CFD SIMULATION AS A TOOL FOR THE IDENTIFICATION OF THE MANIFOLD ELEMENT REACTION TO PRESSURE PULSATIONS EXCITATION (ONE- AND TWO-PHASE FLOW)

PIOTR CYKLIS AND RYSZARD KANTOR

Faculty of Mechanical Engineering M-5,
Cracow University of Technology,
Jana Pawła II 37, 31-864 Cracow, Poland
pcyklis@mech.pk.edu.pl, pmkantor@cyf-kr.edu.pl

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Abstract: The periodicity of compressor operation is a source of pressure pulsations in volumetric compressor manifolds. An analysis of pressure pulsations is important for several reasons.

The Helmholtz model, applied in all commercial programs offered by the companies professionally dealing with damping pressure pulsations, contains numerous simplifying assumptions; a straight pipe segment, with an ideal gas isentropic flow assumption, substitutes each element of the piping system. In many cases this model is insufficient. The existing experimental methods could not be used in the design of a muffler. The aim of this paper is to show a new method to identify an arbitrary fragment of a manifold, *i.e.* a method of identification of the appropriate complex transmittance matrix elements using CFD simulation. This method allows the liquid phase dispersed in the compressed gas and non ideal gas as a working medium to also be considered.

The most important conclusion of this work is that identification of acoustic element parameters in the manifold, based on multi-dimensional simulation model, is feasible. The author obtained much better results from the developed method than those yielded by the classic Helmholtz model. A comparison between pure gas and gas with oil contamination is also shown in the paper.

Keywords: pressure pulsations, two-phase flow, CFD

Notation

Scalar values
b – the flow damping coefficient,
c – sound velocity,
$\dot{m}$ – the mass flow rate,
L – length,
p – pressure,
S – cross-section area,
w – velocity.

Greek letters
τ – time,
ω – frequency.

Complex values
$j=\sqrt{-1}$ – imaginary unit,
P – complex pressure (after FFT),
M – the complex mass flow rate (after FFT),
T – transmittance.

Complex matrices
$\mathbf{A}=\{a_{ij}\}$ – four-pole matrix,
$\mathbf{Z}=\{z_{ij}\}$ – impedance matrix.

1. Introduction

The periodicity of compressor operation is a source of pressure pulsations in volumetric compressor manifolds. This problem concerns mainly piston type compressors, but – to a lesser degree – also screw compressors and other types of volumetric compressors [1, 2].

Theoretical methods for analysis of the interaction between a manifold component and the propagation of a pressure pulsation wave can be divided into two groups:

a) Helmholtz analysis and a solution of the telegraph equation in the complex space [3–5];
b) One-dimensional (differential, characteristics) or multi-dimensional simulation methods based on CFD simulation (Computational Fluid Dynamics - Finite Element Method, Boundary Element Method, Finite Difference Method).

The advantage of CFD methods is that they capable of simulating any geometry and that they take into account the non-linearity of propagation of realistic amplitude pulsations. However, including full manifold geometry in a CFD program is unrealistic, since such a manifold may be composed of many diverse elements and their total length can be of the order of several hundred meters. On the other hand, the Helmholtz method contains numerous simplifying assumptions. Each element of the manifold is replaced by a straight section of a pipeline of a known length and diameter or by a lumped volume. In the case of oil separators, pressure pulsation mufflers of special design, and even for collectors, such simplifications are inadmissible for geometrical reasons. Attempts at theoretical analysis of other shapes were undertaken in [3–5], but only simple geometries (sphere, cylinder) were considered. Additional difficulty lies with a medium that may not always be treated as an ideal gas. This is due to the fact that a compressed gas may be contaminated with oil droplets, and change occurs in the refrigerant compressors phase.

The method presented here allows one to include in the calculated transmittances the influence of any geometrical shape, two-phase flows, as well as a real gas model instead of the isentropic ideal gas assumption.

2. Theoretical basis of the method

The classic Helmholtz model is based on a solution of the wave equation [1] for a straight section of a pipeline. As a result, a four-pole matrix of form (1) is obtained.

This four-pole matrix $\{a_{ij}\}$ has a general form defined by Equation (2). Concurrently with the four-pole matrix, a complex impedance matrix $\mathbf{Z}$, having elements $\{z_{ij}\}$, is defined by relation (3):

$$\begin{bmatrix} P_1 \\ M_1 \end{bmatrix} = \begin{bmatrix} \cosh\gamma L & Z_f \sinh\gamma L \\ \frac{1}{Z_f}\sinh\gamma L & \cosh\gamma L \end{bmatrix} \cdot \begin{bmatrix} P_2 \\ M_2 \end{bmatrix}, \tag{1}$$

$$\begin{bmatrix} P_1 \\ M_1 \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix} \cdot \begin{bmatrix} P_2 \\ M_2 \end{bmatrix}, \tag{2}$$

$$\begin{bmatrix} P_1 \\ P_2 \end{bmatrix} = \begin{bmatrix} z_{11} & z_{12} \\ z_{21} & z_{22} \end{bmatrix} \cdot \begin{bmatrix} M_1 \\ M_2 \end{bmatrix}, \tag{3}$$

where

$$\left.\begin{aligned} \gamma &= \sqrt{(b+j\omega)(\tfrac{j\omega}{c^2})} \\ Z_f &= \tfrac{1}{S}\sqrt{\tfrac{(b+j\omega)}{\frac{j\omega}{c^2}}} \end{aligned}\right\}. \tag{4}$$

In order to generalize the model for an arbitrary geometry, it has been assumed that the forms of matrices $\{a_{ij}\}$, $\{z_{ij}\}$ will not be based on Equation (1), but they will have a completely free form. This assumption means that, for a unique determination of the interaction of an arbitrary component of a manifold, it is sufficient to know four elements of the matrix which must be identified for this component. The aim of the present work has been to develop a theoretical method of identifying these matrix elements.

The concept of the method developed here is as follows: a full multi-dimensional CFD simulation is carried out for the considered element of a manifold, solving the Navier-Stokes set of equations numerically, together with the necessary closing models, *i.e.* gas equation of state, the turbulence model, the boundary conditions. The results obtained are averaged at the inlet and outlet of the analyzed element and then a complex transformation of the results is carried out, so that elements consistent with the generalized form of matrices $\{a_{ij}\}$, $\{z_{ij}\}$ are obtained. In this way, the advantages of both methods can be combined: the Helmholtz model's capability to analyse vast installations and the possibility of studying the geometry of any geometrically complex element without a priori simplifications.

For further considerations it is necessary to define complex transmittances. Complex transmittances are defined below; a flow and flow-pressure transmittance for excitation directed along the direction of the flow in the manifold and opposite to it:

– flow transmittance:

$$T_M(i\omega) = -\frac{M_{i+1}}{M_i}, \tag{5}$$

– flow-pressure transmittance:

$$T_{MP}(i\omega) = -\frac{P_{i+1}}{M_i}. \tag{6}$$

For a symmetrical compressor manifold element, two transmittances would be sufficient. For an asymmetrical one, four transmittances are necessary. These transmittances can be derived from a CFD simulation, as all of them are determined having the flow excitation boundary condition $\dot{m}_i$, $\dot{m}_{i+1}$ or w_i, w_{i+1}. The second boundary condition is a condition of a free outflow or a dead end. For this purpose, complex impedances Z_0 and Z_k have been defined, closing the studied muffler from both sides (see Figure 1):

$$Z_{0,k} = \frac{P_i}{M_i}, \tag{7}$$

where 0 means inlet, and k – outlet of the element along the direction of the flow.

As the transmittances T defined above are determined for specific known values of Z_0 and Z_k, a relation between matrix $\mathbf{Z}$ and transmittances T is unique and can be derived. In the simplest case $Z_k = 0$ or $Z_k = \infty$ or, for an opposite flow, $Z_0 = 0$ or $Z_0 = \infty$.

The four cases chosen for a CFD simulation of the unsymmetrical compressor manifold element are shown in Figure 1. Each simulation allows one transmittance to be determined according to the drawing, referred to as T_a, T_b, T_c, T_d.

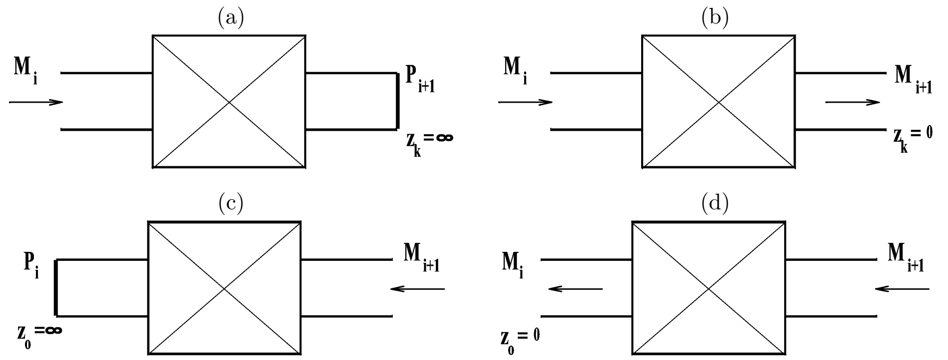

Figure 1. Four cases used for CFD simulation shown with boundary conditions and quantities used for transmittance determination

Writing impedance relations (3) for the a, b, c, d cases as defined in Figure 1 and using Equations (5) and (6), one obtains:

$$\left.\begin{array}{l} P_i = z_{i,i} \cdot M_i \\ P_{i+1} = z_{i+1,i} \cdot M_i \end{array}\right\} \Rightarrow z_{i+1,i} = T_a, \tag{8a}$$

$$\left.\begin{array}{l} P_i = z_{ii} \cdot M_i + z_{i+1,i} \cdot M_{i+1} \\ 0 = z_{i+1,i} \cdot M_i + z_{i+1,i+1} \cdot M_{i+1} \end{array}\right\} \Rightarrow z_{i+1,i+1} = \frac{T_a}{T_b}, \tag{8b}$$

$$\left.\begin{array}{l} P_i = z_{i,i+1} \cdot M_{i+1} \\ P_{i+1} = z_{i+1,i+1} \cdot M_{i+1} \end{array}\right\} \Rightarrow z_{i,i+1} = T_c, \tag{8c}$$

$$\left.\begin{array}{l} 0 = z_{ii} \cdot M_i + z_{i,i+1} \cdot M_{i+1} \\ P_{i+1} = z_{i+1,i} \cdot M_i + z_{i+1,i+1} \cdot M_{i+1} \end{array}\right\} \Rightarrow z_{i,i} = \frac{T_c}{T_d}. \tag{8d}$$

The derived relationships (8) makes possible a unique determination of complex impedance matrix $\mathbf{Z}$ that defines a linear lumped acoustic element.

The evaluation of transmittances $T(i\omega)$ for each of the four cases and for a dozen or so significant harmonics is very arduous and time consuming. Therefore, it is better to use a system response for a unit step function or impulse function excitation. An impulse function is defined as follows:

$$\delta\dot{m}(\tau) = \begin{cases} \dot{m} & \text{for} \quad \tau < \Delta\tau \\ 0 & \text{for} \quad \tau > \Delta\tau \end{cases}, \tag{9}$$

where $\Delta\tau$ is the elementary time step of the numerical calculation.

Using the Laplace transform, we can write:

$$\delta M(s) = \mathscr{L}\{\delta \dot{m}(\tau)\} = 1. \tag{10}$$

System responses for unit step excitation δM_i or δM_{i+1} depending on the case (a, b, c or d as per Figure 1), are δP_{i+1}, δM_{i+1}, δP_i, δM_i, thus:

$$Y(s) = T(s) \cdot X(s), \tag{11}$$

where $X(s)$ denotes excitation, and $Y(s)$ is a system response.

For a known impulse response of a system $\delta Y(s)$ (where Y is P_i, P_{i+1}, M_i, M_{i+1}, respectively):

$$T(s) = \delta Y(s). \tag{12}$$

This means that, based on a system response to a flow impulse excitation in each of the four cases (a, b, c, d), one can determine transmittance of the studied element.

3. CFD identification for a special muffler

The method developed in this work was applied to the pressure pulsation muffler shown in Figure 2, [1, 2].

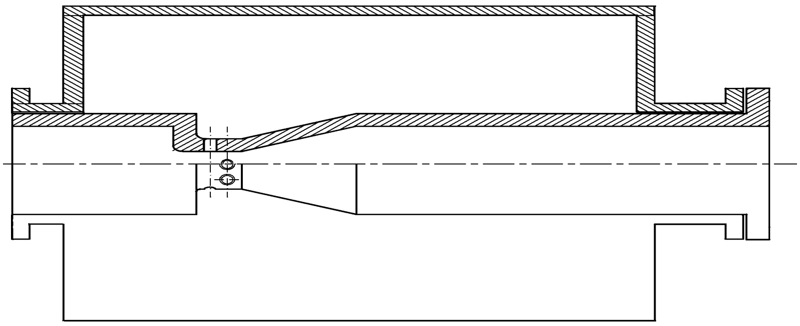

Figure 2. The design of the pulsation damper used for verification

A simulation model of the muffler was created in a cylindrical coordination system using axial symmetry, which reduced the case to a two-dimensional one. A geometric model was introduced into the PHOENICS 3.1 program, based on the Finite Volume Method.

According to the formulas shown earlier, boundary conditions for the investigated case were prepared. At the inlet of the muffler, a step or impulse function excitation was given evenly at the whole inlet cross-section. The velocity condition was applied with the amplitude of 10m/s. At the outlet, a rigid wall or a constant pressure outlet condition was applied.

Along the radius, the grid contained 39 unequal elements resulting from geometry, and 86 along the symmetry axis. The average size of the grid was 1cm, however near the choking part of the inner tube it was more dense. The reaction time of the muffler for the excitation was simulated within the time range of 0–2s, divided unevenly (more densely at the beginning) into 131 parts. This was long enough to obtain the required response. An ideal gas equation of state and the k-ε turbulence model were used for simulation.

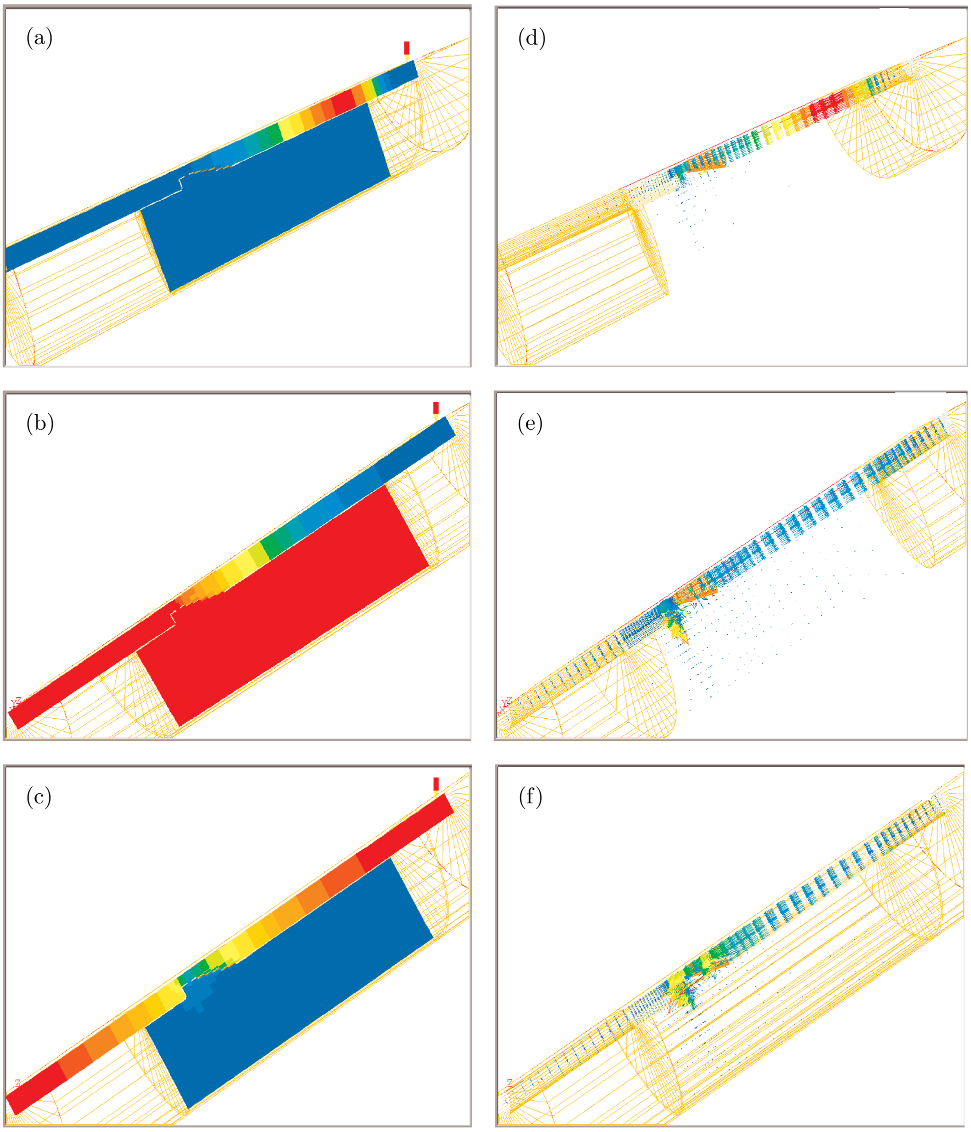

Figure 3. Examples of CFD simulation results: isobars and velocity vectors; (a), (b), (c) – pressure distribution after 0.001s, 0.003s, and 0.01s, respectively; (d), (e), (f) – velocity vectors after 0.001s, 0.003s, and 0.01s, respectively

In Figure 3, waves of pressure and velocity traveling along the symmetry axis of the analyzed element are presented. In order to use the CFD calculation results in the Helmholtz model, the values of the pressure function and the mass flow rate were averaged at the inlet and outlet cross-section. Based on the diagrams obtained for a step and impulse mass inflow excitation (Figure 4), the parameters of the first and second order transmittances were determined.

The first order transmittance (only damping and time lag) has the form:

$$T(s)=\frac{K}{1+s\cdot\zeta}\cdot e^{-s\Delta\tau}. \tag{13}$$

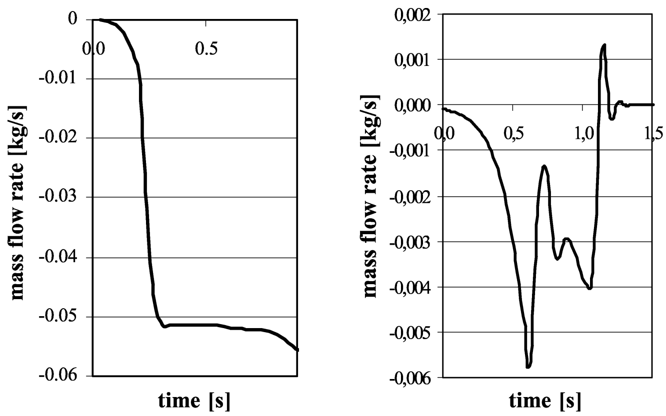

Figure 4. Examples of the calculated acoustic response of the muffler – basis for the transmittance calculation

The second order transmittance has the form:

$$T(s) = \frac{K \cdot \omega_o^2}{s^2 + 2\zeta\omega_o s + \omega_o^2} \cdot e^{-s\Delta\tau}. \tag{14}$$

The coefficients K, ζ, ω_o, $\Delta\tau_1$ were determined graphically from the time response diagrams of the $\dot{m}_i(\tau)$ and $p(\tau)$ functions. The results have been collected in Table 1.

Table 1. Values of coefficients for Equations (13) and (14)

Transmittance	K	ω_o	ζ	$\Delta\tau$
T_a	28154	22	0.053	0.187
T_b	−0.8	35	0.095	0.186
T_c	46154	22	0.060	0.190
T_d	−1.12	35	0.130	0.185

4. Experimental validation

In order to verify experimentally the results of the theoretical identification, an experimental set-up was constructed, based on an S2P216 air compressor operating in the Ward-Leonard system with variable rotational speed [1].

In this test stand, the studied muffler was assembled on the suction line of a compressor. The measurement system consisted of DISA capacity transducers coupled through a converter and amplifier to a transient MC101 recorder, and then to a PC. The measured pressure pulsation curves were compared with the calculated values. A detailed description of the experimental method and set up is given in [1, 2].

Figure 5 presents a comparison of the pressure curves calculated for the muffler using first and second order transmittances, Equations (13) and (14), with the experimental results. A comparison of the significant harmonics obtained from the curves presented in Figure 5 is shown in Figure 6. As can be seen, even first order transmittances calculated on the basis of the CFD simulation are better approximations of the actual pressure pulsations than the classic Helmholtz model.

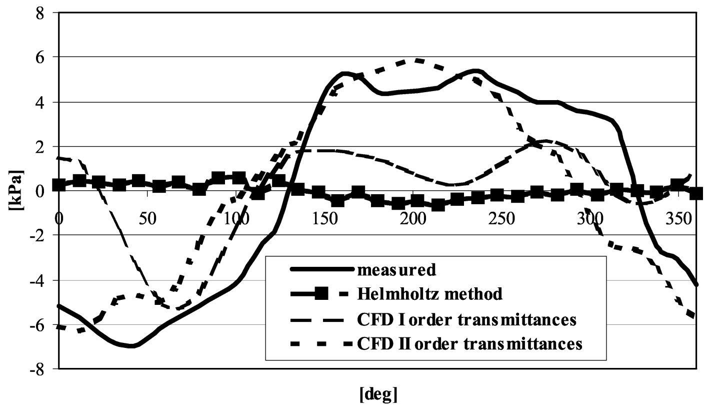

Figure 5. Comparison of the pressure pulsation curves before the muffler obtained by four methods

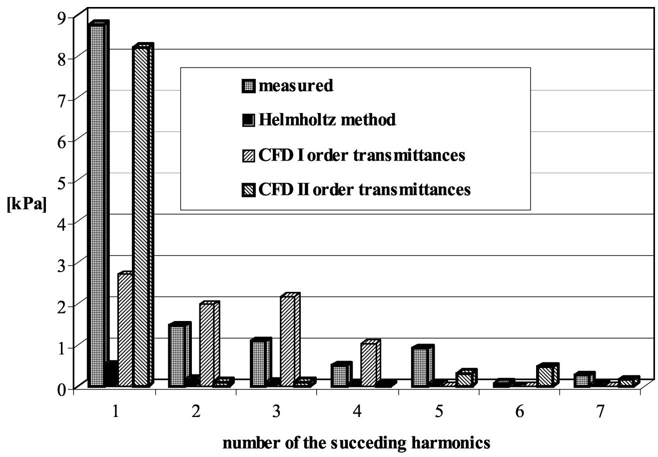

Figure 6. Comparison of the harmonic analysis results for the curves in Figure 5

Table 2. Comparison of the peak-to-peak amplitudes of pressure pulsations

[rpm]	Experiment	Helmholtz method		CFD simulation I order		CFD simulation II order	
	Δp [kPa]	Δp [kPa]	error [%]	Δp [kPa]	error [%]	Δp [kPa]	error [%]
1300	12.3	1.2	90	7.5	39	12.2	1

The results of the calculations are summed up in Table 2. A significantly better agreement with experiment is evident for the CFD-based method, when compared with the classic Helmholtz model.

5. Two-phase flow simulation results

One of the reasons why this method has been worked out, is the possibility to consider the liquid phase presence in the compressed gas. Only a CFD simulation

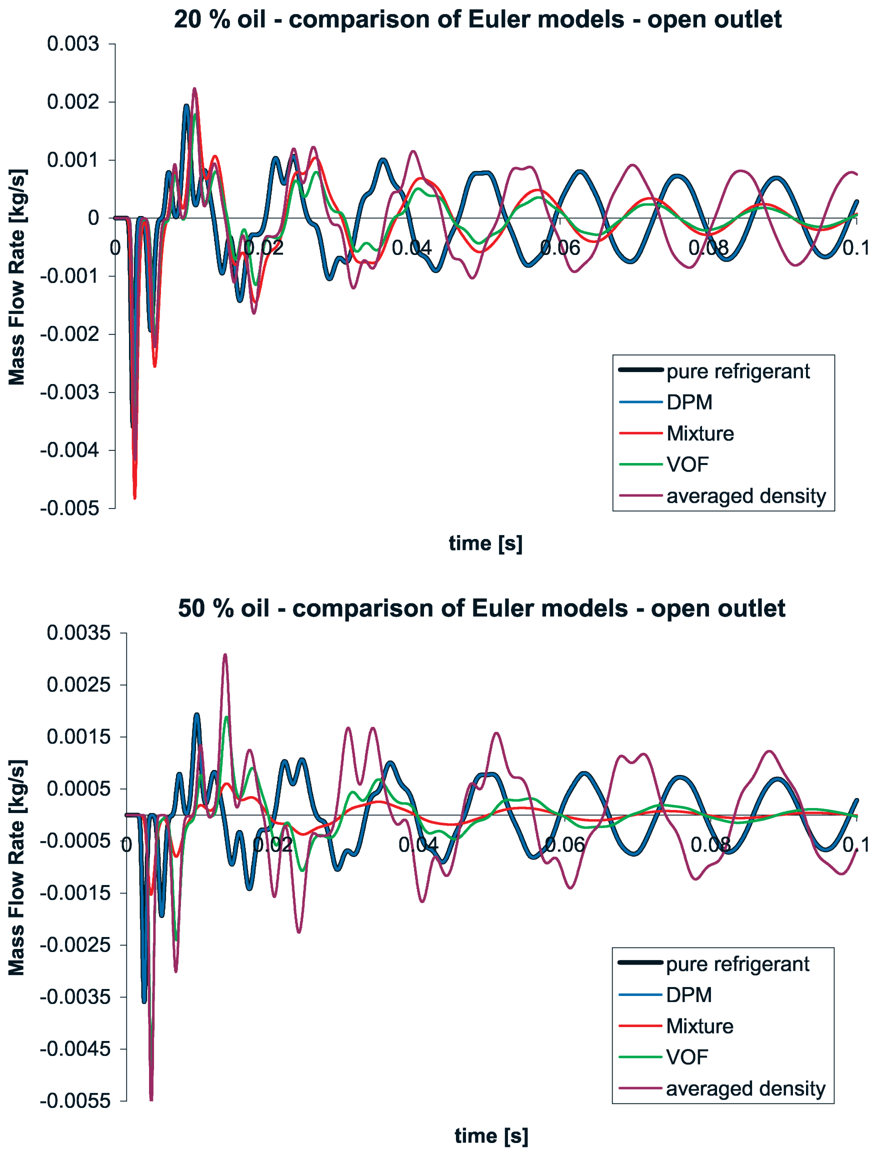

Figure 7. Comparison of simulation results of various two-phase models

makes it possible to determine transmittances in this case without measurements. Due to the extremely long calculation time on a PC using PHOENICS 3.1, we had to switch to FLUENT, which is based on the same FVM method, but the current version is much faster and available on the mainframe server at the CYFRONET Academic Computer Center in Cracow.

In FLUENT, there are several possibilities for second phase modelling. All possibilities for a compressible two-phase flow have been tested. Some benchmark tests have been done to determine the accuracy of two-phase flow models. A nozzle flow has been simulated and compared with standard results, and this comparison have shown good agreement. For a pulsating flow simulation, a simple muffler designed for

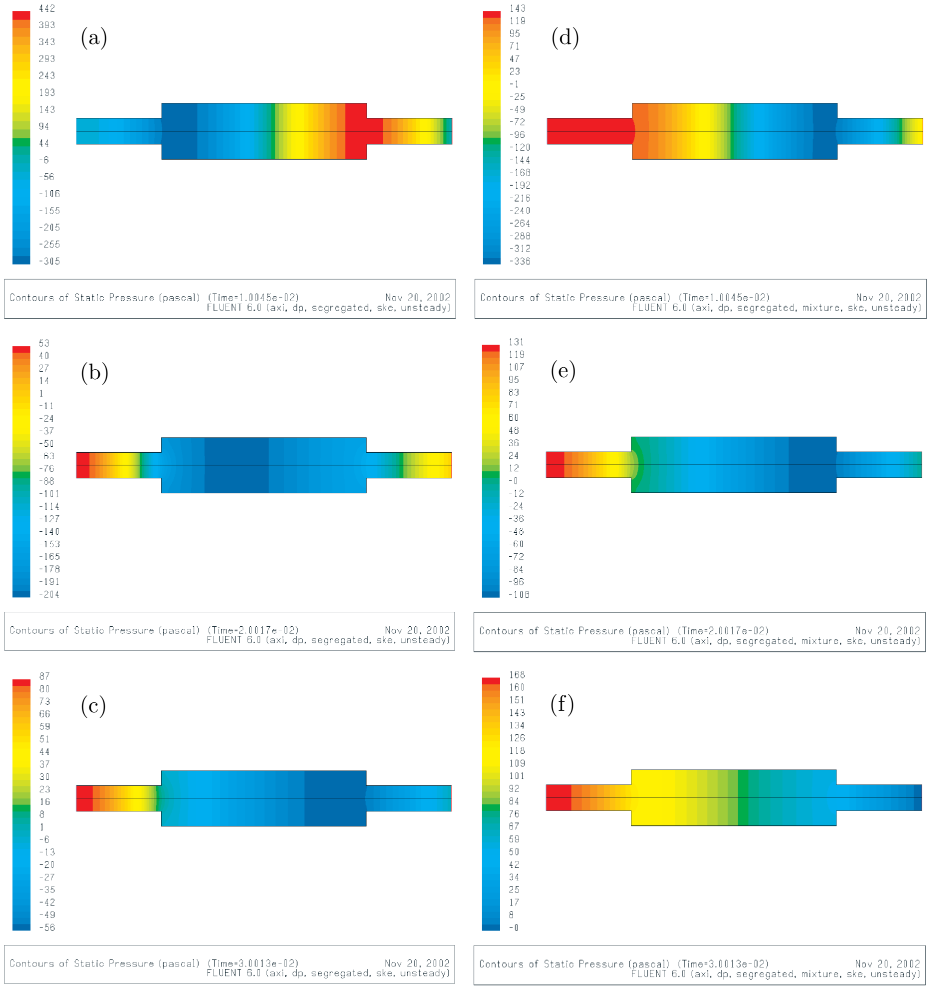

Figure 8. The unsteady pressure contours in the muffler cross-section with pure refrigerant and refrigerant contaminated with oil droplets (20% mass flow ratio); (a), (b), (c) – pure refrigerant, pressure distribution after 0.01s, 0.02s, and 0.03s, respectively; (d), (e), (f) – mixture 20% oil, pressure distribution after 0.01s, 0.02s, and 0.03s, respectively

the refrigerating discharge manifold has been selected. Since the amount of oil in the gas compressed in the compressor manifolds is rather small, the DPM (Discrete Phase Model) was the first one to try. Results of the simulation were promising against the benchmark tests. However, it was found that although the influence of the gas flow on the oil particles seemed accurate, there was no acoustic energy dissipation due to the presence of oil. Therefore, other methods had to be tested: Volume of Fluid (VOF), Mixture Model (MM) and, also, averaged mass, one-phase flow. The results of these simulations are shown in Figure 7. It seems that the MM method is the most accurate for a pulsating flow simulation, since VOF is rather designed for free surface flows.

It can be seen that an amount of acoustic energy is absorbed by oil particles. VOF is less realistic, and in DPM it seems that oil has no influence at all. The pressure contours for this muffler have been shown in Figure 8 for the flow of a clean refrigerant

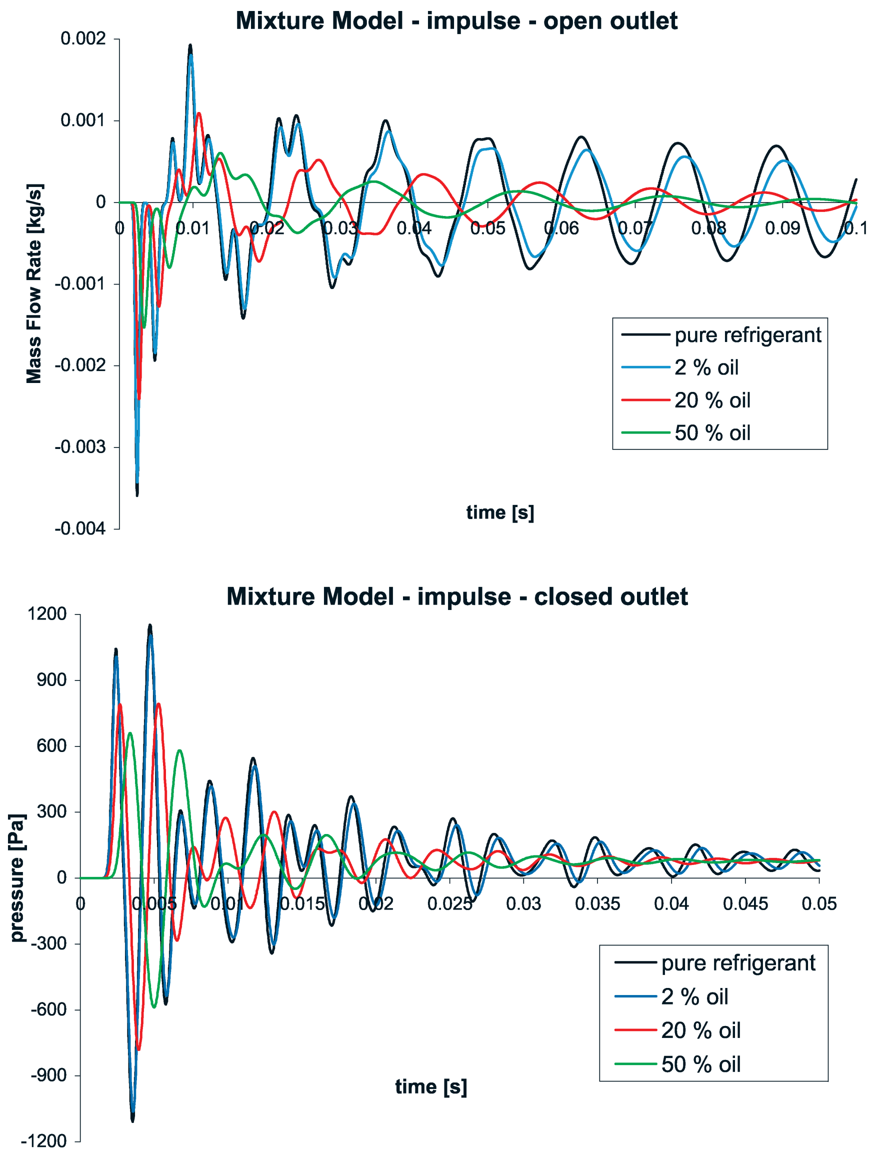

Figure 9. Comparison of pressure pulsation curves and mass flow pulsation curves for closed and open outlet

and the refrigerant contaminated with 20% of oil (mass flow rate oil/gas). In Figure 9, a comparison of pressure pulsation curves as response to the inlet excitation for open and dead end closings of the muffler is shown. The influence of damping due to an increase of oil contamination is clearly visible. This effect occurs even below the oil mass flow rate of 5%. Actually, oil contamination in compressor manifolds is below 5%.

6. Summary and conclusions

The most important conclusion of this work is that identification of acoustic element parameters in the manifold, based on multi-dimensional simulation model

(CFD), is feasible. The obtained results are much closer to reality than those yielded by the classic Helmholtz model.

In the present work, transmittances T_{MP}, T_P were introduced. This approach enabled us to use CFD modelling to identify the parameters of a generalized Helmholtz model.

A direct application of the CFD method to the entire vast manifold is futile, not only because of its computation time but, above all, because introducing geometrical data into the program is a very time consuming task. Therefore, it is more convenient to identify transmittances T_{MP} and T_P. Having obtained these, the four-pole matrix $\{a_{ij}\}$ and impedance matrix $\{z_{ij}\}$ elements can be easily calculated on the basis of formulas (8).

Since the CFD simulation-based method of identification of objects with complex geometry gives better results for the pressure pulsations amplitude than the classic Helmholtz model, the two-phase pulsating flow with impulse function flow excitation has been simulated for a refrigerant installation muffler. Results obtained so far lead to the conclusion that the Mixture Model method is suitable for this problem.

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