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# Predicting acoustic and aerodynamic properties of a silencer mounted on a large chimney using equivalent fluid modeling combined with CFD

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## ABSTRACT

Silencers are a common noise control solution used to decrease the noise radiated from large ducts found in industrial plants. When designing a new silencer, the simplest way to assess the effect of a newly introduced modification is to conduct measurements on an assembled prototype. Unfortunately, this approach is costly and time-consuming. Thus, the ability to predict the acoustic and aerodynamic behavior of silencers (perform virtual experiments) in the initial design stage is of high importance. In this article, we utilize equivalent fluid modeling (EFM) combined with Computational Fluid Dynamics (CFD) to predict the acoustic and aerodynamic properties of a silencer mounted on a large chimney. We use this particular case study to demonstrate that the EFM+CFD approach can serve as an effective virtual prototyping tool. Performing virtual experiments allowed us to design a silencer that introduces 16 dB noise reduction without unwanted effects like higher pressure drop. The silencer was then installed on a real chimney, and in-situ measurements were carried out. Measurement and simulation results were in good agreement..

**Keywords:** *silencer, insertion loss, FEM, TMM, self-noise, total pressure loss coefficient, chimney, CFD*

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## 1. INTRODUCTION

The article presents the issues of designing acoustic silencers in industrial conditions. The subject of the project was a dust collection system, the element of which was a station consisting of 3 industrial fans of the centrifugal type pumping air into a chimney with a diameter of 5 m (Figure 1). The intakes to the chimney were equipped with duct sections intended for the installation of a baffle silencer, but as a result of element usage, the baffle plates were removed and only the air deflectors were left in place. The project was to upgrade the interior of the chimney air intakes by removing the remaining parts of the old silencer and implementing a new silencer (Figure 2) that provides a minimum 16 dB noise reduction from the chimney outlet and no degradation in the flow performance of the system. The flow rate at the chimney outlet was 1.2 mln m3/h.

Acoustic silencers are a common solution to deal with industrial noise. One of the key applications of silencers is to reduce noise propagating through large industrial ducts. From an engineering perspective, the ability to predict insertion loss is a very important aspect when designing acoustic silencers. It saves the time and money needed to build physical prototypes. Various models can be found in the literature to estimate insertion loss [1]: least attenuated mode technique, global attenuation estimation, and mode matching technique. Beranek has proposed a semi-empirical model that takes into account the flow-induced noise [2]. However, these are simplified methods. The most general and accurate method is to solve the full Helmholtz equation in 3D using the finite element method. However, when using the FEM model, a decision must be made on how to model the material forming the core of the silencer. One possibility is the equivalent fluid method, which involves describing the porous material using effective parameters: wave number $k_a$ and characteristic impedance $z_a$. These parameters are parameters typical for fluids, but they can also successfully describe a porous material when the assumption of a rigid frame forming the modeled material is met [3]. In the literature, one can find silencer models where the effective parameters in the FEM model were extracted from the Delany-Bazley-Miki (DBM) formula based on the flow resistivity of the material[4][5]. In the present study, the silencer core parameters ($k_a$,$z_a$) were determined empirically in an impedance tube using the Transfer Matrix Method (TMM)[6].

**Figure 1. Chimney stack with three inlet ducts.**

The issue of designing acoustic silencers in such a situation is a complicated topic and consists of the following steps

1) Study of the sound power level spectrum of the chimney outlet.
2) Study of the flow parameters of the system.
3) Determination of the required acoustic power level reduction value.
4) Determination of the insertion loss (acoustic parameters) of the silencer along with the total pressure loss coefficient of the silencer (flow parameters).
5) Flow simulations of the discharge side of the plant for:
    a. determination of the expected flow velocity distribution upstream of the silencer for evaluation of flow noise,
    b. determination of the required strength of the plate silencer design.
6) Calculation of air flow noise in the stack
7) Calculation of airflow noise in an acoustic silencer

In this article, we present the key acoustic issues affecting the design process in this particular case and present the results of the implementation.

Flow and acoustic measurements were conducted for the following test states:

A. condition without silencers with profiles inside the intake channels (as remains of the old silencer),
B. condition without silencers after removal of profiles inside the inlet ducts,
C. condition with designed acoustic silencers.

## 2. SILENCER DESIGN

The design activities included:

1) Calculations of insertion loss
2) Calculations of the total pressure loss coefficient
3) Calculations of the self noise

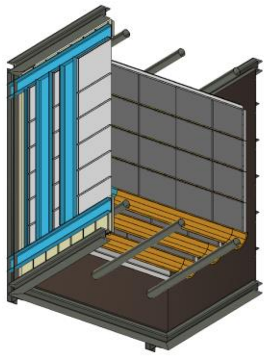

**Figure 2. Drawing of the silencer.**

### 2.1 Silencer insertion loss

In the present study, the silencer IL was determined by performing equivalent fluid FEM simulation, where core parameters ($k_a$,$z_a$) were determined empirically in an impedance tube using the Transfer Matrix Method (TMM)[6]. This method was validated by KFB Acoustics by comparing simulation results and measurements of the insertion loss of plate silencers.

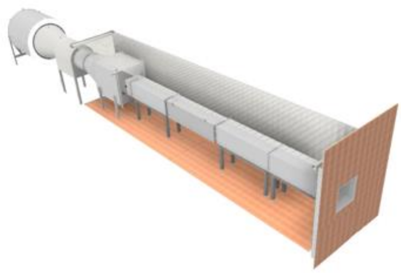

**Figure 3. A 3D model of the test bench setup used for the experimental measurements of acoustic and flowe performance of silencers.**

During the validation, material flow resistivity was measured for comparison purposes and simulation was also performed using the DBM model. Thermoviscous effects were included in the model, as they were crucial in the narrow regions of the silencer (slits between baffles). Then, a series of simulations and acoustic measurements of the insertion loss were carried out. The following factors were tested: silencer type (baffle and bar silencer), core material (foam and mineral wool), presence of housing (with and without), silencer length, width, and shape. In all cases, good agreement was obtained between simulation and measurement results. In the validation process, the ability of the FEM modeling to accurately predict the pressure drop [7] and flow-induced noise [8] was also confirmed. In order to validate the created models, a dedicated test stand was built (Figure 3) for the empirical determination of silencer parameters. Before carrying out the actual measurements, the limits of applicability of the test bench were determined. This limit is determined by the maximum measurable insertion loss. The measurement of this limit was carried out on a duct tightly filled with a thick layer of mineral wool introducing high insertion loss. In this way, it was possible to determine the influence of all side paths, which define the limits of applicability of the test bench.

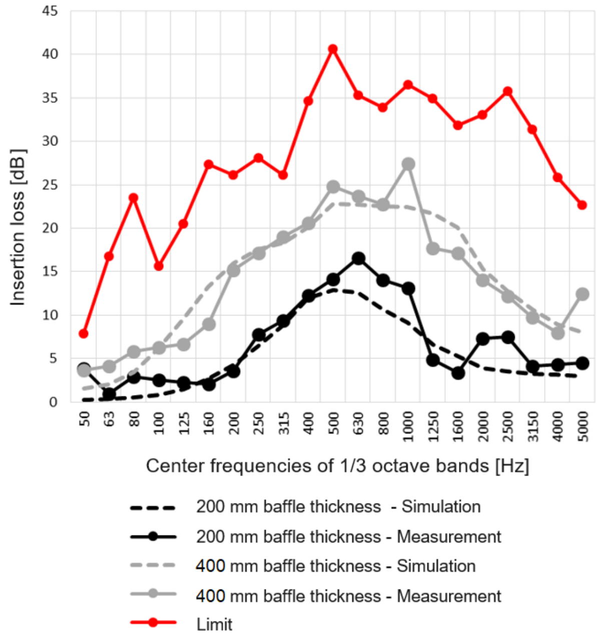


**Figure 4. Experiment vs simulation of the insertion loss of the baffle type silencer to validate virtual model.**

Figure 4 shows an example plot comparing the measurement result (measurement) and the result based on the transition matrix measurement (simulation). The limit of applicability of the measurement bench is represented on the graph by a red line.

In the present project, we utilized the validated equivalent fluid model to simulate insertion loss of silencer which is the subject of the presented design. It is assumed that the insertion loss is determined with a high level of agreement with the actual acoustic behavior of the silencer. The target insertion loss of the silencer depends on:

- The thickness of the attenuation plate and the parameters of the material used,
- The width of the gap between the plates,
- Length.

The calculated insertion loss value for the subject silencer under no-flow conditions is shown below (Figure 5).

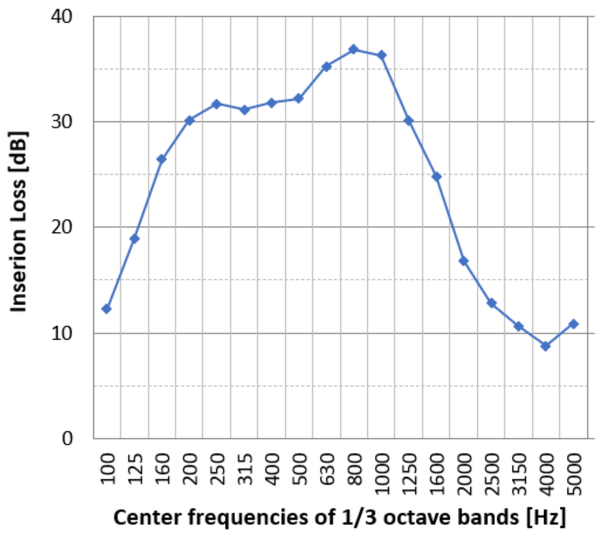


**Figure 5. Calculated insertion loss of designed silencer.**

### 2.2 Total pressure loss coefficient

Total pressure loss coefficient calculations were performed according to the method in ISO 14163 where experimentally verified local resistance coefficients $\zeta$ were used according to the method presented in the author's previous work [7]. The coefficient depends on the geometry of the silencer and, above all, the shape of the front surfaces of the damping plates. For the attenuator in question, a total attenuator resistance coefficient of $\zeta = 4.8$ was obtained.

It should be noted that with a given coefficient ζ, the total pressure loss of the attenuator $\Delta p_t$ will depend on the flow parameters according to the relation [10].

$$\zeta = \frac{\Delta p_t}{\frac{1}{2}\rho_1 v_f^2}$$

ρ – density[kg/m$^3$]
$v_f$ – face air velocity [m/s]

### 2.3 Silencer self-noise

Self-noise was calculated according to the method in ISO 14163 where the air velocity in front of the baffles was assumed to have an average value of 6 m/s. As a next step self-noise was also calculated using the airflow velocity distribution in front of the silencer obtained from CFD simulations where a maximum value equals to 19 m/s (which was taken as the least-favorable variant). The results can be shown on Figure 6.

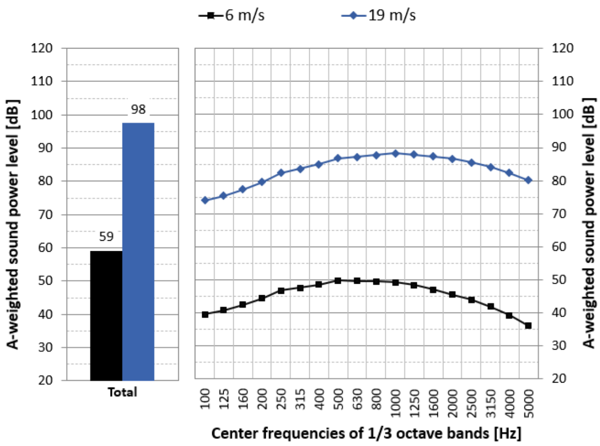


**Figure 6. Self-noise of the silencer dependent on face air velocity.**

CFD simulation was performed using Comsol Multiphisics software for the initial boundary conditions shown below (Figure 7).

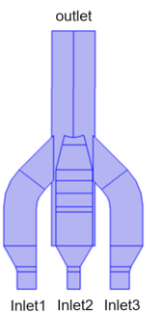


**Figure 7. CFD model setup. outlet: 0 Pa. inlet 1, 2, 3: 330 000 m3/h each.**

The following air velocity distributions were obtained for three operating states A, B ,C.

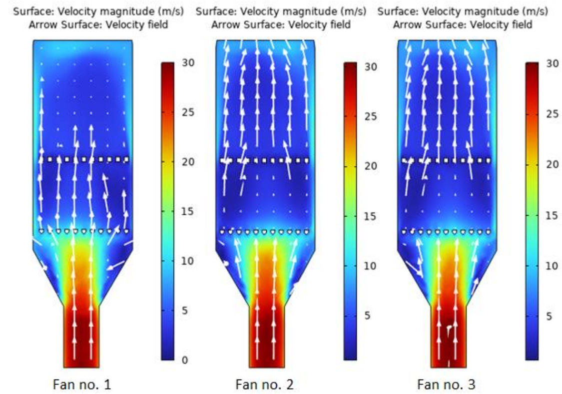


**Figure 8. CFD results: Flow velocity - state A.**

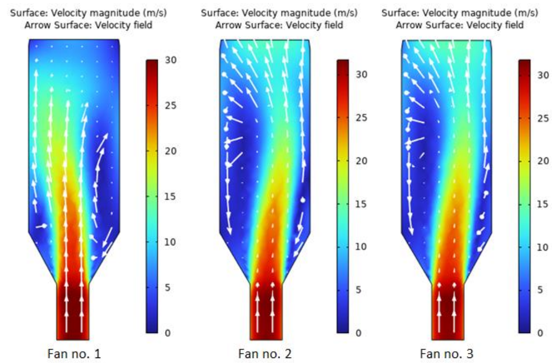


**Figure 9. CFD results: Flow velocity - state B.**

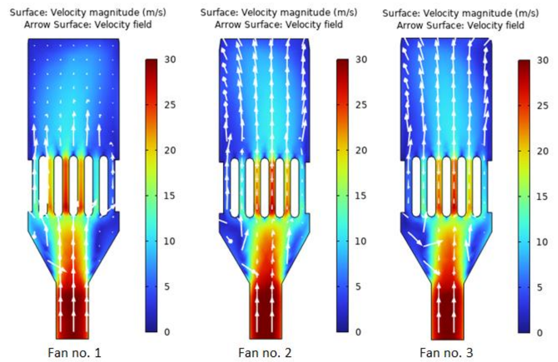


**Figure 10. CFD results: Flow velocity - state C.**

## 3. MEASUREMENT RESULTS

### 3.1 Flow rate measurements

Tests were performed on the volumetric flow rate [m$^3$/h] in the chimney duct for the three test states A, B, C. The results are shown in the graph. Based on the measurements, there was a significant increase in the discharge of the system after the removal of elements of the old silencer from inside the duct. For the condition after the installation of the new silencer insert, no deterioration in flow parameters was noted.

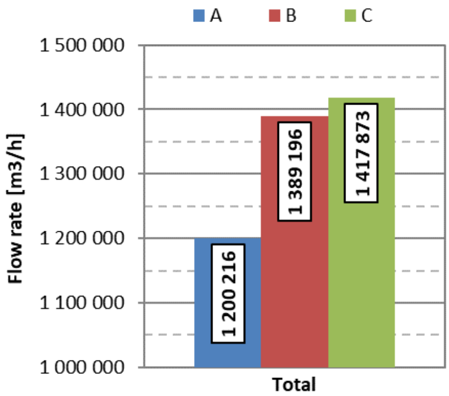


**Figure 11. Flow rate measurement results.**

### 3.2 Sound power measurements

Noise emission measurements were made at the chimney outlet (the tests are carried out in accordance with the guidelines of ISO 3740 standards and taking into account DIN 45635-47 [9]) for three test states A, B, C.

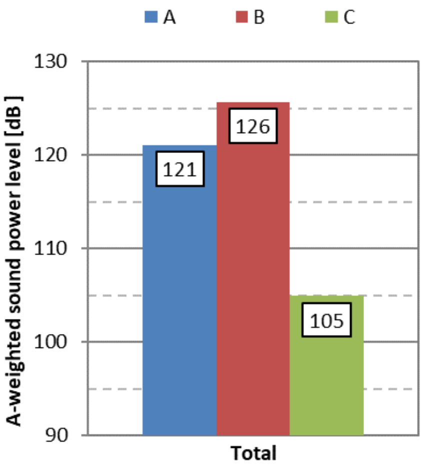


**Figure 12. Results of sound power level measurements.**

Based on the measurements, the attenuation of the acoustic silencer insertion loss is 21 dB. The damping was determined taking into account the second (state B) and third (state C) measurement sessions, since in these sessions the operation of the fans was similar, and thus also the output of the fans. The following graph (Figure 13) shows the noise spectrum for all test states.

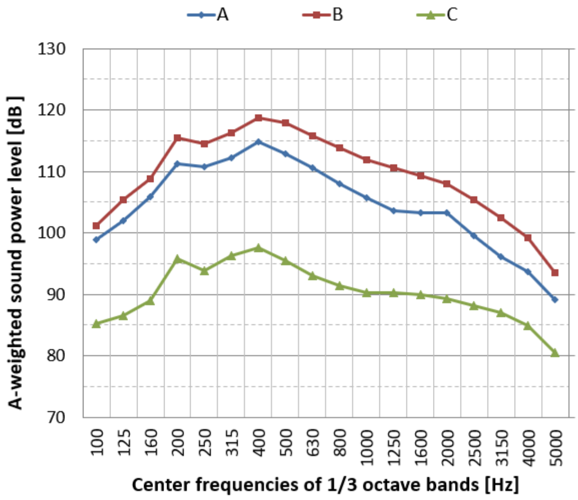


**Figure 13. Results of sound power level measurements - 1/3 octave bands.**

### 3.3 Discussion of measurement results

From the results of the measurements, it can be seen that the increase in flow rate between state B and state A (Figure 11) impacts the noise emission increase (Figure 12). This is as well because of the change in noise emission of the fan due to different operation points of the installation and because of the increase of flow velocity in the chimney stack and the bigger importance of noise emission caused by turbulent flow.

The results of state C versus state B (Figure 11) should not be interpreted as the installation of the silencer caused an increase in flow rat in the chimney. It should be assumed that the installation of silencers has little effect on the change in operating points in a given situation and the recorded values are within the range of measurement uncertainty. It can be concluded that the removal of the remains of the old silencer and the installation of a new silencer has improved the flow rate in the chimney stack.

## 4. COMPARISON OF MEASUREMENTS WITH CALCULATION RESULTS

The difference in insertion loss based on calculations and measurements is mainly due to the lack of consideration of noise generated by turbulent airflow. The following calculation results are presented:

**C.** measured state

**B-IL.** B state with subtracted values of attenuation of the insertion of the attenuator with no flow

**B-IL+SN1.** B-IL state where the self-noise of the silencer is included, where it was assumed 6 m/s avarage air flow velocity (according chapter 2.3).

**B-IL+SN2.** B-IL state where the self-noise of the silencer is included, where air flow velocity is derived from the CFD simulations (19 m/s).

**B-IL+SN2+adj.** B-IL+SN2 state where a correction is added related to difference in flow rate for states B and C (derived from the relationship for flow noise according to VDI_2081).

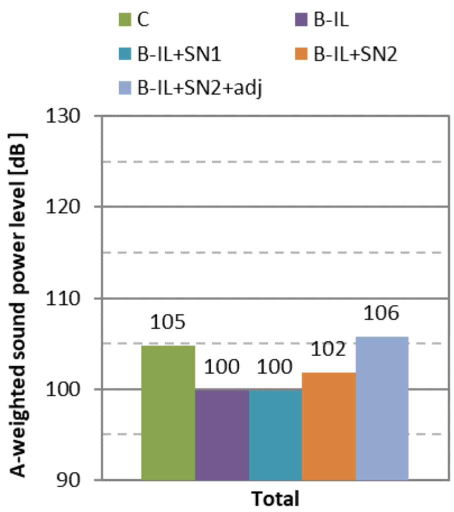


**Figure 14. Results of sound power level measurements vs calculations.**

By compering [B-IL+SN2] to [B-IL+SN1] using figure 15 it can be noted that taking into account the air flow velocity obtained by CFD simulation allowed to approach the characteristics obtained by measurement.

The normalization of noise emissions to the same flow conditions proved to be an important aspect in evaluating the measurement results.

Investigating the difference in frequency characteristics between state C and state B-IL+SN2+adj it can be seen that there is a mismatch in low and high-frequency regions. This may be due to the assumption regarding the frequency characteristics of the silencer's self-noise and the noise caused by the turbulent flow in the section between the silencer and the chimney outlet. For identifying high-frequency mismatch further study is needed. This could be related to the insertion loss characteristic of the silencer in the flow condition.

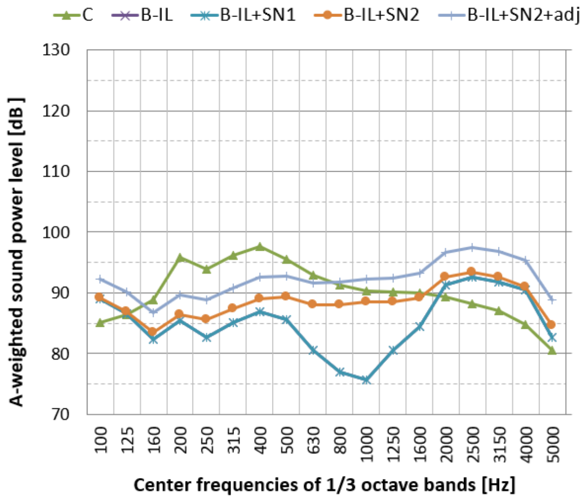


**Figure 15. Results of sound power level measurements vs calculations - 1/3 octave bands.**

## 5. CONCLUSIONS.

It should be noted that the presented approach to determining the insertion loss of plate dampers under no-flow conditions has a very high agreement with laboratory experiments. In real industrial situations, however, methods for determining the noise caused by airflow become crucial.

Information about the distribution of air velocity in a duct can be obtained from CFD simulations, and they are becoming an essential part of the design process, where by using the right velocity values a better correlation with reality can be obtained.

From the perspective of the subject under discussion in this article, the development of computational methods for predicting flow noise in complex systems is becoming crucial.

One of the key issues in the process of noise reduction for air handling systems is to determine the operating point for which protection is to be designed.

## 6. ACKNOWLEDGMENTS

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