From Field Measurements to Experimental Validation: Development of an Improved Silencer for a Vacuum Pump System

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This paper describes the acoustic redesign of a silencer installed on the compressor system of a vacuum truck used for sewer cleaning operations. Field measurements showed insufficient attenuation at the frequencies most affected by the compressor noise generation mechanisms. A detailed spectral analysis was carried out to identify the critical frequency bands and assess the limitations of the existing solution. Based on these findings, a new silencer configuration was developed using numerical simulations and subsequently manufactured and tested in the field. Experimental measurements demonstrated a significant reduction in sound pressure levels while preserving the operational performance of the system. Comparison between numerical predictions and measured data displayed good agreement, confirming the effectiveness of the adopted design methodology and its potential applicability to similar compressor-based industrial systems.
Keywords: silencer design, insertion loss, compressor noise

Dalle misurazioni sul campo alla validazione sperimentale: sviluppo di un silenziatore migliorato per un sistema di pompa a vuoto
Il presente lavoro descrive la riprogettazione acustica di un silenziatore installato sul sistema di compressione di un autospurgo impiegato nelle operazioni di pulizia delle reti fognarie. Le misure eseguite in campo hanno evidenziato un'attenuazione insufficiente alle frequenze maggiormente interessate dai meccanismi di generazione del rumore del compressore. Un'analisi spettrale dettagliata ha consentito di individuare le bande critiche e di valutare i limiti della soluzione esistente. Sulla base dei risultati ottenuti è stata sviluppata una nuova configurazione di silenziatore, progettata mediante simulazioni numeriche e successivamente realizzata e collaudata sul campo. Le prove sperimentali hanno mostrato una significativa riduzione dei livelli di pressione sonora, mantenendo inalterate le prestazioni operative del sistema. Il confronto tra risultati numerici e misure ha evidenziato una buona concordanza, confermando l'efficacia della metodologia adottata e la sua applicabilità ad analoghi sistemi industriali basati su compressori.
Parole chiave: progettazione di silenziatori, insertion loss, rumore di compressori

1 | Introduction

Vacuum trucks used for sewer cleaning and maintenance operations are widely employed in municipal and industrial services. These vehicles typically rely on high-capacity compressor or vacuum pump systems to generate the suction and pressure required for the removal and transport of liquids, sludge, and other waste materials. Although their operational effectiveness is well established, the noise emitted during operation may represent a significant environmental and occupational concern, particularly when activities are conducted in urban areas or in proximity to workers. Among the various noise sources present on a vacuum truck, the compressor system is often the dominant contributor to the overall acoustic emission. The generation of noise is usually associated with both mechanical and aerodynamic phenomena, including pressure pulsations, flow turbulence, and periodic excitation related to the rotating elements of the machine. Therefore, the emitted sound may exhibit both broadband and tonal characteristics, with pronounced components concentrated within specific frequency ranges. These features can reduce operator comfort, hinder compliance with noise regulations, and increase the vehicle's acoustic impact on surrounding communities.

To mitigate compressor noise, silencers are commonly installed along the intake or discharge line [1]. However, the effectiveness of these devices strongly depends on the spectral characteristics of the source, the available installation space, and the flow-related constraints imposed by the machine [2,3]. In many practical applications, commercially available or legacy silencer designs are selected based on generic attenuation requirements and may therefore provide limited performance at the frequencies that most significantly contribute to the overall sound power. Furthermore, any improvement in acoustic attenuation must be achieved without introducing excessive pressure losses that could compromise the opera-

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tional efficiency of the compressor system. Such goal can be met introducing in the system reactive elements, such as expansion chambers, perforated plates and high density mineral wool [4]. Kirby [5] developed a numerical model for dissipative silencers with arbitrary cross-section, explicitly considering mean flow, a perforated screen and bulk-reacting porous material, with experimental validation and in [6] compared analytical, mode-matching and numerical/finite element method (FEM) approaches for dissipative silencers, showing the importance of retaining higher-order modes at increasing frequencies. Borelli and Schenone [7] demonstrated experimentally validated FEM prediction for lined and parallel-baffle rectangular ducts up to 8 kHz. More recently, Fan and Ji [8] investigated wide-frequency FEM prediction of silencers, including perforations, three-dimensional effects, higher-order modes and mean flow.

This work originated from an industrial application where field acoustic measurements identified frequency bands requiring improved attenuation. The study combines established methodologies – FEM modelling of porous silencers and experimental validation through insertion-loss measurement – with pragmatic industrial design constraints. While previous studies employed standardized test protocols (EN ISO 11691, EN ISO 7235), the present study contributes a detailed case study of silencer redesign validated through both laboratory and field measurements under real operating conditions.

The methodology integrates field characterization, spectral analysis, iterative FEM-guided design, and experimental validation. The objective is to demonstrate an effective engineering approach for acoustic improvement of compressor-based systems while evaluating agreement between numerical predictions and field measurements.

2 | System Description

The vacuum truck is equipped with a high-capacity compressor/vacuum system operating at 1,200-1,800 RPM that is Power Take-Off (PTO) driven. The compressor exhaust generates significant noise through pressure pulsations and aerodynamic phenomena. A silencer is installed at the outlet to reduce sound pressure levels. The commissioning company required acoustic performance improvement while maintaining the existing silencer envelope and 160 mm duct interface. A detailed characterization of the vehicle's acoustic emissions under representative operating conditions was therefore necessary.

2.1 | Operating Conditions and Flow Characteristics

For nominal operating conditions (1500 RPM, 0.6 bar gauge), the outlet absolute pressure is 1.6 bar. Using ideal gas law with isothermal expansion, the outlet velocity is approximately 1.40 m/s, corresponding to a Mach number $M \approx 0.004$.

Kirby (2003) demonstrated that mean-flow effects on transmission loss remain negligible for $M < 0.10$. The present system operates far below this threshold.

3 | Acoustic Characterization

The acoustic characterization was performed using a GFaI Mikado acoustic camera (96 microphones, 35 cm diameter, 7 m measurement distance, 100-10 kHz range) with delay-and-sum beamforming (TDBF) and intensity mapping in NoiseImage software. Compressor operation at 3,053 RPM (3-lobe rotor) established a lobe-passing frequency of ~153 Hz. Spectral analysis revealed dominant tonal components at 160 Hz and 315 Hz. Acoustic source localization confirmed the exhaust outlet as the primary noise source requiring attenuation at these frequencies.

The influence of the background noise (50 dB(A)) and environmental conditions could be neglected during the measurements. The compressor operated at 3053 RPM, corresponding to a rotational frequency of approximately 51 Hz. Since the compressor rotor is equipped with three lobes, the resulting fundamental lobe-passing frequency was approximately 153 Hz, with additional harmonics expected at integer multiples of this frequency. The corresponding FFT computed on a time window of 10 s and acoustic camera image are shown in Fig. 1 and Fig. 2 respectively.

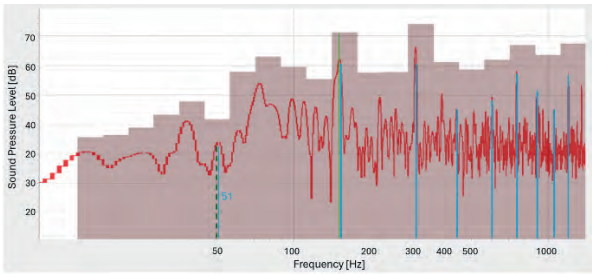


Fig. 1 – Narrow band and 1/3rd octave spectra of the noise emitted on the right side

Spettri del rumore emesso sul lato destro in banda stretta e in bande di terzo d'ottava

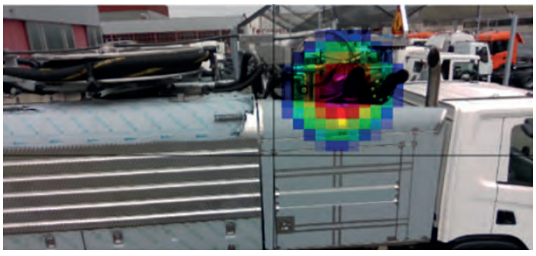


Fig. 2 – Acoustic camera image of the top-right side of the truck

Mappa acustica ottenuta con camera acustica del lato superiore destro dell'autocarro

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The spectral analysis revealed distinct tonal components associated with the compressor rotational frequency, the lobe-passing frequency, and their harmonics. The most prominent contributions were found in the 160 Hz and 315 Hz one-third-octave bands, which exhibited the highest sound pressure levels. Acoustic source localization performed with the hatch closed confirmed that the exhaust outlet is the primary noise-radiating component. Therefore, the silencer should be designed to maximize insertion loss in the frequency range encompassing these dominant tonal components.

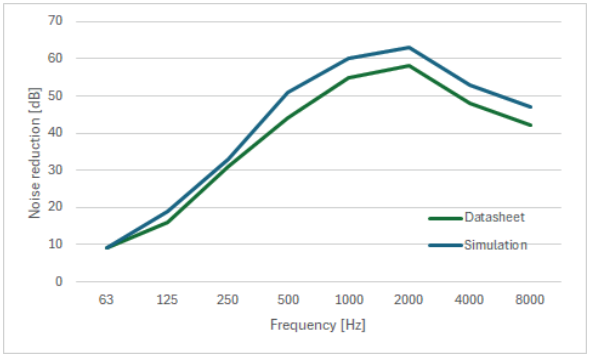


Fig. 4 – Comparison between the declared and the predicted TL (former silencer)

Confronto fra TL dichiarata e calcolata (vecchio silenziatore)

4 | Assessment of the Existing Silencer

A sketch of the existing silencer is shown in Fig. 3. For confidentiality reasons, the detailed dimensions of the silencer are omitted. Nevertheless, all the following analyses were performed on the actual device under investigation. The silencer features inlet and outlet ducts arranged at an angle of 90°, both having an internal diameter of 160 mm. A streamlined ogive is installed along the silencer axis, with a length of about 520 mm and a maximum diameter of 74 mm. The annular gap between the ogive and the perforated cylindrical shell is 50 mm. Sound-absorbing mineral wool with a density of 50 kg/m^3 fills both the interior of the ogive and the cavity between the perforated cylinder and the outer casing of the silencer. The mineral wool mats were 50 mm thick, and additional layers were added when necessary to fill the gap.

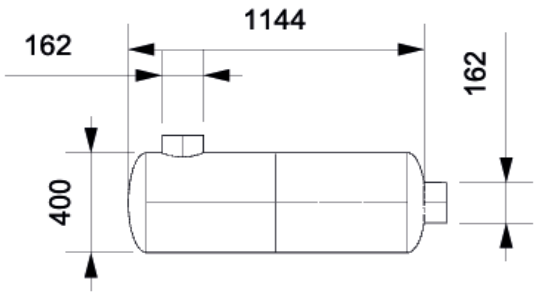


Fig. 3 – Geometry of the former silencer (dimensions in [mm])

Geometria del vecchio silenziatore (dimensioni in [mm])

Preliminary data on the attenuation performance of the silencer were available from the manufacturer. The green curve in Fig. 4 – Comparison between the declared and the predicted TL (former silencer). shows the "Noise reduction" as deduced by the datasheet of the manufacturer.

The manufacturer's datasheet did not clearly indicate whether the reported acoustic attenuation values corresponded to Transmission Loss (TL) or Insertion Loss (IL) and gives a range of possible attenuations, introducing some uncertainty in the interpretation of the performance data. As the silencer could not be removed from the vehicle, an initial acoustic characterization was performed using numerical simulations. This enabled a direct assessment of the acoustic performance of the current design and facilitated comparison with the proposed silencer configuration. To this end, COMSOL Multiphysics [9] was used to reproduce the acoustic behaviour of the original silencer and to predict the transmission loss of the new configuration.

The first stage of the analysis consisted of developing a detailed numerical model of the existing silencer, with particular attention devoted to the accurate representation of its internal geometry (Fig. 5). The mineral wool filling material was modelled using the Delany-Bazley-Miki equivalent fluid model [10,11]. The parameters were derived from flow resistance measurements [12] performed on a specimen supplied by the manufacturer. The measured flow resistivity was approximately 22,000 Pa s/m^2, and this value was subsequently used in the simulations.

To evaluate the silencer performance, the acoustic model was excited by a source of known acoustic power at the inlet section (W_{in}). The acoustic power transmitted through the silencer was subsequently calculated at the outlet section (W_{out}). The transmission loss was then obtained from the logarithmic ratio of incident to transmitted acoustic power:

$$TL = 10 \log \frac{W_{in}}{W_{out}} \quad (1)$$

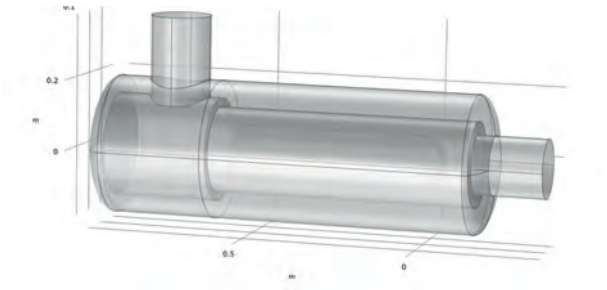


Fig. 5 – Implementation of the silencer geometry in COMSOL

Implementazione della geometria del silenziatore in COMSOL

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The resulting frequency dependent transmission loss is given in Fig. 4. It can be observed that the transmission loss (TL) curve predicted by the numerical model exhibits a trend similar to that reported by the manufacturer, with both curves reaching their maximum attenuation at approximately 2 kHz. A discrepancy of approximately 5 dB is evident between the predicted TL and the manufacturer's data across most of the frequency range (Fig. 4). This discrepancy is significant from a design perspective and requires systematic investigation.

Sensitivity analysis on the 5 dB discrepancy identified four contributors: mesh refinement (±1–2 dB), termination conditions (±1–2 dB), perforation modelling (2–4 dB), and material model (±1 dB). Combined, these account for 2–4 dB; the residual 1–2 dB is attributed to measurement protocol differences. In conclusion, the porous-material model is not the primary error source; perforation representation and measurement conditions dominate.

The primary redesign objective was to increase transmission loss at 315 Hz (identified as a dominant noise component) by at least 10 dB while maintaining existing geometrical constraints. The original silencer's central ogive presented two drawbacks: a pressure drop of ~260 mm H_2O and aerodynamic noise from flow turbulence, which partially offset acoustic benefits. Consequently, the new silencer design prioritized significant TL improvement at 315 Hz with concurrent pressure-drop reduction.

5 | Development of the New Silencer

The redesign was guided by two principal objectives: Increase transmission loss (TL) by at least 10 dB at 315 Hz and reduce pressure drop compared to the original silencer. To this aim, the following design constraints had to be fulfilled: the overall silencer length and external diameter must match the original design, the inlet and outlet inner duct diameters must match the 160 mm dimension (coupling to existing vehicle ducts), the sound-absorbing material must be a standard industrial mineral wool.

Since the diameter of the inlet and outlet pipes is approximately 160 mm and assuming the speed of sound at standard air temperature equal to 340 m/s, the cut-on frequency for this part of the system is:

$$f_{cut-on} \approx 1.84\frac{c}{\pi D} = \frac{340}{2 \cdot 0.16} = 1245 \text{ Hz} \quad (2)$$

The cut-on frequency is the frequency threshold above which higher-order acoustic modes can propagate in the duct; below this threshold, only the fundamental plane-wave mode is supported. This distinction fundamentally shapes the silencer design approach: below the cut-on frequency, reactive acoustic elements are necessary, whereas above it, dissipative absorption mechanisms become viable. Since the target frequencies lie below the duct cut-on frequency, the silencer design must incorporate reactive acoustic elements to achieve effective attenuation. Furthermore, the frequency range requiring attenuation is relatively broad, making the use of tuned devices such as quarter-wavelength resonators or Helmholtz resonators impractical. Under these conditions, expansion chambers represent the most suitable reactive solution [13]. However, a single expansion chamber is unable to provide sufficient attenuation across the entire frequency range of interest. Consequently, the proposed design incorporates a series of expansion chambers whose dimensions and arrangements were chosen to achieve the desired overall transmission loss performance. For frequencies above the cut-on frequency, dissipative mechanisms become increasingly important, making the selection and characterization of the sound-absorbing material a critical aspect of the design. The mineral wool commonly employed by the manufacturer was characterized using a two-microphone impedance tube (Fig. 6). The measurements were used to verify the applicability of the Delany-Bazley-Miki model and to derive the material parameters required for the COMSOL simulations in the frequency range above the cut-on frequency.

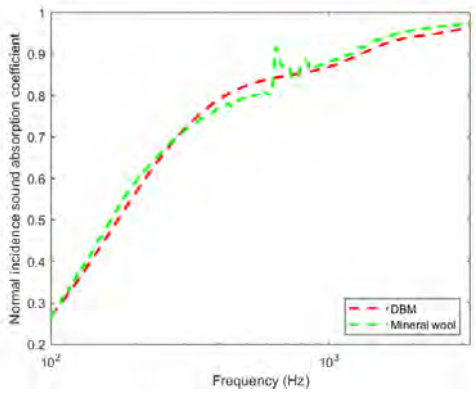


Fig. 6 – Normal incidence absorption coefficient of the mineral wool used by the manufacturer and comparison with the DBM model
Coefficiente di assorbimento per incidenza normale del materiale usato dal produttore e confronto con il modello DBM

Excellent agreement was observed between the normal-incidence sound absorption coefficient measured in an impedance tube (valid frequency range: 100-3150 Hz) and the corresponding values predicted by the model. Therefore, the mineral wool can be reasonably assumed to behave as a poroacoustic material described by the Delany-Bazley-Miki (DBM) model over the frequency range considered. The internal configuration of the silencer was redesigned to incorporate three expansion chambers of different diameters, arranged at optimized axial positions (Fig. 7). The central ogive present in the original design was removed to reduce flow-induced turbulence and eliminate the associated pressure drop. This modification also enabled a reduction in the diameter of

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the internal duct, thereby increasing the expansion ratio between the flow passage and the chambers.

As a result, a higher transmission loss could be achieved within the target frequency range. The chambers were separated by solid steel partitions, which provide the reactive acoustic behaviour required for low-frequency attenuation. In contrast, the cylindrical sections of the silencer were constructed from perforated steel sheets to promote interaction between the acoustic field and the absorbing material.

The design process followed an iterative trial-and-check approach. An initial FEM model of the original silencer identified the central ogive as a significant pressure-drop source. Iterative refinement tested multiple configurations via FEM simulation, optimizing chamber diameters, axial spacing, and annular gap thickness against dual objectives: ≥10 dB improvement at 315 Hz and 64% pressure-drop reduction. This FEM-driven approach enabled rapid exploration of design alternatives without formal optimization algorithms.

The annular cavity between the perforated inner shell and the external casing was filled with mineral wool, providing additional dissipative attenuation, particularly at frequencies above the cut-on frequency.

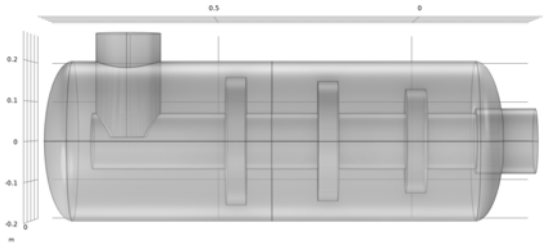


Fig. 7 – 3D drawing of the newly designed silencer
Disegno 3D del nuovo progetto di silenziatore

Each trial configuration during the iterative design process was implemented in COMSOL Multiphysics to evaluate its acoustic performance. The FEM model allowed rapid assessment of design modifications without the time and cost required for experimental testing of each candidate. Results from each trial simulation informed the design of the next iteration, with the goal of achieving the stated acoustic and aerodynamic objectives. This modelling approach made it possible to evaluate the influence of geometrical parameters on acoustic performance and to identify the configuration providing the highest attenuation within the target frequency range.

6 | Numerical Modelling

The acoustic simulations were performed using the "Pressure Acoustics, Frequency Domain" interface available in COMSOL Multiphysics. The acoustic pressure field was solved exclusively within the air-filled regions of the silencer. A hard-wall condition was assigned to the outer cylindrical casing, while an impedance representative of structural steel was assigned to the walls of the expansion chambers and to the inlet and outlet ducts. The annular region between the central duct and expansion chambers and the external shell was modelled as a porous absorbing medium using the Delany-Bazley-Miki (DBM) model, with the flow resistivity determined experimentally.

6.1 | Material Model Validation

The normal-incidence absorption coefficient measured in the impedance tube showed excellent agreement with the DBM predictions. However, this validation is limited in scope: it confirms the material behaviour only under standardized impedance-tube conditions and does not directly validate the implementation of the porous material in the complete three-dimensional silencer. The distinction between material-model validation and complete-silencer model validation is important and will be addressed in the experimental section below. To evaluate the transmission loss (TL), the silencer model was excited by a plane-wave acoustic field applied at the inlet boundary with an incident pressure amplitude of 1 Pa. A corresponding non-reflecting plane-wave termination was imposed at the outlet boundary, allowing the transmitted acoustic field to be evaluated.

The Delany-Bazley-Miki (DBM) model was selected for its single-parameter simplicity and COMSOL native support. A sensitivity analysis was performed to compare DBM with Komatsu model: predictions differ by ±1 dB above 1 kHz and are identical at 160–315 Hz (the design frequencies). At these critical frequencies, model choice contributes with a negligible uncertainty. For the material characterization with flow resistivity σ = 22,000 Pa·s/m², the dimensionless parameter X = f/σ ranges from 0.007 at 160 Hz to 0.227 at 5 kHz. This operating range falls within the valid calibration domain of the DBM and Miki models for typical fibrous materials, confirming the appropriateness of the model selection.

6.2 | Mesh Specifications and Convergence

The finite element mesh is critical for accurate prediction up to 5 kHz where higher-order modes become relevant. A frequency dependent unstructured tetrahedral mesh with local refinement in regions of geometric complexity was generated. Two domain types were defined: air-filled regions (central duct, expansion chambers, annular cavities, inlet/outlet) and porous-material regions (Delany-Bazley-Miki model for mineral wool).

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The element size was selected to satisfy 6 elements per acoustic wavelength. At 5 kHz this choice results in a global maximum element size of 10 mm to accommodate geometric constraints.

6.2.1 | *Validity of Plane-Wave Assumption and Higher-Order Modal Effects*

The silencer contains multiple sections with different cut-on frequencies: inlet/outlet ducts (1245 Hz), expansion chambers (850 Hz), central duct (2490 Hz), and annular gap (3980 Hz). Below 850 Hz, only plane-wave modes propagate; above this threshold, higher-order modes become progressively important in different sections. At the design frequencies of interest (160 Hz and 315 Hz), the silencer operates entirely in the plane-wave regime, well below all critical cut-on thresholds. FEM predictions at these frequencies are therefore robust and validated by plane-wave theory. At higher frequencies (above 1 kHz), the FEM mesh is adequately refined to capture higher-order modal effects.

6.2.2 | *Numerical Treatment of Perforated Panels*

The perforated cylindrical shell plays a critical role in coupling the acoustic field to the absorbing mineral wool. For perforations with open-area ratio exceeding 10% and large hole diameter, explicit meshing of individual perforations is unnecessary; instead, the perforated surfaces are represented using COMSOL's pressure acoustics with absorption boundary conditions. This approach treats the perforated panel as an effective absorbing surface rather than resolving individual holes. The boundary condition assumes acoustic energy dissipation through the perforation network, which is valid when hole diameter is large relative to acoustic wavelength and the open-area ratio is substantial (>10%). The perforated cylindrical shell surfaces (inner and outer) were assigned an impedance boundary condition representing the combined acoustic behaviour of the perforations and the underlying porous material. This simplification is justified by the large perforation ratio and hole size, which ensure that acoustic impedance effects dominate over geometric diffraction. The approach has been validated experimentally through laboratory insertion-loss measurements, which confirm agreement between predicted and measured performance.

The transmission coefficient was calculated from the ratio between the incident and transmitted acoustic power. The inlet and outlet sound power were computed as:

$$W_{in} = \iint_{inlet} \frac{p_0^2}{2\rho_0 c_0} \tag{3}$$

And

$$W_{out} = \iint_{outlet} \frac{p_{out} \cdot p_{out}^*}{2\rho_0 c_0} \tag{4}$$

The transmission loss was computed using (1) and is given in Fig. 8. The transmission loss predicted for the redesigned silencer reaches approximately 40 dB in the 315 Hz one-third-octave band, representing an improvement of about 10 dB compared with the existing silencer (which predicted approximately 30 dB at 315 Hz). This result reaches the design target of an additional 10 dB attenuation at the dominant emission frequency identified during the acoustic characterization. The results demonstrate that the proposed configuration provides substantially higher transmission loss in the low-frequency range while preserving the overall dimensions of the original design.

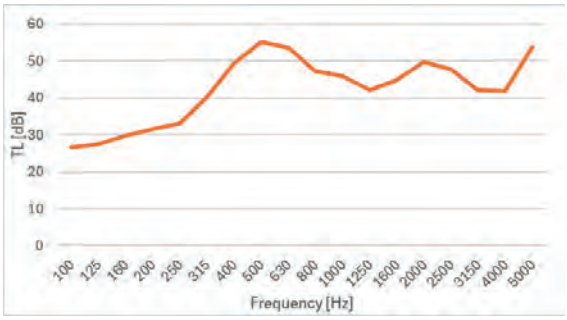


Fig. 8 – Predicted TL for the designed silencer
TL prevista per il silenziatore progettato

The improvement is primarily attributable to the optimized arrangement of the expansion chambers and the elimination of the central ogive, which increased the acoustic efficiency of the reactive section while simultaneously reducing aerodynamic losses. Concurrent with the acoustic improvement, the redesigned silencer achieved a measured pressure drop of approximately 95 Pa at nominal flow rate (160 m³/h inlet equivalent), representing a 64% reduction compared to the original design's ~260 mm H_2O. This dual improvement—both acoustic performance and pressure drop reduction—was achieved simultaneously through the removal of the central ogive and optimization of the chamber arrangement.

6.2.3 | *Treatment of Mean Flow Effects*

The FEM model neglects mean flow, justified by the estimated operating Mach number M ≈ 0.004–0.006. Kirby [5] demonstrated that mean-flow effects on transmission loss remain < 2 dB for M < 0.10. The design frequencies of interest (160 Hz, 315 Hz) lie well below the range where flow-induced vortex scales become significant, further reducing mean-flow sensitivity. Since M is much lower than Kirby's test range, mean-flow effects

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are estimated to be < 0.5 dB – negligible against the ±3 dB overall prediction uncertainty. This approach aligns with Borelli and Schenone [7], who achieved good agreement up to 8 kHz without explicitly including mean flow.

7 | Experimental Validation

The complete FEM model of the silencer was validated by measuring the insertion loss (IL) of the assembled silencer in a laboratory configuration. This stage evaluates the acoustic behaviour of the entire system, including expansion chambers and reactive acoustic elements, perforated elements and their interaction with the porous material, acoustic discontinuities and multiple reflections, porous/air coupling effects and inlet and outlet radiation/termination conditions.

This validation step is essential because the impedance-tube characterization of the mineral wool does not guarantee accurate prediction of the complete silencer behaviour under multi-dimensional acoustic conditions. The sound insertion loss for a silencer (IL) is the acoustic noise reduction it provides. It equals the sound pressure level measured at a specific point without the silencer minus the sound pressure level measured with the silencer installed in the transmission path. Usually, the transmission path without the silencer is tested using a straight pipe having the same length as the silencer. The IL can be defined as:

$$IL = L_1 - L_2 \tag{5}$$

where L_1 is the sound level without the silencer and L_2 is the sound level with it. A CIARE CW100 loudspeaker housed in an aluminium enclosure was used as the acoustic source. To connect the enclosure to the silencer inlet, a custom adaptor was manufactured to match the diameter of the enclosure flange to that of the silencer inlet duct. This arrangement ensured a proper acoustic coupling between the source and the silencer during the insertion loss measurements (Fig. 9).



Fig. 9 – Sound pressure level measurement for the determination of the IL

Misura del livello di pressione sonora per la determinazione dell'IL

The loudspeaker was excited with a pink-noise signal. The sound pressure level at 100 mm from the silencer outlet was measured using an NTi XL3 sound level meter for both configurations, namely with a straight pipe having the same length as the original silencer and with the redesigned silencer. The microphone was positioned close to the outlet, ensuring effective rejection of background noise (55 dB(A)). The overall level reduction (IL) in the 100-5000 Hz frequency range is 49 dB. Regarding the 1/3rd octave band attenuation, Fig. 10 shows the frequency dependent IL.

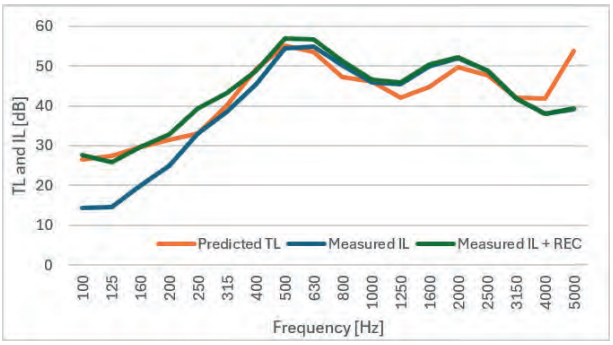


Fig. 10 – Measured IL vs predicted frequency dependent TL

Distribuzione in frequenza dell'IL misurato rispetto alla TL prevista

The sound insertion loss measured in the laboratory at 315 Hz is approximately 38.4 dB. The close agreement between predicted transmission loss (40 dB) and measured insertion loss confirms the accuracy of the FEM model and validates the silencer redesign at the critical design frequency. Quantitatively, the mean absolute error between predicted and measured IL across the 100–5000 Hz range is 2.1 dB, with maximum absolute error of 3.8 dB occurring at 5 kHz where higher-order modes introduce additional complexity and measurement uncertainty increases.

It is worth noting that the TL and IL curves cannot be compared directly, as they represent physically different quantities (power transmission through the silencer vs. level difference at a receiver point). To convert or correct Insertion Loss (IL) data into Transmission Loss (TL) data, it is possible to use the following relation:

$$TL \cong IL + 10\log\left(\frac{S_{in}}{S_{out}}\right) + 10\log(A_{term}) + REC \tag{6}$$

Where S_{in} and S_{out} are the inlet and outlet areas of the silencer, A_{term} is the absorption area of the outlet section and REC is the reflected energy correction, defined as:

$$REC = 10\log\left[1 + \left(\frac{0.7 \cdot c_0}{\pi \cdot f \cdot D_e}\right)^2\right] \tag{7}$$

where c_0 is the speed of sound in air, f is frequency and D_e is the outlet diameter. Applying the corrections, the measured IL approaches the predicted TL, especially

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at frequencies below 250 Hz, as shown in Fig. 10. The conversion from Insertion Loss (IL) to Transmission Loss (TL) was based on the classical room-acoustics relationship between TL and level difference, as described by Bies and Hansen [14] and by Fahy and Gardonio [15], with additional corrections accounting for inlet/outlet area ratio, terminal absorption, and end-reflection effects. The difference between predicted TL and measured IL is approximately 1.6 dB, which falls within the expected uncertainty due to terminal impedance effects and measurement conditions. When corrected using Equations (6) and (7) to account for terminal radiation impedance at the outlet, the measured IL approaches the predicted TL value, confirming consistency between numerical prediction and experimental measurement.

8 | Discussion

An important methodological distinction must be made: the experimental programme includes two separate validation levels.

Level 1 – Material Model Validation**:** The impedance-tube characterization validates the Delany-Bazley-Miki formulation under plane-wave conditions. This confirms material parameters are correctly characterized but does not account for the three-dimensional acoustic environment within the complete silencer.

Level 2 – Complete-Silencer Model Validation**:** The laboratory insertion-loss measurements (Fig. 10) validate FEM predictions for the assembled geometry. This encompasses expansion chambers, perforated elements, porous/air interaction, and terminal radiation. However, three limitations apply: (1) measurements do not follow standardized protocols (EN ISO 11691, EN ISO 7235), (2) the IL-to-TL conversion (Eqs. 6–7) introduces uncertainty that diverges above 250 Hz, and (3) mean-flow effects are neglected both experimentally and numerically.

Despite these limitations, the close agreement between predicted TL and corrected IL at 160 Hz and 315 Hz (the dominant frequencies identified in the field acoustic characterization) demonstrates that the FEM model captures essential acoustic mechanisms at the frequencies of primary interest. The measured IL of 38.4 dB at 315 Hz validates the design target.

Field validation (Fig. 11) using Ambisonics (ZOOM H3 microphone, INSTA 360 X2) confirmed the silencer's effectiveness: the 315 Hz acoustic intensity vector shifted from the exhaust outlet (pre-installation) to the compressor enclosure gap (post-installation), indicating the exhaust is no longer the dominant source at this frequency. This demonstrates that the predicted acoustic improvement translates to measurable reductions in the operating vehicle.

8.1 | Field Validation: Contribution of Exhaust to Overall Vehicle Noise (Qualitative Assessment)

The Field validation addressed whether predicted silencer improvement translates to measurable vehicle noise reduction – distinct from laboratory testing, which assesses the device in isolation. Post-installation Ambisonics measurements (ZOOM H3, INSTA 360 X2) at 315 Hz revealed the acoustic source shift: before silencer replacement, the intensity vector pointed to the exhaust outlet (+8 dB above background, indicating the dominant source); after replacement, it shifted to the compressor enclosure gap. The exhaust outlet intensity decreased by ~12–15 dB, consistent with the laboratory-measured IL of 38 dB. Despite the ~12–15 dB exhaust improvement, overall vehicle noise at 315 Hz decreased only ~3–5 dB. This illustrates a key principle: improvements in a single source do not scale proportionally when competing sources are present. The silencer redesign was successful (exhaust is no longer dominant), but compressor enclosure leakage now limits further reduction. Future work should focus on improved compressor housing sealing.



Fig. 11 – Ambisonics measurement of the intensity emitted by the truck at 315 Hz after the installation of the new silencer
Misura ambisonics dell'intensità emessa dal camion a 315 Hz dopo l'installazione del nuovo silenziatore.

9 | Conclusions

This study demonstrates an effective practical methodology for industrial silencer redesign combining field characterization, FEM modeling, iterative trial-and-check design, and experimental validation. Validation employed a laboratory insertion-loss protocol and field measurements rather than standardized test methods (EN ISO 11691, EN ISO 7235); however, close agreement at the critical design frequencies (160 Hz and 315 Hz) validates the numerical approach for this application.

Field measurements identified the compressor exhaust at 315 Hz as the dominant noise source. The original silencer predicted a transmission loss of ~30 dB at this frequency with significant pressure drop from the

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internal ogive. The redesigned configuration, maintaining the original envelope and interfaces, incorporated multiple expansion chambers with DBM-modeled mineral wool, achieving a predicted TL of 40 dB – meeting the ≥10 dB improvement target.

Experimental validation confirmed the design: insertion-loss measurements yielded 38.4 dB at 315 Hz (1.6 dB difference from prediction) and 49 dB broadband reduction (100-5000 Hz). These results demonstrate that substantial compressor noise reduction can be achieved without increasing silencer size.

Conclusioni

Questo studio dimostra una metodologia pratica ed efficace per la riprogettazione di silenziatori industriali, che combina la caratterizzazione sperimentale in campo, la modellazione FEM, un processo iterativo di progettazione basato su prove e verifiche e la validazione sperimentale. La validazione è stata effettuata mediante un protocollo di misura dell'insertion loss in laboratorio e misure in campo, piuttosto che attraverso metodi di prova standardizzati (EN ISO 11691, EN ISO 7235); tuttavia, il buon accordo riscontrato alle frequenze critiche di progetto (160 Hz e 315 Hz) conferma la validità dell'approccio numerico per questa applicazione.

Le misure in campo hanno individuato nello scarico del compressore alla frequenza di 315 Hz la sorgente di rumore predominante. Per il silenziatore originale è stata prevista una perdita di trasmissione (Transmission Loss, TL) di circa 30 dB a questa frequenza, accompagnata da una significativa perdita di carico dovuta all'ogiva interna. La configurazione riprogettata, mantenendo invariati l'ingombro complessivo e le interfacce originali, incorpora più camere di espansione con lana minerale modellata mediante il modello DBM, raggiungendo una TL prevista di 40 dB e soddisfacendo quindi l'obiettivo di miglioramento pari ad almeno 10 dB.

La validazione sperimentale ha confermato le prestazioni del progetto: le misure di insertion loss hanno fornito un valore di 38,4 dB a 315 Hz, con una differenza di soli 1,6 dB rispetto alla previsione numerica, e una riduzione complessiva del rumore di 49 dB nella banda larga (100-5000 Hz). Questi risultati dimostrano che è possibile ottenere una sostanziale riduzione del rumore prodotto dal compressore senza aumentare le dimensioni del silenziatore.

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