

NUMERICAL SIMULATION OF SUCTION MUFFLERS WITH THE MCCORMACK METHOD

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Abstract. *The pulsating fluid flow through acoustic mufflers has an important role in the performance of reciprocating compressors, regarding acoustic noise attenuation and volumetric efficiency. Several authors have dedicated attention to the modeling of this phenomenon, proposing different solution alternatives, such as linear acoustic models, method of characteristics, finite difference schemes and finite volume methodologies. In this work, a one dimensional fluid dynamics model based on a finite difference scheme is developed to solve the compressible fluid flow through suction mufflers. The chosen technique to solve the partial differential equations is the two-step version of the McCormack method, which is an explicit predictor-corrector type scheme with second order accuracy in both time and space. The model is employed to simulate the dynamic behavior of a typical muffler suction geometry adopted in refrigeration reciprocating compressors. Numerical results for pressure pulsation in the muffler are presented and compared with predictions returned by a finite volume methodology.*

Keywords*: McCormack Method, suction muffler, reciprocating compressor.*

1. INTRODUCTION

The efficiency of reciprocating machines, such as IC engines and compressors, are affected to a great extent by the suction and discharge systems. Such systems play a crucial role on the performance of compressors, affecting directly the volumetric and energy losses, as well as the compressor noise level. Reed valves, which are typically found in reciprocating compressors, open and close depending on the pressure difference between the cylinder and the suction/discharge chamber. Once such valves are opened, the pressure flow field is responsible for the resulting force acting on the reed. For this reason, it is crucial to predict the pulsating fluid flow through acoustic mufflers correctly because of its influence on the valve dynamics.

Because several parameters affect the flow in acoustic mufflers, a systematic experimental investigation of such a phenomenon is difficult and expensive. An alternative commonly adopted is the theoretical flow analysis by numerically solving the differential governing equations. In this respect, the one-dimensional formulation of the unsteady gas dynamics equations has been used to describe the flow in intake and exhaust systems, allowing a better understanding of the problem and decreasing the requirement of time to develop a new muffler design. The fact that valve dynamics is affected by pressure pulsations in the muffler, and vice-versa, requires both problems to be solved simultaneously, increasing the computational effort. Therefore, several numerical solution methods have been proposed in the literature, such as acoustic models, method of characteristics, finite difference schemes and finite volume methodology, with the aim of offering a good compromise between accuracy and computational cost.

If pulsations are small compared to the mean pressure, its behavior can be approximated by the acoustic theory (Elson and Soedel, 1974). In the case of large pressure amplitudes, the non-linear partial differential equations governing the unsteady one-dimensional compressible flow have to be solved (McLaren et al, 1975). More recently, Liu and Soedel (1994) presented a numerical model to solve a set of non-linear differential equations describing the unsteady one-dimensional compressible flow in an intake system, considering wall friction and heat transfer. Pérez-Segarra et al. (1994) developed an unsteady one-dimensional model for the whole compressor using a finite volume technique and solved the fluid dynamics and heat transfer in a very detailed manner. Ignatiev et al. (1996) presented a model for the suction system and applied it to analyze the effect of geometric parameters on the compressor performance. In the work of Bassi et al. (2000) an unsteady one dimensional model was proposed to solve the flow in mufflers through a discontinuous Galerkin method. Their methodology and that proposed by Pérez-Segarra et al. (1994) have shown results in agreement with experimental data.

In the present work, a one-dimensional fluid dynamics model based on a finite difference scheme is developed to predict the compressible fluid flow through suction mufflers. The chosen technique to solve the partial differential equations is the two-step version of the McCormack method, which is an explicit predictor-corrector type scheme with second order accuracy in both time and space. The model is employed to simulate the dynamic behavior of a typical muffler suction geometry adopted in refrigeration reciprocating compressors. Numerical results for pressure pulsation in the muffler are presented and compared with predictions returned by a finite volume methodology by Pereira *et al.* (2005).

2. MATHEMATICAL MODELING

The suction/discharge systems of a hermetic reciprocating compressor can be modeled as a succession of cylindrical ducts and chambers, as indicated in fig. 1 and fig. 2. Considering the hypothesis of transient, compressible flow and a one-dimensional formulation for the conservation equations for mass, momentum e energy, the following system of equations results:

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u)}{\partial x}=0 \tag{1}$$

$$\frac{\partial(\rho u)}{\partial t}+\frac{\partial(\rho u^2+p)}{\partial x}+\rho G=0 \tag{2}$$

$$\frac{\partial(\rho e)}{\partial t}+\frac{\partial(\rho u h_0)}{\partial x}-\rho \dot{q}=0 \tag{3}$$

where ρ, ρu and ρe are the primitive variables of the equation system, respectively named as specific mass, specific momentum and specific total energy. The operating conditions of the compressor allow one to consider the refrigerant fluid inside the muffler as a perfect gas ($p = \rho R\ T,\ c_p = c_v + R,\ \gamma = c_p/c_v$). Therefore, the static pressure p and the stagnation enthalpy h_o can be expressed, respectively, as $p = (\gamma - 1)(\rho e - \rho u^2/2)$ and $h_o = (\gamma)(\rho e)/\ \rho - (\gamma\text{-}1)(\rho u)^2/2\rho^2$, completing the system of equations required to mathematically model the flow.

Wall effects on the flow are included through an account of viscous friction and heat transfer. The viscous shear stress per length unit is written as $G = fu|u|/2D$, where f is the friction factor, which is based on classical correlations available for steady fully developed flow. Accordingly, f = 16/Re is adopted if the flow is laminar, whereas Blasius correlation is adopted in the case of turbulent flow. The heat flux per area unit is expressed by $\dot{q} = h\ [T_b(x,t)\text{-}T_\infty]$, where h is the heat transfer coefficient obtained from correlations developed for pipe flow. However, all simulations reported in the present work considered thermally insulated walls.

Mufflers are formed by a sequence of ducts and chambers of different geometries. Traditionally, the chambers are modeled as singularities for which a special treatment is required. In this work, such singularities are modeled through an incompressible-transient correlation presented by Pérez-Segarra *et al.* (1994).

Boundary conditions suitable for this problem are: (i) static pressure and temperature conditions at the inlet, related to stagnation conditions in the plenum through an isentropic flow hypothesis; (ii) transient mass flow rate at the muffler exit, characterizing the operating condition of a suction valve.

3. NUMERICAL METHODOLOGY

The governing equations for the flow in the muffler were numerically solved using two different approaches: a finite difference method and a finite volume methodology.

In the finite volume methodology the solution domain is divided into small control volumes (Fig. 1) using a staggered grid arrangement. The governing differential equations are integrated over each control volume with the use of Gauss's theorem. The flow properties at the control volumes faces are interpolated with a first order upwind interpolation scheme. A fully implicit time discretization scheme was applied to discretize the unsteady terms in the governing equations. The system of algebraic equations was solved with the Tridiagonal Matrix Algorithm (TDMA) following a segregated procedure. The coupling between pressure and velocity fields was handled through the SIMPLEC algorithm extended to flows of arbitrary Mach number (Van Doormal et al., 1987).

The finite volume methodology is numerically stable and results for pressure pulsation generally are in good agreement with experimental data (Pereira et al., 2005). However, the computational cost is usually high due to two aspects: (i) adoption of low-order temporal and spatial discretization schemes, which requires a considerable grid refinement and a small time step in order to guarantee sufficient accuracy; (ii) use of implicit schemes for time discretization, requiring an iterative procedure to solve the system of equations. Further details on the finite volume methodology can be found in many references, such as Ferziger and Peric (1996).

In the present work, the McCormack finite-difference scheme was applied to solve flow through a simplified geometry of suction muffler (Payri e Torregrosa, 1996), with the main purpose of comparing its performance with the finite volume methodology, in terms of computational cost and accuracy.

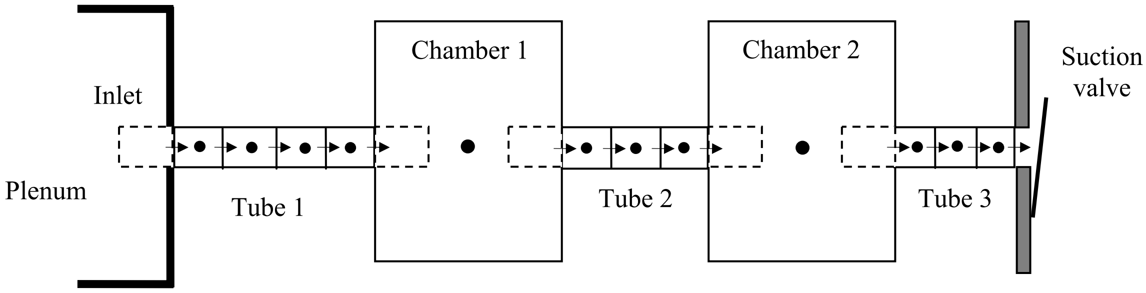


Figure 1. Staggered finite volume grid arrangement adopted in the finite volume methodology.

The McCormack finite-difference scheme is an explicit two-step-predictor-corrector method, with second order accuracy both in space and time. The duct is discretized following a non-staggered arrangement for flow properties, as may be seen in Fig. 2. In this method, the solution in the present time step is obtained from the solution in the previous time step due the introduction of a virtual intermediate time step (predictor step). The properties of the flow at the virtual predictor time step (denoted by a $\overline{n+1}$ super-index) at the position $i\Delta x$ of the duct are obtained using the solution of the flow at the $i\Delta x$ and $(i+1)\ \Delta x$ sections at time n (subsequent time). After that, in the corrector step ($n+1$ time level), the predictor time step solution at the sections $(i-1)\Delta x$ and $i\Delta x$ are employed in order to obtained the final estimate for the flow properties at the $i\Delta x$ section. Therefore, it is convenient to express the system of equations given by Eq. (1), Eq . (2) and Eq. (3) in a compact form, as follows:

$$\frac{\partial \{U\}}{\partial t} + \frac{\partial \{V\}}{\partial x} + \{W\} = 0 \tag{4}$$

where $\{U\}$, $\{V\}$ e $\{W\}$ are represented by:

$$\{\vec{U}\} = \left(\rho; \rho u; \rho e\right) \tag{5}$$

$$\{\vec{V}\} = \left(\rho u; \rho u^2 + p; \rho u h_0\right) \tag{6}$$

$$\{\vec{W}\} = \left(0; \rho G; -\rho \dot{q}\right) \tag{7}$$

Finally, as aforementioned, the predictor and the corrector steps of the McCormack scheme may be written as:

$$\{U_i^{\overline{n+1}}\} = \{U_i^n\} - \left(\{V_{i+1}^n\} - \{V_i^n\}\right)\Delta t / \Delta x - \left(\{W_i^n\} + \{W_{i+1}^n\}\right)\Delta t / 2 \tag{8}$$

$$\{U_i^{n+1}\} = \left(\{U_i^n\} + \{U_i^{\overline{n+1}}\}\right)/2 - \left(\{V_i^{\overline{n+1}}\} - \{V_{i-1}^{\overline{n+1}}\}\right)\Delta t/(2\Delta x) - \left(\{W_i^{\overline{n+1}}\} + \{W_{i-1}^{\overline{n+1}}\}\right)\Delta t / 4 \tag{9}$$

The main advantages of the McCormack finite-difference method over the finite volume methodology presented herein are: (i) smaller computational processing cost, since the method is explicit in time; (ii) greater numerical solution accuracy, because it adopts high order discretization schemes for space and time derivatives; (iii) simplicity of implementation, due to its explicit character.

The numerical modeling applied for chambers is quite different from that applied to ducts. A chamber is considered as an arrangement formed by two main parts: chambers interior and interfaces with the ducts (w and e). The flow in each of these flow regions has a specific dynamics, whose description is given below.

At the chambers interior, the momentum conservation is neglected and, as a consequence, only conservation equations for mass and energy are considered. Therefore, the equations necessary to describe the flow evolution in the chambers interior assume the following form:

$$\rho_{C_j}^{n+1} = \rho_{C_j}^{n} + \left(\Delta t / \Delta\forall_{C_j}\right)\left((\rho u A)_{w_j}^{n+1} - (\rho u A)_{e_j}^{n+1}\right) \tag{10}$$

$$(\rho e)_{C_j}^{n+1} = (\rho e)_{C_j}^{n} + \left(\Delta t / \Delta\forall_{C_j}\right)\left((\rho u h_o A)_{w_j}^{n+1} - (\rho u h_o A)_{e_j}^{n+1}\right) \tag{11}$$

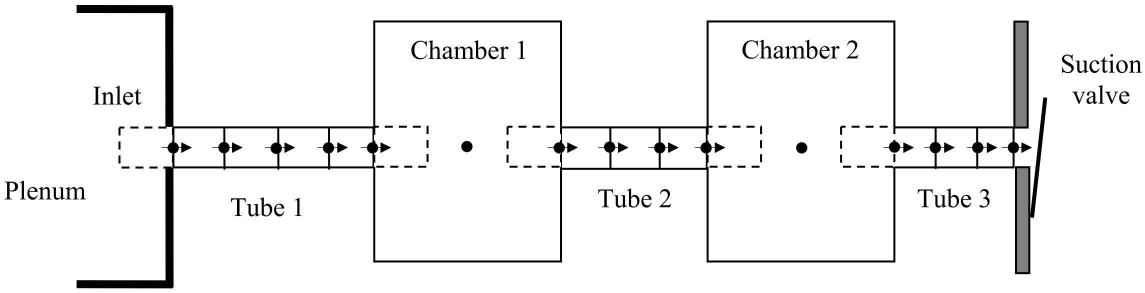


Figure 2. Non-staggered grid arrangement adopted in the finite difference methodology.

where the subindices C_j, w_j, and e_j stand for, respectively, the chamber interior region, the west interface and the east interface of the jth chamber. The total number of chambers is k (j=1, ... , k). The plenum and the suction chamber at the two extremities of the suction muffler are represented by $j = 1$ and $j = k$, respectively. The momentum conservation equation is not solved for the chambers, since the velocity there is virtually zero. However, a momentum balance is carried out at the interfaces between chambers and ducts, according to Eqs. (13), (16), (19) and (22), so as to accommodate the inertia of a small region in the chamber (identified by the dashed line) and viscous friction losses.

At the west and east interfaces of each chamber, the temporal evolution of the specific momentum, mass and energy is evaluated. For the intermediate chambers (j = 2, ..., k-1), the corresponding equations to evaluate the flow properties at the interfaces w_j and e_j are given by:

Interface w_j:

for $(\rho u)_{w_j}^n \geq 0$;

$$\rho_{w_j}^{n+1} = \rho_{w_j}^n - \left(\Delta t / \Delta x_{w_j}\right)\left((\rho u)_{w_j}^n - (\rho u)_{ww_j}^n\right) \tag{12}$$

$$(\rho u)_{w_j}^{n+1} = (\rho u)_{w_j}^n - \left(\Delta t / L_{eff_w}\right)(\rho u)_{w_j}^n \left|(\rho u)_{w_j}^n\right| \Gamma_w^+ / (2\rho_{w_j}^n) + \left(\Delta t / L_{eff_w}\right)\left(p_{w_j}^n - p_{C_j}^n\right) \tag{13}$$

$$(\rho e)_{w_j}^{n+1} = (\rho e)_{w_j}^n - \left(\Delta t / \Delta x_{w_j}\right)\left((\rho u h_o)_{w_j}^n - (\rho u h_o)_{ww_j}^n\right) \tag{14}$$

for $(\rho u)_{w_j}^n \leq 0$;

$$\rho_{w_j}^{n+1} = (1/2)\left\{\gamma(\rho e)_{w_j}^n / (h_o)_{C_j}^n + \left[\left(\gamma(\rho e)_{w_j}^n / (h_o)_{C_j}^n\right)^2 - 2(\gamma - 1)\left[(\rho u)_{w_j}^n\right]^2 / (h_o)_{C_j}^n\right]^{1/2}\right\} \tag{15}$$

$$(\rho u)_{w_j}^{n+1} = (\rho u)_{w_j}^n - \left(\Delta t / L_{eff_w}\right)(\rho u)_{w_j}^n \left|(\rho u)_{w_j}^n\right| \Gamma_w^- / (2\rho_{w_j}^n) + \left(\Delta t / L_{eff_w}\right)\left(p_{w_j}^n - p_{C_j}^n\right) \tag{16}$$

$$(\rho e)_{w_j}^{n+1} = (\rho e)_{w_j}^n - \left(\Delta t / \Delta x_{w_j}\right)\left((\rho u h_o)_{w_j}^n - (\rho u h_o)_{ww_j}^n\right) \tag{17}$$

Interface e_j:

for $(\rho u)_{e_j}^n \geq 0$;

$$\rho_{e_j}^{n+1} = (1/2)\left\{\gamma(\rho e)_{e_j}^n / (h_o)_{C_j}^n + \left[\left(\gamma(\rho e)_{e_j}^n / (h_o)_{C_j}^n\right)^2 - 2(\gamma - 1)\left[(\rho u)_{e_j}^n\right]^2 / (h_o)_{C_j}^n\right]^{1/2}\right\} \tag{18}$$

$$(\rho u)_{e_j}^{n+1} = (\rho u)_{e_j}^n - \left(\Delta t / L_{eff_e}\right)(\rho u)_{e_j}^n \left|(\rho u)_{e_j}^n\right| \Gamma_e^+ / (2\rho_{e_j}^n) + \left(\Delta t / L_{eff_e}\right)\left(p_{C_j}^n - p_{e_j}^n\right) \tag{19}$$

$$(\rho e)_{e_j}^{n+1} = (\rho e)_{e_j}^n - \left(\Delta t / \Delta x_{e_j}\right)\left((\rho u h_o)_{ee_j}^n - (\rho u h_o)_{e_j}^n\right) \tag{20}$$

for $(\rho u)_{e_j}^n \leq 0$;

$$\rho_{e_j}^{n+1} = \rho_{e_j}^{n} - \left(\Delta t / \Delta x_{e_j}\right)\left((\rho u)_{ee_j}^{n} - (\rho u)_{e_j}^{n}\right) \tag{21}$$

$$(\rho u)_{e_j}^{n+1} = (\rho u)_{e_j}^{n} - \left(\Delta t / L_{eff_e}\right)(\rho u)_{e_j}^{n}\left|(\rho u)_{e_j}^{n}\right|\Gamma_e^{-}/(2\rho_{e_j}^{n}) + \left(\Delta t / L_{eff_e}\right)\left(p_{C_j}^{n} - p_{e_j}^{n}\right) \tag{22}$$

$$(\rho e)_{e_j}^{n+1} = (\rho e)_{e_j}^{n} - \left(\Delta t / \Delta x_{e_j}\right)\left((\rho u h_o)_{ee_j}^{n} - (\rho u h_o)_{e_j}^{n}\right) \tag{23}$$

where $\Gamma_{\mathrm{w,e}}^{\pm}$ and $L_{\mathrm{eff_{w,e}}}$ stand for the modeling of net momentum flux and rate of momentum change at the singularities, as proposed by Perez-Segarra *et al.* (1994).

The plenum (j=1) is treated as a reservoir of infinite volume, for which the thermodynamic properties can be assumed to remain constant, representing the properties associated with the compressor operating conditions. Therefore, the plenum thermodynamic properties correspond to the inlet boundary conditions, represented by Eqs. (24) and (25):

$$p_{C_{j=1}}^{n} = p_o \tag{24}$$

$$(h_o)_{C_{j=1}}^{n} = \gamma R T_o / (\gamma - 1) \tag{25}$$

For such a plenum, only the flow dynamics at the interface e_1 is considered to be important. Thus, thermodynamic properties in $e_{j=1}$ are governed by Eqs. (18)-(23). The stagnation enthalpy at the duct inlet is considered to be equal to the plenum stagnation enthalpy (Eq. 25), but the static temperature varies according to the kinetic energy value at each time step. The plenum stagnation enthalpy is used in Eq. (23) for j = 1 and, in addition to that, $(h_o)_{e_{j=1}}^{n} = (h_o)_{C_{j=1}}^{n}$. The plenum pressure is used in Eq. (22), following Eq. (24).

The suction chamber (j = k) is modeled in a similar way as carried out for the intermediate chambers. Flow properties inside the chamber, C_k, is given by Eqs. (10) and (11), whereas at the west interface, w_k, such properties are obtained from Eqs. (12)-(17). The suction valve is represented by the east interface, denoted by e_k., of the suction chamber, represented by C_k The prescribed mass flow rate at the suction valve, $\dot{m}$, is used as boundary condition, with the specific momentum at e_k, $(\rho u)_{e_{j=k}}^{n+1}$, being the ratio between $\dot{m}$ and the suction valve orifice area, A_{orif}. The other flow properties at e_k are obtained by considering the stagnation enthalpy to be constant between the suction chamber interior and the suction valve. Therefore, the thermodynamic properties at the suction valve, represented by the e_k interface, are described by the following equations:

$$(\rho u)_{e_{j=k}}^{n+1} = \dot{m}/A_{orif} \tag{26}$$

$$\rho_{e_{j=k}}^{n+1} = \rho_{C_{j=k}}^{n}\left(1 - \left[(\rho u)_{e_{j=k}}^{n} / \rho_{e_{j=k}}^{n}\right]^2 / \left[2(h_o)_{C_{j=k}}^{n}\right]\right)^{1/(\gamma-1)} \tag{27}$$

$$(\rho e)_{e_{j=k}}^{n+1} = \rho_{e_{j=k}}^{n+1}(h_o)_{C_{j=k}}^{n} / \gamma + (\gamma - 1)\left[(\rho u)_{e_{j=k}}^{n}\right]^2 / (2\gamma\rho_{e_{j=k}}^{n}) \tag{28}$$

While numerical models based on finite-volume methodologies have as a main goal to guarantee the flow properties conservation based on a integral formulations, it is important to mention that finite-difference methodologies seek for a numerical solution in which the conservation of the flow properties is strongly satisfied in a differential manner. Therefore, the application of finite-difference methods usually makes the numerical procedure less stable.

It is possible to state that numerical instability of a given finite difference method is affected by the explicit character of scheme adopted and increase proportionally with the order of the numerical approximations used to express space/time derivatives. Therefore, the higher accuracy and the totally explicit nature of the McCormack scheme penalize the numerical solution methodology, making it even more restrictive from the stability point of view.

At this stage of implementation, there are still important questions related to the boundary conditions treatment, for which a better account needs to be addressed. The current approach is a result of an extensive investigation of alternatives and its configuration was chosen due to adequate stability characteristics.

4. RESULTS AND DISCUSSION

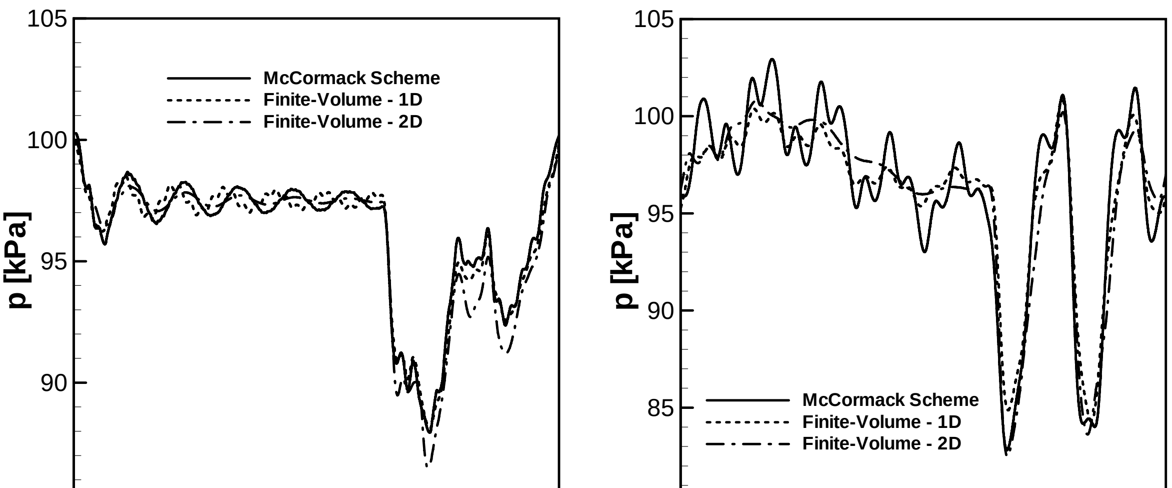
Initially, transient mass flow rate at the suction valve orifice was obtained from a numerical simulation of a hermetic compressor, operating with a frequency of 60 Hz. Such a data was then adopted as the boundary condition for the suction valve orifice and was also employed as boundary condition for other simulations aimed at analyzing pressure pulsation in the muffler at different operating frequencies.

For all situations presented in this work, the numerical simulation was performed for a time period, corresponding to 10 compressor cycles, in order to establish a fully developed cyclic condition for the flow. Therefore, all results to be presented are related to the last compressor cycle. In addition to the results obtained with the McCormack finite-difference methodology, results given by finite volume methodologies based on one-dimensional and two-dimensional formulations are also included. The two-dimensional simulation was carried out by using a commercial code (ANSYS, 2008) with the main purpose of complementing the comparative analysis of both one-dimensional approaches. After choosing the time step needed to satisfy the stability criterion for the McCormack scheme, the same time step was also applied in the simulations performed with the finite-volume methodologies.

Figure 3 shows results for pressure pulsation inside the suction chamber as a function of compressor crank angle position, ωt, considering two operation frequencies (10 Hz and 60 Hz) and the aforementioned numerical solution methodologies. For the lower frequency condition (fig. 3a), the 1D finite-volume methodology (dashed line) returns a result for pressure pulsation that is out phase in comparison in relation to predictions of the other two methodologies when the valve is closed. Despite some differences regarding amplitude, pressure pulsation phases predicted by the McCormack methodology (solid line) and the 2D finite-volume methodology (dot-dashed line) are in close agreement. When the valve is open ($1.25\pi \leq \omega t < 2\pi$), all methodologies predict the same phase for pressure pulsation. This can be justified by the fact that pressure level in the suction chamber is directly related to the mass flow rate prescribed at the suction valve. Concerning the pressure pulsation amplitude, both 1D methodologies give very similar results, although with some disagreement in relation to the 2D finite volume methodology (dot-dashed line). Significant discrepancies can be noticed between the results, especially for the 60 Hz condition (fig. 3b) when the suction valve is closed ($0 \leq \omega t < 1.25\pi$). In this case, the McCormack scheme predicts the highest amplitudes during the period in which the valve is closed ($0 \leq \omega t \leq 1.25\pi$), with both finite volume methodologies predicting pressure pulsations similar to each other. Again, when the valve is open, the phase of pressure pulsation returned by all methodologies present a satisfactory agreement.

From the acoustics point of view, a muffler is designed to dampen pressure pulsation at the inlet originated by the transient flow that takes place in the compressor valve and, as a consequence, to reduce the intensity of pressure waves transmitted to the plenum. Figure 4 shows pressure pulsations in the suction chamber and at the muffler inlet, predicted with the McCormack methodology, as a function of the compressor cycle position. Here, the pressure pulsation in the suction chamber is considered as the excitation source and the pressure at the inlet is the system response. As can be seen in Fig. 4, levels of pressure pulsation at the inlet, represented by the dashed line, are quite low in comparison with the pressure pulsation in the suction chamber, giving a measure of how effective the muffler is in this condition.

The pressure attenuation provided by the muffler can also be analyzed with the assistance of Fig. 5. In that figure, results for pressure distribution along the suction muffler length are shown for different crank angle positions when the valve is either closed (Fig. 5a) or closed (Fig. 5b). Considering that the muffler inlet is at x = 0 mm and the suction muffler is located at x = 320 mm, the muffler efficiency can be assessed with reference to the ratio between pressure amplitudes at the inlet and in the suction chamber.



(a) 10 Hz (b) 60 Hz

Figure 3. Results for pressure pulsation in the suction chamber.

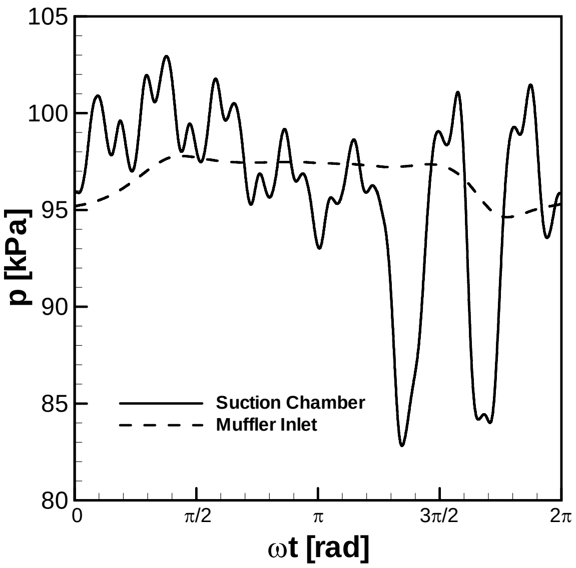
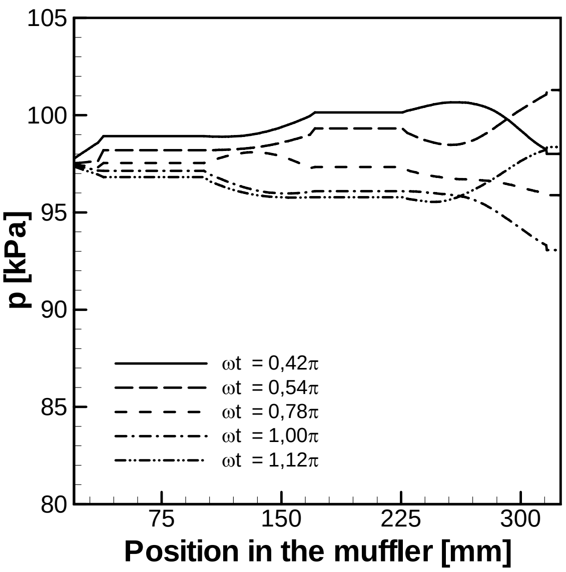


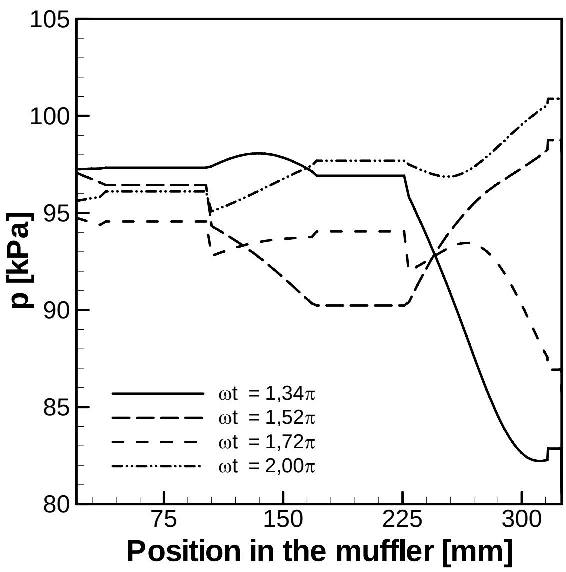
Figure 4. Pressure attenuation in the muffler.

In addition to their acoustic role, mufflers are also designed to provide good volumetric efficiency. In this regard, it is highly recommended that the pressure inside the suction chamber reaches its maximal value near the opening of the suction valve, increasing the pressure difference between the chamber and the cylinder. This allows a high mass flow rate through the valve and, as a consequence, a greater volumetric efficiency. Both simulation methodologies here described can be adopted to find geometric dimensions for ducts and chambers that return an optimal performance for the suction muffler.

The computational processing cost associated with the McCormack scheme is approximately six times less than that of the one-dimensional finite volume methodology. It should be mentioned that although the cost of a one-dimensional simulation is not excessive, this is a very important issue when several geometric alternatives have to be tested for a design optimization purpose. Numerical instability was found in simulations carried out with McCormack scheme, which could not be totally removed in some flow situations. For this reason, the McCormack scheme developed in the present work is still behind the finite volume approach as a simulation methodology for the compressor design. The methodologies adopted here are not capable to predict three-dimensional effects, such as secondary flow due to duct curvature and turbulence generated at the interfaces between chambers and ducts.



(a) Valve closed.



(b) Valve open.

Figure 5. Pressure distribution along the muffler length.

5. CONCLUSIONS

The present work considered the application of the two-step McCormack finite difference method to predict the transient, compressible flow in suction mufflers, under typical operating conditions of reciprocating compressors. Additionally, a finite volume model was also applied to allow a comparative analysis between different simulation approaches. Overall, although all results showed the same fundamental characteristics for pressure pulsation in the muffler, some discrepancies were verified between the methodologies. The computational cost associated with the McCormack scheme was seen to be six times lower than that required for the finite volume model. On the other hand, numerical instability was found in some flow situations with the McCormack scheme. Although the finite difference simulation procedure presented herein is attractive for optimization purpose, it is still requires further development

6. ACKNOWLEDGEMENTS

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