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LARGE EDDY SIMULATION AND AEROACOUSTIC ANALYSIS FOR SIMPLE EXPANSION SILENCER LIKE GEOMETRIES

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ABSTRACT

This study includes compressible large eddy simulations performed for flow predictions inside circular pipes resembling automobile exhaust pipes. Compressible large eddy simulation approach is selected because of its ability to resolve complex turbulent flow fields along with the wave motions present. Three geometries including the pipe alone and the pipes which include simple expansion chambers with circular and elliptical cross-sections are studied. The acoustic performances of the geometries are analyzed by recording the pressure fluctuations at the exit of the pipes. Predictions showed that the presence of a simple expansion chamber can reduce the pressure fluctuations not only by the expansion they provide but also by the wave reflections from the open ends and solid surfaces.Power spectral densities of the pressure fluctuations at the exitsof the pipes showed that the noise improvements by the expansion chambers are more evident at lower frequencies which become more pronounced when an A-weighting procedure was applied.

Keywords: *Engine Exhaust Noise, Simple Expansion Silencer, Compressible Large Eddy Simulation, OpenFOAM, Aeroacoustics.*

INTRODUCTION

Due to increasing number of automobiles in urban areas, the noise emitted by the engines of the automobiles may be a serious source of annoyance for the people. As a result European Union has been issuing tighter noise regulations on automobiles [1]. The exhaust of an internal combustion engine is one of the noise sources of an automobile and if not silenced the noise levels can reach to quite high values [2] which can be considered to be painful for humans [3]. Therefore, silencers are employed on the exhaust pipe to reduce the noise levels. Many different types of silencers such as reactive, absorptive or reactive/absorptive, etc. can be used for this purpose.[2]. Among these, the reactive type silencers employ an expansion chamber [2], which basically acts as a low-pass filter[4]. Here, the expansion chamber creates flow separation and wave reflection in order to generate an interference which would reduce the noise levels.

Acoustic performance of simple expansion silencers have been numerically analyzed using several methodologies. Suwandi et.al.[5]used transmission line theory to predict the aerodynamic performance of simple expansion silencers and were able to obtain reasonable agreement with the measurements. Kumar [6] used a linear acoustic modeling based on the frequency domain acoustic solver BOOST-SID. In reference [7] transmission loss across a simple expansion silencer is predicted using a boundary element methodology. Middelberg et.al.[8]used a pressure based computational fluid dynamics (CFD) solver with a k-ε turbulence model in order to obtain the acoustic performance different simple expansion chamber silencers. Singh [4], employed large eddy simulation (LES) for the acoustic analysis of ducts and mufflers (silencers). Since the flow through the expansion chamber includes sudden expansion which leads to separation, high turbulence and vortex shedding; LES, which is capable of handling such complex flow conditions [9], is selected as the simulation methodology in this study. Moreover, since the acoustic propagation involves wave motion, a compressible LES approach which is very suitable not only to capture the important features of the flow but also for aeroacustic predictions in internal flow is used [10].

The paper includes flow simulations andaeroacoustic analyses through a simple expansion silencer like geometry. For this purpose first a circular pipe geometry, with 5 cm diameter and 2.25 m length, which resembles the exhaust pipe of an automobile is generated. Then two different expansion sections; one with a circular and one with an elliptical cross-section, are added to the mid-section of this pipe. The length of the expansion chamber was 0.75 m. This way, the chamber divided the pipe into three equal length sections. The diameter of the circular expansion chamber is three times the diameter of the pipe while for the elliptical chamber, the longest and the shortest diameters are three and two pipe diameters, respectively. The geometries of these three configurations are displayed in Figure 1.

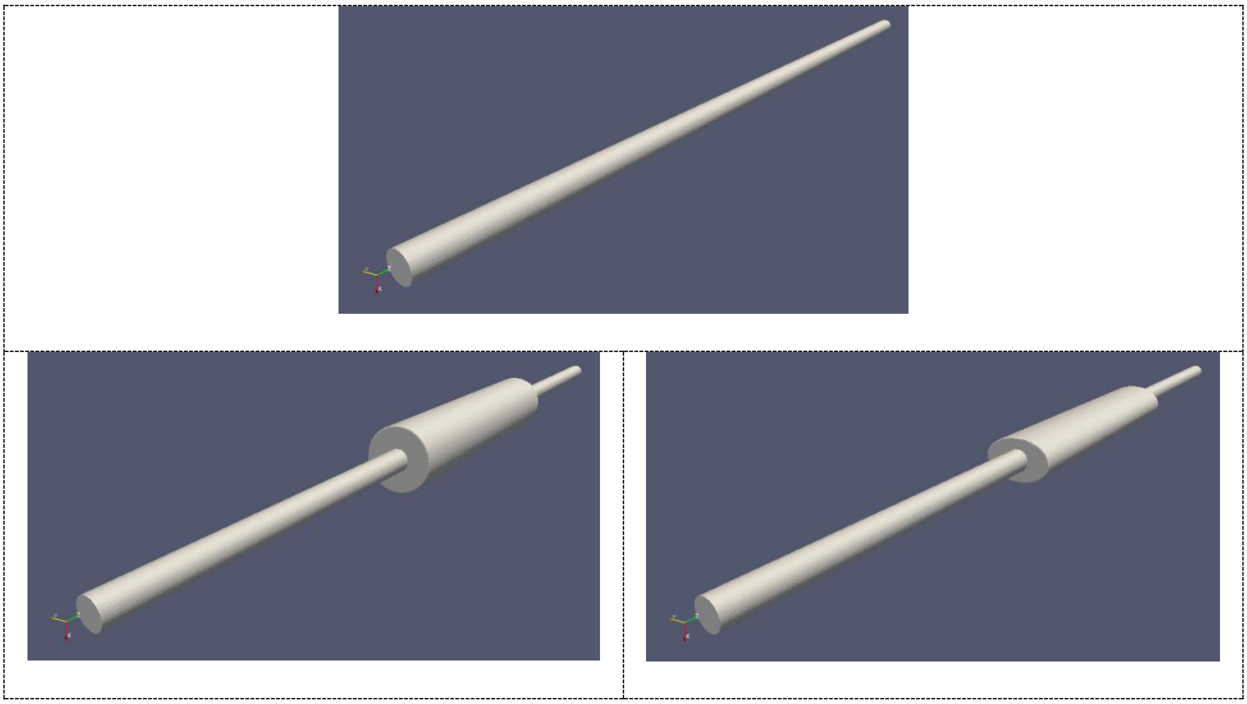


Figure 1. Studied geometries (Pipe alone (top), pipe with the silencers (bottom)

METHODOLOGY

Numerical simulations are performed using the open source computational fluid dynamics solver OpenFOAM[11]. Here, the rhoPimpleFoamsolver of the software is selected. This solver is a pressure based transient compressible solver especially suitable for HVAC and similar applications [12].OpenFOAM’s stabilized linear-upwind[13]discretization (LUST) is used for the fluid velocity and energy, while the second order accurate linear scheme[13] is selected for the rest of the variables.LES is used to simulate the turbulent flow inside the pipe and the chamber. Here, the sub-grid scale (SGS) stresses are modeled using the kEqn model of OpenFOAM which approximates the SGS stresses using a one-equation eddy viscosity model. In order to perform numerical solutions, structured meshes with a maximum cell length of 2mm are constructed using the mesh generation library cfMesh[14]. Here, the mesh of the pipe alone geometry has 603984 cells, while the meshes for the geometries with circular and elliptical geometries have 2119140 and 1549730 cells, respectively.Aeroacoustic analyses are performed by recording the pressure fluctuations at the exit of the pipes. Then the one-sided sound power spectral density [15] for each case is obtained by taking the fast Fourier transform of the unsteady pressure fluctuation data. Here, the pressure fluctuations are obtained by subtracting the mean pressure predictions from the instantaneous pressure predictions. The pressure data at the exit is recorded at every 0.2 ms so that frequencies up to 2500 Hz could be resolved. The signal processing package of the open source scientific programming language Octave [16] is used for constructing the sound power spectral densities.

In order to mimic the engine exhaust process, a periodic inflow velocity is defined at the inlet of the pipe. Here the velocity is defined as:

$$V(t) = V_0 + A \ \sin(2\pi f t) \tag{1}$$

Where V_0 and A are both selected to be 25 m/s, while the frequency f is set to 50 Hz. Here, the V_0 and A values are selected so that the inlet velocity changes between 0 and 50 m/s, where the latter is assumed to be the average discharge velocity for a typical cylinder with a 80 mm stroke and the frequency of 5 Hz is selected to mimic an automobile engine running at 3000 rpm.

Non-reflecting boundary conditions are applied at the exit boundary while no-slip conditions are applied on the solid surfaces.

RESULTS AND DISCUSSION

Numerical solutions were performed for a real time of 0.02 seconds which correspond to the period of the periodic inflow velocity given in equation (1). Here, the time step was chosen such that the maximum Courant number in the flow domain never exceeds 0.5. Simulations were performed in parallel using the high performance message passing library Open MPI [17].

Variation of pressure fluctuations recorded at the exits of the pipes with time is plotted in Figure 2. It is clear that the presence of the silencer clearly decreased the magnitude of the pressure fluctuations reaching the exit. However, the speed of the shock wave was not affected much by the silencers because all the wave fronts seemed to reach the exit almost at the same time. The effects of the two silencer geometries on pressure fluctuations were similar with some noticeable differences both in magnitude and oscillation frequency.

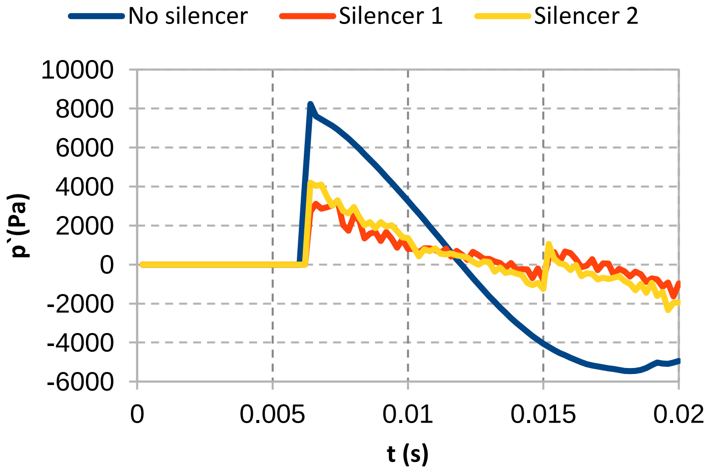


Figure 2. Pressure fluctuations at the exits of the three geometries.

In order to see the effect of the silencer on the wave structure, pressure fluctuation contours at the center planes of the three geometries are plotted in Figure 3, Figure 4 and Figure 5. Here, the snapshots are taken at t = 2, 4, 6 and 8 ms. This time range includes the motion of the shock wave inside the pipe and its reflection from the exit. When no silencer is used the shock wave generated at the inlet moves almost without disturbance along the whole pipe and then reflects from the open end. Therefore large pressure fluctuations are emitted from the exit, which can also be seen from Figure 2. When Figure 4 and Figure 5 are analyzed it can be seen that the shock wave loses its strength once it enters the expansion chamber (at t = 4 ms). Moreover, the wave reflections from the walls of the chamber also decreases the magnitude of the fluctuations however, the fluctuation pressure field becomes more non-uniform compared to the no silencer case. This also explains the smooth versus the oscillatory variations observed in Figure 2 for the geometries without and with silencer. In addition to this, the snapshot at t = 4 ms clearly shows that once the shock wave reaches the inlet of the expansion chamber, it is reflected as an expansion wave moving toward the inlet of the geometry. This further decreases the magnitude of the fluctuations. However, when this expansion wave is reflected from the inlet, it leads to an increase in the magnitude of the fluctuations.

Sudden expansion at the inlet of the expansion chamber also generates vortices shedding from the corner of the chamber inlet. The growth of these vortices as they move downstream is evident from the snapshots taken at t = 4 and 6 ms. However, when the shock wave inside the expansion chamber is reflected from the end walls and moves toward upstream, it interacts with the shed vortices and decrease their strength as can be seen from the snapshot at t = 8 ms.

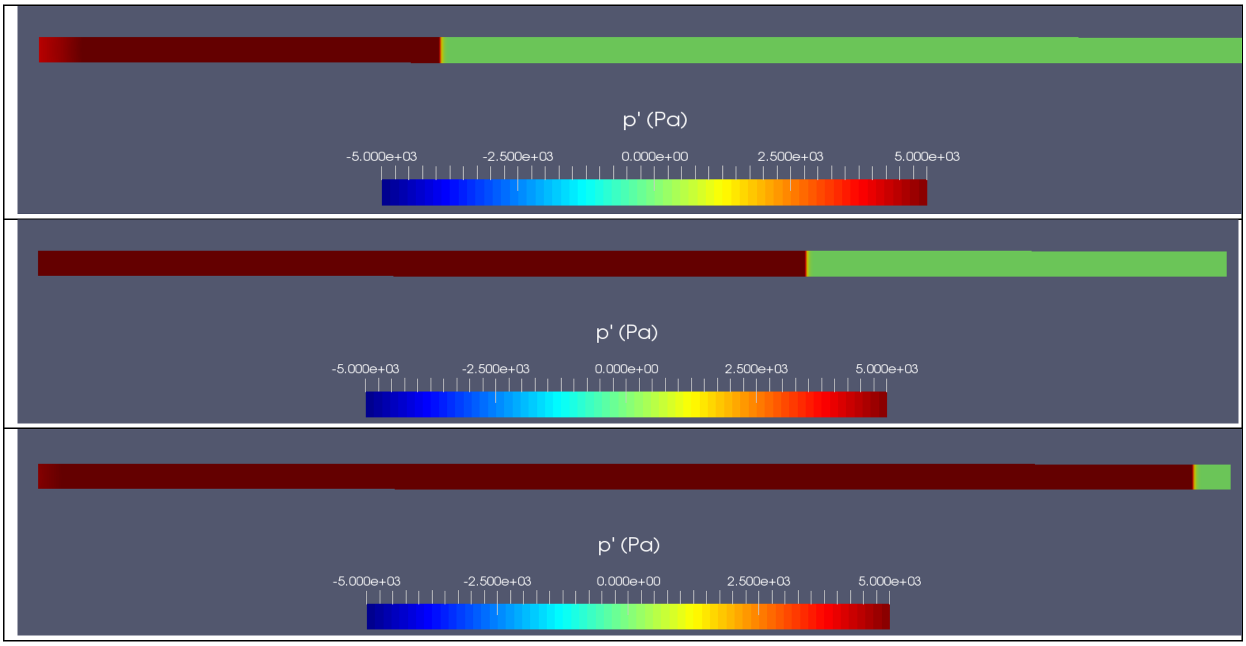
When the flow fields of the two silencer geometries are compared it is observed that the shock wave becomes weaker as it enters the silencer with the circular cross-section. This is mainly due to the larger cross-sectional area of this geometry which provides more space for expansion. Overall, the magnitude of the pressure fluctuations is larger inside the pipe with elliptical silencer compared to the pipe with the circular one.

The power spectral density of pressure fluctuations at the exit of the pipes are displayed in Figure 6. The reference pressure in these power spectral density calculations was taken as 20 μPa. Here, one can see the effect of the silencers especially at lower frequencies. However, acoustic performances of all geometries are observed to be similar at high frequencies. In this study the inlet velocity was spatially uniform and it did not contain any high frequency acoustic pulse. If such a pulse was added the effect of the silencers could be better observed as shown in reference [4].

In order to include the sensitivity of human ear to sound frequencies the A-weighting procedure [18] is applied to the predictions and the resulting sound power spectral density is displayed in Figure 7. The improvements by the silencers at low frequencies became more evident after this weighting procedure is applied.

According to the predictions obtained in this study, the simple expansion chambers analyzed provided moderate improvements on the exhaust noise emissions. Of course, in order to reach more meaningful conclusions the simulation times should be much larger than the oscillating velocity period used in this paper. In addition to this, the silencers used in automobiles have more complex internal structures including porous regions, bafflers [4],[5], [6]and/orsound absorbing materials [2]. Future work will include these additions to the simple expansion chambers studied.

One of the important parameters related to exhaust silencers is the transmission loss [4], [5], [6]. The pressure fluctuations by the waves moving in the downstream direction should be included in the calculations of transmission loss while the effect of the reflected waves should be kept out of the calculations [5]. In some numerical solutions the length of the pipe upstream and downstream of the expansion chamber was kept very long compared to the length of the chamber in order to collect enough data before the reflected wave reaches the probe locations [4], [5]. Since this was not the case in the geometries studied here, transmission loss calculations were not be able to conducted.



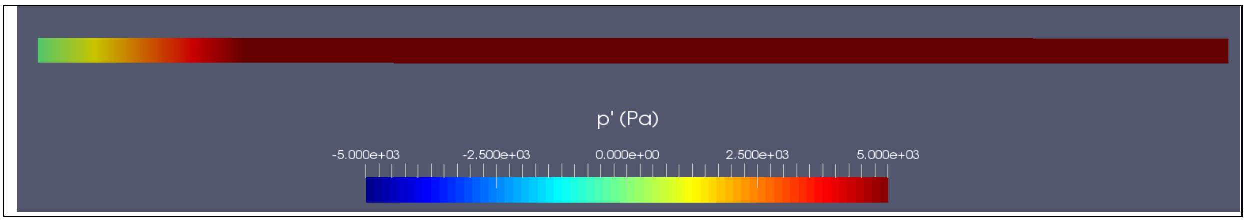


Figure 3. Pressure fluctuations at the center plane of the circular pipe (t = 2, 4, 6, 8 ms)

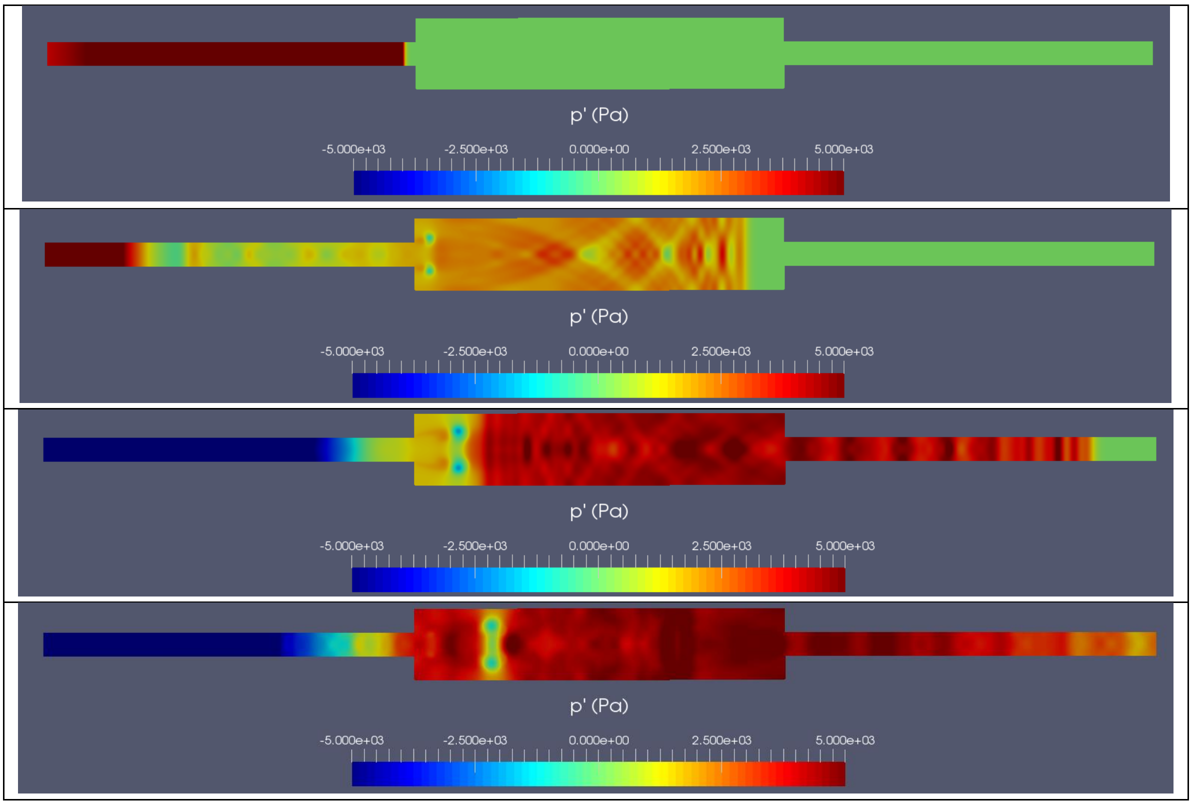
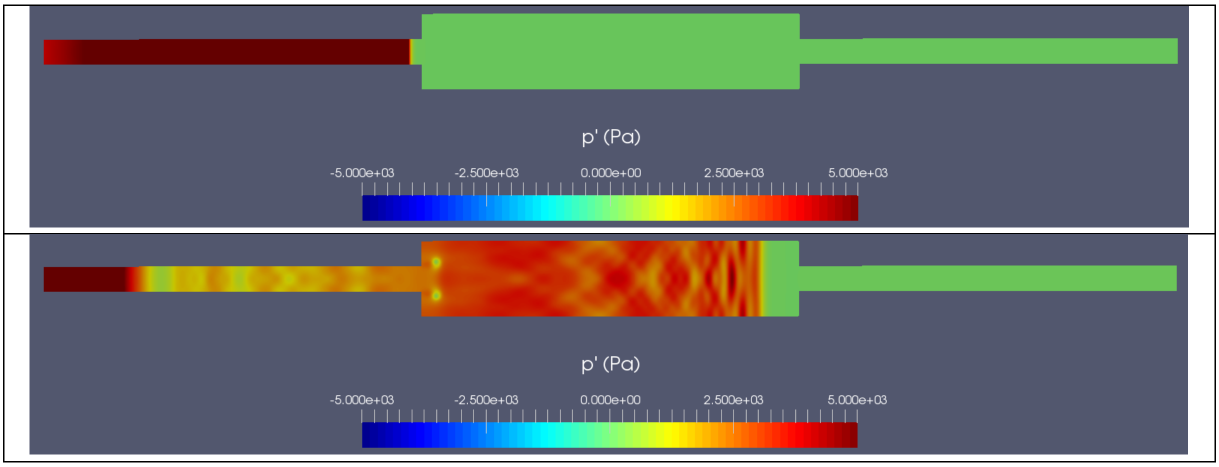


Figure 4. Pressure fluctuations at the center plane of the circular pipe with circular silencer (t = 2, 4, 6, 8 ms)



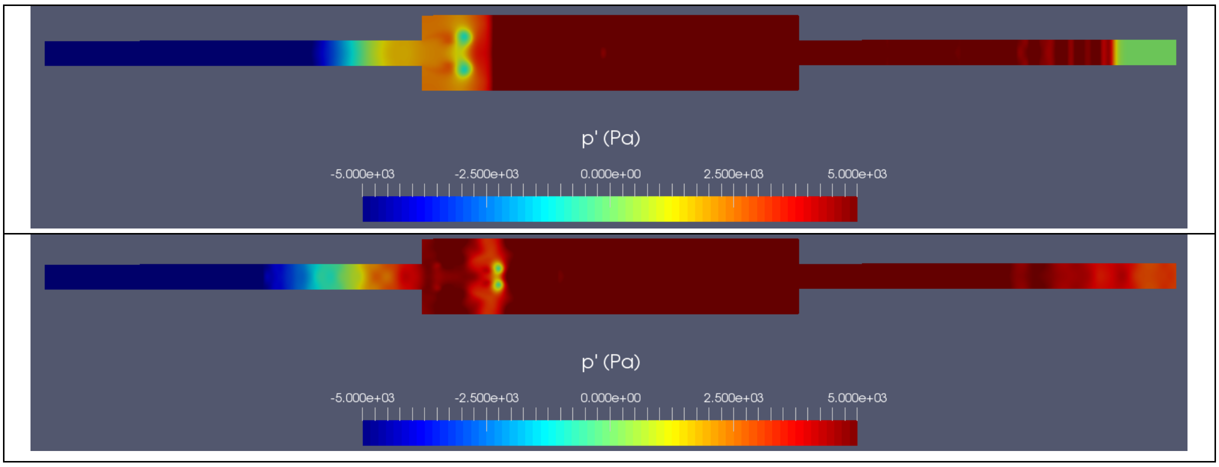


Figure 5. Pressure fluctuations at the center plane of the circular pipe with elliptical silencer (t = 2, 4, 6, 8 ms)

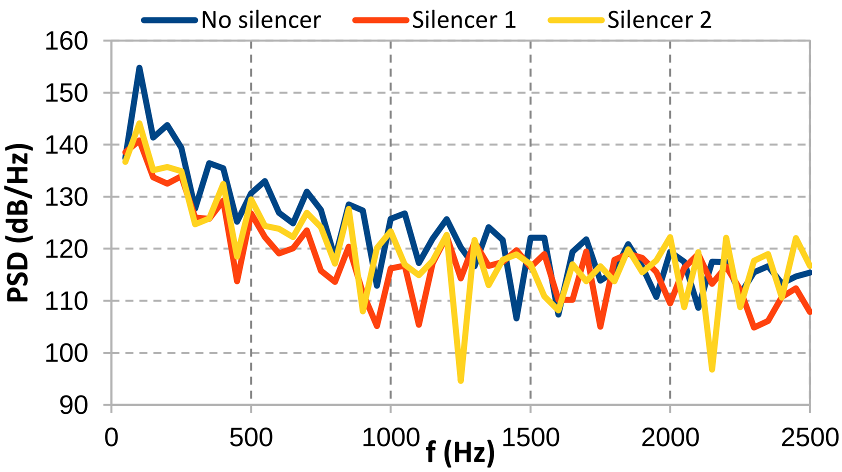


Figure 6. Power spectral densities of pressure fluctuations at the pipe exits.

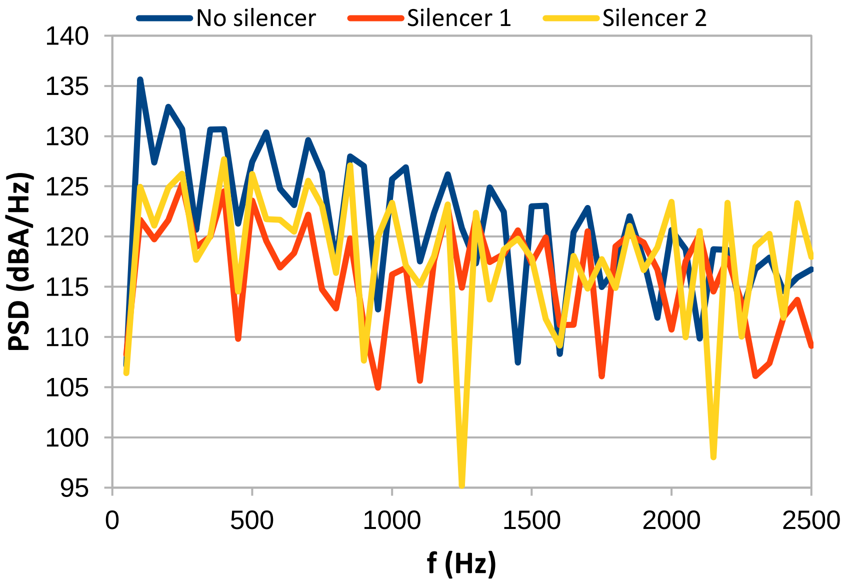


Figure 7. A-weighted power spectral densities of pressure fluctuations at the pipe exits.

CONCLUSIONS

Compressible LES of turbulent flow in circular pipes, resembling the exhaust pipe of an automobile, with and without simple expansion chamber type silencers were performed using the open source CFD software OpenFOAM. Two different expansion chamber geometries with a circular and an elliptical cross-section, without any internal structures were considered. A periodic velocity was assigned at the inlet boundary of the geometries in order to mimic the engine exhaust procedure. The pressure fluctuations were calculated by subtracting the mean pressure from the instantaneous one.

LES predictions showed that the presence of the expansion chamber successfully reduced the pressure fluctuations at the exit of the pipe not only by reducing the shock wave strength with the expansion but also with the wave reflections from the open ends and solid surfaces. Among the two chamber geometries analyzed the one with the circular cross-section provided with the most amount of drop in the fluctuations mainly due to its larger cross-sectional area.

Power spectral densities of the pressure fluctuations at the exit showed that the noise improvements by the expansion chambers were more evident at low frequencies. The improvements become more pronounced when A-weighting, which includes the sensitivity of human ear to various frequencies, was applied to the power spectral density predictions.

Even though the predictions presented here were not validated with experimental data nor a grid independency study was performed, it is believed that the predictions provide a useful insight for ranking the geometries in terms of acoustic performance because of the similar grid spacing employed.

As future work, additional internal structures such as porous regions are planned to be included in to the expansion chambers to analyze their effects on sound attenuation.

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