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Influence of shell shape on flow and acoustic parameters of a steam silencer

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Abstract Steam discharge produces noise due to rapid expansion and a temperature drop of ejected steam. This is why steam silencers are used to change one-stage into multi-stage expansion, which reduces the intensity of pressure and temperature drop during this process and shifts emitted noise into higher frequencies, which are easier to dampen. This paper presents a flow-acoustic numerical model of a steam silencer. It is meant to help to obtain a precise analysis of phenomena occurring in steam silencers and improve the process of designing this type of device. The model described in this paper was based on the parameters of a real working unit manufactured in the Institute of Power Engineering – Thermal Technology Branch. Most of the steam silencers are designed based on construction guidelines that have not been changed for a long time. This restrained an increase in the acoustics efficiency of the steam silencers. An improvement of their flow and acoustic properties allows for the development of smaller, more efficient, and lighter construction. The current version of the model was used for the analysis of flow and acoustic changes which occur after modifying the lower region of a shell of the steam silencer. The proposed modification allowed for a 19% increase in mass flow rate through the silencer and noise reduction in the low-frequency range.

Keywords: Coustics; CFD; Noise emission; Steam silencer

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Nomenclature

c_0	–	speed of sound in the surrounding medium, m/s
D	–	diameter of an outlet cross-section, m
IL	–	insertion loss, dB
L_p	–	sound pressure level, Pa
LD	–	sound level difference, dB
M	–	Mach number
$\dot{m}$	–	mass flow rate, kg/s
P	–	acoustic power of a jet, dB
p_{inc}	–	acoustic power of an incident pressure wave, dB
p_{trans}	–	acoustic power of a transmitted pressure wave, dB
S_{in}	–	inlet cross-sectional area, m
S_{out}	–	outlet cross-sectional area, m
T	–	absolute temperature of steam at the edge of exhaust pipeline, K
TL	–	transmission loss, dB
u_j	–	discharge velocity, m/s

Greek symbols

ξ	–	effectiveness constant
ρ_c	–	characteristic impedance of a medium, $\text{kg/s} \cdot \text{m}^2$
ρ_j	–	fluid density, kg/m^3
ρ_0	–	density of the surrounding medium, kg/m^3

Acronyms

CFD	–	computational fluid dynamics
LES	–	large eddy simulation
RANS	–	Reynolds-averaged Navier–Stokes
TPP	–	thermal power plant

1 Introduction

Noise is unwanted or undesirable sound produced by many process control equipment. At the thermal power plants (TPP) for example, noise is emitted from almost all equipment. The source of the most intense noise at TPPs is steam emissions. At TPPs steam is used because of high availability, high heat capacity, large range of operating temperatures, clear pressure-temperature dependence. An additional advantage is the possibility of using steam as a heat carrier in the form of wet steam or as an insulator in form of superheated steam. Process steam is produced in steam boilers (in power plants and combined heat and power plants), special municipal installations or clean steam generators. Steam produced in this manner is characterized by high pressure, often higher than 10 MPa and temperature often exceeding 500°C [1]. Ejecting steam with such high pressure and temperature into the atmosphere leads to rapid expansion which is accompanied by a high level of noise around 140 dB. Because of the placement at the elevation (tens of meters above the ground) of the steam ejection installation, the noise is spread omnidirectional in large areas. The noise spectrum of steam discharge consists mostly of low frequencies noise, which is very poorly dampened by the air [2]. It also poses a significant threat to the life and health of people [3].

There are several approaches for calculating the noise from steam discharges. Lighthill's theory is the most widely used. According to this theory, the acoustic power of a jet can be calculated as

$$P = \frac{k(\mathrm{M})\rho_j^2 u_j^n D^2}{\rho_0 c_0^m}\,, \tag{1}$$

where ρ_j is the fluid density, u_j is its discharge velocity, D is the diameter of the outlet cross-section, $\mathrm{M} = u_j/c_0$ – is the Mach number, ρ_0 is the density of the surrounding medium, $k(\mathrm{M})$ is the coefficient that takes into account the convection effect (determined experimentally), and c_0 is the speed of sound in the surrounding medium, n and m are the acoustic power coefficients experimentally determined dependent on Mach number (e.g., $n = 6$, $m = 3$ when $\mathrm{M} < 0.5$ – 'sixth power' law) [4].

Lighthill's analogy is based on the conversion of the kinetic energy of turbulent moles (acoustic quadrupoles) into the acoustic energy, but for the steam jet, the relative role of noise components has not been fully understood. This is why an exact expression for the total acoustic power from a steam jet cannot be written without involving experimental data. One of these approaches is based on the dependence of acoustic energy on the jet mechanical energy [5]

$$P_{\mathrm{acoustic}} = \xi P_{\mathrm{mech}}\,, \tag{2}$$

where ξ is the effectiveness constant.

However, Dragun *et al.* [6] introduced the acoustic efficiency, η, instead of the effectiveness constant and defined the formula for calculating the mechanical power, P_{mech}, of the jet as

$$P_{\mathrm{mech}} = 0.5\,\dot{m}u_j^2\,, \tag{3}$$

where $\dot{m}$ is the jet mass flow rate.

This dependency was used in the following formula [7]:

$$L_p = 10\log\left(0.5\eta\dot{m}\frac{u_j^2}{10^{-13}}\right), \tag{4}$$

where $\eta = 0.003{-}0.006$.

The other approach for determining the level of acoustic power is to calculate it using the following formulas, which were obtained from processing experimental data [8,9]:

$$L_p = 17\log\dot{m} + 50\log T - 24.5\,, \tag{5}$$

$$L_p = 81.4 + 10\log\dot{m} + 20\log u_j\,, \tag{6}$$

where T is the absolute temperature of steam at the edge of the exhaust pipeline.

The traditional control of noise emissions is realized by passive techniques based on dissipative and reactive silencers. Dissipative components (using sound-absorbing material) provide balanced noise reduction over a broad frequency range. The reactive components of silencer, using resonant reflections within tuned multihole chambers, provide peak noise reduction in a lower frequencies band. Steam silencers are used to reduce noise emitted by steam jet, especially in a low-frequency spectrum. The principle of steam silencer work is to replace single-stage expansion outflow to the atmosphere into a multi-stage process with subcritical pressure drops inside the silencer. Due to the wide range of applications, the silencers for steam jet must usually be customized to meet the needs of customers. Nowadays, nearly 40% of designed silencers require customization. Such a high degree of customization makes the product design a demanding task.

The effects of construction parameters on silencer performance can be assessed by computational fluid dynamics (CFD) simulations before a prototype is produced. Several studies focused on using CFD methods to investigate the aerodynamic noise in steam jet silencers. Different configurations of simple expansion chamber silencers were modeled to determine gas flow and also their acoustic parameters [10]. A CFD analysis was also carried out to investigate the relations of porosity, flow velocity, and diameter of the holes with the pressure loss in a cross-flow perforated silencer [11].

The specific mechanism of noise formation and a new method of predicting the noise characteristics of steam jets are presented in [12]. In this paper, the large eddy simulation (LES) method was recommended instead of the

Reynolds-averaged Navier–Stokes (RANS) method tested for different turbulence models (e.g., SST – shear stress transport). According to this study, the LES model can calculate the coherent vortex structures of steam emission and allows to determine the area of generation of noise and the acoustic center of the steam ejection. However, resolving a part of the turbulence spectrum scales makes this approach demanding computationally [13].

The model studied in this work is based on the functional construction of the steam jet silencer manufactured in the Institute of Power Engineering – Thermal Technology Branch. The most significant flaw of this type of device is its high weight. The silencer, in particular the expander, works in a high-pressure environment, so its walls need to be made of thick, durable, and consequently heavy material. Improving the flow and acoustic properties can lead to a reduction in the size and weight of a silencer, which is required to obtain a desired sound pressure level emitted by the installation. In this study, the influence of the construction change of the shell of a steam jet silencer on the acoustic parameters was tested. Two variants of silencers were studied (Fig. 1). The first one had a horizontal wall of the shell bottom, and the second design had the modified shell having a 30° slope in the lower region.

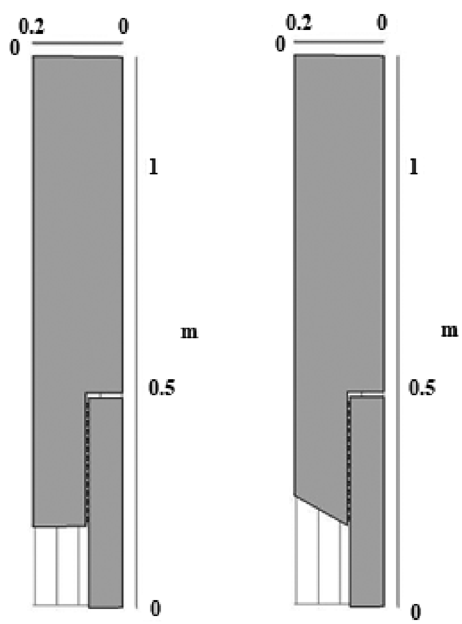

Figure 1: Domains of studied silencers variants.

2 Methodology

The numerical simulations of the steam flow and noise emission were carried out using the commercial finite element analysis, solver and general-purpose simulation software Comsol Multiphysics version 5.6 [17]. The diameter of inlet pipe of the steam jet silencer was equal to 152 mm and the diameter of outlet pipe was 432 mm. The diameter of the sieve holes was equal to 8 mm, the thickness of the wall of the expander was equal to 5 mm, and the height of the expander was equal to 478 mm. Calculations were performed on a domain that represents 1/25th circumferential section of the silencer. This simplification was possible due to the fully repeatable geometry of the domain. The computational mesh consists of tetrahedron cells, which were refined in the holes of the sieve (Fig. 2). The prismatic mesh elements were used in the wall vicinity to properly model the boundary layers. The yplus parameter of the model was in a range from 10 to 60, however, in the most crucial area of the expander, it was around 10.

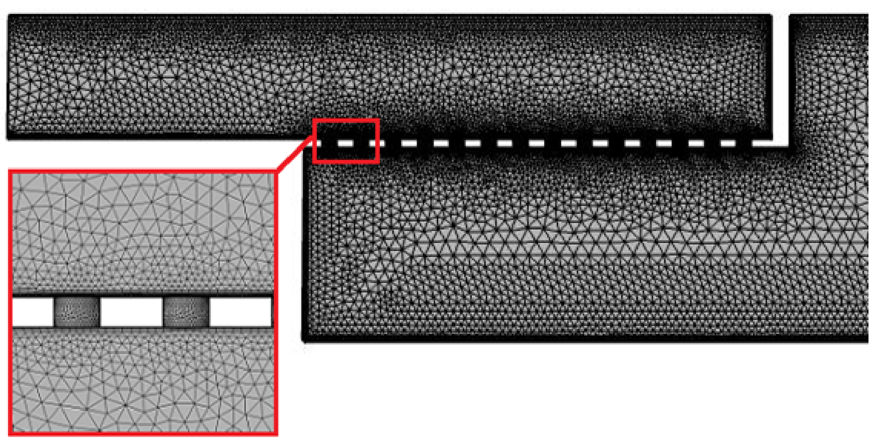

Figure 2: Mesh of the steam silencer model.

Calculations were performed assuming stationary conditions. Steam was considered as compressible gas to simulate properly fluid moving with a velocity higher than Mach number of 0.3. The RANS approach was used because of turbulent fluid flow in a domain. The k-ε turbulence model for high mach number flow was selected with the automatic wall treatment.

Boundary conditions were set according to the working conditions of the steam silencer on which the model was based. The flow at the inlet and outlet was subsonic. Two separate inflow parameter variants were used to reproduce conditions present in the real working unit:

1) total pressure $p_{\text{inlet}} = 3.5$ bar;
2) mass flow rate $\dot{m}_{\text{inlet}} = 0.0667$ kg/s for the 1/25th section of the silencer.

Those parameters were selected to obtain critical and subcritical flow downstream of the sieve holes. In both cases the temperature inlet boundary condition was set to 135°C and the turbulence intensity of 5% was selected. At the silencer outlet, a pressure of 1.02 bar was applied. The boundary condition at the walls was set as no-slip and adiabatic. At the domain sides flow symmetry was set for the considered 1/25th section.

The second-order schemes were used for the discretization of the mass, momentum, and energy equations. Good quality of the iterative solution was reached, with the residuals of governing equations below 10^{-4}.

To simulate the acoustic parameters for the studied silencer taking into account CFD solution results, the linearized Navier–Stokes equations in the frequency domain were applied. Background acoustic field, whose parameters were obtained from the flow part of the simulation was selected as noise generator. The flow fields of pressure, velocity, density, temperature, and turbulent viscosity were mapped onto the acoustic mesh. The generated mesh was designed taking into consideration that results were to be obtained for 1/3 octave bands and the highest frequency tested was equal to 10000 Hz.

The dependence of the mesh on calculated values was verified using three different mesh sizes and checking the mass flow rate and the total acoustic pressure at the outlet. For this test the inlet conditions were set for the total pressure equal to 3.5 bar and temperature of fluid equal to 135°C.

Table 1: Mass flow rate and total sound pressure level values for the mesh comparison.

Mesh No.	Number of elements	Mass flow rate (kg/s)	Total sound pressure level (dB)
I	1322669	0.20309	73.7
II	2213472	0.20309	73.9
III	3458198	0.20314	74.5

It can be seen that the mesh refinement has a negligible influence on mass flow rate. We can observe a small increase of the total sound pressure level present at the outlet of the silencer, however referring to current regulations for in-field acoustic measurements the discrepancies can reach 2.7 dB, so the obtained differences may be treated as negligible [14]. Taking this into consideration mesh I (presented in Fig. 2) was selected for further examination to reduce the simulation time.

3 Results and discussion

3.1 Aerodynamic studies

Two silencer configurations were examined numerically. The basic variant was designed according to the guidelines currently used in the Institute of Power Engineering – Thermal Technology Branch. It was also used for the mesh validation test. The second model was created with modification in the lower region of the shell in form of a slope. This modification is assumed to reduce a recirculation present in the steam silencer resulting in a more uniform flow in the expansion chamber.

In Figs. 3 and 4 it can be observed that for both inlet boundary conditions, the recirculation is present in the lower part of the basic silencer design. For the pressure inlet condition $p_{\text{inlet}} = 3.5$ bar, the supersonic flow can be observed downstream of the sieve holes. The simulation also confirms that the proposed modification of the lower part of the silencer shell highly reduces the size of the recirculation for the subsonic expansion case (Fig. 3). For the case with the pressure inlet condition $p_{\text{inlet}} = 3.5$ bar, where the supersonic flow is observed downstream of the sieve holes, the recirculation was completely removed (Fig. 4). There is no significant mod-

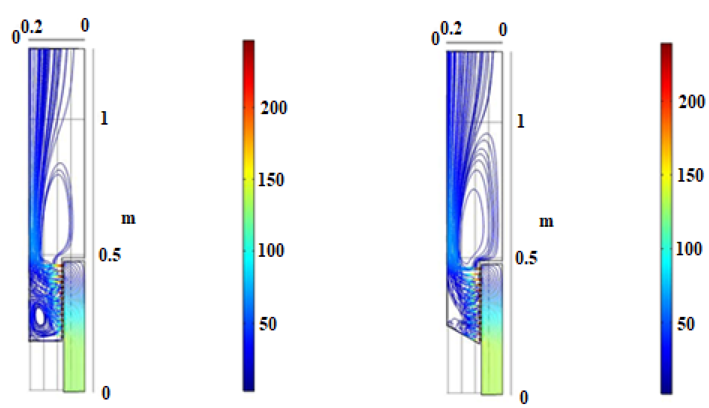

Figure 3: Streamline plots of velocity (in m/s) for the basic (left) and modified (right) design for inlet mass flow rate $\dot{m}_{\text{inlet}} = 0.0667$ kg/s.

ification to the flow structure in the upper part of the silencer, where the steam main flow is restricted to the region near the wall and the center is occupied by a huge recirculation.

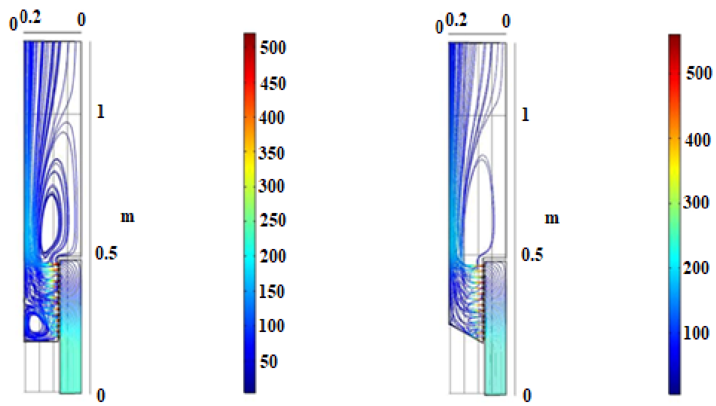

Figure 4: Streamline plots of velocity (m/s) for the basic (left) and modified (right) design for the total pressure at the inlet $p_{\text{inlet}} = 3.5$ bar.

Velocity distribution at the outflow from the holes of the sieve to the expander is presented in Fig. 5 for the supersonic expansion case. The recirculation present in the basic design choked flow in the lower section of the expander. In the modified version, it can be seen that the distribution of steam flow is much more uniform, especially in the lowest 5 holes. Additionally, one can observe a noticeable increase in the flow velocity in all

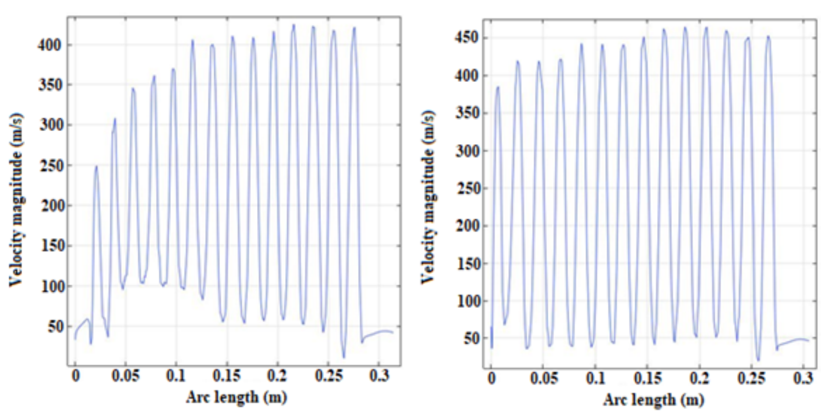

Figure 5: Velocity distribution behind the sieve holes for the basic (left) and modified (right) designs for the total pressure at the inlet $p_{\text{inlet}} = 3.5$ bar.

holes of the sieve. It resulted in a significant (19%) increase in the mass flow rate in the silencer.

3.2 Acoustic studies

The distribution of the total sound pressure level is presented in Fig. 6 showing the contour plots for 20 Hz and 315 Hz. There is a visible sound pressure level reduction with the shift to the higher frequencies. This tendency is not significant for higher frequencies, where sound waves are considerably shorter.

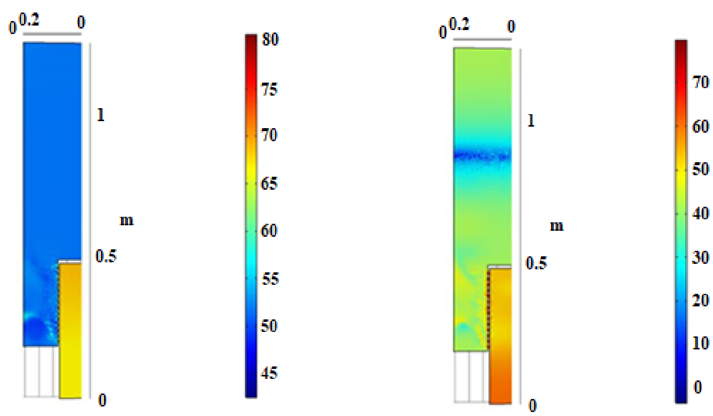

Figure 6: Decay of lower sound spectrum frequencies behind the expander (L_p in dB) for frequency of 20 Hz (left) and 315 Hz (right) for $p_{\text{inlet}} = 3.5$ bar.

The acoustic performance of the silencer can be characterized in a variety of ways. Transmission loss (TL), insertion loss (IL), and level difference (LD) are three mostly used performance parameters of the silencers. The TL is the most commonly presented parameter. This parameter is a function of the muffler alone, so the source and termination properties do not need to be defined. The IL is the difference in acoustic powers radiated from the exhaust system with the silencer attached and with the silencer replaced by a straight pipe. The IL is measured during work, so the source properties must be known. The LD or commonly referred to as 'noise reduction' (RD) is the overall decrease in sound pressure levels at any two arbitrary points in the exhaust system with the silencer attached [15].

The TL is the difference in the acoustic power of the forward traveling 'incident' pressure wave (p_{inc}) at the inlet of the silencer and the forward traveling 'transmitted' pressure wave at the outlet (p_{trans}). The subscripts

1 and 2 in Eq. (7) represent the total measured sound pressure (including reflective waves) at the inlet and outlet, respectively:

$$\mathrm{TL} = 20\log\frac{p_{\mathrm{inc}}}{p_{\mathrm{trans}}} + 10\log\frac{S_{\mathrm{out}}}{S_{\mathrm{in}}} = 20\log\left(\frac{p_1+\rho_c v_1}{2p_2}\right) + 10\log\frac{S_{\mathrm{out}}}{S_{\mathrm{in}}}\,, \quad (7)$$

where ρ_c represents the characteristic impedance of the medium, v_1 is the applied particle velocity at the inlet, S_{out} and S_{in} represent the cross-sectional areas of the inlet and outlet, respectively [16].

The TL for studied silencers with the same pressure at the inlet is presented in Fig. 7. The chart shows changes in the transmission loss for the modified silencer compared to the basic one. The parameter is the function of frequency and depends on the shell shape of the silencer under study. In the TL spectrum for the modified silencer more peaks are observed. The first peak is already observed at 12.5 Hz. The next intensive peak is observed at around 100 Hz for both studied silencers, but for the modified one the peak has much higher values of damping. The damping effect of the modified silencer is also higher at 400 Hz, 1250 Hz, 2500 Hz, and 4000 Hz, but slightly decreases at 800 Hz.

The LD was calculated as follows:

$$\mathrm{LD} = 20\log\frac{p_{\mathrm{in}}}{p_{\mathrm{out}}}\,. \quad (8)$$

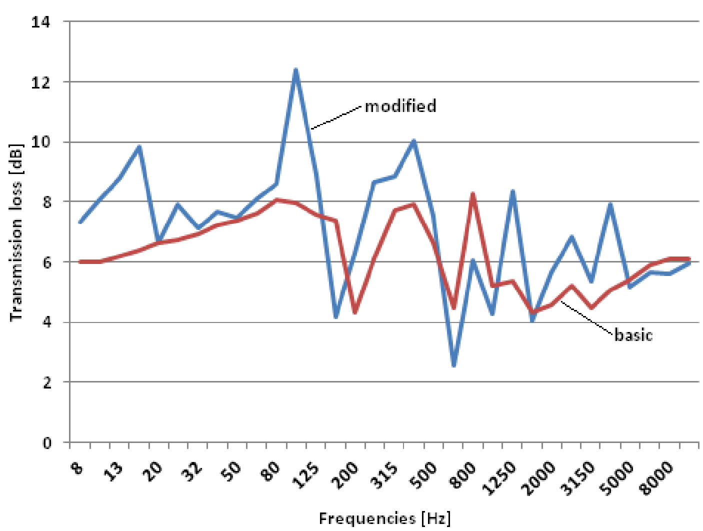

Figure 7: Estimated acoustic transmission loss for the basic and modified silencer design for the total pressure inlet parameter $p_{\mathrm{inlet}} = 3.5$ bar.

The silencer noise reduction was also calculated using the difference between average total sound pressure levels at the inlet and outlet of the silencer – ΔL_p (Table 2). There is no significant difference in ΔL_p and LD for silencers simulated with the mass flow boundary condition, because of the lower flow velocity and lack of supersonic flow areas in the model. The differences are visible for studied silencers with the total pressure boundary condition because of higher intensity of flow and clear supersonic flow regions at the outflow of the expander. More uniformly distributed flow in the expansion chamber and lack of recirculation in the lower parts of the silencer result in higher noise reduction. The modification of the silencer shell, which leads to a more uniform flow improves the acoustic parameters of the device, which is also observed on the sound pressure level spectrum of the studied silencers presented in Fig. 8.

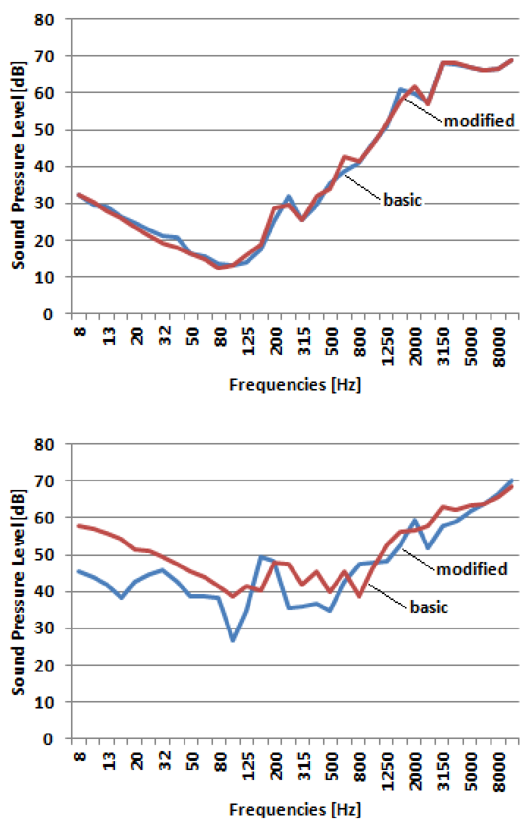

Figure 8: Sound pressure level at the outlet of the basic and modified silencer – upper for $\dot{m}_{\text{inlet}} = 0.0667$ kg/s and lower for $p_{\text{inlet}} = 3.5$ bar.

Table 2: Effect of the silencer geometry modification on the average ΔL_p and LD.

Model variation	Inlet parameter	ΔL_p (dB)	LD (dB)
Basic	0.0667 kg/s	6.6	0.7
Modified	0.0667 kg/s	6.7	0.7
Basic	3.5 bar	12.5	1.4
Modified	3.5 bar	17.1	1.7

4 Conclusions

An influence of shell shape modification on the flow and acoustic parameters of a steam silencer was studied. The numerical model was developed and computational fluid dynamics and acoustics simulations were performed for subcritical and critical flow conditions in the expander.

The numerical study results showed that the proposed modification of the shell had an effect on both the flow pattern and the acoustic performance of the silencer. It resulted in a more uniform flow distribution in the holes of the expander sieve due to a reduction of recirculation in the lower parts of the silencer compared to the basic variant. It caused a significant increase of the mass flow rate for the critical conditions.

The transmission loss of studied silencers was determined. The modified silencer had better noise reduction properties at the low-frequency than the basic one. It was also shown that the presence of supersonic flow in the domain of the silencer increases the sound pressure for low frequencies.

The silencer performance was tested for the specific pressure and mass flow rate conditions and it can vary for other values of these parameters what will be the subject of future studies. The presented model will be further improved. The in-field and bench tests are planned to verify the results obtained from the numerical model.

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