Liquid Contamination Effect on the Pressure Pulsations Simulation in Volumetric Compressor Manifold

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ABSTRACT

In the compressor installation oil or other liquid contamination may occur in the compressed gas. Liquid particles have high density, so it seems that part of acoustical energy of the pulsating flow, should be absorbed by those particles. This should influence the results of calculation of the installation element acoustic transmittances.
In this paper authors deals with the problem of the oil contamination effect on the pressure pulsations calculated using CFD simulation. The FLUENT software is used for the 3D simulation of the oil separator.
The method of simulation allows to calculate acoustic transmittances for the installation element on the basis of the CFD simulation. The inflow to the element of the installation is generated by the impulse of mass. The unsteady flow calculation is worked out for open and closed end of the analyzed element. The method has been shown earlier in the Purdue Compressor Conference Proceedings 2002.
Different levels of oil contamination have been considered and compared from 1 % up to 50 % of the mass flow rate liquid/gas phase. It occurs that below 5 % of the mass ratio liquid/gas (witch is usually the case in the compressor manifolds), the liquid phase has very small influence on the pressure wave propagation. So in this case simple one-phase model is suitable.
Above 20% of the mass ratio the oil contamination has significant influence on the transmittances of the installation element.

INTRODUCTION

The phenomenon of pressure pulsation significantly influences the functioning of the entire compressor manifold. The analysis of the manifold element response for pulsating flow excitation is the subject of many experimental investigations (Abom, Boden, 1988); (Chung, Blaser, 1980); (Cyklis, 2001); (Lung, Doige, 1993). However great part of pressure pulsation analysis is based on the theoretical investigations. An important issue in theoretical analysis of the pressure pulsations is the model the working gas (Liu, Soedel, 1994). The use of the proper equation of state for the gas, and the description of the real gas properties, as well as dispersed liquid contamination influence not only the accuracy of the obtained results, but also quality of the solution. Dealing with pressure pulsation, it is important to know both: the real velocities of wave propagation, and the values of the thermal properties of the gas. Liquid contamination influences the wave propagation taking away part of the pulsation energy. The most common approach in analysing the phenomenon of pressure pulsation is the use of the ideal gas equation of state. However, in certain ranges of gas contamination, the error of results obtained through this equation, as compared with real values, becomes considerable. A need thus arises to carry out calculations using alternative models describing the behaviour of real gas with impurities.

1. THEORETICAL BASIS OF THE METHOD

The classic Helmholtz model is based on a solution, for a straight section of a pipeline, of the wave equation of the form (1). As a result, a four-pole matrix or an impedance matrix of the form (2) is obtained. Elements of this matrices $\{a_{ij}\}\,\{z_{ij}\}$ are determined only for a segment of a straight, constant cross-section pipeline.

$$\begin{cases} -\dfrac{\partial p}{\partial x} = \dfrac{1}{S}\dfrac{\partial \dot{m}}{\partial \tau} + \dfrac{b}{S}\cdot \dot{m} \\ -\dfrac{\partial \dot{m}}{\partial x} = \dfrac{S}{c^2}\dfrac{\partial p}{\partial \tau} \end{cases} \tag{1}$$

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

where: p – pressure, m – mass flow rate, S – cross section area, b – damping factor, τ - time, x – Cartesian coordinate, L – pipe length, c – speed of sound, , P – pressure in complex space, M- mass flow in complex space.
In order to generalize the model for an arbitrary geometry it has been assumed that matrices $\{a_{ij}\}$, $\{z_{ij}\}$ are not to be based on equation (1) solution, but that they will have a generalized form. This form has to be determined on the basis of an identification method. In the method developed here, for a considered element of a manifold a full multi-dimensional CFD simulation is carried out, solving the Navier-Stokes set of equations numerically together with the necessary closing models, i.e. gas state model, turbulence model, boundary conditions. The results obtained are averaged at the inlet and outlet of the analysed element and next 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 possibility of analysis of extended installations and the possibility of studying geometry of an arbitrary element, without a priori simplifications(Cyklis, 2002).
For further considerations it is necessary to define complex transmittances. The complex transmittances are defined: a flow and flow-pressure (3) respectively, both for forward and reverse flow.

$$\overline{T}_M(i\ ) = -\frac{M_{i+1}}{M_i} \qquad \overline{T}_{MP}(i\ ) = -\frac{P_{i+1}}{M_i} \tag{3}$$

For a symmetrical compressor manifold element, two transmittances would be sufficient, for unsymmetrical four transmittances are necessary. These transmittances can be derived using CFD simulation, as all of them are determined having the flow excitation boundary condition: $\dot{m}_i$, $\dot{m}_{i+1}$. The second boundary condition is a condition of a free outflow or a total closing.
The four cases prepared for the CFD simulation of the unsymmetrical compressor manifold element are shown on the Figure 1.

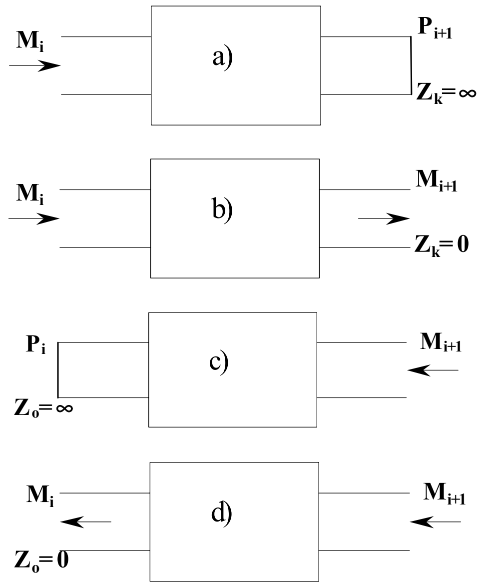

Figure 1: Four cases of simulation essential to complete determine of acoustic parameters

Each simulation allows to determine one transmittance according to the drawing, named respectively $\overline{T}_a, \overline{T}_b, \overline{T}_c, \overline{T}_d$.
Writing down impedance relations for the cases a, b, c, d as defined in Figure 1 one obtains:

$$\left.\begin{aligned} P_i &= z_{ii} \cdot M_i \\ P_{i+1} &= z_{i+1,i} \cdot M_i \end{aligned}\right\} \Rightarrow z_{i+1,i} = \overline{T}_a \tag{4a}$$

$$\left.\begin{aligned} P_i &= z_{ii} \cdot M_i + z_{i+1,i} \cdot M_{i+1} \\ 0 &= z_{i+1,i} \cdot M_i + z_{i+1,i+1} \cdot M_{i+1} \end{aligned}\right\} \Rightarrow z_{i+1,i+1} = \frac{\overline{T}_a}{\overline{T}_b} \tag{4b}$$

$$\left.\begin{aligned} P_i &= -z_{i,i+1} \cdot M_{i+1} \\ P_{i+1} &= -z_{i+1,i} \cdot M_{i+1} \end{aligned}\right\} \Rightarrow z_{i,i+1} = \overline{T}_c \tag{4c}$$

$$\left.\begin{aligned} 0 &= -z_{ii} \cdot M_i - z_{i,i+1} \cdot M_{i+1} \\ P_{i+1} &= -z_{i+1,i} \cdot M_i - z_{i+1,i+1} \cdot M_{i+1} \end{aligned}\right\} \Rightarrow z_{i,i} = \frac{\overline{T}_c}{\overline{T}_d} \tag{4d}$$

The derived relationships (4) allow a unique determination of complex impedance matrix Z that defines a linear lumped acoustic element. As the excitation for the CFD simulation the impulse or/and unit step function for mass flow rate is used, allowing for general element response calculation.
The general method for calculation all transmittances is the use of an impulse or a unit step function as flow input excitation, for each of four cases a, b, c, d on Figure 1.
Since pressure and flow pulsations have damped oscillation characteristic it is obvious to use a general damped harmonic form for transmittances. The acoustic response of the installation element for an impulse inflow excitation have general form in the complex domain:

$$T(s) = \frac{K\,\omega_0^2}{s^2 + 2\,\zeta\,\omega_0\,s + \omega_0^2} \cdot e^{-s\,\tau} \tag{5}$$

and in the real domain:

$$y(\tau) = K \cdot u_0 \cdot \frac{\omega_0}{\sqrt{1-\zeta^2}} e^{-\zeta\omega_0\tau} \cdot \sin\left(\omega_0\sqrt{1-\zeta^2}\tau + \tau_0\right) \tag{6}$$

The damping coefficient may be derived as:

$$\zeta = B \cdot \sqrt{\frac{1}{\left(\frac{2\pi}{\tau_c}\right)^2 + B^2}} \tag{7}$$

and the amplification:

$$K = \frac{A_0 \cdot \sqrt{1-\zeta^2}}{u_0 \cdot \Delta\tau \cdot \omega_0} \tag{8}$$

The natural harmonic frequency:

$$\omega_0 = \frac{2\pi}{\tau_c\sqrt{1-\zeta^2}} \tag{9}$$

The values τ_0, τ_c, B may be easily determined from the analysis of real system response shown on Figure 2.

u_o – the height of the impulse excitation function, during $\Delta\tau$- the elementary numerical "time mesh" dimension.

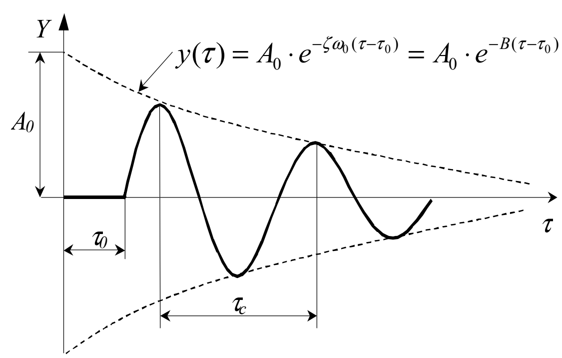

Figure 2: Curve of oscillating damped impulse response

However usually the system response for an impulse inflow excitation tends to form more then one basic natural frequency. In this case the response is decomposed using Fourier analysis, and each frequency is consequently analysed using above shown algorithm.

2. DIFFERENCE BETWEEN TWO-PHASE MODELS

In most commercial packages for CFD simulations, three main two-phase flow models may be introduced:

– DPM (Discrete Phase Modeling) – where gas phase is calculated in continuous basis (Euler model), and for the liquid phase discrete Lagrange model is used,
– VOF (Volume of Fluid) – both phases are continuous (Euler model), the common momentum equation is solved for both phases and results are divided accordingly to the volume ratio of each phase,

– Mixture model - also both phases are continuous, but additionally equations for each phase content and diffusion equation for concentration are included. The comparison of models is shown in Table 1.

Table 1: Multiphase flow models

	DPM	Mixture	VOF
Variables	Euler-Lagrange	Euler	Euler
Particle size determination	yes, (also Rosin-Rammler distribution available)	yes (only uniform sized particles)	no
Modelling of static systems	no	yes	yes
Sliding	yes	yes	no
Phase interaction	to weak influence of the liquid phase on gas	yes	yes

The comparison of three simulation methods shows that VOF can be rejected since there is no sliding and particle size determination. At the beginning it was obvious that DPM method shall be employed. This method makes it possible to introduce real oil particle size distribution which seems very convenient. Unfortumetely after numerous simulations we found that the feed-back from liquid phase to gas is unrealistic (too small). No pulsations energy have been transferred to liquid phase. So finally the Mixture model have been choosen.

3. CFD SIMULATION AND IDENTIFICATION

Described model was used for numerous numerical experiments for nozzle, muffler and oil removers simulations. Since only the Mixture model was able to account for taking over part of pulsation energy by liquid phase dispersed in gas. Only this model results will be shown here. In this paper the oil remover for a refrigerant compressor is shown as an example. The geometry of this element is shown on the Figure 3 together with the finite volume mesh.

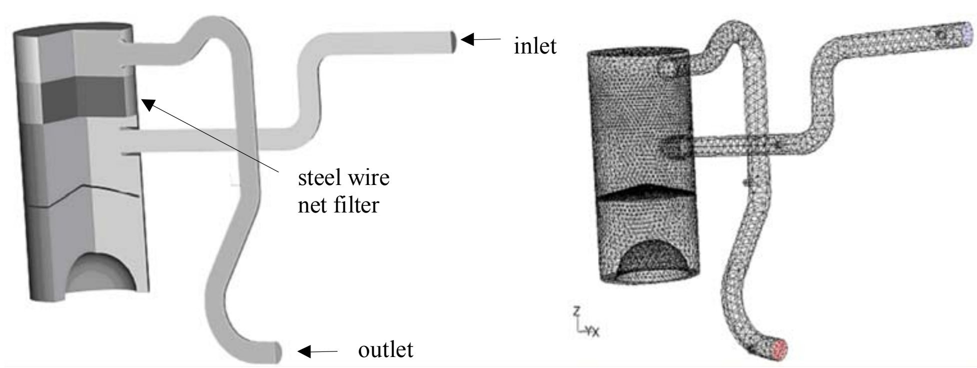

Figure 3: 3D geometry of oil remover (83 792 of finite volume elements)

The results of simulation for pure gas and 1, 5, 20, 50 % of oil mass content in gas are shown. The results of 3D simulation are averaged within the inlet and outlet cross-section area in order to obtain one-dimensional mass flow and pressure curves. On the fig. 4 one example of such averaging is shown. The transmission matrix for this case is T_a according to fig.1.

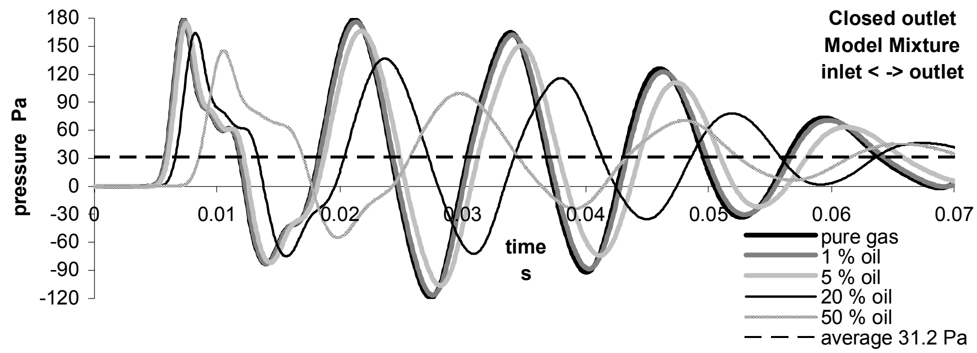

Figure 4: An example of the impulse reaction (pressure) on closed outlet of oil remover

They are then used to calculate complex transmittances and four-pole matrix for the oil remover. On the Figure 4 it's clearly visible that the rise of oil contamination reduces the main response frequency. The calculated parameters of each transmittance have physical meaning. The Figure 5 shows four parameters for T_b oil remover flow transmittance.

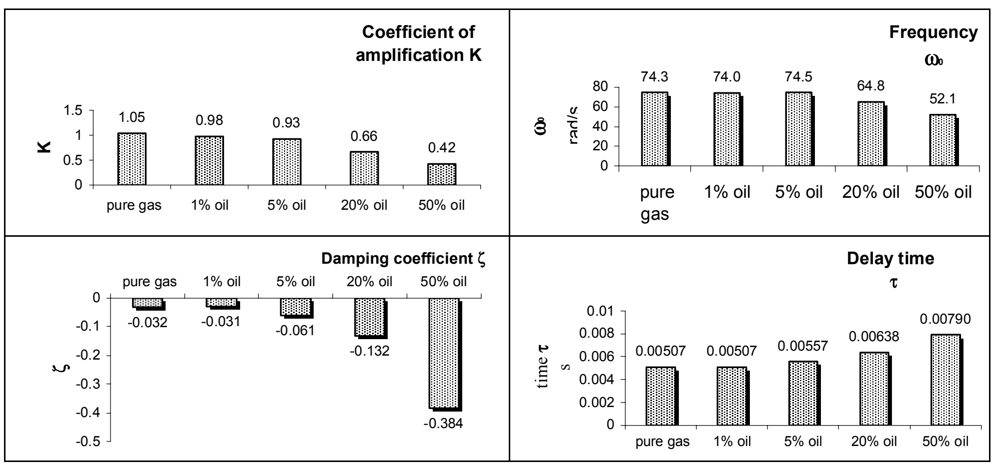

Figure 5: The comparison of transmittance T_b parameters for different oil mass fraction for 1-st harmonic frequency

The important conclusion which may be derived is that the significant influence of the oil contamination on the acoustical response of the oil remover occurs close to 20 % of the oil to gas mass fraction. For lower values the tendency of changes is visible, but the quantitative difference are small enough to be neglected in the simplified technical analysis. Generally the changes of transmittance parameter values with the respect to oil mass contamination is proportional to the oil mass percentage in gas. The linear characteristic is very convenient, while it can be used to estimate transmittance parameters for different oil content.

4. EXPERIMENTAL VALIDATION

The flow case through the oil remover (Fig. 3) with oil mass fraction about 1% for a refrigerant compressor has been also used for experimental identification. The comparison of the experimental results with classic Helmholtz model and the model described in this paper is shown on the Figure 6 and in Table 2. On the fig. 7 harmonic analysis of pressure pulsations from fig.6 is presented.

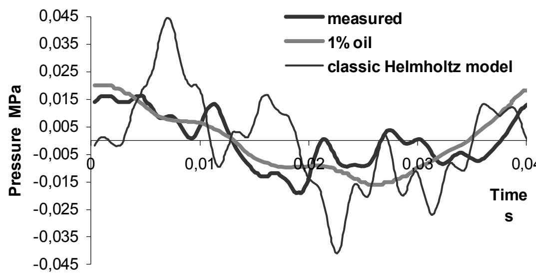

Figure 6: Pressure pulsations curves averaged in inlet cross section of oil remover

Table 2: Peak-to-peak amplitudes comparison

	Experiment	1% of oil mass fraction	Helmholtz ideal model
Δp_{max} [MPa]	0,035	0,036	0,085
Error [%]	–	2,9%	143,7%

Although there are discrepancies between presented model and experimental results, they are comparable. The classic Helmholtz method in this case gave unrealistic results. Main cause of this is the oil remover geometry, which in case of 3D CFD simulation is properly modelled, but also oil contamination has its influence. Slight differences between the present model and experiment may be due to the fact that the oil contamination is not constant: higher at the oil remover inlet and lower at the outlet.

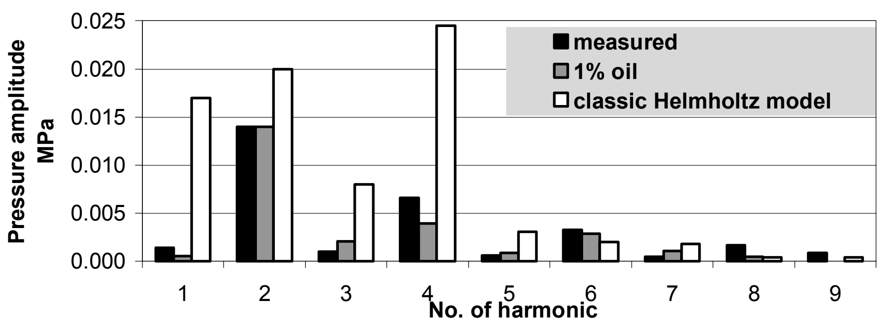

Figure 7: Harmonic analysis of the pressure pulsations averaged spatially in the inlet cross section of oil remover

5. SUMMARY AND CONCLUSIONS

In the paper only small part of numerous simulation results is shown. However on this basis some of important conclusions may be drawn. First, the CFD identification method proved to be a proper tool for the analysis of pressure pulsation attenuation in the installation element. Using CFD identification method it is relatively simple to introduce real gas model or oil contamination. It is also possible to use this method for phase change simulation and its influence on pressure pulsation attenuation. Second, the oil contamination in the compressed gas may influence the pressure pulsations propagation lowering its frequency response increasing dumping and time delay. The quantitative results have nearly linear dependency on the oil mass fraction. It is also important that in modern compressor installations where oil mass fraction is below 1 %, the oil contamination influence on pressure pulsations may be neglected. But if any other liquid is present (e.g. phase change) its influence on the pulsations may not be neglected.

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