

Investigating different acoustic wall treatment materials for attenuation of noise and vibration in a supersonic wind tunnel

A Mogwera¹, D A Desai^{1,3}, F Fameso¹ and S Dikgale²

¹ Tshwane University of Technology, Department of Mechanical and Mechatronics Engineering, Staatsartillerie Road, Pretoria West, Pretoria, 0183, South Africa

² Council of Scientific & Industrial Research (CSIR) , Meiring Naudé Rd, Brummeria, Pretoria, 0001, South Africa

³ Author to whom any correspondence should be addressed
Email: desaida@tut.ac.za

Abstract. The demand for high-performance testing environments with reduced noise levels has driven increased focus on the aerodynamics and acoustics of supersonic wind tunnels. Operating at supersonic speeds, these wind tunnels generate significant noise and vibrations that can interfere with measurement accuracy, cause structural degradation, and disrupt operations. Traditional approaches to attenuate these problems often focused on airflow dynamics, somehow neglecting the impact of compressor fan vibrations and dynamics. However, acoustic wall treatments placed where fans are commonly located in the wind tunnel, can minimize these sounds and vibrations but such investigations seem under researched. Hence, this study investigates the vibro-acoustic performance of three materials: ceramic fibre, mineral wool, and melamine foam using finite element analysis (FEA). The simulated results indicate that ceramic fibre achieved the lowest A-weighted SPL of 24.04 dBA, demonstrating its strong sound-absorbing properties compared to melamine foam and mineral wool, with SPLs of 48.687 dBA and 35.357 dBA, respectively. Analytical validation further confirmed the accuracy of the simulated harmonic response analysis results. These results highlight ceramic fibre as a highly effective material for acoustic wall

treatments in minimizing compressor fan-induced noise and vibrations in supersonic wind tunnel applications such as those found at the CSIR. By providing a comparative analysis of material responses specific to compressor fan acoustics, this research contributes to the design of quieter, more efficient supersonic testing environments.

1. Introduction

In supersonic wind tunnel operations, controlling noise and vibrations is critical, particularly when the wind tunnel operates at Mach numbers between 0.2 and 1.4. Wind tunnels at these speeds generate extreme aerodynamic forces, and the compressor fan, which circulates and increases the pressure of air, becomes a primary source of high-frequency vibrations and intense noise levels. These intense vibrations generated by compressor fans within these tunnels can adversely affect the performance and accuracy of the results. Managing these challenges using acoustic treatments is important for maintaining the structure of the wind tunnel and ensuring smooth operations during tests.

To solve this problem, researchers have studied ways to reduce vibration and noise acoustic wall treatments in other applications [1]. These studies may not directly apply to compressor fan sections in wind tunnels, but they provide valuable insights into the potential effectiveness of acoustic wall treatments for vibration reduction. However, research on acoustic treatments for compressor fans in supersonic wind tunnels is limited. When the compressor fan in the wind tunnel operates, it produces high-frequency vibrations that result in significant noise levels. If these noise levels are sustained for long periods, they can potentially harm the hearing of operators working nearby.

There are a lot of studies on aerodynamic and aeroacoustics wind tunnels that focus more on computational fluid dynamics (CFD) simulations and research studies based on FEM are limited. For example, Cattafesta et al. [2] believe that wind tunnels play a crucial role in aerodynamics research and model testing, allowing researchers to obtain data related to flow over-scaled or full-scale models. They enable the measurement of aerodynamic forces, pressure distributions, and fluid flow behaviour. The researchers further emphasize the fact that CFD has become increasingly powerful, and wind tunnels remain vital tools for engineering research and understanding aerodynamic phenomena.

Santana et al. [3] in their research, they provide solutions to improving aerodynamic wind tunnels to be able to perform tests of aeroacoustics for vibrations and noises experienced during the operations of wind tunnels.

They did this by applying high-performance acoustic materials. They analysed three high-performance acoustic materials using CFD, at three different flow speeds, 15 m/s, 31 m/s, and 37 m/s. The improvements resulted in a reduction of broadband noise by up to 5 dB and a nearly complete elimination of spectral tones. Additionally, the turbulence level at the test section decreased from 0.25% to 0.21% due to the acoustic treatment, resulting in broadband-noise reduction of up to 5 dB. As mentioned above, most studies on aerodynamic and aeroacoustics wind tunnels focus more on CFD simulations than FEA.

Hence, this work investigates the vibration response of three materials that can be used in supersonic wind tunnel acoustic wall treatments to minimise noise and vibrations in the compressor fan section using FEA.

1.1 Modelling Assumptions

This study specifically focuses on the vibro-acoustic behaviour of acoustic treatment materials using FEA and does not incorporate CFD-based airflow simulations. This simplification allows the study to isolate the acoustic performance of the materials under idealised harmonic excitation conditions, without the influence of complex aerodynamic flow typically associated with active airflow in supersonic wind tunnels. Other delimitations include the use of a simplified cylindrical duct geometry in place of a wind tunnel model, and the assumption of isotropic, homogeneous material behaviour without temperature dependence. These choices were made to balance computational cost while focusing on the comparative acoustic performance of materials.

2. Physical description and geometry of the model

The geometry of the wind tunnel compressor fan section is a complex, multi-layered structure that includes a three-stage compressor fan system, as shown on the left in Figure 1. This detailed duct model is provided to illustrate the actual structure of the wind tunnel duct. However, it was not used for analysis, as this study does not consider a coupled acoustic-structural analysis involving the fan, duct, and acoustic materials together. Instead, a simplified model of the wind tunnel duct, shown on the right in Figure 1, was used for numerical analysis to reduce computational effort and costs. The core component of this geometry is a cylindrical duct made of structural steel, of dimensions shown in Table 1.

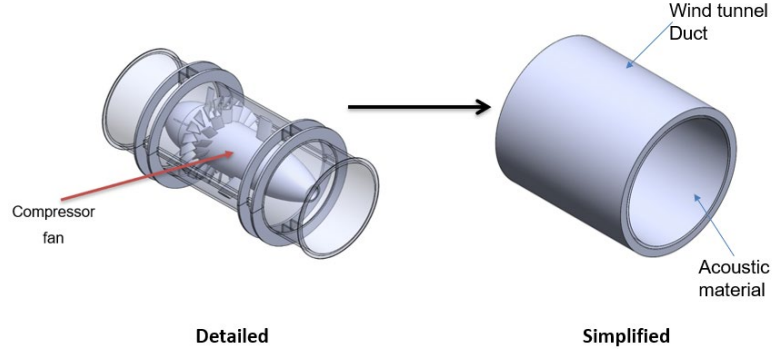


Figure 1. Detailed and simplified wind tunnel compressor fan sections.

Table 1. Wind tunnel duct dimensions.

Outer diameter (m)	Inner diameter (m)	Wall thickness (mm)	Length (m)	Material
3.2	3.07	124	5.02	Structural steel

3. Mathematical models

3.1 Acoustic wave propagation

Sound waves in air are typically modelled by the acoustic wave equation, derived from the Navier-Stokes equations under the assumption of small perturbations and linear behaviour in a homogenous, isotropic medium. This wave equation, [4] is expressed as follows

$$\frac{\partial^2 p}{\partial t^2} = c^2 \nabla^2 p \quad (1)$$

where p represents the acoustic pressure fluctuation, c is the speed of sound in air, approximately 344 m/s at room temperature, and t denotes time. The term $\nabla^2 p$ is the Laplacian of the pressure field, representing the spatial variation of pressure. This equation assumes that sound waves are propagating in a lossless medium without viscous or thermal losses.

3.2 Noise reduction coefficient (NRC)

The NRC is an empirical measure used to describe the average sound absorption capability of a material at mid-range frequencies, 250, 500, 1000, and 2000 Hz [5]. NRC is calculated by averaging the sound

absorption coefficients at these frequencies, providing a single value that simplifies material comparison. It is an important equation to consider using since it impacts the values of the SPLs. The equation is as follows:

$$NRC = \frac{\alpha_{250} + \alpha_{500} + \alpha_{1000} + \alpha_{2000}}{4} \quad (2)$$

where α is the sound absorption coefficient.

3.3 Material sound absorption coefficient

Mathematically, the absorption coefficient is expressed as [4]:

$$\alpha = 1 - \left(\frac{p_r}{p_i}\right)^2 = 1 - \left|\frac{Z_1 - Z_0}{Z_1 + Z_0}\right|^2 \quad (3)$$

where p_r is the reflected sound pressure and p_i is the incident sound pressure. Z_1 is the specific material surface impedance and Z_0 the characteristic impedance of air. Values of α range from 0 which is complete reflection to 1 which is complete absorption.

3.4 Sound pressure level (SPL)

SPL is calculated as:

$$SPL = 20 \log_{10} \left(\frac{p_{rms}}{p_0} \right) \quad (4)$$

where p_{rms} is the root-mean-square sound pressure and p_0 is the reference sound pressure.

$$p_{rms} = \frac{p_{max}}{\sqrt{2}} \quad (5)$$

The A-weighting filter adjusts the sound pressure level based on how sensitive human hearing is to that frequency. Yi et al. [6] said the principle of using the A-weighting filter is that each frequency f must be multiplied by an A-weighting factor $A(f)$, which is a standardized set of values that modify the raw sound pressure to better align with human hearing. A-weighted SPL is calculated as:

$$L_{p,A} = 10 \log_{10} \left(\frac{p_A^2}{p_0^2} \right) \quad (6)$$

where $L_{p,A}$ is the A-weighted SPL in decibels, p_A^2 is the mean squared A-weighted pressure and p_0 is the reference sound pressure 20 μPa in air.

3.5 Acoustic impedance of the medium

Empirical models like the Delany-Bazley model are used to calculate impedance for porous materials. These models take into account the material's properties, such as flow resistivity, porosity, and thickness to estimate the impedance. Delany-Bazley Equation for porous materials is given by [7]:

$$Z = \rho_0 C_0 \left(1 + 0.0571 \left(\frac{f}{\sigma} \right)^{-0.732} - j 0.087 \left(\frac{f}{\sigma} \right)^{-0.732} \right) \quad (7)$$

f is the frequency Hz , σ the flow resistivity $Pa \cdot s/m^2$ and ρ_0 and c_0 are the density and speed of sound in air.

3.6 Acoustic pressure

For the analysis in this project, the wind tunnel duct geometry is cylindrical. According to Singhal et al. [8] in their study on acoustic properties in cylindrical structures, acoustic pressure within a cylindrical cavity with rigid boundaries containing a homogenous fluid is governed by the wave equation expressed in cylindrical coordinates. This equation defines the behaviour of acoustic pressure, considering the shape and boundary conditions of the cavity, which are critical to accurately modelling sound propagation and pressure fluctuations in cylindrical ducts like those used in wind tunnels. The wave equation in cylindrical coordinates in the radial direction is as follows:

$$P_{i,m,n}(r) = \rho_f \omega^2 A_r J_n \left(\frac{r_i}{R} r \right) \quad (8)$$

where ω is the coupled natural frequency, A_r is the pressure amplitude $J'_n(r_i)$ is the Bessel function, r_i is the Bessel root function and R is the radius of the cylindrical structure.

3.7 Natural frequency response

In acoustics and structural dynamics, fluid coupling describes the interaction between a solid boundary such as the cylindrical wall of a duct and the adjacent fluid such as air, which affects how sound and vibration propagate through the system [9]. This interaction can be represented through coupled and uncoupled frequency equations. In the uncoupled approach, the cylindrical wall vibrates independently of the fluid's influence and coupled is dependent on the fluid's influence. The uncoupled and coupled cylindrical natural frequency equation given by Singhal et al. [8] are as follows

a) Uncoupled cylindrical natural frequency equation

$$\omega_{uncoupled} = \frac{1}{2\delta_0} \left[\sqrt{2} \times \sqrt{\delta_0(\delta_1 + \sqrt{\delta_2 + \delta_3})} \right] \quad (9)$$

b) Coupled cylindrical natural frequency equation

$$\omega_{coupled} = \frac{1}{2\chi_0} \left[\sqrt{2} \times \sqrt{[\chi_0(\chi_1 + \sqrt{\chi_2 + \chi_3})]} \right] \quad (10)$$

where, $\delta_{0,1,2,3}$ and $\chi_{0,1,2,3}$ are the three roots of the characteristic polynomial, $m_{m,n}$ is the Mass element from the mass matrix, $k_{m,n}$ is the stiffness element from the stiffness matrix and m, n are the desired mode shapes.

4. Numerical modelling

The FEA setup shown in Figure 2 involves creating two CAD models of a wind tunnel duct, both with and without acoustic materials. For each configuration, modal analysis is first performed to determine natural frequencies and mode shapes. This is followed by harmonic response and harmonic acoustic analyses to assess structural vibrations and acoustic behavior. In the wind tunnel duct-only model, frequency responses and acoustic pressure levels are calculated, while in the model with acoustic materials, A-weighted sound pressure levels are also evaluated. The following subsections provide a detailed breakdown of this process, including the geometry of the models, material properties, applied loads and boundary conditions, and mesh settings.

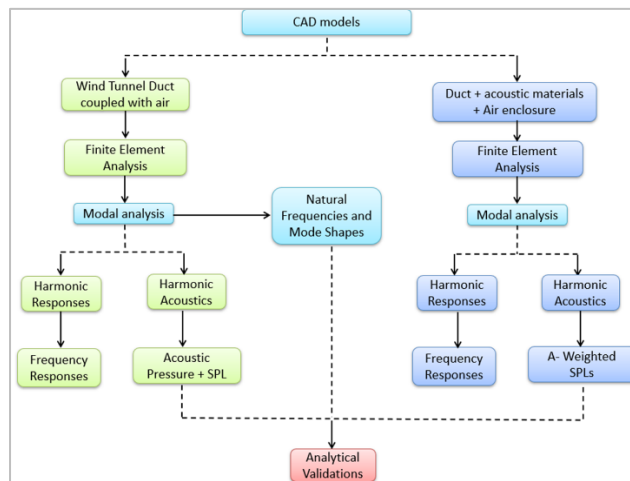


Figure 2. Methodology of the FEA Setup.

4.1 Geometry

A full harmonic acoustic analysis is executed which was divided into two sub-analyses: analysis of the wind tunnel duct coupled with air as shown on the left as part 1 in Figure 3 and analysis of the wind tunnel duct with a layer of acoustic material enclosed with air as shown in the right as part 2.

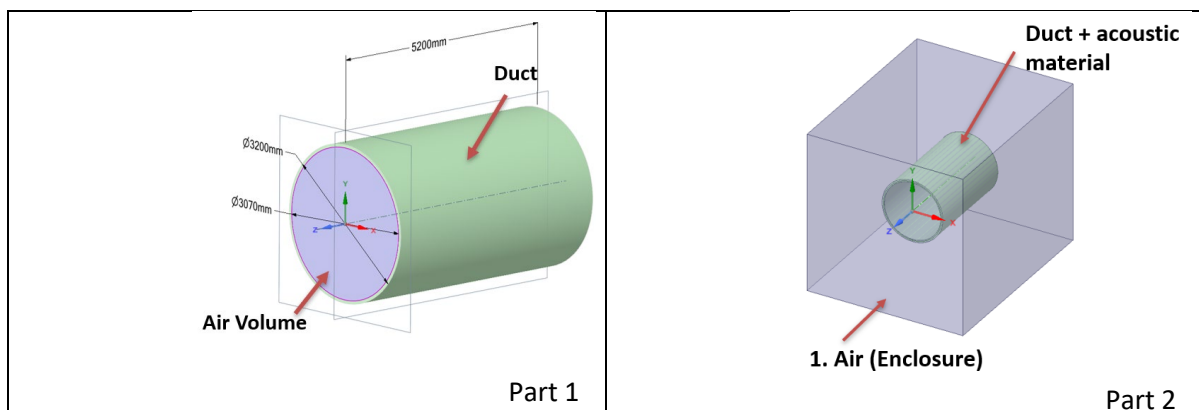


Figure 3. Numerical modelling geometries.

4.2 Loads and boundary conditions

Referring to Figure 4, the following load and boundary conditions were applied as follows:

1. A pressure of 250 kPa was applied to the internal surface of the wind tunnel duct, representing the maximum operating pressure generated by the compressor fan during standard operation at the CSIR. To simulate the vibrational response of the duct under operational conditions, a frequency-swept harmonic excitation was applied, capturing the dynamic loading effects induced by the rotating fan system.
2. To replicate real-world mounting conditions, fixed support boundary conditions were applied along the duct's outer surface, restricting all translational and rotational degrees of freedom. These conditions simulate rigid wall connections and structural constraints of the physical system.
3. Firstly, the radiation boundary condition was applied on the whole surface of the air volume to simulate sound waves radiating into free space, which is the cavity of the wind tunnel duct. Then, “imported velocity”, which is the excitations based on the vibration modes from the harmonic response, is applied to the outer surface of the air volume geometry. Secondly, impedance boundary conditions were set for the interaction between the air and the structural walls of the duct. This boundary condition accounts for the damping effects of the acoustic material. The acoustic boundary conditions used for testing each acoustic material (Mineral wool, Ceramic fibre and Melamine foam) are impedance boundary conditions and the impedance values were obtained using the Delany-Bazley equation (7).

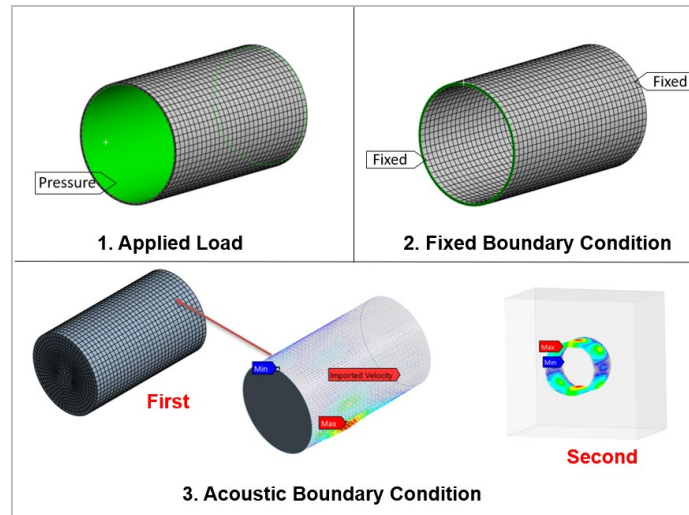


Figure 4. Loads and Boundary Conditions Applied.

4.3 Mesh design

The mesh convergence study was conducted to help obtain accurate simulation results. The mesh element type used for the wind tunnel duct coupled with air was a hex-dominant mesh with a converged seed size of 0.11 m. For the wind tunnel duct with a layer of acoustic material enclosed with air, the element type used is tetrahedron mesh with a seed size of 0.17 m. Element quality protocols were satisfied.

4.4 Finite element analysis procedures

The FEA procedure used in this work was a full method of harmonic analysis as shown in Figure 5, using ANSYS FEA software which involves multiple steps that sequentially build on each other to provide a comprehensive understanding of how a system behaves under dynamic loading, with specific emphasis on structural vibrations and acoustics. This process was divided into three steps which are A: modal analysis, B: harmonic analysis and C: harmonic acoustics as shown in Figure 3. Both parts of the analysis followed the same sequence shown in Figure 3.

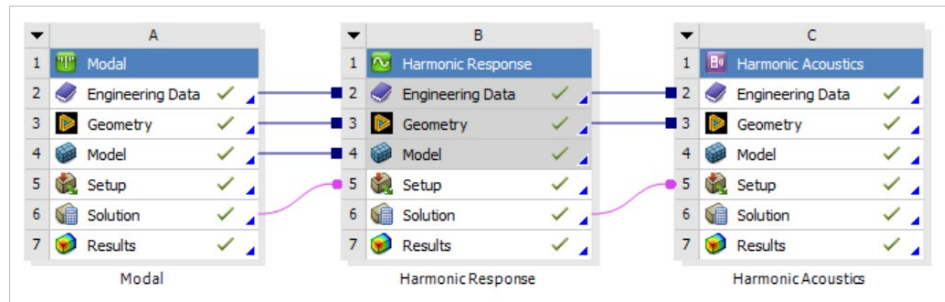


Figure 5. FEA procedure diagram.

The modal analysis shown in Figure 3 was conducted to determine the natural frequencies, while the harmonic response analysis was used to calculate the vibration amplitudes and velocities at different frequencies. These results were then imported as loads into the harmonic acoustic analysis to determine the acoustic pressures and A-weighted SPLs.

5. Results and discussion

5.1 Acoustic pressure and A-weighted SPL

Figure 6 is a graphical summary representation of the results of acoustic pressure, SPL, and A-weighted SPL for three acoustic materials, the results of which were validated analytically using mathematical models discussed in section 3. The contour plots illustrate variations in acoustic pressure and SPLs within the air domain and the wind tunnel duct cavity. For melamine foam, high-pressure regions (red zones) are minimal, indicating limited absorption. In contrast, mineral wool and ceramic fibre exhibit broader green and blue regions, reflecting higher acoustic energy dissipation. The most effective materials, mineral wool and ceramic fibre, reduce red/orange areas and increase blue/green regions, implying better sound absorption and lower SPL values.

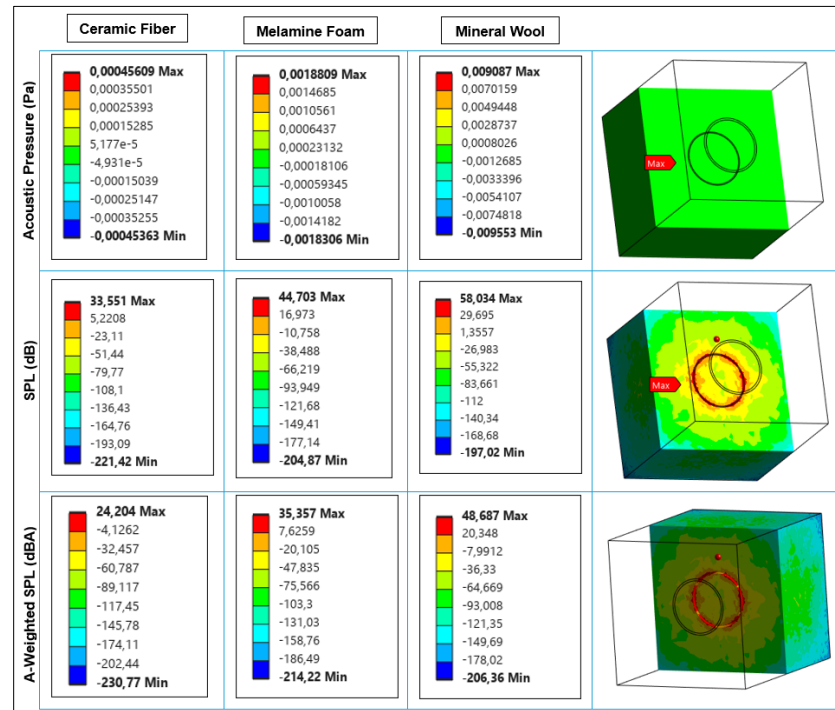


Figure 6. Comparison between the three acoustic materials.

As can be seen in Figure 7, the material with high density and damping ratio, namely ceramic fibre, has superior SPL attenuation properties. Hence, it is a material that is ranked better suitable for reducing vibrational energies in the wind tunnel duct than melamine foam and mineral wool, in this order.



Figure 7. A-weighted SPL results.

To further understand the vibro-acoustic attenuation behaviour of the proposed materials, a frequency-domain analysis of the A-weighted Sound Pressure Level (SPL) was performed across a range of excitation frequencies. Figure 8 below presents a comparative frequency spectrum analysis of A-weighted (SPLs) for ceramic fibre, mineral wool, and melamine foam. As illustrated, ceramic fibre consistently demonstrates superior attenuation across the frequency range (50–500 Hz), maintaining lower SPL values around 24 dBA. Mineral wool performs moderately, with SPLs in the mid-30s dBA range,

while melamine foam exhibits the highest SPLs, close to 49 dBA at higher frequencies. This spectrum visualisation supports earlier findings and provides a clearer understanding of how each material behaves across a range of vibrational frequencies typical in supersonic wind tunnel environments.

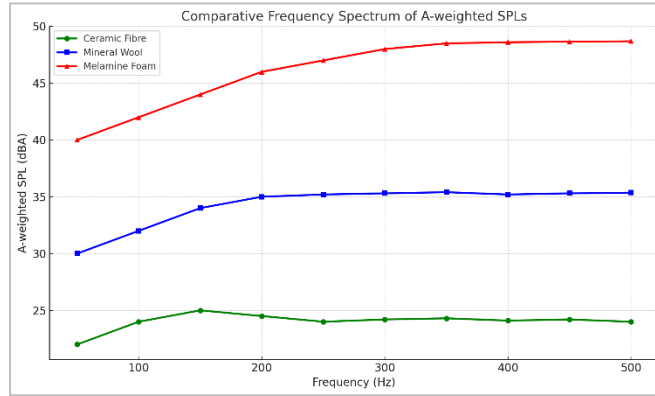


Figure 8. Comparative frequency spectrum of A-weighted SPLs.

5.2 Analytical validation results discussion

The calculated natural frequency of the wind tunnel duct alone was first validated through analytical methods, and the result of 98.23 Hz. ANSYS simulations produced a comparable natural frequency of 95.1 Hz, resulting in a minor percentage difference of 2.39%. Similarly, the calculated acoustic pressure of 0.0119 Pa was compared with the simulated value of 0.0109 Pa, showing a 6.77% difference. Although slightly higher, the percentage difference is still within acceptable engineering tolerance limits and can be attributed to the detailed modelling of Fluid Structural Interaction (FSI) in ANSYS, which was not fully accounted for in the analytical calculations. The A-weighted SPL, which reflects human hearing sensitivity across frequency ranges, was calculated to be 53.02 dBA and compared with a simulated value of 56.58 dBA, resulting in a 6.5% difference. This variation can be considered acceptable, displaying the suitability of the developed models.

5.3 Comparative physical properties of the acoustic materials discussion

The variation in acoustic performance among the tested materials can be attributed to their quantified physical properties. Ceramic fibre, with a density of 128 kg/m³ and a flow resistivity of 50 kPa·s/m², exhibited the lowest A-weighted SPL (24.04 dBA). Its high density and resistivity result in increased acoustic impedance, enabling effective viscous and thermal damping of incident sound waves. Mineral wool, with a density of 125 kg/m³ and flow resistivity of 10 kPa·s/m², showed moderate performance (35.36 dBA), aligning with its intermediate impedance and absorption capacity. Melamine foam, the lightest at 10 kg/m³ and with the lowest flow resistivity 4.5×10^3 kPa·s/m², recorded the highest SPL (48.69 dBA), indicating limited low-frequency attenuation. These quantitative metrics substantiate the acoustic results and demonstrate that material density and flow resistivity are key factors influencing sound absorption efficiency in supersonic wind tunnel applications.

6. Conclusion

This research aimed to investigate and compare three different acoustic wall treatment materials to minimise noise and vibrations in the compressor fan section of a supersonic wind tunnel using FEA. The A-weighted SPL of the duct without acoustic material was 56.58 dBA. Among the three materials tested, ceramic fibre achieved the lowest SPL at

24.04 dBA, demonstrating the highest sound-absorbing performance. Hence, this study successfully identified ceramic fibre as a suitable material for noise control, supporting the potential of targeted acoustic treatments to improve supersonic wind tunnel performances.

While acoustic performance was the primary focus of this study, long-term durability and thermal resistance are essential for supersonic wind tunnel applications operating at Mach 1.4 to 5.0, where internal duct temperatures can reach between 153°C and over 1100°C, depending on flow conditions. Ceramic fibre, with a working temperature threshold exceeding 1200°C, is designed for thermal insulation and structural integrity in extreme conditions. Its non-combustibility and low thermal conductivity under thermal cycling make it ideal for sustained exposure in the compressor fan section. In contrast, despite its acoustic absorption properties, melamine foam is less durable and may degrade under sustained thermal loading. These considerations reinforce the selection of ceramic fibre for its acoustic effectiveness and suitability under realistic operational and thermal conditions.

7. References

- [1] Gomes J P, Bergmann A and Holthusen H 2019 Aeroacoustic wind tunnel design *CEAS Aeronaut J.* **10**, pp. 231–49
- [2] Cattafesta L, Bahr C and Matthew J 2010 *Encyclopaedia of aerospace engineering* (Wiley)
- [3] Santana L D, Carmo M, and Catalano F M 2014 The update of an aerodynamic wind-tunnel for aeroacoustics testing *Journal of Aerospace Technology and Management* **6**, no. 2, pp. 111–8
- [4] Peters R J 2013 *Acoustics and noise control* (Routledge)
- [5] Zhu W, Chen S, Wang Y, Zhu T, and Jiang Y 2018 Sound absorption behaviour of polyurethane foam composites with different ethylene propylene diene monomer particles *Archives of Acoustics* **43**, no. 3, pp. 403–11
- [6] Yi W *et al.* 2021 Design and characterization of a multifunctional low-speed anechoic wind tunnel at HKUST *Aerospace Scientific Technology* **115**
- [7] Taraldsen G 2005 The Delany-Bazley Impedance Model and Darcy's Law
- [8] Singhal R K, Guan W and Williams K 2002 Modal analysis of a thick-walled circular cylinder *Mechanical System Signal Process* **16**, no. 1, pp. 141–53
- [9] Mathur G, Chin C, Simpson M and Lee J 2001 Structural acoustic prediction and interior noise