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Experimental Characterization of Acoustic Liners with Extended Reaction

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Suppressing of jet engine noise by inlet and exhaust duct liners and internal combustion engine (ICE) noise by intake and exhaust systems is an important part of developing environmentally acceptable vehicles. The acoustic liner is designed to provide an impedance boundary condition in the engine duct that reduces the propagation of engine noise through the duct. An accurate impedance boundary condition is necessary to optimally suppress the noise at different conditions. The goal of the research presented in this paper is to present a new technique to Educe and characterize the acoustic liner impedance for cases with extended reaction. This technique is depending on comparing both the measured and predicted 2-port transfer matrices. The measurement of the transfer matrix is performed using the two microphone technique, while the prediction of the transfer matrix is obtained assuming plane waves in the inner pipe and outer chamber coupled by a perforated wall impedance. By using a regression process the unknown wall impedance is then educed. The method is applied to investigate the effect of flow on the impedance of so called Micro-perforated panels (MPP). A MPP consists of a panel (here a plate made of Al or steel) with small perforations distributed over its surface. When these perforations are of sub-millimeter size they provide by themselves enough acoustic resistance and low acoustic mass reactance necessary for a wideband absorber.

Nomenclature[3]

c = speed of sound
D = hydraulic Diameter (duct area/perforated perimeter)
M = flow Mach number
K = complex wave number
L = length of chamber
U_o = average axial flow speed
u = fluctuation in axial velocity
u_w = velocity fluctuation through the wall
p = fluctuation in pressure
q = fluctuation in axial volume velocity
$\mathbf{T}_P$ = predicted transfer matrix
$\mathbf{T}_M$ = refers to the measured transfer Matrix
Z = characteristic wave impedance
Z_w = normal acoustic impedance
ρ = fluctuation in density
ρ_o = density
ρ_w = density in impedance layer

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I. Introduction

Acoustic liners and perforated mufflers are used to suppress noise in turbofan engine ducts and internal combustion engine (ICE) exhaust noise, respectively, by providing appropriate impedance boundary conditions. Current single degree of freedom liners typically possess a perforated face sheet backed by honeycomb cells and a rigid plate while, perforated mufflers can be found in different configurations such as perforated pipes backed by an air cavity volume, which can be filled with a porous material, and a rigid outer wall. These perforated face sheets and pipes exhibit a nonlinear impedance characteristic that are usually parameterized as a function of sound pressure level (SPL) and grazing flow Mach number. Perforated plates have been studied extensively to design liners for sound attenuation in lined engine ducts[1,2,3,4]. Konishi et al.,[1] investigated micro-perforated plates for the frequency tuning of Helmholtz resonators used as absorbers. They however studied only the resonance frequency change as a function of *thickness/hole diameter* and not the impedance. Studies conducted by Bielak et al.,[3] provided comparisons of the impedance of conventional perforates with micro-perforates for various grazing flow velocities and SPL values. For the aeronautical side the studies done so far are mainly on liners with a local acoustic reaction, i.e., with a response that can be described by a normal impedance. But liner configurations with extended reaction or wave propagation behind the liner perforated face sheet is also of interest and is commonly used, e.g., in automotive mufflers. In this paper a method to educe the liner face sheet normal impedance for a configuration with extended reaction will be investigated. The method is then applied to measure the impedance for two commercial types of micro-perforated face sheets with a thickness around 1 mm and perforation close to 1%. Such plates are designed to have a normal impedance close to the characteristic wave impedance at NTP and are used, e.g., as panel absorbers for noise control[5,6]. Compared to the "thick" micro-perforated face sheets investigated in Refs [2, 3] the sheets (plates) investigated here are "thin", i.e., hole size/thickness <= 1, which imply that they are less linear than the "thick" ones.

A. Earlier work on liner characterization

The normal incidence acoustic impedance of liner materials can be experimentally determined in several ways. The classical method is the standing wave tube method which has been a commonly used procedure for impedance measurements for over 80 years[7]. The development of Fast Fourier algorithms and its implementation in laboratory analyzers provided new possibilities. The standing wave method evolved to an easier and more accurate method using fixed microphone positions, the two-microphone technique. Seybert&Ross[8] were able to separate the incident and reflected wave spectra from measurements of auto- and cross-spectral densities between the microphones. This method has been used extensively and has become an ISO standard method[9]. Another very widely used method is the In-Situ technique. Dean[10] introduced this method in 1974. Since then, it has been extensively used to measure the impedance of locally reacting acoustic liners with and without flow. Several drawbacks and problems have been reported with the use of this method. There are usually strong near field effects which influence the reading of the surface microphone, which is subject to the incident sound wave. These effects are even stronger with grazing flow. Moreover, the in-situ method fails when there is, for example, porous material inside the liner so that the sound propagation inside the cavity cannot be predicted.

A continuing measurement problem in liner treatment technology is the accurate determination of normal incidence impedance of acoustic materials in grazing flow environments. All the above discussed problems motivated the evolution of indirect (inverse) approaches, which depend only on the measurement of the acoustic pressure at selected locations in the test rig. These indirect methods have an extra advantage of not destroying the sample by drilling holes for the microphones. The so-called infinite-wave-guide method, involves measuring the sound attenuation properties of an assumed single, unidirectional propagating mode, in a wave-guide lined with the acoustic material over a sufficient length to be effectively infinite. This data is then used with the solution to the wave equation in an infinite wave-guide to establish the impedance of the material at the boundary. The evolution of wave-guide models for this purpose began over 25 years ago with a uniform mean flow model[11]. Infinite wave-guide models are applicable, in a very straightforward manner, to situations for which a single mode propagates within the wave-guide containing the unknown material. However, many conventional liner concepts generate more complex acoustic fields. Thus, the measured data must be interpreted as the superposition of many propagating modes.

Watson et al.[12] developed this approach further and presented a finite-element-based, numerical method for educing the impedance of an acoustic material placed at the wall of a two dimensional duct that conveys a multi-modal sound field. Their proposed method depends on the measurement of acoustic pressure at selected locations at the upper wall of a rectangular duct to determine the normal incidence impedance of the material placed at the lower wall. These measurements are then used as input to a Finite Element propagation model to extract the impedance of the material. They first tested the method with input data computed from modal theory and then contaminated by random error. Then they validated the impedance extraction method using measured data and a finite-length liner[13]. Later[14], they used the Davidon-Fletcher-Powell optimization algorithm to educe the impedance that minimizes the difference between the measured and the numerically computed upper wall pressure. After developing and validating the method for the no flow case, they could reproduce the measured normal incidence impedance in a uniform flow[15]. They later showed that the liner resistance educed in the presence of mean flow can differ from that educed when only uniform flow is modeled[16]. They developed their code further to be able to handle segmented liners in the circumferential and axial directions (checkerboard liners)[17]. Leroux et al.[18] proposed another method based on the measurement of the scattering matrix of the lined section using the two-source method. The transmission and reflection coefficients before and after the lined section can be calculated from the scattering matrix. They proposed a new theoretical model called "Scattering Matrix by Multimodal Method (S3M)" to compute the scattering matrix of a lined duct with flow.

A similar indirect method is presented by Elnady and H. Bodén [19]. The liner sample is placed inside a rectangular duct. The amplitude of the plane wave incident towards the lined section is measured using the two-microphone technique. The reflection coefficient at the exit plane is also measured using the same technique. These measured values are fed to an analytical model for sound propagation through the lined section, which is constructed using mode matching. A solution of the impedance value is educed in the complex plane to match the calculated sound field to the measured one. Two cost functions are tested and compared. The first is the summation of the differences (magnitudes) between the measured and calculated complex pressures at the microphone positions. These measurement positions are already used for the two-microphone technique. The second cost function is the difference between the measured and calculated complex amplitude of the scattered plane wave.

II. Description of the proposed method

A. The 2-port model

It is assumed that a liner configuration as in Figure 1 is given. This configuration is for instance typical to a muffler used for automotive exhaust noise control. The issue is now, assuming that the state in the fluid is known everywhere, to determine the properties of the impedance layer (surface) which couples the inner (1) and outer regions (2).

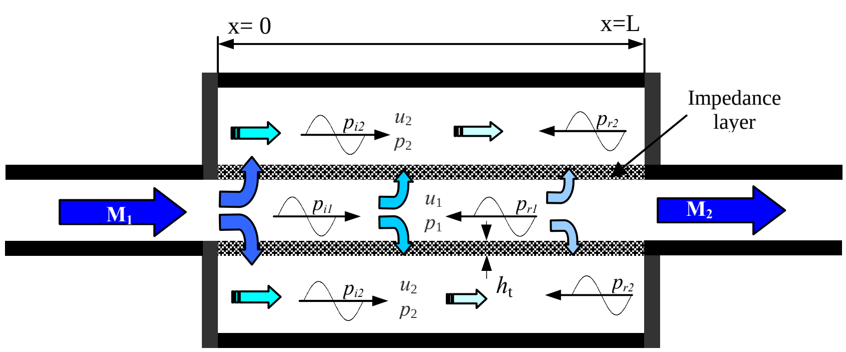


Figure 1. Flow and acoustic waves in the assumed liner configuration with extended reaction.

Based on an earlier published paper by the authors[20], the governing equations for 1D acoustic waves in this system are:

$$\frac{\partial \rho_j}{\partial t} + U_{oj}\frac{\partial \rho_j}{\partial x} + \rho_{oj}\frac{\partial u_j}{\partial x} = (-1)^j \frac{\rho_w}{D_j} u_w, \tag{1}$$

$$\rho_{oj}\left(\frac{\partial}{\partial t}+U_{oj}\frac{\partial}{\partial x}\right)u_j=-\frac{\partial p_j}{\partial x}. \tag{2}$$

Here $j=1,2$ denotes the inner pipe and outer chamber see Figure 1. The coupling between the fields; inner pipe and outer camber 1 and 2 is done via an acoustic impedance

$$Z_w=\left(p_1-p_2\right)/u_w. \tag{3}$$

To solve the problem a propagating wave ansatz is made and harmonic space and time dependence are introduced. Suppressing the harmonic time dependence ($e^{i\omega t}$) the fluctuating quantities can then be written as

$$p_j\,(\mathrm{x})=\hat{p}_j\,\mathrm{e}^{-iKx},\ u_\mathrm{j}\,(\mathrm{x})=\hat{u}_j\,\mathrm{e}^{-iKx},\ \hat{p}_j=c_j^2\,\hat{\rho}_j,\ \hat{p}_j=Z_j\hat{u}_j \tag{4}$$

Substituting equations (3) and (4) into equation (2) gives

$$\rho_{oj}\left(i\omega+U_{oj}\,(-iK)\right)Z_j^{-1}=iK. \tag{5}$$

From equation (5) the characteristic wave impedance can be obtained

$$Z_j=\frac{\rho_{oj}\omega-\rho_{oj}U_{oj}K}{K}=\frac{\rho_{oj}c_j\left(k_j-M_jK\right)}{K}, \tag{6}$$

where $M_j=U_{oj}/c_j$, and $k_j=\omega/c_j$. Substituting equations (3) and (4) into equations (1) and with the help of equation (6) gives

$$\left(\frac{ik_j}{c_j}\right)\hat{p}_j-\left(\frac{iM_jK}{c_j}\right)\hat{p}_j-\frac{iK^2}{c_j\left(k'_j-M_jK\right)}\hat{p}_j=(-1)^j\,\frac{\rho_w}{D_j\,Z_w}\left(\hat{p}_1-\hat{p}_2\right). \tag{7}$$

This equation is simplified by Multiplying with $i\,,c_j$ and putting $B_j=\dfrac{c_j\rho_w}{D_jZ_w}$ gives

$$-\left(k_j-M_jK\right)\hat{p}_j+\frac{K^2}{\left(k_j-M_jK\right)}\hat{p}_j=(-1)^j\,iB_j\left(\hat{p}_1-\hat{p}_2\right). \tag{8}$$

Equation (8) represents a pair of homogenous linear equations which have non-trivial solutions (eigenvalues) for the wave-numbers K corresponding to free waves in the two channels. This linear equation system can be written as

$$\begin{pmatrix} K_1^2+iB_1\left(k_1-M_1K\right) & -iB_1\left(k_1-M_1K\right) \\ -iB_2\left(k_2-M_2K\right) & K_2^2+iB_2\left(k_2-M_2K\right)\end{pmatrix}\begin{pmatrix}\hat{p}_1\\ \hat{p}_2\end{pmatrix}=\begin{pmatrix}0\\0\end{pmatrix} \tag{9}$$

where $K_j^2=K^2-\left(k_j-M_jK\right)\left(k_j-M_jK\right)$. Equation (9) defines a forth order algebraic equation for the wave-numbers K_n, n =1,2,3,4. To each of the wave-numbers there is a corresponding 2-D mode (eigenvector) $\mathbf{e}_n$. The eigenvalues and corresponding modes can be calculated numerically for instance by using Matlab. Using these eigenvalues and modes a general expression for the sound field can be obtained in the form of a 4x4 matrix **H**(x), which defines the relationship between *p* and *q* and the modal amplitudes at a cross-section *x*. Applying this result to *x*=0 and *x*=*L* and solving the modal amplitudes from the second of these equations and putting the result into the first, the four-port transfer matrix $\mathbf{S} = \mathbf{H}(0)\mathbf{H}^{-1}(L)$) is calculated. This four-port matrix is then reduced to a two-port matrix $\mathbf{T}_p$ by using the rigid wall boundary conditions in pipe 2, i.e., $\hat{q}_2(0) = 0$ and $\hat{q}_2(L) = 0$. A straightforward derivation reveals that[20]

$$\begin{pmatrix} \hat{p}_1(0) \\ \hat{q}_1(0) \end{pmatrix} = \mathbf{T}_p \begin{pmatrix} \hat{p}_2(L) \\ \hat{q}_2(L) \end{pmatrix}, \text{ with } \mathbf{T}_p(Z_w) = \begin{pmatrix} S_{11} - S_{12}S_{41}/S_{42} & S_{13} - S_{12}S_{43}/S_{42} \\ S_{31} - S_{32}S_{41}/S_{42} & S_{33} - S_{32}S_{43}/S_{42} \end{pmatrix}. \tag{10}$$

B. The measured 2-port

If there are no internal sources inside the two-port element the transfer-matrix could be written in the following form[21]

$$\begin{pmatrix} \hat{p}_u \\ \hat{q}_u \end{pmatrix} = \begin{pmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{pmatrix}_M \begin{pmatrix} \hat{p}_d \\ \hat{q}_d \end{pmatrix}, \tag{11}$$

where (*u, d*) is just up and downstream the test object.The transfer matrix is measured using the 2-microphone technique and experiments were carried out at room temperature using the flow acoustic test facility at The Marcus Wallenberg Laboratory, KTH. The test ducts used during the experiments consisted of standard steel-pipes with a wall thickness of 3 mm. The duct diameters were chosen to fit the test objects with 57 mm inner diameter. Eight loudspeakers were used as acoustic sources, as shown in Figure 2. The loudspeakers were divided equally between the upstream and downstream side. Each loudspeaker was mounted in a short side-branch connected to the main duct. The distances between the loudspeakers were chosen to avoid any minima at the source position. Fluctuating pressures were measured by using six condenser microphones (B&K 4938) flush mounted in the duct wall. The measurements were carried out using two different types of signals; swept-sine and random noise with different number of averages in the frequency domain. The two-port were obtained using the source switching technique as described in reference[21]. The flow velocity was measured using a pitot-tube connected to an electronic manometer (Swema Air 300). It was measured at a distance ten times the duct diameters from the loudspeakers and six times the duct diameters from the test object diameter in order to avoid any flow disturbance. The flow up and down stream of the test object was measured separately before and after the acoustic measurements and the average result was used. In order to suppress flow noise the transfer functions between the reference signal (voltage to the loudspeakers) and the microphone signals were measured.

By comparing the predicted (which is a function of the unknown wall impedance, Z_w) and the measured transfer matrices $\mathbf{T}_p$ and $\mathbf{T}_M$ a non-linear equation can be obtained:

$$\mathbf{f}(Z_w) = \left[\mathbf{T}_{P11}(Z_w) \quad \mathbf{T}_{P12}(Z_w) \quad \mathbf{T}_{P21}(Z_w) \quad \mathbf{T}_{P22}(Z_w)\right]^T - \left[\mathbf{T}_{M11} \quad \mathbf{T}_{M12} \quad \mathbf{T}_{M21} \quad \mathbf{T}_{M22}\right]^T \tag{12}$$

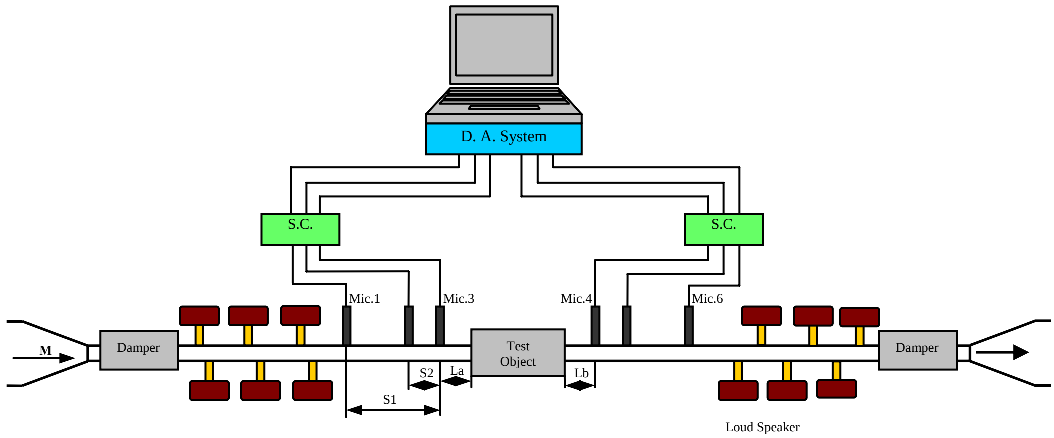


Figure 2. Measurement configuration for 2-port measurements at MWL.

The goal is to find the root of this equation $\mathbf{f}(Z_W) = \mathbf{0}$, and since it difficult to find the derivative of **f** and to use Newton's Method , the secant[22] method was used to compute the roots

$$Z_{w,n+1} = Z_{w.n} - \left(Z_{w,n} - Z_{w,n-1}\right)\left(\mathbf{f}\left(Z_{w,n}\right) - \mathbf{f}\left(Z_{w,n-1}\right)\right)^{-1} \mathbf{f}\left(Z_{w,n}\right) \tag{13}$$

The initial starting value $Z_{w,n}$ is calculated from the empirical formula for perforates published in Ref.[23], while $Z_{w,n-1}$ can be set to (say) 90% of $Z_{w,n}$. It was found that this procedure converged without a problem except for cases where the perforate impedance is close to the characteristic impedance in air (less 0.1 normalized).

In this study the six different MPP plates which are commercially available and shown in Figure 3 were tested. Data for the plates are summarized in Table 1. Three plates had circular holes (*C*) and three had slit-shaped holes (*S*), the slit-shaped hole sample S1 is used in two directions, first when the edge of the slit faces the flow and then in the opposite direction as shown in Figure. For the S2&S3 samples only the results when the plates were tested with the slits parallel to the flow direction are presented. For the S3 plate with a flat surface, unlike S1&S2 with slightly corrugated surfaces, the orientation of the slit should have small or no influence on the result.

Sample	Hole diameter	Thickness	porosity	Sample	Slit width	Thickness	porosity
C1	1 mm	1 mm	2 %	S1	0.5 mm	1.5 mm	1 %
C2	1 mm	1 mm	1 %	S2	0.3 mm	1.5 mm	2 %
C3	1 mm	1 mm	0.5%	S3	0.3 mm	1 mm	5 %

Table 1. Details of the six MPP:s for which the normal acoustic impedance was determined experimentally using the proposed method.

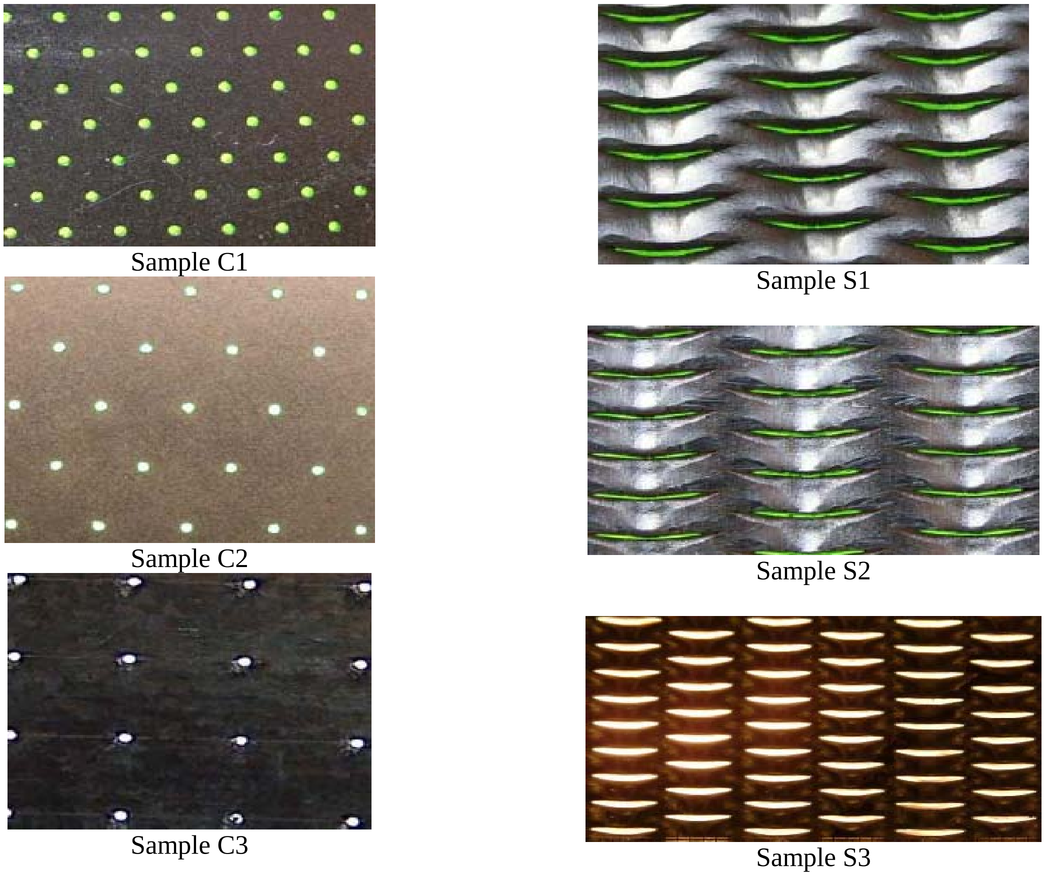


Figure 3. Photos of the tested MPP:s: C with circular holes and S with slit-shaped holes.

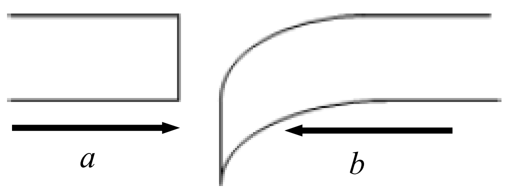


Figure4. Typical shape of the slit-shaped holes in sample S1 and S2: (*a*) and *b*) refers to opposite and with flow direction, respectively.

III. Results and Discussions

A. Validation test with no flow

The educed results using the proposed technique is compared with that measured using the two microphone technique for the no flow case with the sample installed across the duct to verify the accuracy of the new technique. As can be seen from Figure 5, the agreement between the two results is excellent.

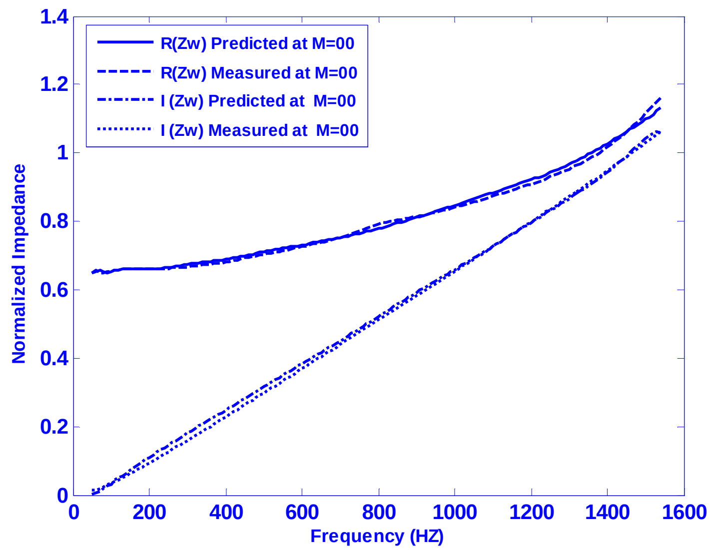
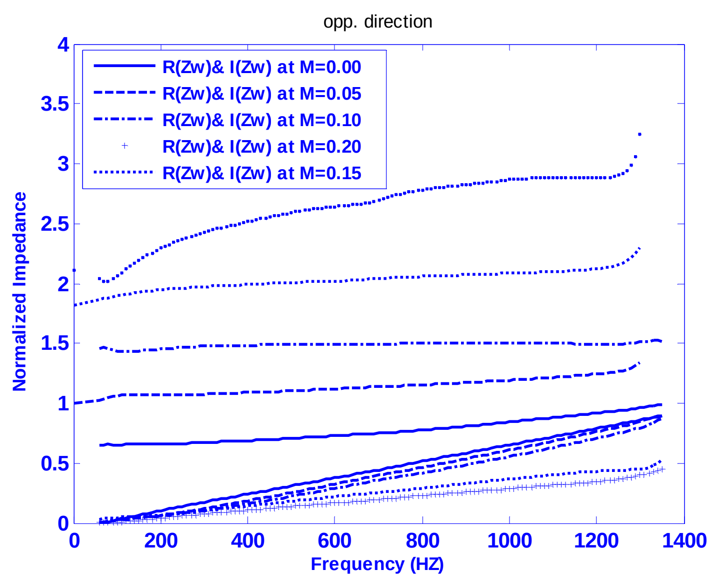


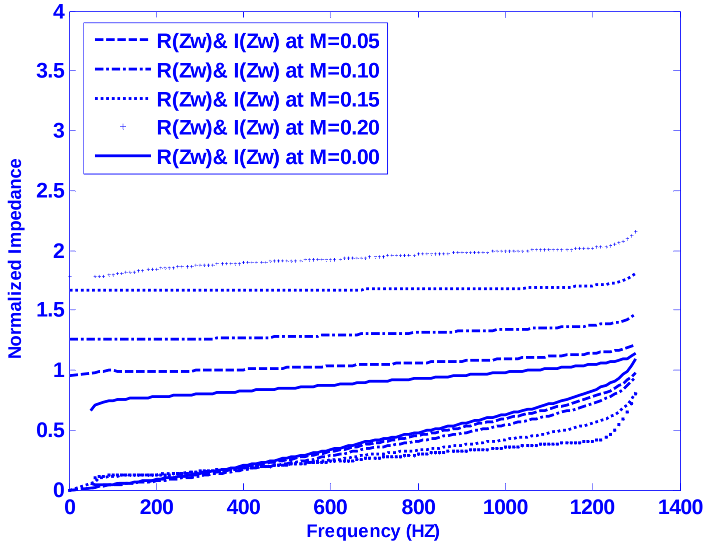
Figure 5. Normalized impedance versus Frequency for Measured (ref case) and Educed Values using the new method.

B. The effect of flow

The effect of flow on the wall impedance is presented in Figure 6 to Figure 11. It is clear from these figures that the resistive part (approximately constant) increases with the flow and that the reactive part (mass plug ~ proportional to the frequency) slightly decrease.



(a)



(b)

Figure 6. Educed Normalized impedance versus Frequency at different flow Mach numbers for the sample *S1* with *slit-shaped holes.* Porosity=1 %, width of slit =0.5 mm, plate thickness=1.5 mm. (a) and (b) refers to slit opposite and with flow direction respectively.

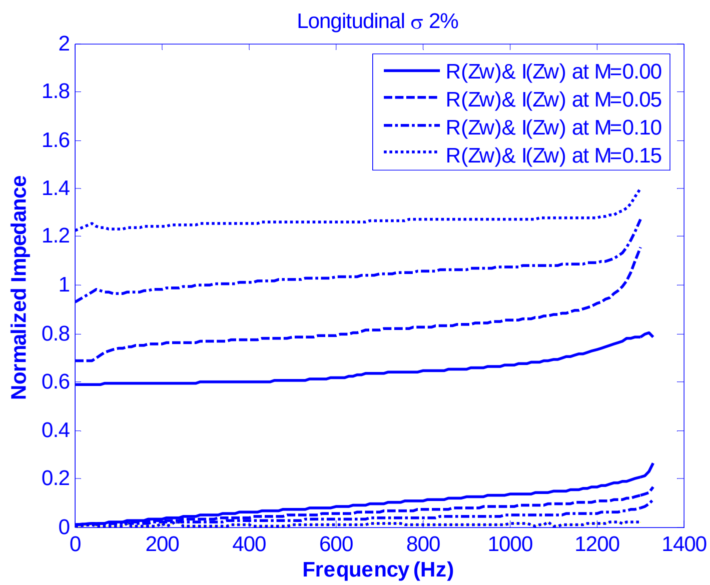


Figure 7. Educed Normalized impedance versus Frequency at different flow Mach numbers for the sample *S2* with *slit-shaped holes.* Porosity=2 %, width of slit =0.3 mm, plate thickness=1.5 mm.

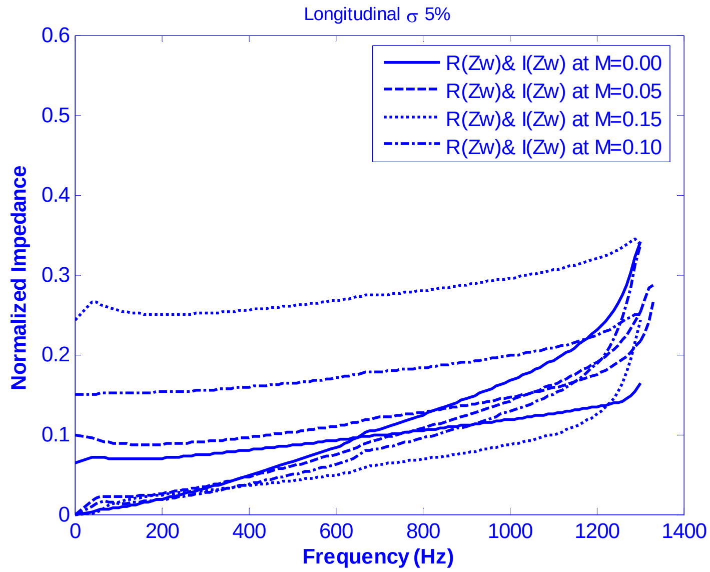


Figure8. Educed Normalized impedance versus Frequency at different flow Mach numbers for the sample *S3* with *slit-shaped holes.* Porosity=5 %, width of slit =0.3 mm, plate thickness=1mm.

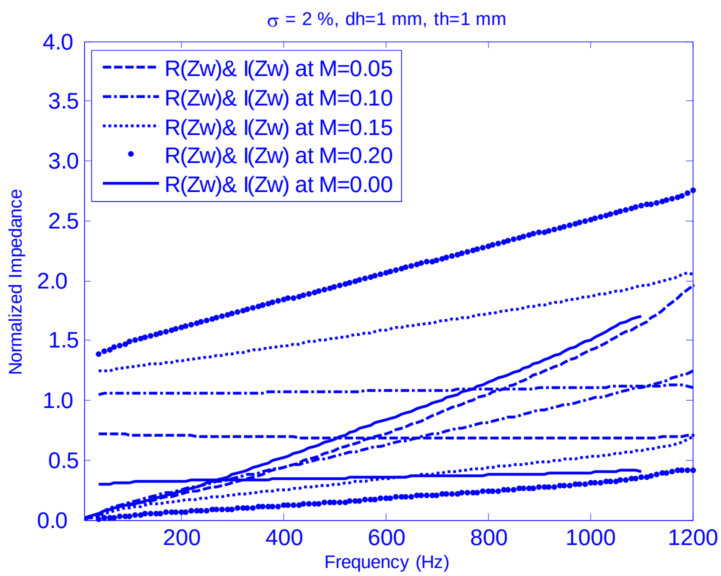


Figure 9. Educed Normalized impedance versus Frequency at different flow Mach numbers for the sample *C1* with *circular holes*. Porosity =2 %, hole diameter=1 mm, plate thickness=1 mm.

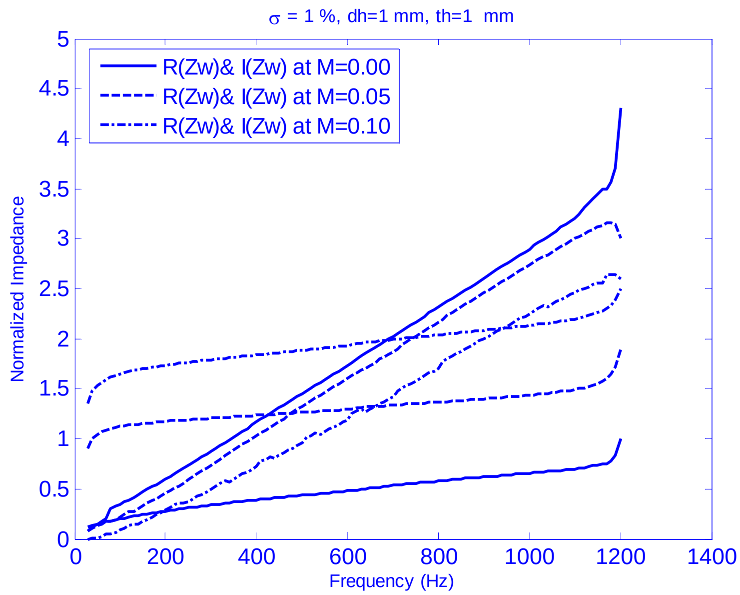


Figure10. Educed Normalized impedance versus Frequency at different flow Mach numbers for the sample *C2* with *circular hole*s. Porosity = 1 %, hole diameter=1 mm, plate thickness=1 mm.

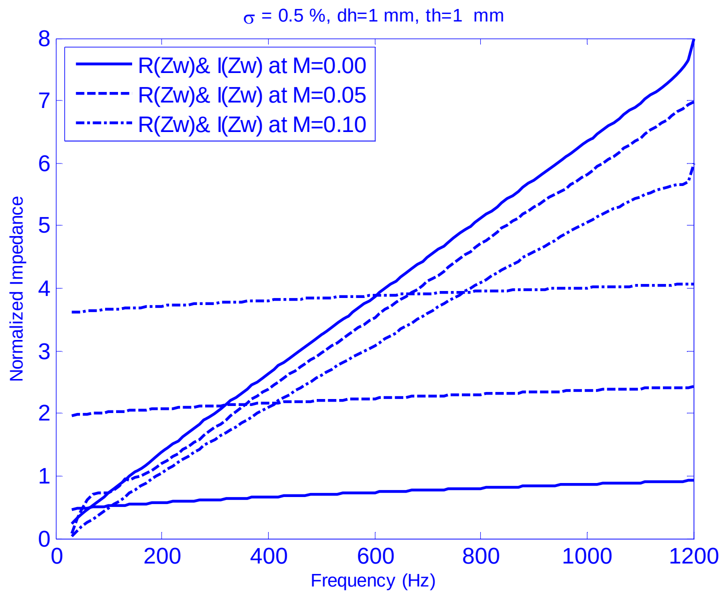


Figure 11. Educed Normalized impedance versus Frequency at different flow Mach numbers for the sample *C3* with *circular hole*s. Porosity = 0.5 %, hole diameter=1 mm, plate thickness=1 mm.

C. Characterization of MPP wall impedance

The normalized impedance ξ of a perforate with combined grazing and through flow (neglecting non-linear effects) is according to the model suggested by Bauer[23]

$$\xi = \left[\left(\frac{\sqrt{8\mu\rho_o\omega}}{\rho_o c\sigma C_D} \right) \left(1 + \frac{th}{dh} \right) \right] + \frac{\beta M_g}{\sigma} + \frac{\alpha M_b}{\sigma C_D} + \frac{jk}{\sigma C_D} \left[t_h + \delta \times F_{\text{int}} \times F_\delta \right] \qquad (14)$$

where $\alpha = 1.15$, $\beta = 0.3$ are semi-empirical constants, μ is the dynamic viscosity, σ is the perforate porosity, M_g is the grazing flow Mach number, M_b is the bias (through) flow Mach number, C_D is the orifice discharge coefficient, t_h is the wall thickness and d_h is the hole diameter. The factor δ is the acoustic end correction for both side of the hole and put equal to $0.62d_h$ and F_δ is the flow effect on acoustic reactance assumed to be $1/(1+305M^3)$ according to Rice[24]. The interaction between the holes is taken into account by $F_{\text{int}} = 1 - 1.47\sqrt{\sigma} + 0.47\sqrt{\sigma^3}$, but for this case with small σ this factor can be put to 1.

To apply the equation to the present case it is assumed that the bias flow is negligible. Based on equation (14) plotting: $\left(\text{Re}\left(\xi(M,f)\right) - \text{Re}\left(\xi(M=0,f)\right) \right) \sigma / M_g$, will give the value for the constant β, see Figure 12 for the MPP:s with circular holes. Excluding the two cases with the sample C1 at M=0.15 and 0.2, which deviate from the rest, it seems that the average value of β is close to 0.15. The reason that the sample C1 deviates is probably that there is effect of bias flow for this plate which has a higher porosity than C2 and C3. Figure 13 illustrates the prediction accuracy obtained when this β value is used in equation (14).

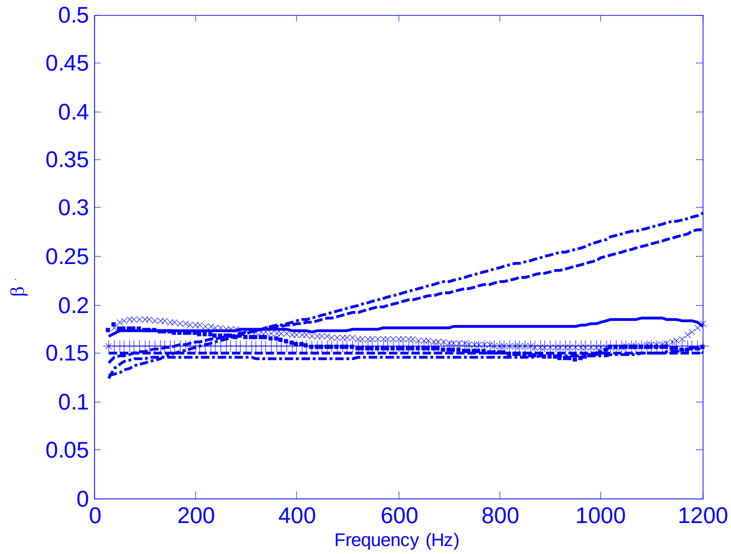


Figure 12. Estimated β value for the tested MPP:s with circular holes, see eq. (20). The value is smaller than the values typically used in the literature which is 0.3-0.5. Two curves deviate from the proposed average value (M=0.15 and 0.2 for Sample C1).

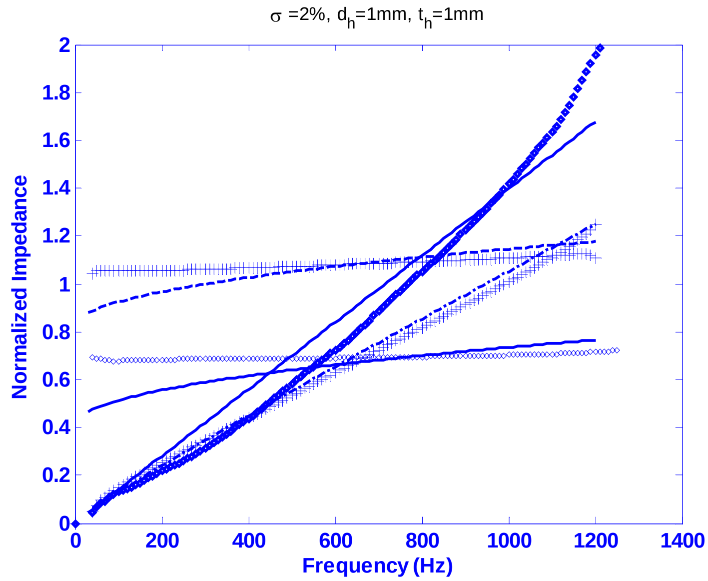


Figure 13. Comparison between educed and predicted impedance versus Frequency at different flow Mach numbers for a sample *C1* with *circular holes*.
Educed results: +, real(Z) & Im(Z) at M=0.1; ◊ , real(Z) & Im(Z) at M=0.05.
Predicted results (with β=0.15): --, real(Z) & Im(Z) at M=0.1; ▬ , real(Z) & Im(Z) at M=0.05.

IV. Summary and Conclusions

In this paper a new simple (1D) technique is presented to educe and characterize an acoustic perforate or liner cover sheet impedance for cases with extended reaction. The technique is depending on comparing both the measured and predicted 2-port transfer matrices. The measurement of the transfer matrix is performed using the two microphone technique, while the prediction of the transfer matrix is obtained assuming plane waves coupled by the perforate or cover sheet impedance. By comparing the measured and predicted transfer matrices using a regression process the unknown impedance is educed.

The method can be developed further to handle more complicated configurations, e.g., axial or circumferential variation of the impedance. Circumferential variation can be included in the present formulation by recalculating the wave numbers in the lined duct, while axial variation implementation requires extension to mode matching at each impedance step. The method has been applied to study the effect of flow on the normal impedance of two types of commercially available micro-perforated plates (MPP:s) with hole size/thickness around 1 mm and perforation ratios around 1%. The results are compared with semi-empirical formulas from the literature[23]. And it appears that the effect of a grazing mean flow on the resistance is smaller than suggested by these formulas.

Acknowledgments

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