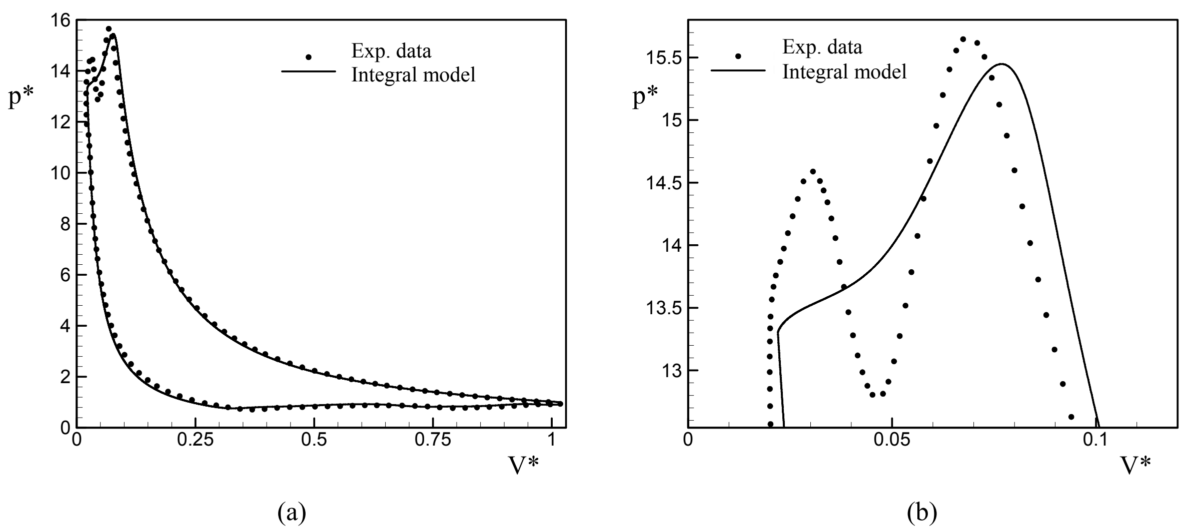


Figure 2: Numerical and experimental results for the compressor indicator diagram.

This study considers the development of a simplified methodology to solve the flow inside the cylinder as the piston approaches the TC crank position, by combining differential and integral formulations for the governing equations. Differential equations are solved to capture flow details along the clearance between the piston surface and the cylinder head, during the opening of the discharge valve, whereas for the remainder of the compression cycle, including the opening of the suction valve, an integral formulation is adopted. In some respect, the present methodology can be regarded as a compromise between integral methods and the model developed by Matos *et al.* (2006).

2. MATHEMATICAL MODEL

2.1 Integral formulation

The integral methodology adopted to simulate the compressor is a simplified version of the model proposed by Ussyk (1984). The code accounts for piston displacement as a function of crankshaft angle, the thermodynamic process inside the cylinder, mass flow rate through the valves, valve dynamics, gas pulsation inside the mufflers and refrigerant thermodynamic properties. Several parameters are calculated during the compressor cycle, such as instantaneous pressure throughout the compressor, mass flow rate, valve dynamics, energy and mass losses, refrigerating capacity, etc.

The transient equations associated with the compressor simulation code are solved via a fourth-order Runge-Kutta method. Thermodynamic properties can be evaluated using the perfect gas hypothesis or through a program link to the REFPROP database (Gallanger *et al*., 1993). Valve displacement is modeled by a one-degree of freedom mass-spring model, in which the natural frequency and damping coefficient have to be specified for the valves. Valve stiction can be obtained from the analytical model proposed by Khalifa and Liu (1998).

The resultant force acting on the reed and the mass flow rate through the valve are obtained with reference to effective force area A_{ef} and effective flow area A_{ee}, respectively. From the pressure difference across the valve, Δp_v, A_{ef} is determined from $A_{ef} = F/\Delta p_v$. The effective force area can be understood as a parameter related to how efficiently the pressure difference Δp_v opens the valve. On the other hand, for the same pressure drop, A_{ee} expresses the ratio between the actual mass flow rate through the valve and that given by an isentropic flow condition. If specified, gas pulsation in mufflers can be modeled following a methodology developed by Deschamps *et al*. (2002).

The thermodynamic process for the gas inside the cylinder can be evaluated either through a polytropic model or by the first law of thermodynamics. The internal temperatures of the compressor have to be supplied as inputs for the simulation program. This is accomplished by an interface with a second simulation code, which evaluates the temperature in eight control volumes through energy balances and using some of the compressor simulation program outputs. The control volumes considered are: gas in the suction muffler, cylinder walls, gas in the discharge muffler, discharge gas, internal environment, compressor housing, electric motor and bearings. Steady state condition is assumed for all temperatures with the exception of the in-cylinder gas. The control volume balance equations are simultaneously and iteratively solved since they depend on all compressor energy fluxes. More details on the compressor simulation program can be obtained in Fagotti *et al*. (1994).

2.2 Differential formulation

The conservation equations for the one-dimensional compressible flow through a typical control volume shown in Fig. 2 are those related to mass, momentum and energy:

$$\frac{\partial m}{\partial t} + \dot{m}_o - \dot{m}_i = 0 \tag{1}$$

$$\frac{\partial\, mv}{\partial t} + \left[\dot{m}v\right]_o - \left[\dot{m}v\right]_i = (p_i - p_o)A_s - \tau_w A_\ell \tag{2}$$

$$\frac{\partial\, m(h + v^2/2)}{\partial t} + \left[\dot{m}\,(h + v^2/2)\right]_o - \left[\dot{m}\,(h + v^2/2)\right]_i - \forall \frac{\partial p}{\partial t} = \dot{Q} \tag{3}$$

where m, v, p and h are the mass, velocity, pressure and enthalpy, respectively. The sub-indices “o” and “i” denote quantities at the outlet and inlet sections of each control volume. All other variables appearing in the above equations are listed in the nomenclature section.

The effect of the wall on the flow was taken into account through estimates of friction force, implied by the no-slip and impermeable wall boundary condition. Shear stress τ_w was evaluated from standard correlations for laminar and turbulent flow regimes. On the other hand, the wall heat transfer $\dot{Q}$ was estimated through the correlation proposed by Lawton (1987). A state equation for the gas, $p = p(\rho, T)$, completes the system of equations.

The aforementioned equations were applied to solve the flow along the clearance between the piston and the cylinder head, but for the discharge orifice an integral conservation formulation had to be used instead. Pressure drop at the geometric singularity between the clearance and the valve orifice was estimated with reference to area ratio (Potter and Wiggert, 1991).

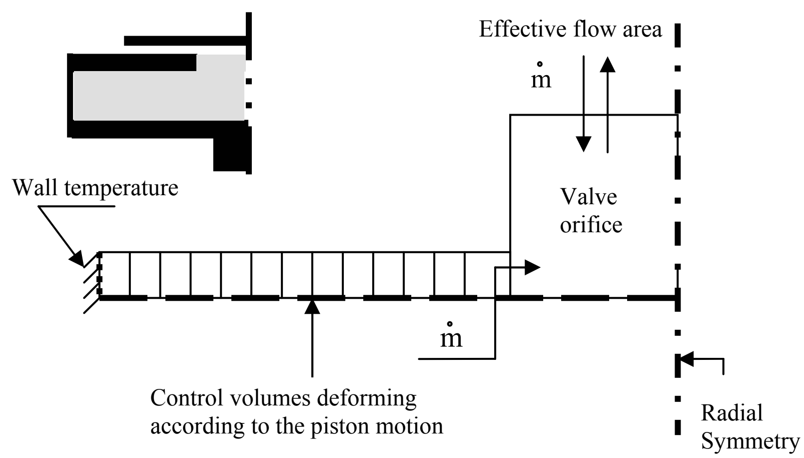


Figure 3: Solution domain and boundary conditions for the differential model.

3. SOLUTION PROCEDURE

The discretization of the governing equations was obtained via the finite volume methodology and using a staggered grid scheme. The convection at the control volume faces was approximated with the Power Law Differencing Scheme (PLDS). A fully implicit time discretization scheme was applied to unsteady terms in the equations.

The system of algebraic equations was solved with the Tridiagonal Matrix Algorithm (TDMA). The coupling between pressure and velocity was handled through the SIMPLEC algorithm extended to flows of arbitrary Mach number. A very important aspect addressed was the validation of the numerical solution by means of sensitivity tests with respect to grid refinement and time step. The grid level used in all simulations had approximately 80 equally distributed nodes.

The compressor simulation starts with the piston at the bottom center (BC) crank position with the integral formulation, as represented in the illustrative diagram shown in Fig. 4. A cylinder pressure value of 10 bar was specified for changing from one formulation to the other, after undertaking a sensitivity analysis of this choice in relation to the results.

Thus, during the compression stroke the integral formulation is changed to the differential formulation when the pressure reaches a value set at 10 bar. At this point, average values for pressure, p, temperature, T, and density, ρ, are directly transferred and used as initial fields for the differential formulation. Other quantities given are the instantaneous piston position and cylinder volume V, both related to the crankshaft angle, wt, where $w = 2\pi f$ and $f =$ 60 Hz. No information is available on the velocity field and its initial value is set to zero.

After the gas has been pushed out of the cylinder, the discharge valve has been closed and the compressor is in the expansion stroke, the procedure is changed back to the integral formulation when the pressure again reaches 10 bar.

Because flow quantities are available for each control volume of the domain, average values can easily be obtained for pressure, temperature and density.

The iterative procedure evaluates properties for each time step, here corresponding to 10^{-3} rad. The convergence of the procedure is assessed by examining whether the compressor operation conditions are cyclically repeated. By using such time steps, 4 cycles were required to establish a periodic condition.

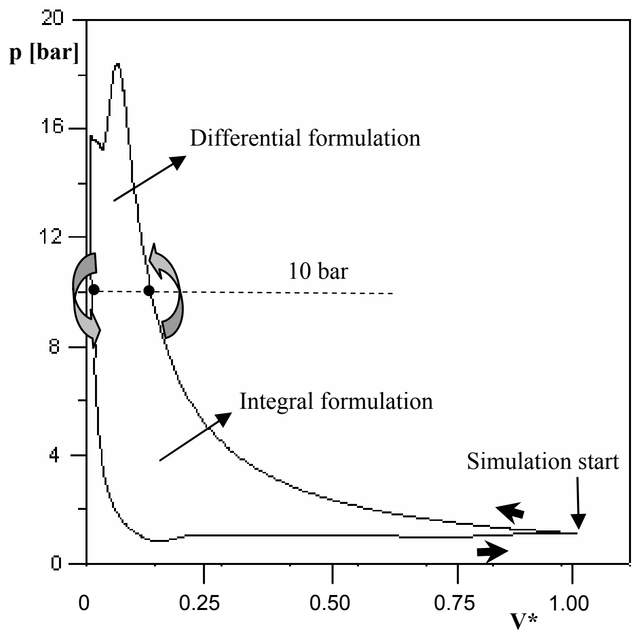


Figure 4: Schematic diagram of the interaction between integral and differential formulations.

4. RESULTS

Figure 5 shows how the velocity and pressure vary along the cylinder clearance as the piston approaches the top center (TC) crank position. It can be noted that for a clearance value c/D = 0.048 the model predicts the presence of gas flow towards the valve orifice, even with the valve still closed. The pressure drop is associated with the viscous friction and the flow acceleration as the gas moves to the valve orifice. As would be expected, when the flow enters the orifice the velocity decreases and the pressure shows a partial recovery.

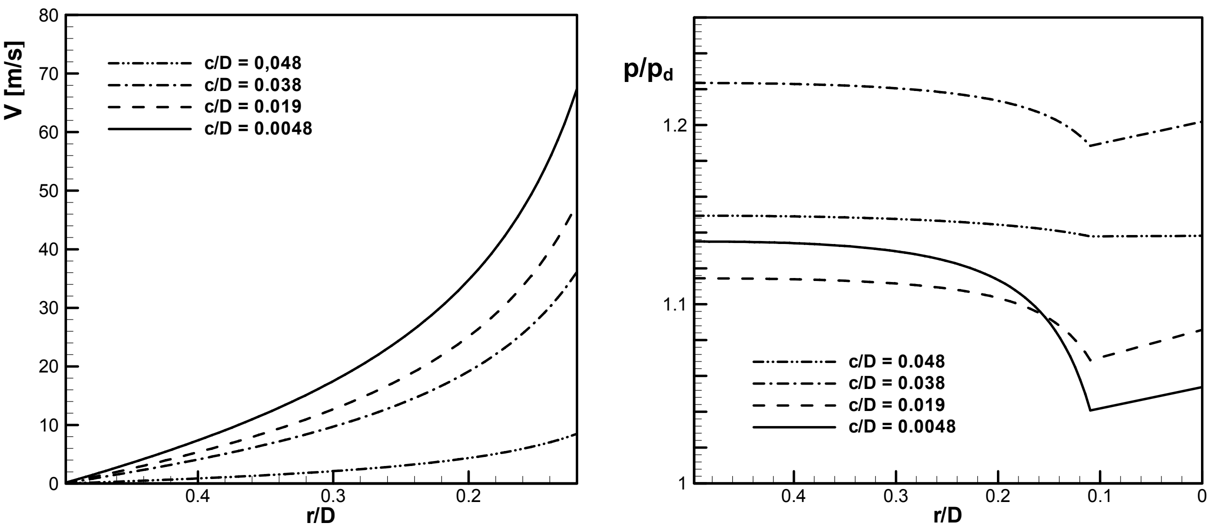
At c/D = 0.038, velocity levels are much higher because the valve is already open, as becomes evident from Fig 5a and the pressure drop in Fig. 5b. The increase in pressure inside the cylinder occurs because the piston is closer to the TC position. However, as the valve continues to open, velocities increase and the cylinder pressure level eventually decreases (c/D = 0.019).

An interesting phenomenon occurs when the piston reaches c/D = 0.0048. There, one can see a pressure rise in the clearance region, which is due to the great flow restriction caused by the proximity of the piston and cylinder head, making it difficult for the gas to leave the clearance area. The viscous friction is further increased by the compressibility effects since, as the density decreases along the clearance region, the velocity levels become even higher.

Figure 6 was prepared for a comparative analysis between the present methodology and the integral approach. As can be seen, the present methodology gives a compressor indicator diagram which is in closer agreement with the experimental data, especially with respect to the presence of two peaks in the overpressure region. The first peak is a consequence of the valve dynamics since the cylinder pressure keeps rising until the valve lift is great enough to discharge the gas. The second peak, also observed by Matos *et al.* (2006), is directly linked to the flow restriction in the clearance area as the piston approaches the TC position, as explained in Fig. 5.

It is interesting to note that, despite the contrast shown in Fig. 6, if average values are used in the comparison the difference between the two methodologies is less evident, as illustrated in Fig. 7 where average values for pressure and temperature are presented. This is so because the mass inside the clearance area is much smaller than that in the orifice and, therefore, will have less impact on the results for a weighted mass average quantity. Naturally, this is an aspect of the analysis that may vary according to each compressor design.

Despite the interesting results offered by the present methodology, the necessity for data on force and flow effective areas to characterize the flow through the discharge valve is considered to represent a major drawback. An alternative, though much more computationally expensive, is to adopt the model developed by Matos *et al.* (2006), solving the flow along the complete compression cycle and through the compressor valves.



(a) (b)

Figure 5: Variation in pressure and velocity along the clearance region.

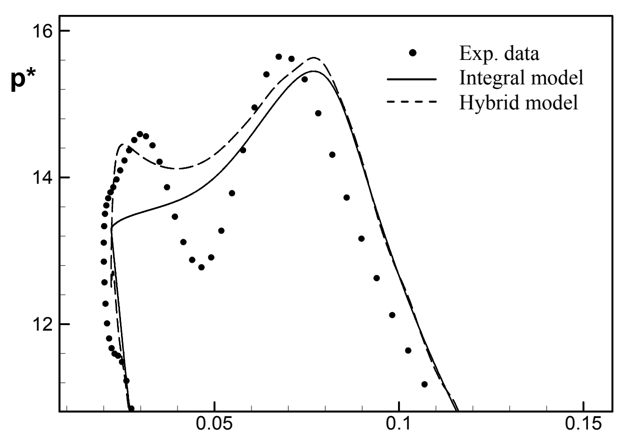
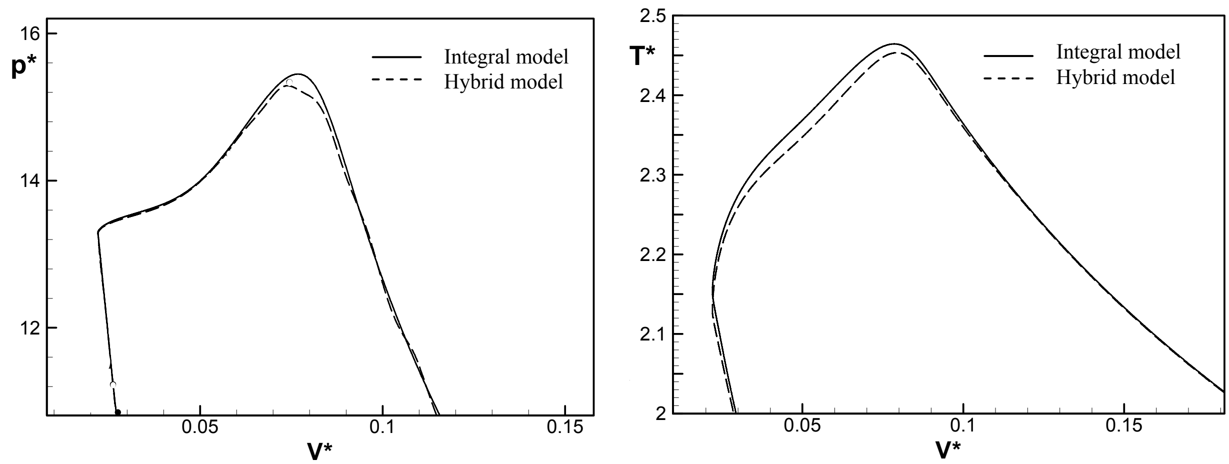


Figure 6: Comparison between the integral and hybrid approaches; cylinder wall pressure.



(a) (b)

Figure 7: Comparison between integral and hybrid approaches; average quantities.

6. CONCLUSIONS

This paper presented a simplified methodology to simulate reciprocating compressors, in which integral and differential formulations were combined. The compressible flow between the piston and cylinder head during the discharge process was predicted via a one-dimensional model, so that variations in flow quantities along the clearance region could be predicted. Despite the model simplicity, results are seen to be physically consistent, describing several important flow features found in compressors, such as pressure overshooting at the cylinder wall due to flow restriction in the cylinder clearance, as observed in the more elaborated methodology of Matos *et al.* (2006).

NOMENCLATURE

A_s	flow cross section area	p_d	discharge pressure
A_ℓ	flow lateral area	$\dot{Q}$	heat transfer
c	cylinder clearance	r	local radial position
D	cylinder diameter	T*	temperature normalized by the suction condition
h	enthalpy	v	mean radial velocity in the cylinder clearance
$\dot{m}$	mass flow rate	$\forall$	volume
m	mass	V*	volume normalized by the piston swept volume
p	pressure	ρ	density
p*	pressure normalized by the suction condition	τ_w	wall shear stress

REFERENCES

Aigner, A., Meyer, G., Steinrück, H., 2005, Valve Dynamics and Internal Waves in a Reciprocating Compressor, *Proc. 4th Conference of the EFRC - European Forum for Reciprocating Compressors*, Antwerp, Belgium.

Deschamps, C.J., Possamai, F.C., Pereira, E.L.L, 2002, Numerical Simulation of Pulsating Flow In Suction Mufflers, *Proc. Compressor Engineering Conference at Purdue*, West Lafayette, paper C-11, 8 p.

Fagotti, F., Todescat, M.L., Ferreira, R.T.S., Prata, A.T., 1994, Heat Transfer Modeling in a Reciprocating Compressor, *Proc. Compressor Engineering Conference at Purdue*, West Lafayette: p. 605-610.

Gallanger, J., McLinden, M., Morrison, G. and Huber, M., 1993, NIST Thermodynamic Properties of Refrigerants and Refrigerants Mixtures Database (REFPROP), Thermophysics Division, National Institute of Standards and Technology, US Department of Commerce.

Khalifa, H.E. and Liu, X., 1998, Analysis of Stiction Effect on the Dynamics of Compressor Suction Valve, *Proc. Int. Compressor Engineering Conference at Purdue*, West Lafayette: p. 87-92.

Lawton, B., 1987, Effect of Compression and Expansion on Instantaneous Heat Transfer in Reciprocating Internal Combustion Engines, Proc. Instn Mech. Engrs, Part A, *Journal of Power and Energy*, v. 201, pp. 175-186.

Machu, E.H., 1988, Problems with Modern High Speed Short Stroke Reciprocating Compressors: Increased Pocket Losses Due to Piston Masking and Eccentric Gas Load on the Piston, *Proc. Gas Machinery Conference*, Denver, 20 p.

Matos, F.F.S., Deschamps, C.J., Prata, A.T., 2006, A Two-Dimensional Simulation Model for Reciprocating Compressors with Automatic Valves, *Proc. Int. Compressor Engineering Conference at Purdue*, West Lafayette, paper C-053, 8 p.

Potter, M.C. and Wiggert, D.C., 1991, *Mechanics of Fluids*, Prentice-Hall.

Rovaris, J.B., Deschamps, C.J., 2005, A Hybrid Simulation Methodology for Reciprocating Compressors, *Proc. International Conference on Compressors and Their Systems*, London: p. 183-192.

Ussyk, M. S., 1984, Numerical Simulation of Hermetic Reciprocating Compressor Performance (in Portuguese), *M.Sc. Thesis*, Federal University of Santa Catarina, Brazil.

ACKNOWLEDGEMENT

This study forms part of a joint technical-scientific program of the Federal University of Santa Catarina and EMBRACO. Support from the Brazilian Research Council, CNPq, is also acknowledged.