TECHNICAL TRANSACTIONS | CZASOPISMO TECHNICZNE
MECHANICS | MECHANIKA
1-M/2016

PIOTR CYKLIS, PRZEMYSŁAW MŁYNARCZYK*

CFD IMPULSE FLOW SIMULATION THROUGH SHAPED NOZZLES

SYMULACJA CFD PRZEPŁYWU IMPULSOWEGO W DYSZACH KSZTAŁTOWYCH

A b s t r a c t

Pressure pulsations in volumetric compressor manifolds are important issues in the compressor plant operation. For pressure pulsation attenuation, mufflers designed using the Helmholtz resonator approach are applied. Nowadays, the variable revolution speed compressors requires new design methods for pressure pulsation damping elements. One of the possibilities to attenuate the pressure pulsations is the introduction of specially shaped nozzles in the gas duct flow. In this paper, the impulse flow simulations conducted in FLUENT/ANSYS. The experimental validation of the simulation results is also presented.

Keywords: pressure pulsations, nozzle, simulation

S t r e s z c z e n i e

Pulsacje ciśnienia są ważnym problemem w instalacjach sprężarek wyporowych. Do projektowania tłumików pulsacji ciśnienia wykorzystuje się metodę Helmholtza Aktualnie coraz większe zastosowanie znajdują sprężarki o zmiennej prędkości obrotowej. Jedną z możliwości jest zainstalowanie dyszy tłumiącej. W artykule tym przedstawiono wyniki symulacji przepływu impulsowego w oprogramowaniu FLUENT/ANSYS Uzyskane wyniki zostały skonfrontowane z wynikami uzyskanymi na stanowisku badawczym.

Słowa kluczowe: pulsacje ciśnienia, dysze, symulacje

DOI:

* Prof. PhD. DSc. Eng. Piotr Cyklis, MSc. Eng. Przemysław Młynarczyk, Institute of Thermal and Process Engineering, Faculty of Mechanical Engineering, Cracow University of Technology.

1. Introduction

Pressure pulsations in volumetric compressor installations are caused by the periodicity of compressor operation. This periodicity induces vibrations, noise and sometimes causes valve failures. In extreme cases, they may cause cracking of the pipelines. The standard method to attenuate pressure pulsation is to apply the Helmholtz theory for the design of mufflers [1, 2]. Such mufflers attenuate only narrow band frequencies close to the designed frequency. In the case of variable revolution speed compressors, this is not efficient since the pressure pulsations have to be attenuated for a wide range of frequencies. This is the reason why other possibilities for pressure pulsation damping are needed [3, 4]. An attenuation method which can be useful is the application of shaped nozzles. It has been found that the shape of the nozzle influences the attenuation of pressure pulsations; however, it also has an impact on the compression power requirement. The nozzle attenuation impact on the pulsation amplitudes is smaller than the Helmholtz resonator. However even 20% of the pulsation attenuation may be sufficient to fill standard requirements in some manifolds. The optimal choice of nozzle dimensions and shape needed to attenuate pressure pulsations requires experimental investigations or the application of a numerical simulation method. Experimental investigations have many disadvantages, and cannot be used in the installation design process for practical reasons. The worked-out CFD-based method consists of two simulations: an impulse excited flow to assess pulsation attenuation; a steady flow through the nozzle to assess increases in pumping power. In the literature, investigations regarding numerical nozzle fluid flow can be found. Some published results are shown in [5, 6 &7]. In this paper, impulse flow simulations using various models and methods conducted in FLUENT/ANSYS are compared. The experimental validation of the simulation results is also described.

2. Numerical model

All of the investigated simulations were conducted using the FLUENT/ANSYS software. For all cases, the 2D axisymmetric, compressible ideal gas isentropic flow model was used with the Reynolds-Stress turbulence model. The simulation was transient with a time step of 2-6 seconds. Impulse flow damping using the QUICK spatial discretisation scheme was conducted for different damping elements. The investigated nozzle shapes were the Venturi orifice, Venturi nozzle and hyperboloidal nozzle with an inner diameter of d = 15 mm. The investigated shapes are shown in Figure 1. The outer diameters of the nozzles are 35mm as the installation pipe has the same diameter. For the purpose of verification of the simulations results, experimental investigations were conducted for the same nozzle shapes.

The results obtained in simulations were spatially averaged at the inlet and outlet cross-section to obtain only time-dependent mass flow fluctuations. The output signal has a dampened harmonic form. In order to determine the damping coefficient, the attenuation curves were described by the equation:

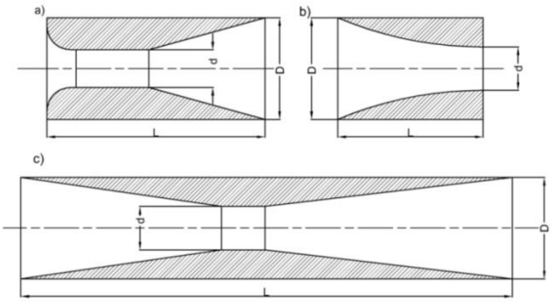

Fig. 1. The investigated geometries: a) Venturi nozzle, b) hyperboloidal nozzle, c) Venturi orifice

$$y(x) = A_0 \cdot \mathrm{e}^{B \cdot t} \tag{1}$$

where

$$B = \frac{\omega}{\zeta} \tag{2}$$

$$\omega = \frac{2 \cdot \pi}{T} \tag{3}$$

The curves have to be matched to the obtained signal.

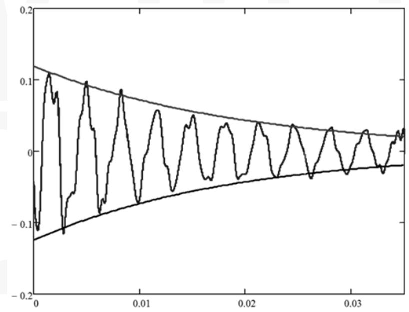

Fig. 2. Signal with both attenuation curves obtained using L-M method

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. The same procedure was applied for minimum values for the lower damping curve. Then, for the selected points, the curve described by equation $y(x) = A_0 \cdot \mathrm{e}^{B \cdot t}$ is fitted using the L-M method. An example of the attenuation curve fit is shown in figure 2.

The signal analysis, using the L-M method, requires further improvement to obtain the damping factor. If the simulation time is too short, the damping curve will not represent the signal attenuation. Besides the first high amplitudes can be misleading and the signal harmonizes after some periods. The signal after the fast Fourier transform was analysed only for a selected number of the first frequencies.

The numerical model has to be simple and fast to compute; therefore, only four necessary boundary conditions were applied:

- Mass flow Inlet – where the mass flow impulse excitation is 0.1 kg/s. The duration of the impulse excitation is equal to the first, one time step. For the rest of these transient simulation steps, the mass flow inlet boundary is defined as zero. The backflow temperature, if it does occur, is defined as 350 K – close to the flow temperature during experiments.
- 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 backflow temperature, if it does occur, is defined as 350 K.
- Wall – where the tangential stresses are included in the momentum conservation equation. There is no slip condition defined.
- Axis – to determine the appropriate physical value for a particular variable at a point on the axis, the software uses the cell value in the adjacent cell.

An appropriate mesh was prepared for each geometry. For the 2D axisymmetric simulations, the model contains the nozzle shape and a straight pipe fragment. The same geometry as on the test stand installation. In Figure 3, the simulated elements of the mesh are presented.

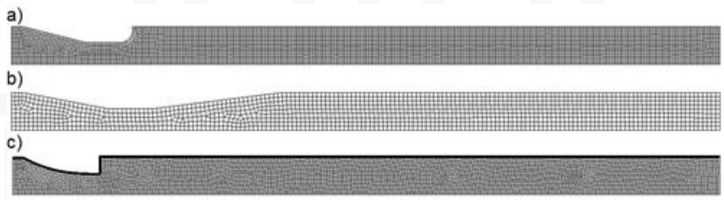

Fig. 3. The mesh models for a) Venturi nozzle, b) Venturi orifice, c) Hyperboloidal nozzle

The geometry and mesh were the same for each shape for all simulations. In Figure 3, three different but still correct meshes are presented. The results of the impulse flow simulation is not strongly dependent on the grid size, the near wall layers addition etc [8]. The most important issue was to obtain the structured mesh, according to the QUICK discretisation scheme requirements. For the presented geometries, a good quality mesh was obtained for approximately 3,500 cells with a cell size ~1 mm.

3. Simulation results

For the selected geometries, numerical simulations were conducted. For each element, the results were spatially averaged at the outlet cross-section to obtain time-dependent mass flow fluctuations [9]. The damping coefficients were estimated using the L-M method with three different approaches. For all approaches, the signal was modified to a number of first frequencies using fast Fourier transform. The data generation for the first 64 frequencies is obtained using two methods, The third method was used only for the first eight frequencies. In the first method (Method 1) the obtained signal is analysed without any limitations, for all periods. The second method (Method 2) has some limitations, the signal is analysed for only nine periods, and the first lower amplitude peak is not taken into consideration as is shown in Figure 4.

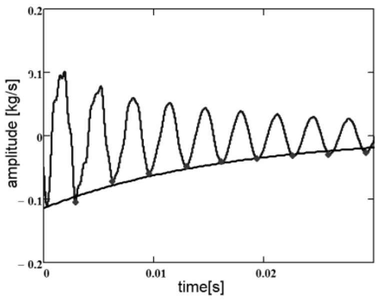

Fig. 4. The points defining the lower attenuation curve for the nine-period analysis, disregarding the first lower amplitude peak

The first eight frequencies had the most significant impact on the signal in the third method (Method 3), therefore only the signals for first eight frequencies were analysed. In the Fig. 5 two signals are presented: the signal obtained in simulation and the signal for the first eight frequencies after FFT.

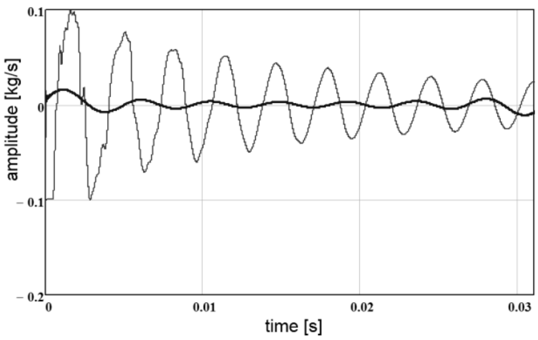

Fig. 5. The signal: from simulations; after the FFT based reduction

The damping factor calculated for the damping curves obtained in the simulations for the different geometries using the described approach are shown in Figures 6-8. In these figures, the results for the impulse propagation through the nozzles are related to the result obtained for the straight, empty pipe geometry. To verify the obtained results, experimental investigations was performed.

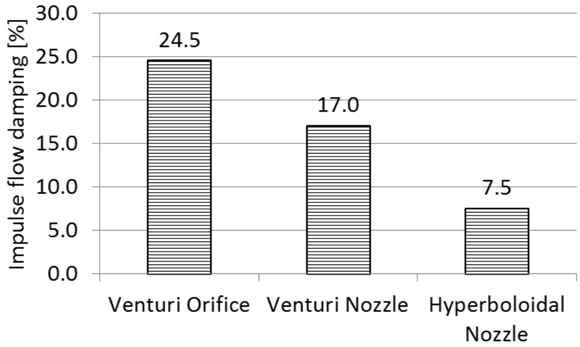

Fig. 6. The impulse flow damping related to the empty pipe (Method 1)

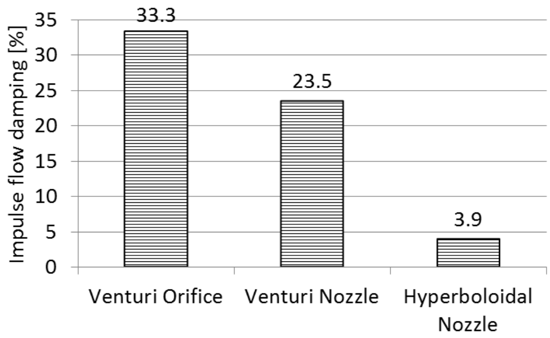

Fig. 7. The impulse flow damping related to the empty pipe (Method 2)

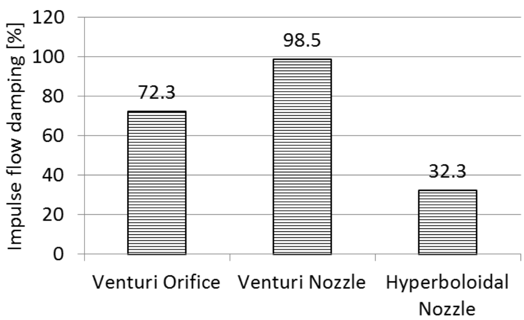

Fig. 8. The impulse flow damping related to the empty pipe (Method 3)

4. Experimental verification

The experimental investigations were performed on a special test stand which was devised to measure, in particular, pressure pulsations in the screw compressor discharge manifold. The test stand and the tests themselves were described in detail in publications [9, 10]. The effect of the nozzle on the pressure pulsations was measured on the test stand for different compressor revolution speeds. In this paper, the average values of the pressure pulsation attenuation for three different geometries are presented. In Figure 9, the pressure pulsations peak-to-peak value decrease for different applied nozzles are shown, related to the empty pipe pulsations for the compressor revolution speed equal to 1615 rev/min.

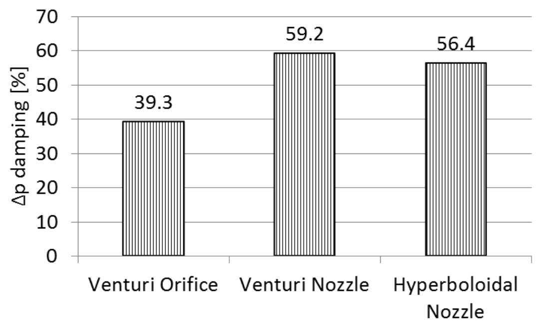

Fig. 9. The pressure pulsations peak-to-peak value decrease compared to the empty pipe pulsations

An exact match of simulation and experimental results cannot be expected in terms of numbers, since the simulation results for the impulse function are related to all frequencies The experimental results are limited only to the chosen investigated frequencies. However, in the case of the third method, the relationships between the compared results for each shape and dimension are similar for both the simulation and experiments. It can be seen that the influence of the hyperboloidal nozzle on the impulse flow damping is under-estimated for all three methods used in the analysis. In this case, the numerical model has to be improved.

5. Conclusions

Numerical simulations of the impulse flow damping coefficient for three different nozzle shapes and three different methods of analysis of the obtained results were presented. The damping factors for the three geometries, calculated using different approaches, were compared with the pressure pulsation damping peak-to-peak values obtained experimentally. It was shown that the method of the 2D simulation results analysis affects the calculated damping coefficient; however, the numerical model still has the highest impact on the coefficient value. The results obtained, using shown numerical simulations and analysis method, corresponds with the experimental results of nozzle pulsations damping. Further method development is necessary to obtain qualitative comparison.

Nomenclature

B – damping coefficient for exponential curve [Hz]
ξ – damping factor [dimensionless]
ω – natural frequency of the system [Hz]
A_0 – initial amplitude value [Pa]
T – basic period of free pulsation frequency [s]

References

[1] Andersen K.S., *Analyzing Muffler Perfomance Using the Transfer Matrix Method*, COMSOL Conference, Hannover 2008.
[2] Ma Y.-C., Min O.-K., *Pressure calculation in a compressor cylinder by a modified new helmholtz modelling*, Journal of Sound and Vibration, vol. 243, 2001, 775-796.
[3] Lokitsangtong M., Murakami S., Iijima T., *Acoustical Characteristics of Single-Resonator-Type Silencers*, Proceedings of the School of Engineering, Tokai University, Series E 31, 2006, 39-44.
[4] Bilawchuk S., Fyfe K.R., *Comparison and implementation of the varoius numerical methods used for calculationg transmission loss in silencer systems*, Applied Acostics, vol. 64, 2003, 903-916.
[5] Sadamoto A., Tsubakishita Y., Murakami Y., *Sound attenuation in circular duct using slit-like short expansion of eccentric and/or serialized configuration*, Journal of Sound and Vibration, vol. 277, 2004, 987-1003.
[6] Herrmann J., Koreck J., Maess M., Gaul L., von Estorff O., *Frequency-dependent damping model for the hydroacoustic finite element analysis of fluid-filled pipes with diameter changes*, Mechanical Systems and Signal Processing, vol. 25, 2011, 981-990.
[7] Lee S.-H., Ih J.-G., *Effect of non-uniform perforation in the long concentric resonator on transmission loss and back pressure*, Journal of Sound and Vibration, vol. 311, 2008, 280-296.
[8] Cyklis P., Młynarczyk P., *Nozzle suppressed pulsating flow simulation issues*, Technical Transactions, 1-M, 2015, 21-29.
[9] Cyklis P., Młynarczyk P., *An innovative simulation method for the estimation of the nozzle pressure pulsation attenuation*, Journal of Vibration and Control, 2015, doi: 10.1177/1077546315619534.
[10] Młynarczyk P., *Experimental investigations of pressure pulsations damping in screw compressor system*, Proceeding of the Student's conference, Prague 2014.

Acknowledgement

The presented research was conducted partly under the **PB 4801/B/T02/2010/39** research grant.