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Insertion Loss of Spiral Ducts – Measurements and Computations

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This work presents measured and computed characteristics of insertion loss (IL) of spiral ducts. Numerical and experimental models of spiral ducts have been investigated. For the numerical modeling, a three dimensional model computed by the use of a finite element method in a COMSOL Multiphysics computer application has been used. For the experimental modeling, there has been made a spiral duct model by using a three dimensional rapid prototyping technique. An acoustic system with a round silencer has been ivestigated, and the spiral duct has been inserted at the inlet. IL is considered in this paper as the difference between the sound pressure level (SPL) [dB] probed at only one outlet point of the acoustic system without and with an acoustical filter (spiral duct), respectively. The results of measured and computed IL of spiral ducts presented in this paper confirm the fact that this newly discovered technical solution for attenuating sound in ducted systems has an applicable potential. There are visible small discrepancies between the measurements and computations. The results can differ due to the non ideal dimensions of the experimental model and the non ideal inlet and outlet surroundings of the experimental acoustic system. However, the IL characteristics of the computed model is almost wholly covered by the IL characteristics of the measured model.

Keywords: spiral duct, insertion loss, measurements, computations.

1. Introduction

Acoustic attenuation performance of different types of spiral ducts is a main purpose of several papers [5–7, 9–11]. All those papers were based on numerical results. Till now, the only one measured parameter was the sound pressure level distribution at the outlet of the spiral duct [8].

Hence, it has been already proven that inserting a just one spiral turn at the inlet circular duct of the round silencer can improve the sound attenuation performance of this silencing system [5].

Obviously, higher acoustical parameters are achieved when the spiral duct has more eligible geometrical parameters fitted to the sound wave length. In that case the highest transmission loss can be observed [13] at the resonance frequency of the spiral duct – Fig. 1, which produces a phenomenal sound pressure level (SPL) distribution at the outlet – Fig. 2.

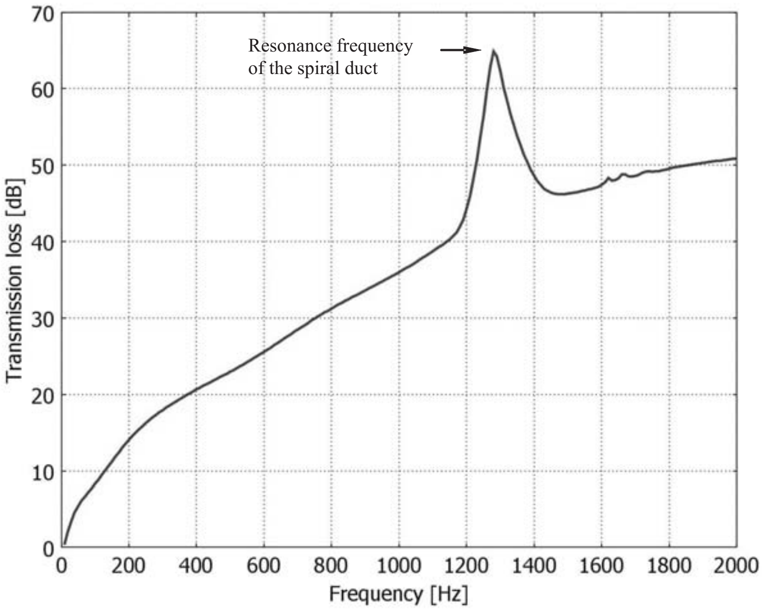

Fig. 1. Transmission loss of the round silencer with one spiral turn of the spiral duct at the inlet [6].

It is simple to achieve the compute transmission loss [5–7, 10, 13] of an acoustic system in comparison by making measurements of this parameter. However, it is very simple to compute and measure the insertion loss (IL) [11, 13]. Computed insertion losses of selected spiral ducts were presented in one paper earlier of the author [11]. Measured and computed IL of spiral ducts are presented in this paper too.

IL is considered in this paper as the difference between the sound pressure level (SPL) [dB] probed at only one outlet point of the acoustic system without and with an acoustical filter (spiral duct) [13], respectively. Symbolically,

$$\mathrm{IL} = \mathrm{SPL}_1 - \mathrm{SPL}_2 \quad [\mathrm{dB}], \tag{1}$$

where the subscripts 1 and 2 denote acoustic systems without filter and with filter, respectively.

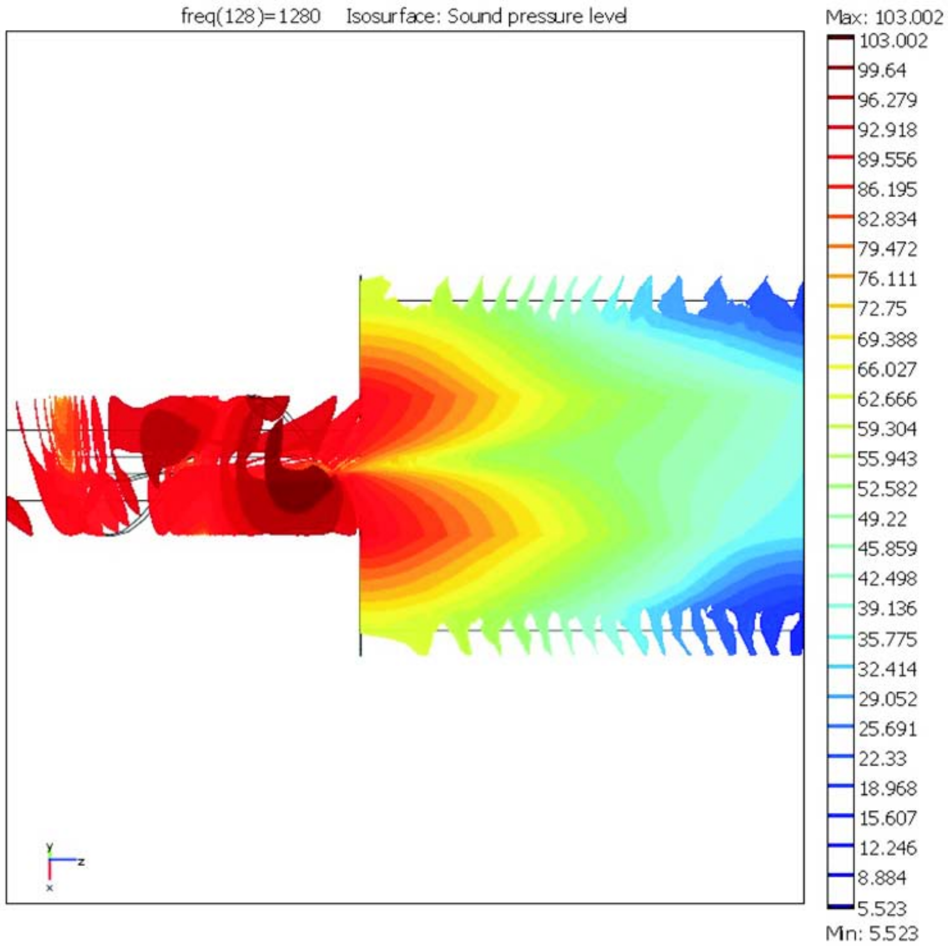

Fig. 2. Sound pressure level distribution at the resonance frequency of one spiral turn of the spiral duct placed at the inlet of theround silencer [6].

2. Numerical and experimental models

The experimental system consists of a circular duct (diameter $d = 125$ mm, length $l = 53$ cm) with a spiral element inside (which forms a spiral duct) from one side connected to the sound source and from a second one connected to the round silencer – Fig. 3. The spiral duct was made using the three dimensional rapid prototyping technique. As the source signal the white noise is adopted. The dimensions of the silencer are as follows: internal diameter $d = 125$ mm, length $l_S = 58$ cm. The silencer is filled with an absorptive material of apparent density of about 90 kg/m^3 which is 7 cm thick in radius – counting up from the internal diameter. An about 1 mm thick steel sheet covers the round silencer at the external side.

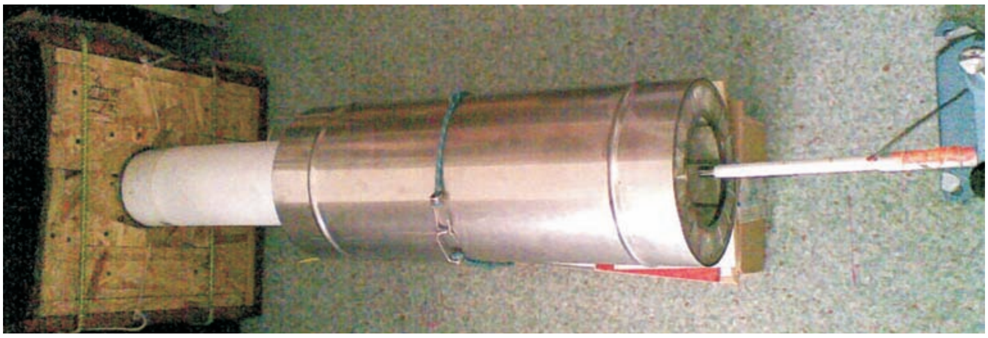

Fig. 3. Round silencer connected to the sound source. Between the silencer and the sound source there are inserted spiral ducts.

An experimental spiral duct is presented in Fig. 4a. Its dimensions are: circular duct diameter $d = 125$ mm, circular mandrel diameter $d_t = 30$ mm, thickness of the spiral profile $g = 3$ mm, the relation between the circular duct diameter d and the spiral lead s is represented as the ratio $s/d = 1.976$.

a)

b)

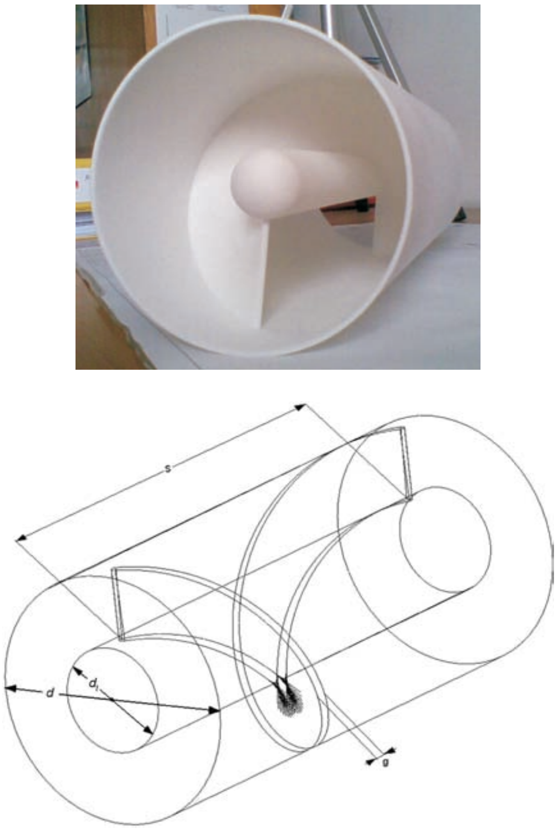

Fig. 4. Experimental spiral duct with ratio $s/d = 1.976$, $d = 125$ mm, $g = 3$ mm, $d_t = 30$ mm: a) real view, b) schematic view.

The measurement point of the sound pressure level (SPL) [dB] was situated at the outlet plane of the silencer in his main axis. A Brüel&Kjær 1/2″ microphone unit type 4190-C-001 was mounted on a tripod and connected to the Brüel&Kjær PULSE platform, as seen partially on the right side of Fig. 3.

To compare the experimental results with those simulated numerically there were made computations for a three-dimensional (3D) acoustic system similar in geometry to this presented in Fig. 3 using the finite element method in a COMSOL Multiphysics – Acoustic Module computational environment [2]. This is an interactive environment for modeling and solving all kinds of scientific and engineering problems based on partial differential equations. The Acoustic Module of this software provides tailored interfaces for modeling acoustics in fluids and solids. The module includes also a modeling support for several types of damping [2].

The 3D numerical model of the acoustical system with a spiral duct inside is presented in Fig. 5. The dimensions are nearly identical with the experimental model, but it should be mentioned that the numerical model takes into account only the ideal environment and the materials. It also reveals that all hard surfaces of the ducts and silencer are simulated as a sound hard boundary (wall) (3) [2]. The 1 m cubic box shown in Fig. 5 is a simulated acoustic free field atmosphere and all external surfaces are set as impedance boundary conditions (5), (6) with a value of the characteristic impedance of air [2].

A 1 m cubic box of air placed at the outlet of the numerical acoustic system brings the computed results closer to the experimental ones.

The problem has been solved in the frequency domain using the time-harmonic Pressure Acoustics application mode [2]. The final parameter of this solution is the acoustic pressure p [Pa], which can be computed by the use of the slightly modified Helmholtz equation:

$$\nabla \cdot \left(-\frac{\nabla p}{\rho_0} \right) - \frac{\omega^2 p}{c_s^2 \rho_0} = 0, \tag{2}$$

where ρ_0 is the density of air ($\rho_0 = 1.23$ kg/m^3), c_s is the speed of sound in air ($c_s = 343$ m/s) and ω is the angular frequency.

For the models investigated in this work, the boundary conditions are of three types [2]. For acoustically hard walls at the solid boundaries, which are the walls of the spiral element profile of mandrel or circular duct, the model uses sound hard (wall) boundary conditions:

$$\left(\frac{\nabla p}{\rho_0} \right) \cdot \mathbf{n} = 0. \tag{3}$$

The boundary condition at the inlet surface (sound source) of a circular duct is a combination of incoming and outgoing plane waves:

$$\mathbf{n} \cdot \frac{1}{\rho_0} \nabla p + ik \frac{p}{\rho_0} + \frac{i}{2k} \Delta_T p = \left(\frac{i}{2k} \Delta_T p_0 + (1 - (\mathbf{k} \cdot \mathbf{n}))\, ik \frac{p_0}{\rho_0} \right) e^{-ik(\mathbf{k} \cdot \mathbf{r})}, \tag{4}$$

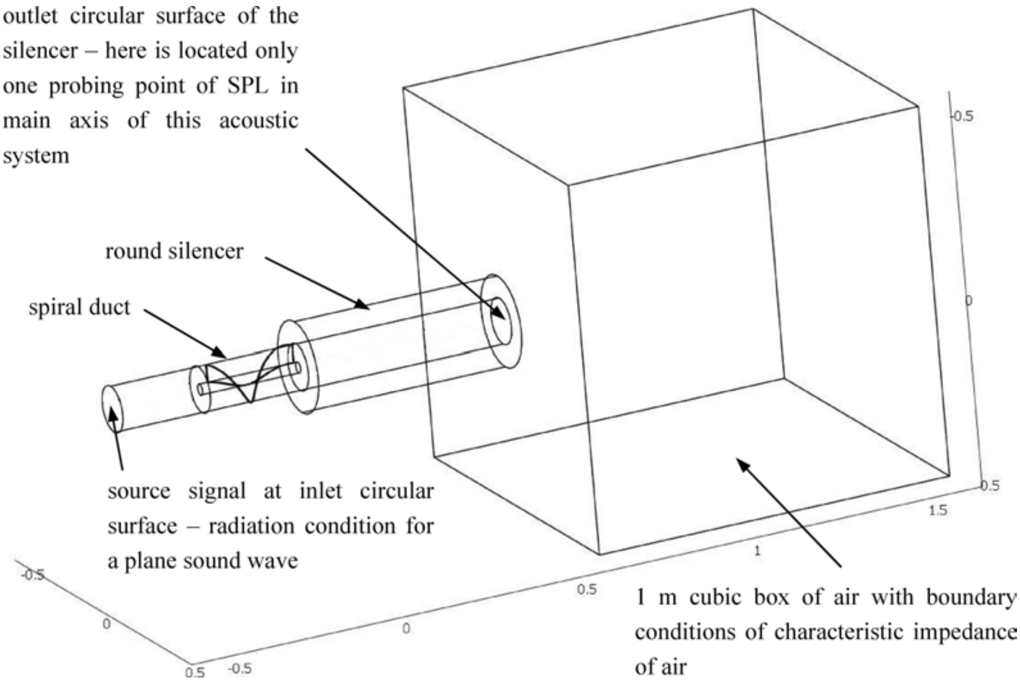

Fig. 5. Numerical acoustic system (the geometrical configuration is same as that of the experimental one) constituting a spiral duct with ratio $s/d = 1.976$ and a round silencer with an absorptive material of apparent density 90 kg/m^3 with the fiber diameter $d_{av} = 6$ µm.

where Δ_T denotes the boundary tangential Laplace operator, $k = \omega/c_s$ is the wave number, $\mathbf{n}$ is the normal direction vector for investigated circular duct, and the wave vector is defined as $\mathbf{k} = k\mathbf{n}_k$, where $\mathbf{n}_k$ is the wave-direction vector. In Eq. (4), p_0 represents the applied outer pressure, and i denotes the imaginary unit [4]. The inlet boundary condition is valid as long as the frequency is kept below the cutoff frequency for the second propagating mode in the cylindrical duct.

At the outlet of the 1 m cubic box, the boundary walls are set as the impedance boundary condition, which is a generalization of the sound-hard and sound-soft [2] boundary conditions:

$$\mathbf{n} \cdot \left(\frac{1}{\rho_0} (\nabla p - \mathbf{q}) \right) + \frac{1}{Z} \frac{\partial p}{\partial t} = 0. \tag{5}$$

In the frequency domain Eq. (5) becomes

$$\mathbf{n} \cdot \left(\frac{1}{\rho_0} (\nabla p - \mathbf{q}) \right) + \frac{i\omega p}{Z} = 0, \tag{6}$$

where t [s] is the time, $\mathbf{q}$ is the dipole source, Z [Pa·s/m] is the acoustic input impedance of the external domain. The input impedance is the ratio of the pressure to the normal particle velocity, which is set in computations as the characteristic impedance inside the domain, $Z = Z_0 = \rho_0 c_s$.

To compute the acoustical damping by absorbion by the glass-wool inside the round silencer, the damping enters the equation as a complex speed of sound, $c_c = \omega/k_c$, and a complex density, $\rho_c = k_c Z_c/\omega$, where k_c is the complex wave number and Z_c is the complex impedance.

For porous materials with a rigid skeleton, the well-known model of Delany and Bazley is used, which estimates these parameters as functions of frequency and flow resistivity. For the original coefficients of Delany and Bazley [3], the expressions are

$$k_c = \frac{\omega}{c_s}\left[1 + 0.0978\left(\frac{\rho_0 f}{R_f}\right)^{-0.7} - i0.189\left(\frac{\rho_0 f}{R_f}\right)^{-0.595}\right], \tag{7}$$

$$Z_c = \rho_0 c_s\left[1 + 0.0571\left(\frac{\rho_0 f}{R_f}\right)^{-0.754} - i0.087\left(\frac{\rho_0 f}{R_f}\right)^{-0.732}\right], \tag{8}$$

where R_f is the flow resistivity, f is the frequency [Hz], $k_a = \omega/c_a$ and $Z_a = \rho_a c_a$ are the free-space wave number and the impedance of air, respectively. For glass-wool-like materials the empirical correlation of Bies and Hansen is used [1]:

$$R_f = \frac{3.18 \cdot 10^{-9} \cdot \rho_{ap}^{1.53}}{d_{av}^2}, \quad \left[\frac{\mathrm{Pa \cdot s}}{\mathrm{m}^2}\right], \tag{9}$$

where ρ_{ap} is the material's apparent density and d_{av} is the mean fiber diameter. The investigated model of the round silencer uses the glass-wool of $\rho_{ap} = 90\ \mathrm{kg/m^3}$ and $d_{av} = 6\ \mu\mathrm{m}$.

The numerical model is computed by the use of finite element method (FEM) by the terms of the element size [12] and maximum element size equals $h_e = 0.2(c_s/f_{\max})$, where $f_{\max}$ is the value of maximum investigated frequency (in this paper $f_{\max} = 2$ kHz).

3. Results

The results for the measured and computed IL of spiral ducts are presented in Fig. 6. The difference between the measured and computed frequencies of maximal attenuation (f_{rC} and f_{rE}) equals 15 Hz. The resonance frequency of the computed spiral duct equals $f_{rC} = 1291$ Hz with a value of IL = 20.2 dB. In experiment, the resonance frequency of the spiral duct is higher than the numerical one and equals $f_{rE} = 1306$ Hz with a value of IL = 26 dB. The ranges of attenuated frequencies for a sound attenuation (IL) higher than 3 dB are very similar in both cases. However, a wider range of attenuating frequencies in is visible in the experiment.

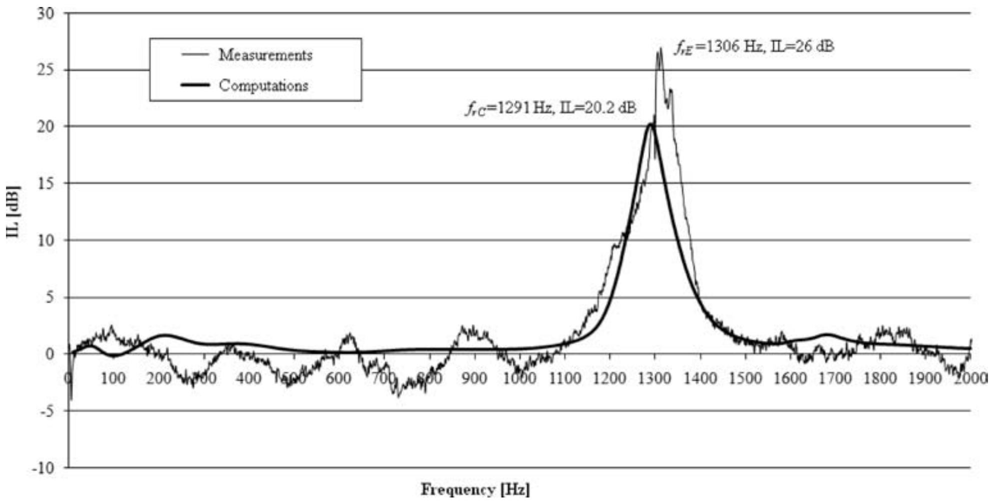

Fig. 6. Measured and computed IL of spiral ducts for a frequency range from 10 Hz to 2 kHz.

For the numerical model, the range of attenuated frequencies for an IL level higher than 3 dB holds between the frequencies from about 1180Hz to about 1430 Hz and equals 250 Hz. For the experimental model, the range of attenuated frequencies for an IL level higher than 3 dB can was determined for frequencies between 1145 Hz and 1422 Hz and equaled 277 Hz in this case.

4. Conclusions

The results of measured and computed IL of spiral ducts presented in this paper confirm the fact that this newly discovered technical solution for attenuating sound in ducted systems has an applicable potential. A small discrepancy between the measurements and computations is visible. The results can differ due to non ideal dimensions of the experimental model and to the non ideal inlet and outlet surroundings of the experimental acoustic system. However, the IL characteristics of the computed model is almost wholly covered by the IL characteristics of the measured model. The range of attenuated frequencies at a IL level higher than 3 dB is more than 25 Hz wider for the experimental model, and the IL for resonance frequency is higher than that for computations by about 6 dB.

Furthermore the spiral duct is an effective sound attenuator which can be considered as a band stop filter in a domain of ducts and mufflers.

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