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CFD ANALYSIS OF THE IMPULSE FLOW DAMPING BY THE SPECIALLY SHAPED NOZZLE

ANALIZA CFD TŁUMIENIA IMPULSU PRZEPŁYWU PRZEZ DYSZĘ SPECJALNEGO KSZTAŁTU

A b s t r a c t

The vibrations and noise caused by pressure pulsations are one of the major problems in volumetric compressors manifolds. There is still no easy solution for this problem. Passive damping of those pulsations is possible using specially shaped nozzle placed in place of the straight tube. Experimental analysis of the pressure pulsations damping caused by a nozzle is possible however only some arbitrary chosen nozzle shapes can be investigated. The analysis of damping of the impulse flow by the nozzle using CFD simulation is more general and gives the possibility to estimate the nozzle influence on the pulsating flow in theoretical way, so many different shapes can be investigated. In this paper examples of impulse flow damping factors for three different nozzles have been shown.

Keywords: *CFD simulations*, *Pressure pulsations damping*

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

Wibracje i hałas powodowane pulsacjami ciśnienia są jednym z głównych problemów pojawiających się przy eksploatacji układów sprężonego powietrza. Wciąż nie ma prostego rozwiązania dla tego typu problemów. Pasywne tłumienie tych pulsacji można osiągnąć wykorzystując dysze o specjalnym kształcie. Artykuł prezentuje próbę symulacji impulsu jednostkowego poprzez trzy typy elementów tłumiących oraz opisuje wpływ kształtu elementu na pulsacje ciśnienia występujące w układzie.

Słowa kluczowe: *symulacje CFD*, *Pulsacje ciśnienia*

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Nomenclature

ξ – damping factor
ω – natural frequency of the system
τ_0 – response time
T – period

1. Introduction

The main problem, addressed by this article, is the possibility of pressure pulsations attenuation in the compressors manifold system. Periodic work flow of the volumetric compressors cause the pressure and mass flow pulsations. The pressure pulsations attenuation in the volumetric compressor manifolds is one of the major design problems both in small refrigerating compressor units due to noise, but also in high capacity volumetric compressors due to vibrations caused by pulsations [1]. Development of the design is, among other things, a result of advances in computer modelling, computational fluid dynamics allows to get answers to questions on how to optimize constructions. This brings up the need to analyse the phenomena of pulsation wave propagation and it's damping. In the smaller compressors we can't use large muffling systems- the one way to suppress pressure pulsations is to install choking elements at the start of the compressor outflow pipe. This attenuates the pulsations but also increases the compression power. So the optimisation is required.

Computer simulation may be used to investigate the influence of different muffler types on pressure pulsations [7, 8]. CFD simulations have become also a working tool for designers and maintenance engineers [2–5]. There are several approaches such as generalised Helmholtz method based on CFD simulations, 2D transfer matrix [6] etc. In this paper simplified analysis of one most important parameter resulting from CFD calculations is shown.

2. Compressor pressure pulsations

Pressure pulsations frequency result from the compressor forced flow, what means that first basic harmonic results from multiplying number of revolutions and amount of volumes extruded during one rotation of the drive shaft. Therefore, the strongest excitation comes from one cylinder, single-acting reciprocating compressor. Extorting is still not critical and sufficient to evaluate the propagation of pressure pulsation. Depending on the system configuration, harmonic waves of pressure pulsations can strengthen or weaken each other. In the system occurs a lot of phenomena such as: wave reflections, interference of reflected waves, pulsation damping, thus all of the phenomena associated with the internal acoustics. A pulsating gas stream significantly influences the functioning of the entire compressor manifold, especially affected in a negative way to the compressor systems. Pressure pulse waves, carry a significant acoustic energy which is a system energy loss, but in the special case in the compressor dynamic charge it can be positively used. The main problems which pressure pulsations cause are [3, 4]:

– System vibrations and sometimes even mechanical damage.
– Aerodynamic and mechanical noise.
– Reduction in displacement chamber filling with wrong pressure phase in the suction cycle.
– Resonance with the work of the valves causing them to wear faster.
– Problems with the operation of the associated equipment, for example: flow meters, pressure gauges etc.

Author performed a series of experiments, using various of choking elements to decrease pressure pulsations. After the experiment, according to the results, author choose three different shapes to use in the computational simulations.

3. Muffling elements

Passive damping pressure pulsations is possible using specially shaped nozzle placed in place of the straight tube. Experimental analysis of the pressure pulsations damping caused by a nozzle is possible however only some arbitrary chosen nozzle shapes can be investigated. Author made a series of measurements for different nozzle shapes.

The three different shapes, which was chosen after this measurements, is venturi orifice with inner diameter ϕ15, venturi nozzle with inner diameter ϕ15 and hyperboloidal nozzle with inner diameter ϕ20. Simulations was performed for the pipe with this three types of choking elements and, obviously, for the empty system pipe.

3.1. Venturi orifice

Ventouri orifice ϕ15 is the longest element used in the simulation, as the geometrical dimension as shown in Figure 1a. The data obtained during the experiment shows that this element damped pressure pulsations well, with not much loss in the system fluid flow rate.

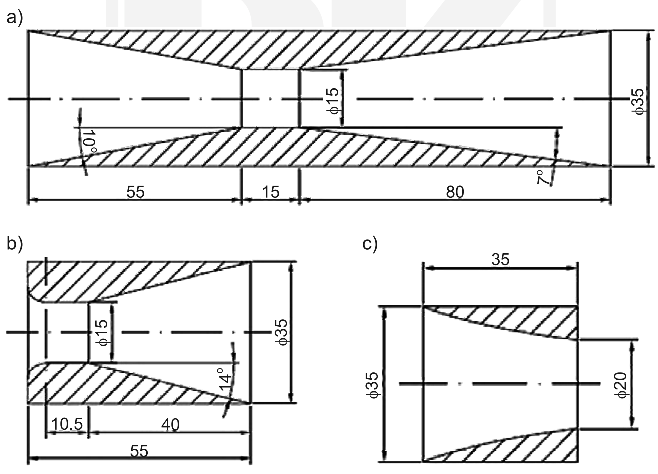

Fig. 1. Shape and main dimensions of the a) venturi orifice, b) venturi nozzle, c) hiperboloidal nozzle
Rys. 1. Kształt i podstawowe wymiary a) zwężki venturiego, b) dyszy venturiego, c) dyszy hiperboloidalnej

3.2. Venturi nozzle

Venturi nozzle ϕ15, from data obtained in the experiment, is characterized by very similar parameters of pressure pulsations damping and flow rate like venturi orifice. Shape of this element is harder to made, however the advantage of this solution is the smaller dimensions which is important in the small compressors systems.

3.3. Hyperboloidal nozzle

The last choking element is hyperboloidal nozzle ϕ20. Experiment shows that this type of choking elements haven't big influence on pressure pulsations. Furthermore, this nozzle have very complicated geometry which is expensive to produce.

4. Simulations results

Computational simulations were performed in the Ansys Fluent environment. The most common approach in analysis of the phenomenon of pressure pulsation is the use of the ideal gas equation of state.

To the simulation were used density-based, transient model, with using energy equations and inviscid fluid model which was chosen as ideal gas. As impulse, which had represented dirac pulse at the pipe inlet in the first time step, was chosen mass flow impulse with a value of 0.1 kg/s. The end of pipe was declared as open pressure outlet. As the velocity of wave propagation is dependent on the temperature, for both, inlet total temperature and outlet backflow total temperature has been declared the same as the environment temperature and it is 350 K. Size of the grid elements was assumed for every one example individually, but as default minimum cell size was chosen 1mm, which gives a grid density of around 1 cell per mm^2. There was used a default Fluent grid type for this type of simulations. According to the cell size and transient time model, time stepping method has been chosen fixed with duration of time step equal to $2 \cdot e^{-6}$ s. To gain desirable simulation time 0.01s was needed 5000 time steps. In the model definition also declared axisymmetric calculations, so geometry used in the simulations was only the profile of nozzle shape in the tube. Convergence of every time step solutions was usually achieved around 60–70 iterations step, with declaration of 100 maximum iterations per time step. For every simulations, geometry of the whole pipe had length 327,3 mm and all elements were placed 5 mm behind pipe inlet. In the Figure 2. are shown results of pressure contours for two elements (venturi orifice and hyperboloidal nozzle).

In the Figure 2. are shown pressure contours in two different elements but in the same time: after 1000 and 4000 time steps. There is visible clear difference in the process of propagation of the pressure pulsations. It could be said, that in hyperboloidal nozzle there is a spatial pressure propagation while in the venturi orifice it looks like a one-dimensional propagation.

In the Figure 3. Is presented attenuation of the pressure pulsations (in the pascal unit) in the impulse flow in four choking elements. Graphs shows pressure value at the pipe outflow. To present how the pressure pulsations are suppressed, on the graphs in Figure 3. are drawn damping characteristic curves. They are characterized by the equation [1, 3]:

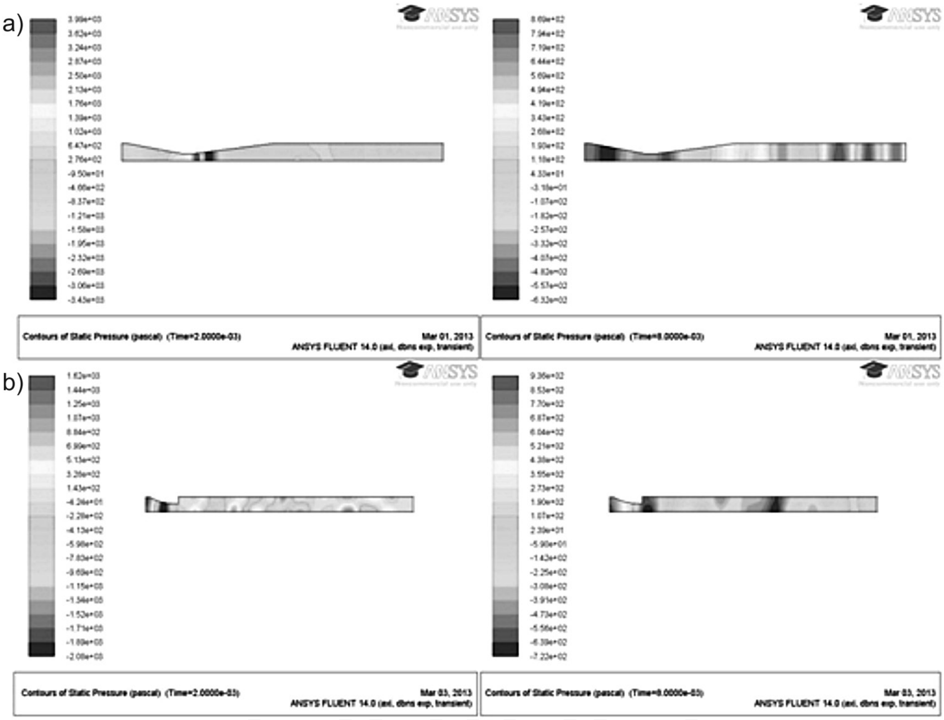

Fig. 2. Pressure map in 0.002 (left) and 0.008 (right) second: a) venturi orifice, b) hyperboloidal nozzle

Rys. 2. Mapa rozkładu ciśnienia w 0.002 (z lewej) oraz 0.008 (z prawej) sekundzie: a) zwężki venturiego, b) dyszy hiperboloidalnej

$$y(\tau) = A_0 \cdot e^{-B(\tau-\tau_0)} \tag{1}$$

where:

$$B = \xi\omega \tag{2}$$

where omega is:

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

Period of every pressure pulsations is known. It can be taken from the graphs or simulations results table.

In the present case author could manipulate the B value to obtain attenuation curves, by matching them to the pressure pulsation signal. When the value of the exponent of damping characteristic equation is known, and the natural frequency of the system is known, we can deduce from equation (3) the damping coefficient ξ. Damping factor and exponent of the damping characteristic equation for every simulations are shown in the Table 1.

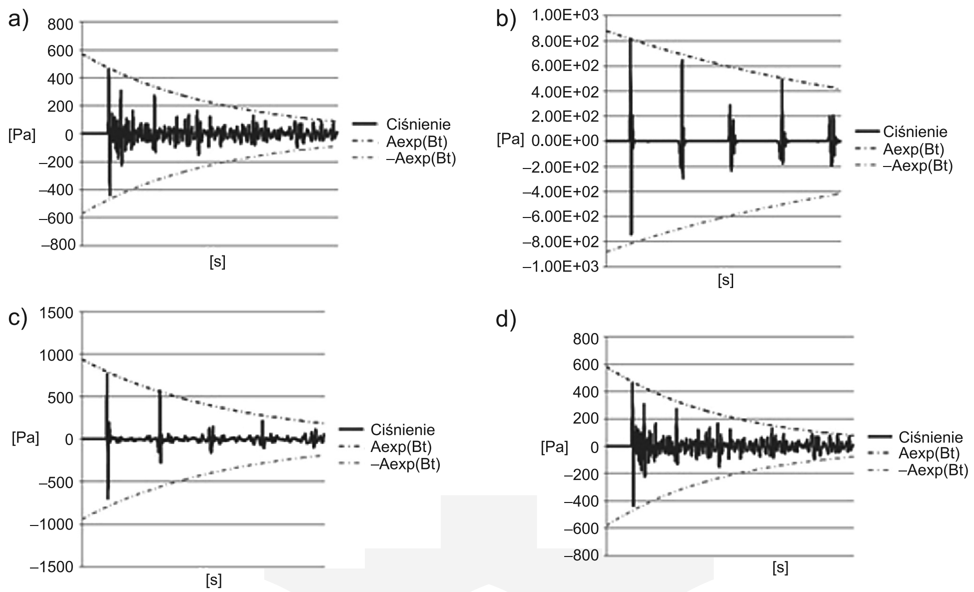

Fig. 3. Pressure pulsations graphs with damping characteristic curves for: a) hyperboloidal nozzle, b) empty pipe, c) venturi orifice, d) venturi nozzle

Rys. 3. Pulsacje ciśnienia z krzywymi charakterystyki tłumienia dla: a) dyszy hiperboloidalnej, b) pustej rury, c)zwężki venturiego, c)dyszy venturiego

Table 1

Value of the exponent of the damping characteristic equation *B* and the damping coefficient ξ

Element	ξ	*B*
Venturi orifice φ15	–0.056	–195
Venturi nozzle φ15	–0.057	–242
Hyperboloidal nozzle φ20	–0.053	–225
Empty pipe	–0.025	–90

The simulation result obtained damping coefficients at a similar level. As it can be noticed damping in the empty pipe is much smaller than in the other elements, which looks natural and correct. Results for two venturi choking elements looks good, because damping percent for this types of mufflers with the same diameter obtained very close to each other also in the experiment. However, if we look at the results of the experiment the ξ coefficient for hyperboloidal nozzle is too close to the other two elements with smaller inner diameter.

5. Conclusions

The main problem in conducting simulations is the discretization and selection of the calculation method. Especially, for hyperboloidal element, the most difficult problem was to precise design the shape of the nozzle, and then make the appropriate discretization of this element. Anyhow the presented calculations shows that the impulse flow propagation through the choking element may by an interesting and simple tool to assess the possibility to utilise specially designed nozzle shape for pressure pulsation attenuation. The experimental investigation carried by the author are not included here due to the space limits and subject of the paper, but there is a correlation between the experimental results and calculated results presented in this paper.

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