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The CFD based estimation of pressure pulsation damping parameters for the manifold element

Piotr Cyklis*, Przemysław Młynarczyk

*Cracow University of Technology, Institute of Power and Process Engineering, Mechanical Faculty, al. Jana Pawla II 37, 31-864 Krakow, Poland

Abstract

Vibrations and noise caused by the pressure pulsations in the volumetric compressor manifolds have a high impact on the compression power requirement and the reliability of manifold operation. For the pressure pulsations attenuation, different types of mufflers are applied using a design based on the Helmholtz resonator model approach. This is particularly effective for the constant revolution speed compressors. For the widely introduced variable revolution speed compressors, other pressure pulsation attenuation methods are required. One of the possibilities to attenuate the pressure pulsations over a wide range of frequencies is the introduction of specially shaped nozzles in the gas duct flow. The CFD impulse flow simulations can be used as a tool for the estimation of the nozzle influence on the pressure pulsations damping. The direct 3D or 2D CFD simulation results are difficult to apply for this estimation. A new method has been worked out to obtain the damping coefficient for the diverse nozzle type, using the numerical simulation results of the impulse flow propagation. In this paper the application of the Levenberg-Marquardt method for damping parameter estimation is shown. This method allows fitting the attenuation curves described by damped equations to obtain damped oscillation functions from CFD results. In this paper the CFD based estimation method of pressure pulsation damping parameters is illustrated with an example of its application for a nozzle.

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Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>b</td>
<td>flow damping coefficient</td>
</tr>
<tr>
<td>c</td>
<td>sound velocity</td>
</tr>
<tr>
<td>j</td>
<td>imaginary unit</td>
</tr>
<tr>
<td>H</td>
<td>Hessian</td>
</tr>
</tbody>
</table>

Corresponding author: pcyklis@mech.pk.edu.pl
1. Literature review

Volumetric compressor installations, under the pressure pulsations excitation caused by the periodicity of the compressor work vibrate, which is the noise and vibration source. The pressure pulsations phenomenon is an important problem presented in a number of articles on compressors, pumps and combustion engines. The pressure pulsations directly affect the power consumption, causing mechanical vibrations of the compressed gas systems, induce the noise and in some cases cause the valve failure [1][2][3][4]. Some attempts to apply the CFD simulations to assess the passive pressure pulsation damping using shaped nozzles can be found in publications such as [5][6][7] where some research results are presented.

2. Theoretical approach

The pulsation wave travels in the complex manifold as one-dimensional. The travelling pressure wave is reflected, damped and transmitted on each manifold element. Therefore one-dimensional reaction of the manifold element on the pulsating flow excitation is crucial for manifold construction. However, the manifold elements such as valves, nozzles, tanks, oil separators are in fact 2D or 3D elements and the pulsation is reflected in all directions, affecting 1D reaction. Nowadays the possibilities of 3D simulation allow determining the element reaction to the flow excitation at the inlet. The most important aim of our work is to find a simple but effective method to assess the element reaction. A detailed description of the transmittance method was shown in [8]. The classic complex domain approach, called here the Helmholtz model, is most commonly used particularly in the USA. The main advantage of this method is the possibility of composing the model of a vast manifold with many branches by simple multiplication of the matrices.

The classic Helmholtz model is based on a solution of the wave equation of the form (1), derived for a straight pipeline. The solution result in the complex domain is a four-pole matrix presented by equations (2) and (3). The elements of this matrix \{aij\} are determined strictly for a segment of a pipeline. Concurrently with a four-pole matrix, a complex impedance matrix having the elements \{zij\} may be used.

\[
\begin{align*}
\frac{\partial p}{\partial x} &= \frac{1}{s} \frac{\partial \dot{m}}{\partial t} + \frac{b}{s} \dot{m} \\
\frac{\partial \dot{m}}{\partial x} &= \frac{s}{c^2} \frac{\partial p}{\partial t} + c_m S(p-p_0)
\end{align*}
\]  

(1)

\[
\begin{bmatrix}
P_1 \\ M_1
\end{bmatrix} = \begin{bmatrix}
cosh y L \\ 1/2f \sinh y L \\
1/2f \cosh y L
\end{bmatrix} \begin{bmatrix}
Z_f \\
cosh y L
\end{bmatrix} \begin{bmatrix}
P_2 \\ M_2
\end{bmatrix}
\]

(2)
where:

\[
\gamma = \sqrt{(b + j\omega) \left( \frac{f_0 a}{c^2} \right)} \\
Z_f = \frac{1}{S} \sqrt{\frac{(b + j\omega) f_0 a}{c^2}}
\]  

(3)

This general approach is applicable for simple shapes. It is also possible to calculate generalized four-element transmittance matrix on the basis of the CFD simulation. The four generalized transmittances may be defined likewise eq. (2). The generalized transmittance in the complex domain has the following form:

\[
T(S) = e^{-S\tau_0} \sum_{i=1}^{n} \frac{K_1 \omega_i^2}{S^2 + 2\xi \omega_i \tau + \omega_i^2}
\]

(4)

which in fact is the solution of the damped oscillation of a gas column.

For simplified estimation of the attenuation of a passive damping element influence on the pressure pulsation damping only one transmittance may be used. This transmittance is assessed using the mass flow impulse at the inlet of the element and after 3D/2D flow simulation the response at the outlet of the element is obtained.

This paper shows how the transmittance can be calculated using 3D simulation coupled with the Levenberg-Marguardt method for damping coefficient estimation

3. CFD simulation method

The key element of the proposed method is the possibility to define the attenuation factors of shaped nozzles on the basis of the computer simulation performed using the FLUENT/ANSYS software. The method was proposed in [8]. The boundary conditions for simulation are as follow:

- At the inlet impulse excitation of the 0.1[kg/s] peak mass inflow is introduced. In numerical application the impulse excitation means that its duration is equal to one time step. The mass flow in all other time steps is zero.
- The pressure outlet where the pressure at the outlet is defined as the arithmetical average between the pressure outside the domain and the last cell inside the domain.
- The wall where tangential stresses are included in the momentum conservation equation. The velocity at the wall equals zero.

The ideal gas isentropic flow with the Reynolds-Stress turbulence model has been applied. The RSM turbulence model was chosen as the most accurate in this case. The flow is turbulent due to unsteady excitation and high peak velocity. Mach number for this case was approximately 0.46.

The results were obtained for a 2D structured mesh using axial symmetry. An example of the model mesh is shown in figure 1.

![Fig. 1. Nozzle mesh details.](image)
The results obtained in simulations were spatially averaged at the inlet and outlet to obtain one dimensional mass flow fluctuations. The important parameter is to choose the solution discretization method. The authors conducted simulations for different discretization methods, one so-called standard and the QUICK discretization scheme – which is the scheme dedicated to the quadrilateral and hexahedral meshes, where the subsequent cells can be specified. As the results, for the First- and Second-Order Upwind schemes, in this case are similar, the First-Order Upwind scheme as a less time-consuming was chosen as the standard discretization method.

4. Methodology for the analysis of CFD simulation results

The purpose of this study is to estimate one-dimensional parameters for the nozzle shape influence on the pressure pulsations damping, using CFD simulations. The 3D simulation of the impulse flow excitation at the inlet gives the mass flow response at the outlet in the shape of damped oscillation. After spatial averaging the results of the mass flow at the outlet, the zero-dimensional, time dependent only, solution of the mass flow rate on the outlet is obtained. The signal has a damped harmonic form. In order to determine the damping coefficient, the attenuation curves which are described by the equation:

\[ y(x) = A_0 e^{Bt} \]

have to be matched to the obtained signal.

To adjust the attenuation curves for the received signal an algorithm using the Levenberg-Marquardt (L-M) method was developed. The algorithm is based on the extraction of signal maximum values points of the time intervals equal to one period for the upper curve, and minimum values for the lower damping curve. Then, for the selected points the curve described by equation \( y(x) = A_0 e^{Bt} \) is fit using the Levenberg-Marquardt method.

4.1. Levenberg-Marquardt method

The curve fitting the dataset can be based on a number of different methods, for example, the method of the steepest descent or the Gauss-Newton method. Levenberg-Marquardt algorithm is in fact an iterative algorithm which combines the attributes of the methods of the steepest descend and Gauss-Newton, and at the same time gives the quickest convergence [9][10][11]. The steepest descent method is an iterative optimization without the limitation described by the scheme:

\[ x_{i+1} = x_i - \lambda \nabla \Phi(x_i) \]

and, after selecting the starting point, the algorithm runs by minimizing directional function \( \Phi(x_k - \lambda \nabla \Phi(x_k)) \) with respect to step \( \lambda \), whose length is selected in the order to the largest decrease of the function value in new point. In general, this method is quite slow to converge. To achieve a faster convergence, the Gauss-Newton method can be used. However, the convergence speed of this method depends heavily on the starting point selection. The Gauss-Newton method is described by:

\[ x_{i+1} = x_i - (H(x_i))^{-1} \nabla \phi(x_i) \]

where \( H \) is the Hessian of the function \( \phi \) and there is no need to know the exact value of the Hessian. Kenneth Levenberg combined both methods and proposed a method described in the form:

\[ x_{i+1} = x_i - (H(x_i) + \lambda I)^{-1} \nabla \phi(x_i) \]

where \( I \) is the identity matrix.

The algorithm is based on the determination of the value of function in \( x_{i+1} \), and then the error value is examined in this point. If the error is growing, then the value of \( \lambda \) increases and the calculations repeat; if the error decreases, the step decreases and the calculations are continued. Donald Marquardt introduced an amendment by turning the
identity matrix to the diagonal of the Hessian matrix. The algorithm obtained based on equation (9) is characterized by a rapid convergence and for the tasks with a small number of parameters is much faster than the steepest descent and the Gauss-Newton methods.

\[ x_{i+1} = x_i - (H(x_i) + \lambda \text{diag}[H])^{-1}\nabla \varphi(x_i) \]  

(9)

4.2. Determination of attenuation parameters using the created algorithm

The algorithm is written in Mathcad 15 software and for determining the attenuation coefficients for the signal received in simulation consists of the following steps:

a) The generation of data for the first 64 frequencies is obtained using the Fast Fourier Transform. The signal obtained in the numerical simulations with the superimposed signal for the first sixty-four frequencies is shown in figure 2.

b) For the obtained signal the algorithm determines, in the intervals equal to the period, the maximum values points for which the upper attenuation curve will be designed.

c) For these points the attenuation curve described by equation (5) is matched using the Levenberg-Marquadt method, as shown in figure 2. By using the “genfit” function (which is based on the L-M method) in Mathcad software, on obtaining the damping curve, the program directly calculates exponent B and initial amplitude A₀ values.

d) The known values of exponent B and signal period T allow calculating the number of parameters of the obtained signal.

e) Similar steps and procedures determines the lower damping curve (Fig. 2), thereby obtaining two attenuation curves.

f) The next step is to calculate attenuate parameters:

- Damping coefficient ζ, described as:

\[ \zeta = \frac{B}{\omega} \]  

(10)

- Amplification factor K, described as:
\[ K = \frac{A \omega}{\Delta t m_0 \omega_0^2} \]  

- Natural frequency \( \omega_0 \), described as:

\[ \omega_0 = \frac{A \omega}{\sqrt{1-\zeta^2}} \]  

In the presented example, the values of individual coefficients for the attenuation curves and the values of the calculated parameters are summarized in table 1.

<table>
<thead>
<tr>
<th>Attenuation curve</th>
<th>Upper</th>
<th>Lower</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial amplitude ( A )</td>
<td>0.119</td>
<td>-0.124</td>
</tr>
<tr>
<td>Period ( T )</td>
<td>0.00355</td>
<td>0.00364</td>
</tr>
<tr>
<td>Exponential ( B )</td>
<td>-50.67</td>
<td>-53.23</td>
</tr>
<tr>
<td>Damping coefficient ( \zeta )</td>
<td>-0.029</td>
<td>-0.03</td>
</tr>
<tr>
<td>Gain coefficient ( K )</td>
<td>336.375</td>
<td>-349.832</td>
</tr>
<tr>
<td>Natural frequency ( \omega_0 )</td>
<td>1773</td>
<td>1773</td>
</tr>
</tbody>
</table>

5. Application example

The passive pressure pulsation damping can be obtained introducing the shaped nozzles to the compressor outflow manifold. For the experimental verification of numerical results a special test stand was set up. The test stand contains a DEMAG screw compressor, special outflow installation (where the damping nozzle and the dynamic pressure pulsations sensors are located), an oil separator and the outflow manifold with appropriate metering devices. The place of the assembly of the investigated elements is located 17mm behind the compressor discharge chamber. The test stand is presented in figure 3.

![Fig. 3. Test stand (left) and the Venturi nozzle shape (right)](image)

The investigated nozzle is the Venturi nozzle shape with dimensions as shown in figure 4. Two different nozzle diameters were investigated, for \( d=15 \) and \( d=20 \). The \( L \) dimension for the narrower nozzle is 55 mm and for the second one \( L=51 \). The outer diameter \( D \) was always equal to the outflow pipe diameter and is 35 mm.

The experimental investigations were performed for various speeds of the compressor from 1400 rev/min to 2300 rev/min. The results for different rotational speeds and as one averaged parameter were compared with the results obtained in the numerical simulations.

6. Numerical simulations results

The numerical simulations were performed for three different shapes corresponding to three different configurations: one for the empty outflow pipe and two simulations with a damping nozzle in the outflow pipe. The attenuation curves were fitted using the algorithm described previously, and next the attenuation parameters were computed. The obtained attenuation curves are shown in figure 4.
The results for averaged damping coefficient for two attenuation curves obtained in the simulations for these three shapes are shown in table 2.

<table>
<thead>
<tr>
<th>Element</th>
<th>Damping coefficient $\zeta$</th>
<th>Impulse signal damping compared with empty pipe [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Empty pipe</td>
<td>-0.0265</td>
<td>0</td>
</tr>
<tr>
<td>Venturi nozzle $\varphi$ 20</td>
<td>-0.0295</td>
<td>11.3</td>
</tr>
<tr>
<td>Venturi nozzle $\varphi$ 15</td>
<td>-0.031</td>
<td>17.0</td>
</tr>
</tbody>
</table>

For the impulse flow transient simulation it can be noticed that the pressure dispersion is much wider in the shaped nozzles than in a straight pipe. This effect is responsible for the energy dissipation and hence the impulse flow damping. The pressure maps in 1000 simulation step are shown in figure 5.

In order to show different energy dissipation only the quantitative comparison is presented. The pressure values are similar and they were left out to obtain a better quality of the figure.

7. Experimental investigation results

The experimental investigations were performed at six different compressor rotational speeds, starting from 1400 rev/min with a step of approximately 180 revs per minute. The pressure pulsations damping was calculated as the decrease in the peak-to-peak value compared with this value for the experiment with an empty outflow pipe, as shown in figure 6.

One of the advantages of using shaped nozzles to attenuate the pressure pulsations is damping effectiveness for a wide range of frequency. In figure 7 the pressure pulsations peak-to-peak values for different installation configurations for different frequencies are shown.
Fig. 7. Pressure pulsations peak-to-peak values for different compressor revolution speed. Installation with empty pipe (left), Venturi nozzle ϕ 20 (center) and Venturi nozzle ϕ 15 (right)

As can be seen in the figures the pressure pulsations peak-to-peak value oscillates up to 110% of the minimum values in the investigated revolution speeds. In table 3 a comparison between pressure pulsations value in the installation without any damping elements and with mounted two investigated elements is shown.

<table>
<thead>
<tr>
<th>Element</th>
<th>Average pressure pulsations value [kPa]</th>
<th>Average pressure pulsations damping compared with empty pipe [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Empty pipe</td>
<td>43.43</td>
<td>0</td>
</tr>
<tr>
<td>Venturi nozzle ϕ 20</td>
<td>24.12</td>
<td>44.46</td>
</tr>
<tr>
<td>Venturi nozzle ϕ 15</td>
<td>17.70</td>
<td>59.24</td>
</tr>
</tbody>
</table>

From the experimental investigation results it can be seen that the application of shaped nozzles allows attenuating the pressure pulsations over all the investigated revolution speeds in a similar way. The results obtained in the numerical and experimental investigations, shown in this paper, cannot be directly compared due to the basics of the methods. In the simulation there is a very simplified model (a pipe with a nozzle), and in the experimental investigation we deal with a complex installation composed of a number different objects, like an oil separator, metering orifices etc. The parameters which can relate to each other can be the damping gain (in %) ratio between the attenuation nozzles. This ratio, defined as an average pressure pulsations damping of the Venturi nozzle ϕ 15 to Venturi nozzle ϕ 20 in the experimental investigations is 1.33. For the numerical simulations the ratio is defined as an impulse signal damping of the Venturi nozzle ϕ 15 to Venturi nozzle ϕ 20 is 1.50, which are very similar values.

8. Conclusions

The application of the shaped nozzles or any other passive damping element to attenuate the pressure pulsations requires a tool to estimate its influence on the pressure pulsation as well as the total energy consumption. The proposed method of the estimation of the transmittance parameters allows assessing the influence of any damping element on the pulsation attenuation in the manifold. The transmittance parameters have to be determined on the basis of time dependence only, averaged on the cross section, pressures and mass flows at the inlet and outlet of the investigated element. The application of the Levenberg-Marquadt method, presented in the paper, allows precise and unique estimation of the transmittance parameters, on the basis of the damped oscillation function obtained as a result of the pressure and mass flow averaged on the cross section. These functions are the results of 3D/2D simulation of the impulse mass flow excitation at the inlet of any damping element. The advantage of this approach is that the damping element may be optimized theoretically, before it is introduced into the manifold.

9. References


