

Research Article

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On the acoustic transparency of perforated metal plates facing a porous fibrous material**

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Abstract: Thin impervious layers, cloths or perforated plates are usually utilized with fibrous absorbing materials in order to avoid small particles, coming from deterioration over time or from flow abrasive effect, becoming dislodged and polluting the environment. These protective facings are to be carefully considered and analyzed, since they can affect the acoustical behavior of the “backing” material. This study addresses this issue through an experimental survey and a theoretical analysis using the Transfer Matrix Method (TMM). Experiments have been performed in the frequency range 160–2,500 Hz, analyzing the different behaviors due to multiple combinations of percentage of open area and air gap between perforated facing and absorbing material. Experimental data have shown a marked effect of the percentage of perforation, at least up to a threshold value of 20%, whereas the air gap slightly affected the acoustic behavior of the covered absorbing material. The TMM was applied to the tested faced absorbing system, and experimental and theoretical results were compared, showing the good accuracy of the model. Several geometrical configurations were then modeled through TMM and the possibility of using this method in order to assess the acoustic transparency of perforated metal plates was assessed.

Keywords: Acoustic transparency, perforated plates, transfer matrix method, fibrous materials, sound absorption

1 Introduction

One of the main problems encountered in noise mappings, and therefore in the modeling of the relative sound field, consists of the difficulty of defining the sound-absorbing properties of the surfaces present within the field, and in particular of the acoustic barriers. It is therefore important to develop knowledge on an experimental basis of the behavior of sound-absorbing materials such as perforated plates, that are often used to make acoustic barriers. Furthermore, dissipative mufflers, absorptive linings and acoustic panels make wide use of fiber materials to absorb noise, adopting different techniques to avoid small particles, coming from deterioration over time or from flow abrasive effect, becoming dislodged and polluting the environment. Thin impervious layers, cloths or perforated plates are usually utilized to this aim, often affecting the acoustical behavior of the “backing” material. In particular, perforated facings have a broad application in lined ducts and parallel-baffle mufflers, where they not only protect the fiber absorbing material from dust or grazing flow, but also act as a rigid support for the material. The effect of this thin protective facing, which frequently cannot be considered insignificant, must be carefully considered and analyzed, since it can deeply modify the acoustic behavior of the absorbing material and change the expected performance of the device. Mainly a reactance is added to the impedance of the absorbing material, but the facing also causes an additional acoustic resistance dependent on the characteristics of the facing. Therefore, the possibility of analyzing and predicting the effect of perforated facings on the sound absorption of fiber material is a key element in a number of technical applications.

In spite of that, while the acoustic characteristics of perforations in contact with air alone have been extensively investigated, literature on perforations facing sound absorbing material is currently not so plentiful, even if the first studies date back more than sixty years. In 1947, Bolt theoretically analyzed the behavior of perforated facings for absorptive materials, only considering the mass reactance of perforations [1]. The approximate analysis was then reduced to a design chart, with the variables: number of holes

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per unit area, hole diameter, thickness of facing, weight of perforated facing and frequency. The study indicated that facings reduce sound absorption at high frequencies, while at low frequency absorptivity increases since the mass reactance of the facing counteracts the stiffness reactance of the material. Sabine [2] analyzed the behavior of a porous material with or without a perforated rigid plate, such as an asbestos cement board, over it. In the following five years, relevant contributions came from Ingård and Bolt [3], Callaway and Ramer [4] and Ingard [5]. In the first paper, the above-mentioned analysis was extended to a number of sound absorptive structures, also including porous material covered by a perforated facing. The effect of partitions was taken into account, together with the angle of incidence of sound. The authors analyzed conditions in which a perforated facing structure acts like a Helmholtz resonator and proposed equations and design charts to calculate their impedance and the absorption coefficient. Callaway and Ramer focused their attention on the combined influence of the backing material and open area ratio and on the effect of a gap between the facing and the porous backing. Their experiments demonstrated the order or magnitude of the increase of the acoustic resistance for a porous material covered by perforated facing with low percentage of open area (lower than 5%); this increase is greater for higher density backing materials. The air gap strongly influences acoustic behavior, so that the effective resistance of a backing material can be varied by adjusting the gap. Ingard theoretically analyzed the effect of perforation and of the air gap between the facing and the absorbing material and concluded that the facing also causes an additional acoustic resistance which is often larger than the acoustic resistance of the porous layer itself. This additional resistance is ascribed to the near field losses in the porous material around the perforations of the facing and is deemed practically zero when the air space between facing and material exceeds approximately one perforation diameter, while the viscous resistance of the facing itself is in most cases considered negligible. The air gap between the perforation and facing material resulted in narrowing the effective frequency range of absorption coefficient.

Twenty years later, Davern [6] experimentally studied the effect of porosity and thickness of the perforated facing, as well as of the density of backing material and of the thickness of the air gap between facing and fibrous material. The perforated facing system impedance was measured for a number of combinations of the characteristic parameters. Davern evidenced the selective behavior of the perforated sound absorbing system and the possibility of using it with the aim of optimizing the design both for a narrow range of frequencies, and for broadband absorbers. In the same

year, Bauer [7] proposed a model to predict the impedance of perforated facing backed by porous material with flow at Mach numbers from 0 to 0.6. A point reacting behavior was assumed in the material, such as that induced by a number of impervious walls subdividing the porous material into a number of independent areas. A correlation for the specific impedance of the covered material was proposed, expressly including the “grazing flow effect”, so relevant to diverse technical fields. In respect of possible applications, Munjal and Thawani [8] theoretically analyzed the effect of the percentage of open area of a perforated panel facing a highly porous fibrous material on the acoustic performance of absorptive ducts. Analytical models for circular ducts and parallel baffle mufflers were presented, with bulk reacting as well as locally reacting absorptive linings protected by a perforated or un-perforated layer. Recommendations for the design of the silencers can be obtained from those models, particularly in respect of the effect of perforated facing characteristics and their porosity. In Selamet *et al.* [9] the impedance of a perforation facing continuous strand fibers in a perforated concentric silencer was investigated theoretically and experimentally. The effects of perforated duct porosity and fiber density were examined. An expression for the acoustic impedance of perforation facing absorbing fibrous material was presented, by modifying the empirical equation proposed by Sullivan and Crocker [10] for perforated holes. Borelli and Schenone [11] also considered the technical problem of perforated facing in dissipative silencers and highlighted the need for an accurate predictive method for facing behaviors in order to calculate their transmission loss.

A simplified approach was proposed by Schultz and Plank [12] and Schultz [13] through the introduction of a parameter called Transparency Index (TI), depending on the percentage of open area of the perforated facing, the thickness of the sheet and the shortest distance between holes. A suitable value of TI which, according to Schultz and Plank, achieves the goal of an acoustically transparent facing material is 10,000. This value would lead to a sound attenuation no greater than 1 dB at 10 kHz, and at lower frequencies the sheet would be almost completely transparent to sound. Nevertheless, Schultz and Plank underline that the TI approach is “a good rule of thumb” [12], and it can be observed that for TI equal to 5,000 the attenuation “is only 1.5 dB” and with TI equal to 2,000 the attenuation “is only 2.5 dB” [13].

Guignoard *et al.* [14] proposed a method to calculate the surface impedance at normal and oblique incidences for sound absorbers consisting of a perforated facing backed with a porous material. Results were obtained with and without an air gap between the porous material and im-

pervious backing. The analytical method was successfully validated through a comparison with experimental data, even if, according to the authors, the imaginary part of impedance for the fiberglass is slightly lower than the corresponding predicted quantity as the laws of Delany and Bazley were not perfectly suited to the fiberglass studied. The effect of perforated plates on porous backing material was also analyzed in Rebillard *et al.* [15], taking into account the flexural stiffness and the elasticity of the facing. This theoretical study led to a validated model for the acoustic field based on a matrix representation obtained from the Biot theory. Noticeable results were achieved under different conditions, considering unbonded and bonded porous facing. These last conditions were modeled by means of the matrix method both as a porous thin plate, and as an elastic porous membrane. Mechel, starting from the assumption that general comments on the influence of perforated metal sheet covers were contradictory, presented an analytical description of the phenomenon for both parallel slits [16, 17] and circular perforations [18]. He noted that the cover behaves quite differently depending on whether there is an air gap between the cover and the absorbent layer or not. When the absorber layer comes into contact with the cover plate, the near field impedance of the inner orifice is changed from a mass reactance to a resistance. Furthermore, there is an impedance transformation due to the dislocation of the boundary plane from the surface of the absorber to the outer surface of the cover. Takahashi [19] treated the effect of interaction between hole and structure and the wave scattering from a boundary surface with impedance discontinuity. In this way, the effect of diffraction phenomena caused by impedance discontinuities of the boundary surface can be considered. The influence of panel vibration was theoretically studied and the results were expressed in terms of field-incidence-average absorption coefficient for a perforation facing absorbing material. Chen *et al.* [20] numerically analyzed the effect of assembly of perforated plates and porous materials on acoustic absorption, for diverse surface shape and percentages of open area. A strong dependence of the acoustic behavior on the porosity of the perforated plate was predicted, particularly for the lower percentage of open area (5%): the acoustic absorption peak shifts towards a lower frequency and its value reduces for the upper-frequency bands.

Kingan and Pearse [21] experimentally investigated the absorption characteristics of porous absorbers in combination with perforated facings by means of measurements in a reverberating room. The effect of the hole size, hole pattern and open area ratio were examined. In respect of percentage of open area, they observed for low frequencies (less than 250 Hz) very little effect on sound absorption. Facings

with open area ratios $\geq 22.7\%$ were effectively acoustically transparent and had little effect on the sound absorption properties of the backing. The results of a facing with an open area ratio of $\geq 22.7\%$ turned out to be very similar to having no facing at all.

In order to predict the acoustic behavior of these perforated assemblies, some models have been proposed in recent years, somehow always deriving from the pioneering works. Kirby and Cummings [22] evidenced the magnitude of the increase of the complex component of the impedance due to a flat perforated facing of a porous material, so that, even for plates with a large percentage of open area, the inclusion of perforates in the modeling of dissipative silencers is necessitated. Thus, they proposed a semi-empirical correlation to calculate the impedance of the perforation when backed by absorbing material in the presence of a grazing mean flow. Their measurements also evidenced that perforation impedance strongly depends on the density of the material immediately adjacent to the holes, indicating that the porous material has only a very localized effect on the orifice impedance. Some further research was successively implemented by Lee *et al.* [23], who experimentally analyzed the acoustic impedance of perforations in contact with fibrous materials, varying porosity and hole diameters of the facing, together with the density of the fibrous material. Their results showed that both resistance and end correction coefficient decrease as the percentage of open area increases, except for values of the percentage of open area higher than 15%. The interaction between the perforations and the fibrous material affects the resistance, which significantly increases, more than the end correction coefficient. The imaginary part of the perforation impedance also proved to depend on the frequency, the hole diameter and the wall thickness, as models would predict. Atalla and Sgard [24] presented a general model that easily and automatically handles the miscellaneous configurations for a faced absorbing system in the context of the transfer matrix method. In particular, they demonstrated that a perforated plate or screen can be modeled as an equivalent fluid following the Johnson-Allard approach with an equivalent tortuosity. The model parameters were obtained in the typical cases where the perforated panel is coupled to free air, air gap or porous layers. This general approach allows one to retrieve classical models for both perforated plates and screens. In addition, the mass of the perforated screen or perforated plate can also be accounted for, by extending the model to include the inertia of the solid phase. Besides, it is not necessary to develop a specific model for a certain perforated system, since all the existing models (macro- and microperforated) can be obtained from an equivalent fluid model by selecting appropriate parameters. More re-

cently, Allard and Atalla [25] dedicated a paragraph of their book to the modeling of porous material with perforated facing, in which these results are generalized. Both normal and oblique incidence were considered in the determination of impedance and results are extended to anisotropic stratified porous media. The geometry of the apertures, the physical properties of the porous layer, the air gap between the facing and the backing are taken into account by this model, which is proposed as a design tool in the analysis and optimization of stratified porous materials covered by perforated facings. Further studies seem to suggest that the above-mentioned model can be used effectively for uncommon fiber materials, like coir fiber. The results were presented by Ayub *et al.* [26], who applied the Atalla-Sgard model in order to estimate the absorption coefficient of multiple perforated plate systems composed of coir fiber and an air gap. Based on this method, several combinations of multilayer assembly with multiple perforated facings, coir fiber and air gap were investigated. Theoretical results were found to be in good agreement with the measured values, so the validation of the modeling was extended.

Two papers from Nakai *et al.* [27] and Nakai and Yoshida [28] examined, from an experimental and theoretical viewpoint, the behavior of perforated plates with and without backing material. In Nakai *et al.* experiments are reported showing that the perforated facings of porous material act as low pass filters and their cut-off frequency depends on the perforation rate and the thickness of the plates. Theoretical analysis through FEM was used in order to compare the behavior of a perforated plate with a single hole plate. An equivalent electrical circuit model allows one to define a proportionality between upper cut-off and perforation rate/thickness. Besides the effect of air space between the plate and glass wool on normal incident absorption coefficients was studied. Nakai and Yoshida presented a method aimed at simulating the normal incidence sound absorption coefficient of perforated plates by transfer line parameters in a two-point network, based on a plane wave propagation. The simulation was extended to PFPs backed with glass wool, showing good agreement with experimental absorption coefficients.

A specific focus in Patraquim *et al.* [29] was on the effect of the fabric lining the perforated plates when fibrous material is backed. Several nonwoven textiles were tested together with mineral wools of different densities and perforated plates of various open areas. The flow resistivity of the nonwoven fabric was found to be a determinant variable. A high resistivity hinders the resonant behavior of the system, drastically lowering the absorption provided between 200 Hz and 1,000 Hz. For small flow resistivity, this behavior was not noted and the resonance peak in the absorption

curve still resulted very pronounced. The theoretical analysis, based on the conversion of the acoustic impedance of a single hole in an average value corresponding to the open area of the perforated panel, revealed good agreement with the experimental data only when the nonwoven textile had a small resistivity.

Right from the very first studies, the main elements of this topic were clearly set and they have accompanied research until today, passing through the whole quoted literature: how do the percentage of open area (or open area ratio), the air gap between the facing and the backing material, the hole geometry, the physical properties of material and facing affect the acoustic behavior of the absorbing material? How can we predict the performance of a perforated facing porous material depending on its physical characteristics? With this study, the authors wish to contribute to tackling these issues, by means of an experimental survey and a theoretical analysis of the results in accordance with the Transfer Matrix Method (TMM). In particular, the aim is to assess when the effect of perforations is negligible, so that they can be considered acoustically transparent and simply operate as a protection for the absorbing layer, without affecting the performance of the system. It is worth studying this topic, for instance, for dissipative silencers and lined ducts, where these conditions frequently occur and it is important to decide how the fibrous material can be faced without changing its acoustic behavior. Mechel [16] points out that information on this topic is mostly qualitative (e.g., Sabine [30]; Doelling [31]; Sharland [32]; Bies [33]) and that the influence of air gaps should be experimentally investigated since they can exist even if they are not intended, for example if the perforated metal panel has some buckling or if it is not correctly mounted.

In this research, the effect of perforated facings on sound absorption characteristics of samples made by polyester fiber was experimentally investigated in the frequency range 160–2,500 Hz. First the acoustic behavior of the porous material was characterized. Then experiments were performed on the basis of the ASTM C384-04 standard [34] by means of two stationary wave tubes with different diameters on perforated facing samples. The different behaviors due to the multiple combinations of percentage of open area and to the air gap from the absorbing material were then analyzed and discussed. Finally, the Transfer Matrix Method (TMM) was applied to the tested faced absorbing system and experimental and theoretical results were compared. Validation was successful and the model allowed us to numerically analyze several different configurations, varying both open area percentage and air gap.

2 Experimental set-ups and operating procedures

Three different sets of experimental equipment were utilized in order to measure the different acoustic properties of the porous fiber material and of the covered samples examined in this study. One set-up was dedicated to measuring the flow resistivity of the porous material; two acoustic tubes were used to determine acoustic impedance and sound absorption coefficient for all the samples.

First, the acoustical behavior of the porous material was characterized by measuring specific flow resistance, R_f , based on the ISO 9053-1:2018 standard [35] (Figure 1). The test section was made with a vertical polymethyl methacrylate tube (height 1.2 m, inner diameter 0.2 m), in which a metallic grid was inserted with the aim of holding up the absorbing material. The grid mesh (10 mm $\times$ 10 mm) was chosen so that it did not appreciably affect the flow resistance of the sample. Two pressure taps, placed upstream and downstream of the sample, were connected through flexible ducts to a capacitive differential pressure gauge, with a measurement range from 0 to 25 Pa and a precision equal to 1%. The air flow through the sample was obtained for continuity by emptying a 100-liter water vessel. A regulating valve allowed us to control the water flow rate, which was calculated by weighing with a precision balance (Mettler PE 12) the liquid mass that flowed down into the vessel during a certain time interval. Theoretical error analysis gave a maximum error lower than 6% of the measured value. In accordance with the ISO 9053-1:2018 standard, the flow

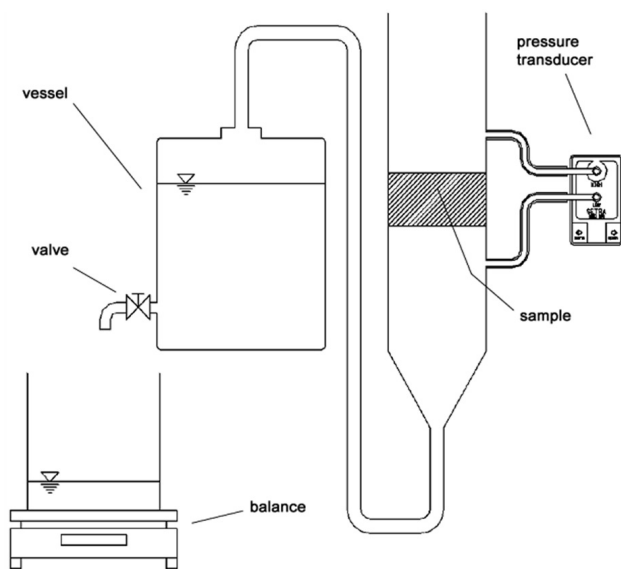


Figure 1: Flow resistance measurement equipment

resistance measurement equipment was validated by comparison with the results of a round robin test [36], giving a percentage difference of +12.5% compared to the average value of the round robin test, within a range of extremes of -17.8% and +18.2%. It must be pointed out that 5 out of 9 laboratory results were farther from the average value.

Sound absorption coefficients and surface impedances were measured utilizing the Standing Wave Ratio method by means of two cylindrical Kundt tubes of different diameters, both of which were made of transparent PMMA. Using two tubes allowed us to increase the frequency range for which the study was performed, as described in the following.

The first is a vertical standing wave tube having a 0.19 m inner diameter and 2 m length overall, compliant with the ASTM C384-04 standard (Figure 2). The sound source comprises a loudspeaker 200 mm in diameter, placed at the top of the tube. The bottom of the tube consists of a metal disk with a thickness of 40 mm which functions as the rigid reflection surface. The microphone and its preamplifier are suspended inside the tube by means of the microphone cable. By acting on the cable, the microphone can be slid along the entire tube length. The loudspeaker is powered by a sinusoidal signal (produced by means of a GW Instek

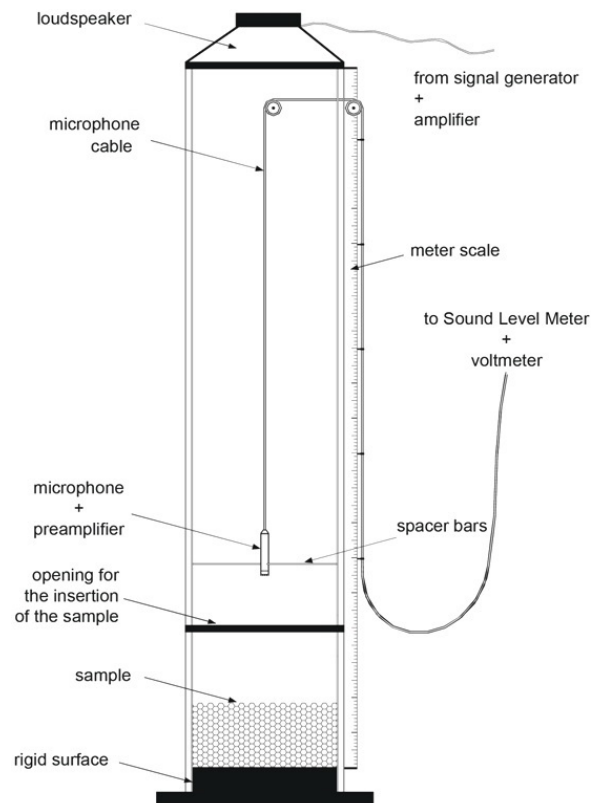


Figure 2: Vertical standing wave tube

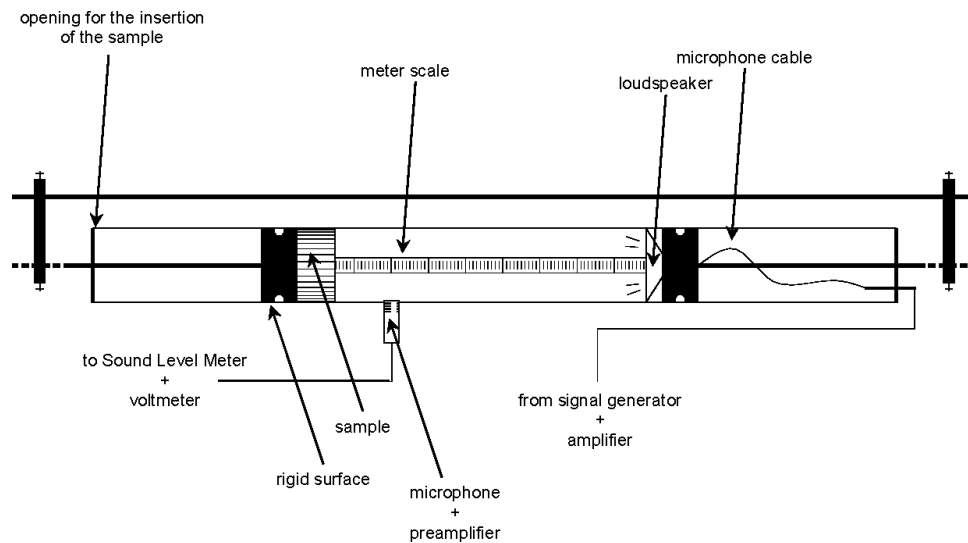


Figure 3: Horizontal standing wave tube with variable length

GFC-8015G function generator). The measurement system also has a sound-level meter analyzer (Larson Davis model 824), which permits frequency analysis in octave bands and third octave bands, an oscilloscope (Tektronix 2213) and a digital multimeter (Fluke 77), used to analyze output voltage. In order to apply the Standing Wave Ratio method, it is necessary to be able to measure at least a minimum and a maximum of the pressure magnitude inside the tube. Hence, a longer length of tube than $\lambda/2$ is needed (where λ is the wavelength of the lowest frequency to be investigated). In accordance with the aforementioned ASTM C384-04 standard, “to ensure that at least two minima can be observed in the tube, its length should be such that $f > 0.75 \cdot c / (l - d)$ ”, where f is the frequency, c the speed of sound in air, l is the tube length and d is its internal diameter. Assuming a value of $c = 343$ m/s, this condition imposes a lower limit to the frequency of measurement that corresponds to 142 Hz due to the minimum useful length (defined in the standard as $l - d$) of 1.81 m. The upper frequency limit is imposed by the necessity of having a plane wave inside the tube. Applying Rayleigh’s theory [37], as reported again by the ASTM C384-04 standard, “for circular tubes, the upper limit of frequency is $f < 0.586c/d$ ”, so that the highest frequency that can be investigated is equal to 1,058 Hz because of the inner diameter equal to 0.19 m.

The second experimental setup, comprising an inner diameter of 70 mm and a useful length of 630 mm, has a horizontal axis and a movable test section (Figure 3). A 1/2" microphone is inserted through a hole, suitably lined with soundproofing material to ensure the test section is acoustically sealed. A metallic disc with a thickness of 40 mm and a width of 70 mm was used as a bracket for the

absorbing material samples at one end of the tube, while a loudspeaker with the same diameter operated as a sound source at the other. Both the bracket and the loudspeaker can be moved separately along the tube by means of rods and so create a test section with a variable length. Besides regulating the relative position of the sound source and the sample, the sliding rods are connected by an external rod, which permits the loudspeaker and the sample to be moved as one. This way, using a fixed microphone, maximum and minimum pressure values can be measured along the tube. This experimental set up allows greater flexibility compared to rigid application of the ASTM C384-04 standard that requires the sound absorption coefficient to be calculated for a tube with a fixed length, depending on the minimum frequency of interest, and with a mobile sliding microphone to measure the Standing Wave Ratio. So as to include the effects of attenuation in the calculation, two minimums were used and thus the lower frequency of the range is 408 Hz.

The upper limit of the usable frequency is imposed by the need for plane waves in the tube. According to Rayleigh’s theory, this condition is verified for frequencies the wavelengths of which are greater than 1.707 times the diameter of the tube; up to 2,871 Hz for this set up. The accuracy of the set-up was tested by comparing the results measured with a similar sample of polyurethane with open cells in the present set-ups with those obtained during an interlaboratory round robin test or RRT [38]. A comparison between the experimental data from the current apparatus and the maximum deviation measured by the eleven laboratories involved in the above-mentioned RRT, shown in Figure 4, indicated that the values measured fall within the

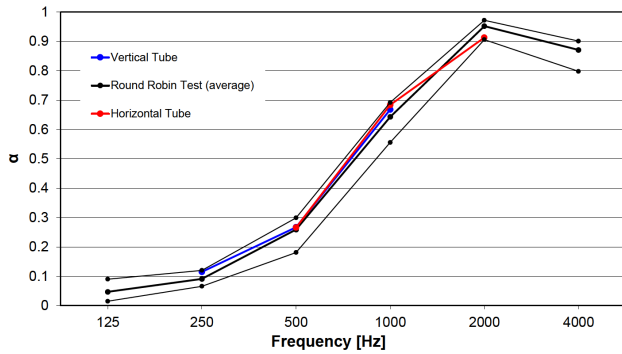


Figure 4: Comparison between experimental data obtained from the current setup and the RRT. The thin black lines represent the maximum and minimum values obtained for each frequency in the RRT

Table 1: Percentage of open area (σ) and Transparency Index (TI) for the different metal plates used as perforated facing

σ	4.9	7.7	10	15	20	30
TI	28.4	83.0	161.0	543.3	1043.2	5173.1

confidence limits of the test, thus indicating satisfactory accuracy [39].

The experiments were performed on polyester (PET) fiber samples with a bulk density of 30 kg/m^3 and melting point at 260°C . The study considered a sample thickness of 100 mm. The samples were faced using different metal plates perforated with circular holes. The hole diameter was equal to 2 mm for all facings, while the percentage of open area varied from 4.9% to 30%. Figure 5 shows a sketch of the different metal plates used as perforated facing for the horizontal and vertical Kundt tube. The perforated panels were either placed adhering to the PET fiber material or at distances of 2, 4, and 6 mm.

In Table 1 the Transparency Index (TI) values are reported for all the different percentages of open area analyzed, according to the following definition [12]:

$$TI = \frac{0.04\sigma}{\pi t a^2} \quad (1)$$

where σ is the percentage of open area of the perforated facing, defined as perforated area/total area (%), t is the thickness of the sheet (in inches) and a is the shortest distance between holes (in inches). It must be noted that tested TI values are lower than the threshold value 10,000 for all the facings analyzed, so a certain effect of the perforations can be expected. The following experiments show whether for TI values ranging between 28.4 and 5,173 a lack of transparency is really observed and in which terms.

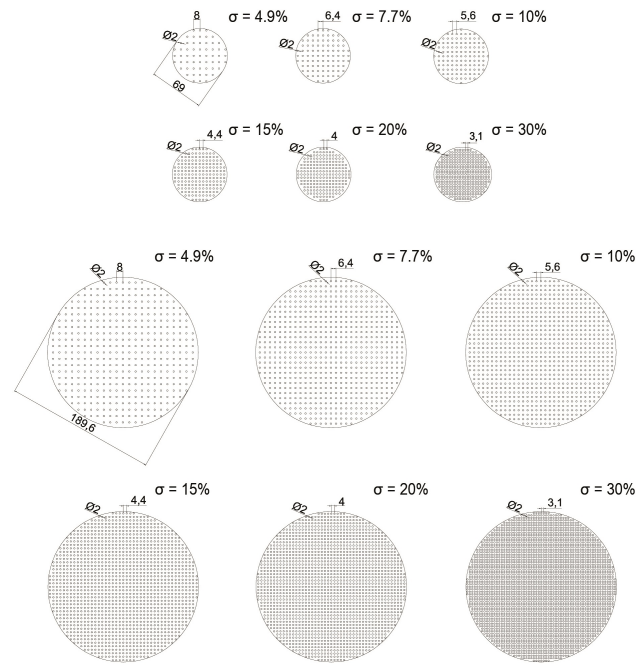


Figure 5: Sketch of the different metal plates used as perforated facing (horizontal and vertical Kundt tube)

3 Properties of the porous fiber material

The acoustical behavior of the un-faced polyester fiber sample was characterized by measuring flow resistivity, R_1 , surface impedance and sound absorption coefficient at normal incidence, respectively Z and α , by means of the above-mentioned setups. The microscopic structure of the polyester fiber material is displayed in Figure 6a. The image is zoomed 10 times compared to real size and shows a structure of the porous material roughly similar to that described in other papers [40, 41].

Figure 6b shows the deviation from the average value (right y axis), and the flow resistivity value (left y axis) for each run performed. Flow resistivity, R_1 , resulted equal to $4285 \text{ Pa}\cdot\text{s/m}^2$, which is a typical value for this material and density.

Curves for the real and imaginary part of the normal-incidence superficial acoustical impedance of the sample tested are reported in Figure 7a, while Figure 7b reports the sound absorbing coefficient at normal incidence. The comparison with literature correlations proposed for porous homogeneous materials by Delany-Bazley [42] and GaraiPompoli [40], shows good agreement between measured and calculated values; in this way, the accuracy of the experimental set-ups and of the operating procedure were further validated. The above measured flow resistivity was intro-

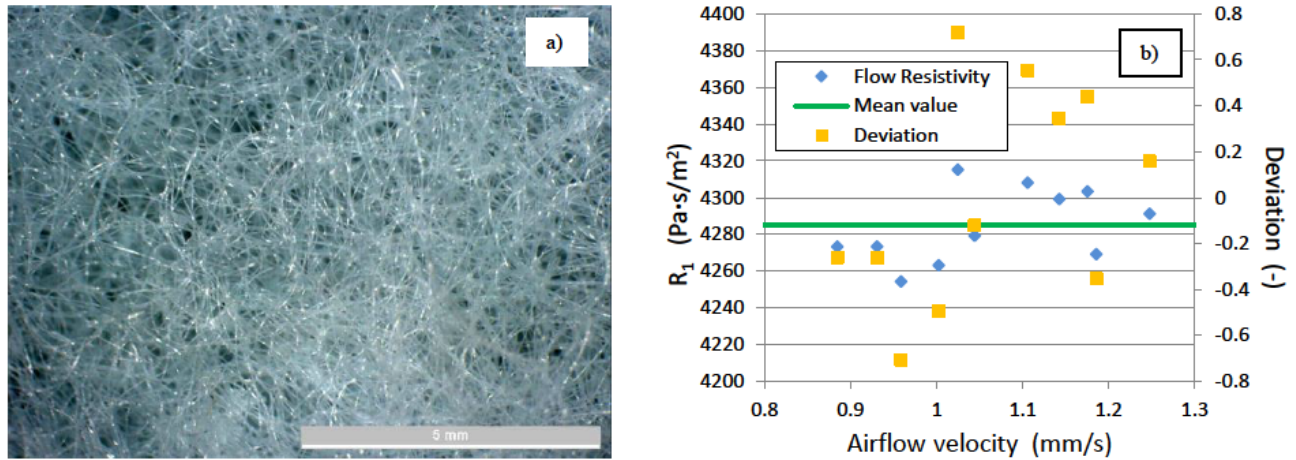


Figure 6: Microscopic structure (a) and R_f measurements (b) for un-faced polyester fiber samples

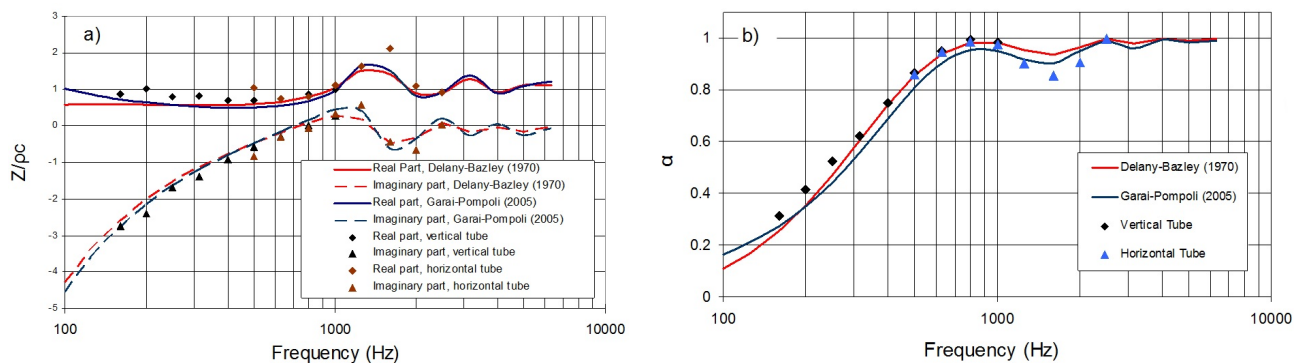


Figure 7: Normalized impedance (a) and absorption coefficient (b) curves for un-faced absorbing material

Table 2: Summary of the models' deviations

	Standard deviation (Real part)	Standard deviation (Imaginary part)	Variance (Real part)	Variance (Imaginary part)
Delany-Bazley	0.367	0.933	0.13	0.87
Garai-Pompoli	0.390	0.984	0.15	0.97

duced into the correlations. For both a real and imaginary part of Z the different correlations give a close prediction. A good ability to describe the experimental performance of uncoated material was noted overall. In general, the Delany-Bazley model has a lower standard deviation and variance if compared with that of GaraiPompoli for the sample tested, both for the real and the imaginary part of acoustic impedance (see Table 2).

These preliminary measurements affirmed that the backing material presents standard characteristics, which are well described by the most widely used correlations. This is a point for the theoretical analysis that is going to be introduced. In the light of the previous results, in the modeling described in the following Section 7, the correlation

of Delany-Bazley was used in order to theoretically analyze the behavior of the faced absorbing system.

4 Facing with perforated plates: effect of the percentage of open area

So as to analyze the effect of perforations on the acoustic behavior of the absorbing material, the samples were faced with different metal plates perforated with circular holes. Throughout the experiments the metal plates were 1 mm

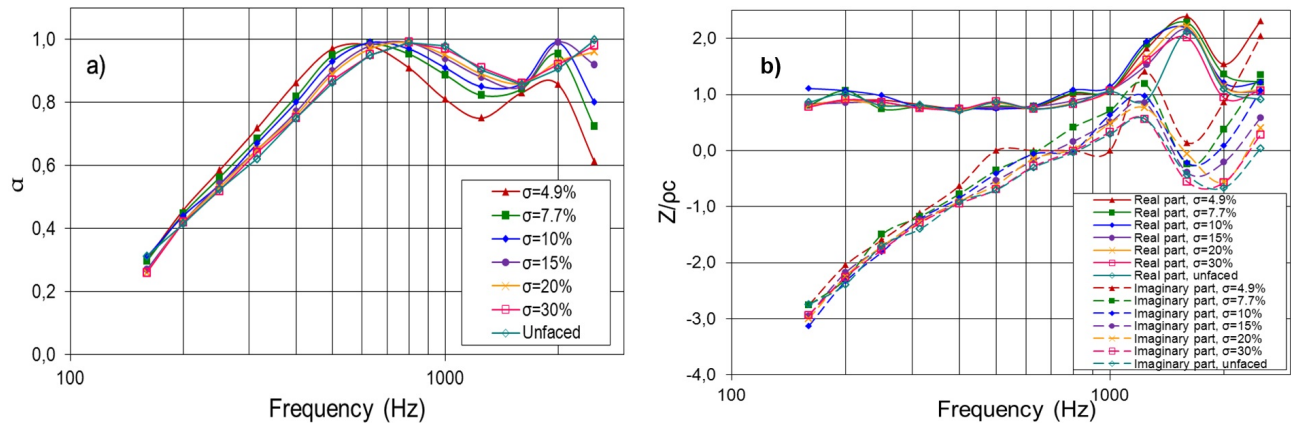


Figure 8: Absorption coefficient (a) and normalized acoustic impedance (b) curves for samples faced with different perforated plates

thick. The hole diameter was 2 mm for all facings but the percentage of open area, σ , ranged from 4.9% to 30%.

Initially, metal plates were placed in close contact with the polyester fiber material. Normal incidence sound absorption and acoustic impedance were measured for all facings.

In Figure 8a, the sound absorption coefficient curves are shown in one-third octave band for the samples faced with the various perforated facings in comparison with the un-faced sample. The diagram shows that, generally, when σ rises, the α curves for faced and un-faced samples tend to be closer, i.e., the effect of facing decreases as the open area increases. In particular, when the percentage of open area equals 30% for both, in relation to the acoustic impedance (Figure 8b), the real part does not depend on σ , or if so, this is just a very small effect probably because flow resistivity is not actually affected by the facing even for the lower percentage values of open area adopted during the experiments. The imaginary part tends to increase when σ reduces, due to the mass added to the facing; this effect decreases as the percentage of open area increases and becomes negligible when it is equal to or greater than 20%. This behavior corresponds precisely to what the models described hereinafter predict for this facing thickness and smaller hole diameter: the real part does not markedly increase until σ is greater than 2.5%, while the imaginary part presents values that increase with the frequency and the open area.

These experimental data essentially agree with results in the literature. Munjal and Thawani [8], at the end of their theoretical analysis, conclude that for highly porous fibrous materials a thin perforated plate with 34.9% of open area is practically as good as 100% of open area. On the other hand, they suggest that 4.9% of open area affects absorption behavior at high frequencies, and state that about 10%

of open area is a good design compromise between acoustical performance and mechanical strength. Similarly, in Ingard [5], the effect of a perforated panel with a 7.7% of open area of is theoretically analyzed and a considerable effect on absorption coefficient curves of porous materials is foreseen.

Such conclusions are substantially confirmed by this study, i.e., protective layers may have a dual function, which is either to work as a mere support for the porous sound-absorbing material or to operate as a real absorbent panel according to the open area. Therefore, the different intended use depends on the percentage of perforation: in case it is equal to or greater than 20% of the surface of the facing [33], it takes on the mere function of supporting and protecting the sound-absorbing material.

5 Facing with perforated plates: effect of the air gap between facing and absorbing material

The effect of a gap between the perforated facing and the absorbing material was then analyzed for the different σ values pondered in the previously described experiments. The perforated facing distorts the sound field in a small area around the holes, for a distance of only about one perforation diameter from the facing [5]. When the facing is close enough to the porous material for the distorted part to fall inside the material, the facing creates not only an inertial but also a resistive effect [16]. In such way, the surface acoustic impedance of the perforation facing the absorbing material is influenced by the air gap in place between the facing and the material. When the air gap is wide enough, the effect of the perforated facing is only a mass reactance

which augments the impedance of the porous layer; in other cases, the facing also acts as a resistance and the normal impedance of the faced porous material is affected.

Therefore, the impedance of the porous material covered by a perforated facing depends on the size of the air

gap, since the real part of the impedance depends on the flow resistivity of the material in contact with the facing.

In order to experimentally analyze such an effect, three different air gaps of 2, 4 and 6 mm were created between the perforated facing and the material. These gaps correspond to 1, 2 and 3 diameters of perforation holes respectively.

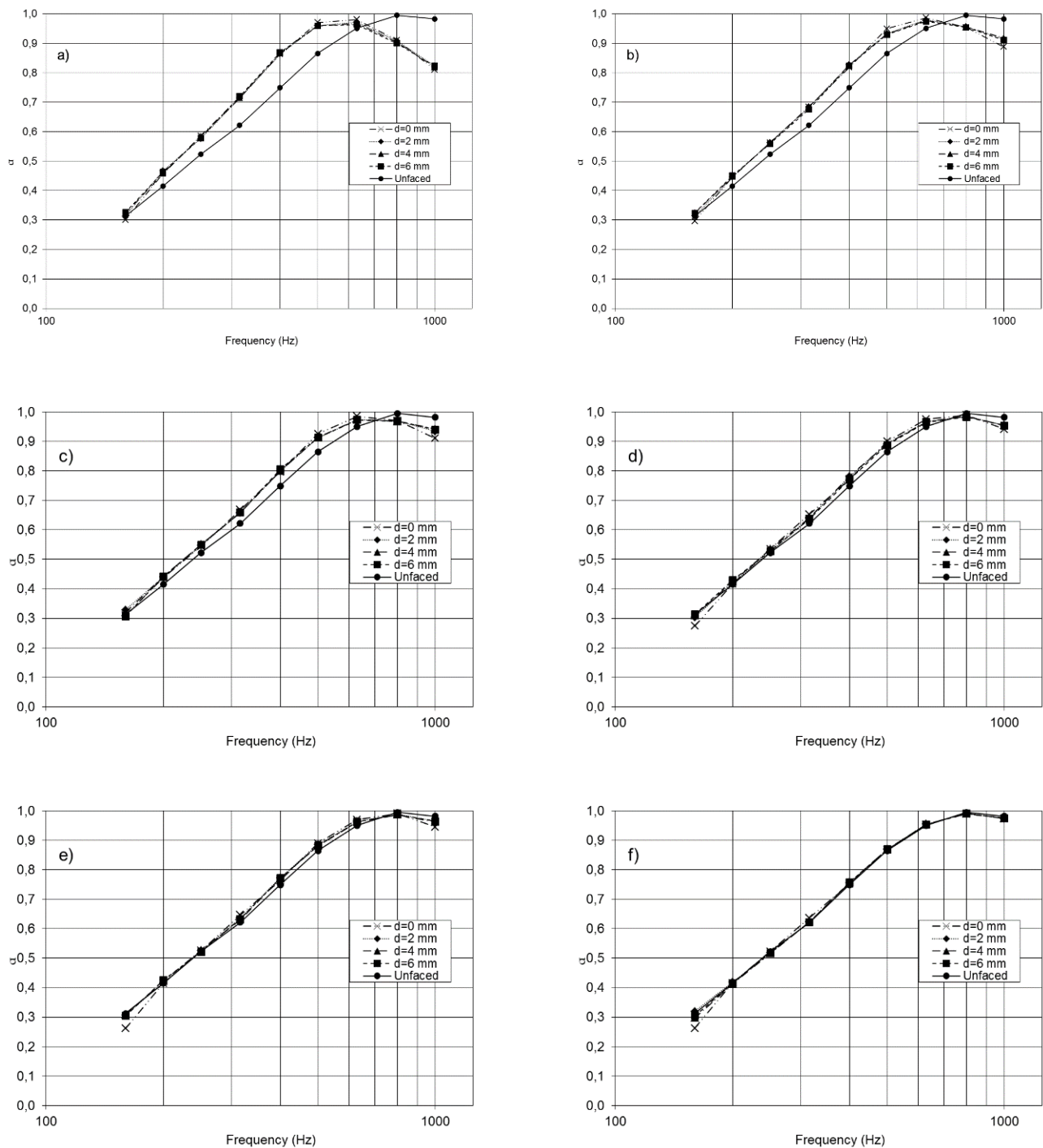


Figure 9: Absorption coefficient curves for different air gaps, d , between the perforation and the absorbing material: a) $\sigma = 4.9\%$; b) $\sigma = 7.7\%$; c) $\sigma = 10\%$; d) $\sigma = 15\%$; e) $\sigma = 20\%$; f) $\sigma = 30\%$

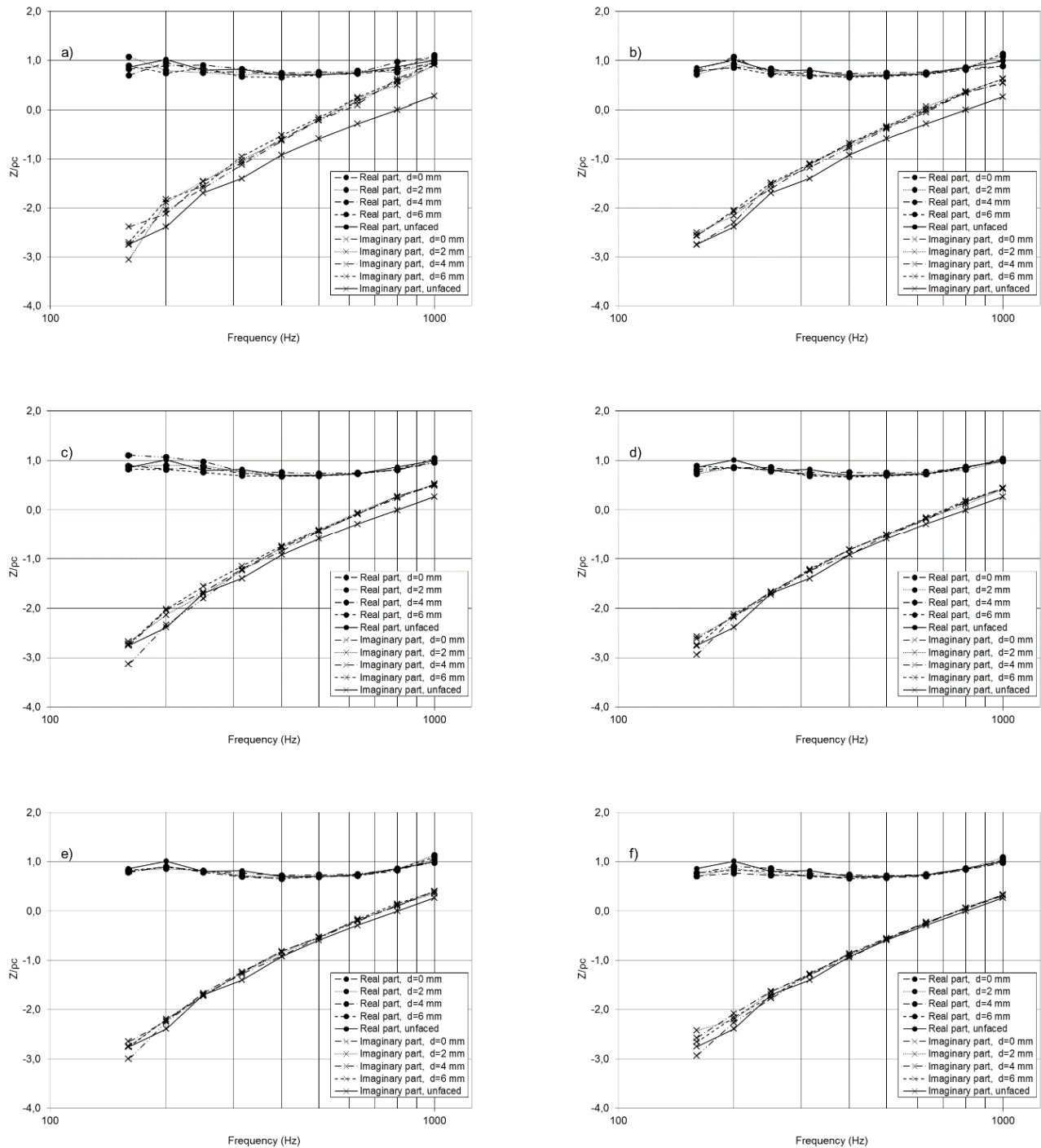


Figure 10: Normalized acoustic impedance curves for different air gaps, d , between the perforation and the absorbing material: a) $\sigma = 4.9\%$; b) $\sigma = 7.7\%$; c) $\sigma = 10\%$; d) $\sigma = 15\%$; e) $\sigma = 20\%$; f) $\sigma = 30\%$

Experiments were performed for all the different σ values tested earlier. Both sound absorption coefficient (Figure 9) and surface impedance (Figure 10) were measured in the range from 160 to 1,000 Hz using the vertical Kundt tube.

Air gaps seem to affect the acoustic behavior of the covered absorbing material only slightly. Along the α curves,

for the different air gaps, the results are very similar for all the percentages of open area tested. In particular, when the dependence of acoustic impedance on air gap is examined, no evident effect is observed for any of the d values considered during the experiments. This result appears to

be valid for all percentages of open area and for both the real and imaginary parts.

These results do not conflict with the model proposed by Allard and Atalla [25] for porous material with perforated facings for the case of circular holes; their model predicts a certain shift in α curves, but of minor entity. Given that in this case the flow resistance $rd/\rho c$ of the porous material is equal to 1.2, the model set out by Ingard [5] would presume a stronger effect of the air gap on sound absorption coefficients of the faced absorbing material but this was not actually observed during the experiments. Quite the opposite, unlike σ , changes in d values neither influenced impedance nor absorption coefficient.

It should be pointed out that whereas the percentage of open area, along with hole diameter and facing thickness, can all be set exactly in manufacturing, the air gap between the perforated plate and the fibrous material is extremely difficult to control, since the surface of the absorbing sample is not smooth, but exhibits undulations with a magnitude close to the nominal air gap. This problem can only be overcome by adopting non-fibrous porous material in the experiments; however, in most technical situations it is precisely this kind of sound absorbing material that is actually utilized (dissipative silencers, sound absorbing panels, acoustic barriers, etc.), so this issue is not to be ignored.

6 Comparison with transfer matrix based models

The experimental values of the acoustical properties were compared with simulations obtained from the TMM. A system made with porous fiber material, a perforated facing and, if any, an air gap was modeled. The method is relatively straightforward and easy to implement if compared with other methods such as finite elements, which are dramatically demanding in terms of computational power and hardware resources [43]. Figure 11 shows a sketch of the multilayer structure analyzed and of the quantities described hereinafter [44].

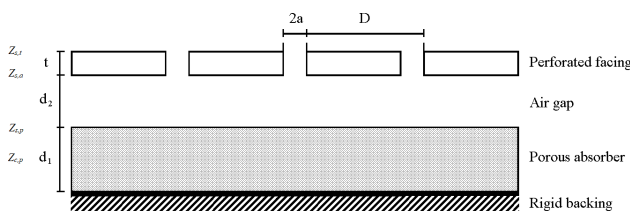


Figure 11: Sketch of the multilayer layout (adapted from [44])

Two cases were examined: the presence or the absence of an air gap.

In the presence of an air gap the assumption that only plane waves are propagating in the porous absorber is made; the surface impedance $Z_{s,p}$ at the top of the porous absorber is then calculated by Eq. 2:

$$Z_{s,p} = -jZ_{c,p} \cot(k_p d_1) \quad (2)$$

where $Z_{c,p}$ and k_p are respectively the characteristic impedance and the wavenumber of the porous absorber and d_1 is its thickness.

The surface impedance $Z_{s,a}$ at the top of the air layer is then given by Eq. 3:

$$Z_{s,a} = \frac{-jZ_{s,p}Z_{c,a} \cot(k_a d_2) + Z_{c,a}^2}{Z_{s,p} - jZ_{c,a} \cot(k_a d_2)} \quad (3)$$

where $Z_{c,a}$ and k_a are respectively the characteristic impedance and the wave number of air and d_2 is the air gap thickness.

Finally, the total surface impedance $Z_{s,t}$ of the whole multi-layer sample is given by Eq. 4:

$$Z_{s,t} = Z_{s,a} + r_m + (2\delta a + t) \frac{j\omega\rho}{\sigma} \quad (4)$$

where $r_m = \frac{\rho}{\sigma} \left(\frac{t}{2a} + 1 \right) \sqrt{8\nu\omega}$ is the resistance term (assuming that the hole radius is not submillimeter in size, to ensure it is larger than the boundary layer thickness: for the same reason the viscous term due to the boundary layer effect can be neglected [45], $\nu = 15 \times 10^{-6} \text{ m}^2/\text{s}$ is the kinematic viscosity of air, ω is the angular frequency (1/s) and δ is the end correction factor. Several studies and formulas are devoted to the end correction factor (see for instance Sivian [46]; Melling [47]; Jaouen and Bécot [48]). In this paper the formula $\delta = 0.8(1 - 1.47\sigma^{1/2} + 0.47\sigma^{3/2})$ is used as suggested by Rscchevkin [49, 50] and experimentally obtained from Fok's function [51] by Nesterov [52]: this formula is valid for every value of σ and also takes into account the interaction effects between adjacent orifices [53].

In the absence of the air gap, the assumption that only plane waves are propagating in the porous absorber can lead to an underestimation of the resistance whose magnitude can be remarkable; for this reason, only in the case of the perforated sheet being in contact with porous absorbent, is the method described by Ingard [5] used. In this case, with the thickness $d_2 = 0$, the surface of the porous sheet is imagined to be split into a series of elementary cells each containing one hole, whose size is $D \times D$; the impedance at the top of the porous absorber, or below the perforated sheet, is then given as a sum of modal terms:

$$Z_{s,a} = \frac{Z_{0,0}\sigma}{\varphi} + \frac{4}{\pi} \sum_m \sum_{n, (n \neq 0 \& m \neq 0)} \frac{\nu_{m,n} Z_{m,n} J_1^2 \left(\frac{2\pi a}{D} \sqrt{m^2 + n^2} \right)}{\varphi(m^2 + n^2)}$$

$$Z_{m,n} = -jZ_{c,p} \frac{k_p}{k_{m,n}} \cot(k_{m,n}d_1)$$

$$k_{m,n} = \sqrt{k_p^2 - \frac{4m^2\pi^2}{D^2} - \frac{4n^2\pi^2}{D^2}}$$

$$v_{m,n} = \begin{cases} 0.5 & m = 0 \text{ or } n = 0 \\ 0 & \text{otherwise} \end{cases}$$

where φ is the porosity of the porous absorber, J_1 is the Bessel function of the first kind and first order, and the sum is carried out over all combinations of n and m when both are not equal to zero.

Knowing the impedance described above, the total surface impedance $Z_{s,t}$ of the whole multi-layer sample is then given by:

$$Z_{s,t} = Z_{s,a} + (2\delta a + t) \frac{j\omega\rho}{\sigma} \quad (5)$$

To calculate the characteristic impedance and the wavenumber of the porous absorber to be inserted in the previous formulas, the empirical method of Delany and Bazley [42] was used:

$$Z_{c,p} = \rho_0 c_0 \left[1 + 0.0571 \left(\frac{\rho_0 f}{R_f} \right)^{-0.754} - j0.0870 \left(\frac{\rho_0 f}{R_f} \right)^{-0.732} \right] \quad (6)$$

$$k_p = \frac{\omega}{c_0} \left[1 + 0.0978 \left(\frac{\rho_0 f}{R_f} \right)^{-0.700} - j0.1890 \left(\frac{\rho_0 f}{R_f} \right)^{-0.595} \right] \quad (7)$$

where f is the frequency (Hz) considered. The Delany-Bazley correlation was chosen for different reasons. The first is that this empirical model is based on knowledge of only one parameter (flow resistivity), thus being very easily implemented in codes and models for engineering purposes; moreover, despite being the oldest model available, it was demonstrated that in its frequency range of validity, *i.e.*, $10^{-2} \leq (f/R_f) \leq 1$, it gives the smallest relative error between measured and predicted data if compared with other predictive methods [54]. The range of validity of the model is respected for the data presented in this paper, since as previously said, the frequency range investigated in this study varies from 160 to 2,500 Hz and the flow resistivity of the porous absorbing material being equal to 4,285 Pa·s/m², the validity of the model is assured.

Finally, the normal incidence sound absorption coefficient is computed by means of the following formula:

$$\alpha = 1 - \left| \frac{Z_{s,t} - \rho_0 c_0}{Z_{s,t} + \rho_0 c_0} \right|^2 \quad (8)$$

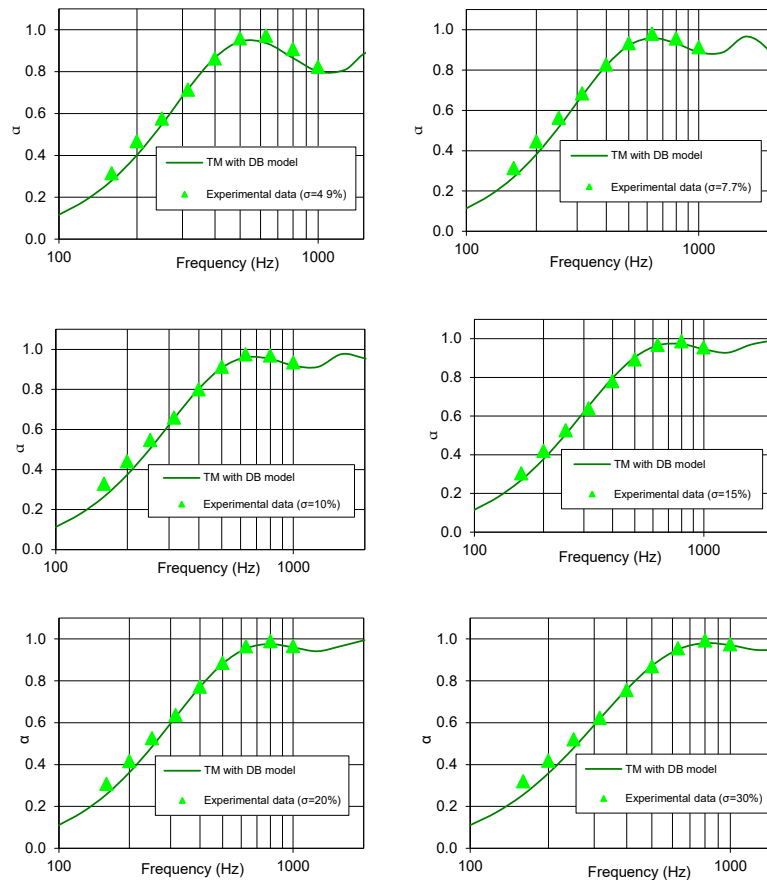


Figure 12: Experimental and predicted normal incidence sound absorption coefficient comparison (2 mm air gap)

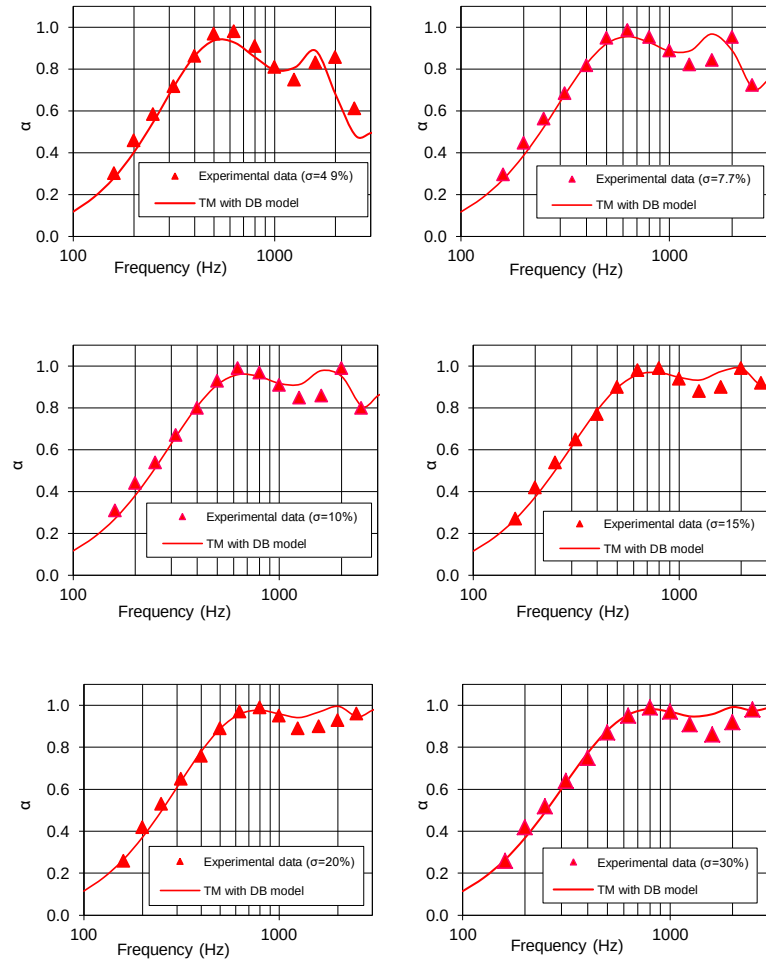


Figure 13: Experimental and predicted normal incidence sound absorption coefficient comparison (no air gap)

In the following Figures 12 and 13, a comparison between experimental and predicted values of the normal incidence sound absorption coefficient is reported for two different cases: with the presence of a 2 mm air gap (Figure 12) and without the presence of the air gap (Figure 13). Numerical modeling confirmed the slight effect of the air gap thickness on system behavior already shown by the experimental data reported in Figure 9, with an actual superimposition of α curves for different air gaps. For the sake of brevity, only the curves for the smaller air gap and diverse percentages of open area have been drawn, even though similar results were obtained for all the clearances.

As can be noted, for an air gap of the same thickness as the diameter of the circular holes of the perforated plate, the simplified transfer matrix model (TMM) using the Delany-Bazley correlation gives excellent agreement with experimental data. This accuracy goes up to the upper limit of the tested frequency range, i.e., 1,000 Hz. The effect of changes in percentage of perforation are well interpreted by the model, which reproduces the slight variation of the α values

measured. The agreement for a perforated plate in direct contact with the porous media, which needs completer and more complex implementation due to multiple waves propagating in the material, is good albeit not so much as for the previous case. The presence of a second peak in the α curve is roughly predicted, but the maximum value and the corresponding frequency do not exactly correspond with the experiments.

For the purpose of quantifying the accuracy of the simulation model in Figures 14 and 15, the comparisons of experimental (x axis) and predicted (y axis) values are reported for all simulations performed. The accuracy of the predictive model has been evaluated according to two criteria: the fraction of data predicted to within $\pm 30\%$, called λ , and the mean absolute percentage error, ϵ :

$$\epsilon = \frac{1}{N} \sum_{i=1}^N \left| \frac{\text{predict.value} - \text{exp.value}}{\text{exp.value}} \right| \% \quad (9)$$

The mean absolute percentage error is equal to 5.2% without an air gap and 4.7% with an air gap, while the overall

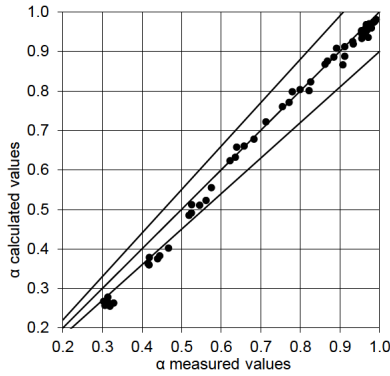


Figure 14: Measured and calculated normal incidence sound absorption coefficient comparison (2 mm air gap)

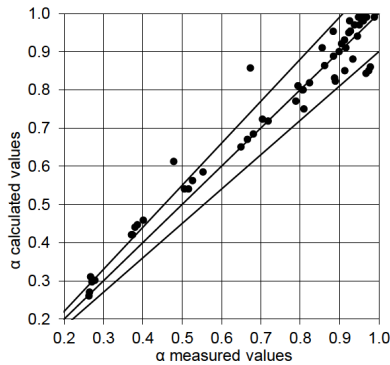


Figure 15: Measured and calculated normal incidence sound absorption coefficient comparison (no air gap)

ϵ value is 5.0%. Using the same set of data, the fraction of data predicted to within $\pm 10\%$ is calculated to be 82%, 80% and 81% respectively for no air gap, an air gap and all data set. This analysis confirmed the modeling effectiveness in simulating the effect of perforated facings on the acoustical behaviour of absorbing material.

In general, the trends of experimental and computed data are very close, thus suggesting that the chosen methodology is capable of predicting the sound absorbing behaviour of a multi-layer sandwich composed of a porous material, an air gap and a perforated facing with acceptable accuracy. One of the research aims was to find a method to predict the performance of such a system depending on its physical characteristics, and numerical modeling through TMM confirmed to be an operative choice, just as recommended by Allard and Atalla [25]. Through TMM the effect of perforations can be accurately predicted, thereby allowing one to evaluate acoustical transparency of facings in an easy and fast way. Indeed, it is worth highlighting that the adopted numerical method is relatively easy and fast to implement, thus being available not only for scientific analysis, but even for technical design [55].

With the adopted approach, through a few experimental tests aimed at model validation, it is possible to analyze a wide range of technical solutions by simulating the variation of each physical parameter. This method allows the user to save time and money, still obtaining accurate results. This conclusion is neither original [44], nor disrupting, but shows that when the transparency of perforations is to be assessed, numerical methods and TMM are a very effective option. By way of example, some simulations have been run in order to analyze some configurations that were not studied with the experimental set-ups previously described.

Figure 16 displays the effect of the perforated panel thickness, assuming an air gap of 0.5 mm, 100 mm polyester fiber and two different percentages of open area equal to 4% and 40%. First, a marked influence of σ on absorption coefficient can be noticed, thus confirming the strong impact of the facing features on the sound absorbing behavior. Furthermore, as expected, increasing the thickness of the panel decreases the absorption coefficient at medium-high frequencies (approximately from 400 Hz up), while at low frequencies (between 50 and 400 Hz) it increases slightly. This effect is more evident for the 4% open area, since the open area equal to 40% ($TI=13,040$) is acoustically very transparent. In this case, the effect of the thickness becomes evident only for frequencies above 1,000 Hz and it is in any case definitely lower.

In Figure 17, the effect of the diameter of the holes (and, consequently, of their spacing) is investigated for a percentage of open area of 12.5%, panel thickness equal to 1 mm and again 100 mm thick polyester fiber. When the diameter increases, with all other parameters held constant, α values tend to reduce for frequencies approximately above 500 Hz, as theoretical and experimental studies expect to happen. In addition, the simulations reported in Figure 17 show that, in accordance with Mechel [16], a small air gap can cause some modification in the absorption at medium and high frequencies, and that this aspect deserves to be taken into consideration. The same parameters are analyzed for a percentage of open area of 25% in Figure 18. This σ value is over the threshold of 20% that limits the plate transparencies actually affecting the sound absorption, as observed in Bies [33] and in this work. As can be noted, given the high transparency of this kind of panel, the effect of all other geometrical parameters is negligible up to 2,000 Hz, where a very small decrease of the absorption coefficient appears as the diameter of the holes increases. All the results are perfectly in accordance with the findings of Schultz [13]. Because of the high transparency of all simulated perforations, the effect of plate thickness or size and geometry of the holes is almost negligible at lower frequencies.

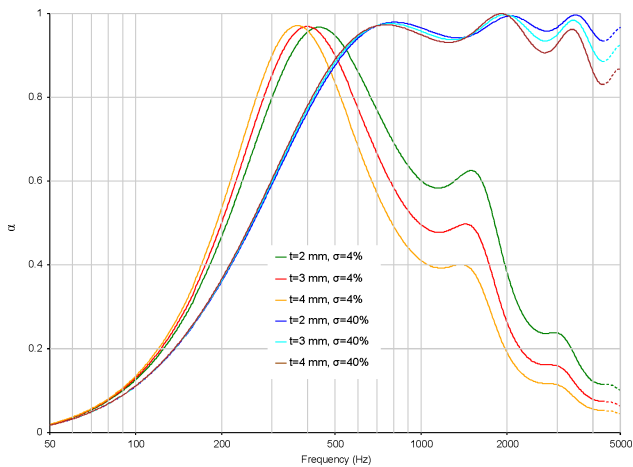


Figure 16: Influence of the plate thickness (0.5 mm air gap)

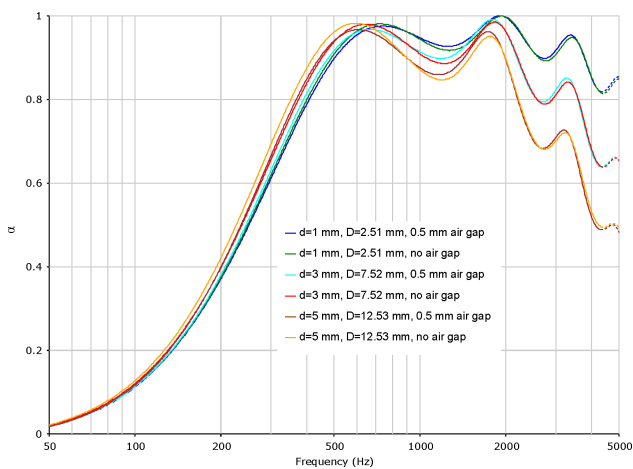


Figure 17: Influence of the hole diameter ($\sigma = 12.5\%$, $t = 1$ mm)

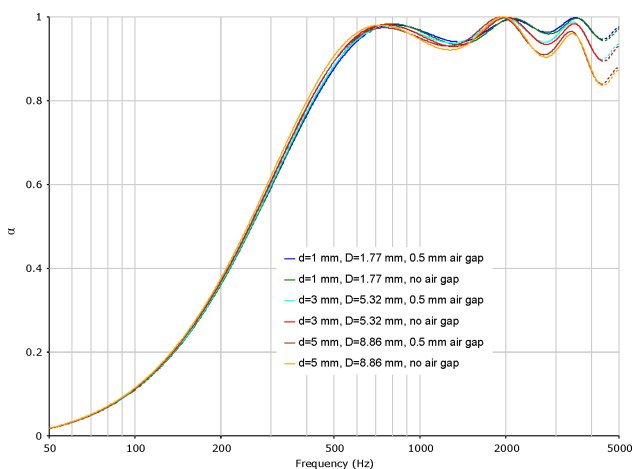


Figure 18: Influence of the hole diameter ($\sigma = 25\%$, $t = 1$ mm)

The simulations that have been performed exemplify how numerical methods can be effectively used to assess the acoustic transparency of perforated metal plates facing

sound absorbing material. Moreover, TMM gives useful information regarding the effect of physical characteristics influencing their behavior. This method is not an alternative to experimenting for the purpose of deep comprehension of sound attenuation mechanisms, but it can be a simple and reliable way to detect whether perforations affect acoustic performance of the backing materials in mufflers, lined ducts, and panels.

7 Conclusions

In this study, the acoustical transparency of perforated facings backed by a sound absorbing material was investigated to assess how perforations can alter the system behavior. The different behaviors due to the multiple combinations of percentage of open area and air gap thickness between perforated facing and absorbing material were analyzed and discussed. The operational objective was to assess when perforations can be considered acoustically transparent and do not actually affect the performance of the backing layer, only playing a role as a rigid support or protective screen for the fibrous material. This knowledge plays a meaningful role, among other things, in designing lined ducts and dissipative silencers.

During the experimental study, first of all, the acoustical behavior of un-faced polyester fiber was characterized by measuring flow resistivity, surface impedance and sound absorption coefficient at normal incidence. The comparison with the most used literature correlations showed good coherence between measured and calculated values. In this way, the precision of the experiments was assessed, as well as the standard features of the fibrous material. The Delany-Bazley correlation was later used in order to theoretically analyze the behavior of the sound absorbing system.

In order to analyze the effect of perforations, polyester fiber samples were then faced using different metal plates perforated with circular holes ($\varnothing = 2$ mm), initially placed in close contact with the absorbing material and then set at a variable distance from it. Experimental data essentially agreed with the literature results, *i.e.*, the acoustic properties were found to depend on the percentage of perforation, with a threshold over which their effect is negligible. In case the percentage of open area is equal to or greater than 20% of the facing surface, the latter takes on the mere role of supporting and protecting the sound-absorbing material. For a smaller percentage of open area, as σ decreases, the α curves for faced and un-faced samples tend to distance, thereby indicating different acoustical behaviors of the lined duct or of the absorbing panel to which the facing

is applied. In particular, the frequency corresponding to the peak value of α tends to decrease, just as the theoretical models predict. The real part of the impedance very slightly depends on σ , while the imaginary part increases when the percentage area ratio reduces, due to the mass added to the facing.

Later, perforated plates were placed at a distance of 2, 4, and 6 mm from the absorbing material, and the variation of the sound absorption coefficient with the air gap thickness was measured. Air gaps seem to slightly affect the acoustic behavior of the covered absorbing material. The differences between the α curves, for a wide range of air gaps, turned out to be very small for all the percentages of open area tested. In particular, when the dependence of acoustic impedance on the air gap is examined, no evident effect is observed for any of the d values considered during the experiments. This result appears to be valid for all open-area percentage values and for both the real and imaginary part.

Finally, a system made of porous fiber material, perforated facing and air gap was simulated by the Transfer Matrix Method. The validation of the model through comparison with the experimental data measured was successful, even when no air gap was considered. The model accurately predicts the behavior of some different configurations, with different open-area percentage and air gaps. The values of mean absolute percentage error, ϵ , and fraction of data predicted to within $\pm 30\%$, λ , confirmed the good accuracy of the numerical model. λ is equal to 82%, 80% and 81% respectively for no air gap, an air gap and all data set, while ϵ values are 5.2% without air gap, 4.7% with air gap, and 5.0% for the whole data set. In order to illustrate the effectiveness of TMM, some simulations were run for different values of the significant physical parameters. The outcome was coherent with roughly what was expected. Experimental validation and physical coherence of the results showed the possibility of using this method to assess the acoustic transparency of perforated metal plates and their effect on the sound absorption of backing material.

Generally, the experiments confirmed that the acoustic behaviour of faced absorbing material strongly depends, among other features like hole geometry, wall thickness and fibers in contact with the facing, on the perforated percentage area and on the air gap of the facing material. Such a variation of sound absorption coefficients and specific surface impedances may greatly affect the performance of dissipative silencers, thus calling for detailed comprehension and accurate modeling of the physical phenomenon. Through numerical modeling the effect of a protective perforated facing in lined ducts and parallel-baffle mufflers or in all those cases in which a perforated screen is aimed to

cover and preserve an absorbing material can be analyzed in an economic and fast way, so that a number of technical issues can actually benefit from this result.

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