

Extended Investigations on Micro-Grooved Elements - A Novel Solution for Noise Control

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ABSTRACT

The goal of this paper is to provide a complete characterization of acoustic performance for a novel type of advanced acoustic material - micro grooved element (MGE). In a previous study, the MGEs have been proved to offer a respectable alternative for the existing and increasingly popular micro perforated elements (MPEs).

The MGEs are multi-layer elements where the acoustic attenuation effect originates from viscous losses taking place in a number of sub-millimeter grooves forming acoustic micro-paths inside the material. This new configuration allows to replace the laser perforation process, used to manufacture the MPEs, with less time consuming and more cost effective technologies. Moreover, such elements preserve low weight and surface roughness. Experiments have demonstrated that the MGEs can be regarded as suitable solution for noise control in a wide range of applications.

In this work, the transfer impedance and the absorption coefficient of a number of MGEs, composed of geometrically different apertures, have been investigated by using the classical two-port method. The effect of the main geometrical parameters controlling the acoustic performance has been analyzed and modeled. Also, the interaction between the micro-paths has been studied. The non-linear behavior, observed in the presence of high excitation levels has been included in the model.

The acoustic impedance of the micro-grooved elements has been derived from the experimentally obtained TL curve in the presence of grazing flow. Consequently, the model developed for MGEs has been further extended to the grazing flow conditions and compared to the well-known models proposed by Rice for micro-perforated elements.

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INTRODUCTION

The first studies on the MPEs have been presented during the '60s by the Chinese scientist Maa and have recently been revised by the same author [1, 2, 3]. Nevertheless, the porous materials, like fibers and foams, have been the most commonly utilized materials in duct acoustics for decades [4]. Nowadays environmental issues have redirected the interest of the market towards absorptive elements where such porous components are avoided.

The main concerns associated with the typical porous materials are the non-renewability, the deterioration of the performance over time and the possible separation of small particles that pollute the surrounding media [6].

Technically, the MPEs are perforated plates with holes typically in the sub-millimeter range and perforation ratios around 1%. Such characteristics provide a normal impedance with a real part close to the characteristic impedance in air

($\sim 400 \text{ Pa}\cdot\text{s/m}$ at standard temperature and pressure - STP), with a constant (frequency independent) resistance and a negligible reactance. This ideal behavior is achieved in the operating range where the acoustic boundary layer thickness is larger than the aperture radius [4]. When the aperture diameter is much higher than the thickness of the boundary layer - typical to ordinary macro perforations -, the resistance provided by the material decreases tremendously and the absorptive properties drop. For this reason the typical macro-perforated metal sheets are often implemented in combination with fibrous materials.

The main disadvantage of the MPEs is related to the manufacturing costs: the micro-perforation process is typically time consuming and expensive.

Lower-cost MP absorbers with irregular slit-shaped apertures are available and usually manufactured by punching the sheet metal [7]. In this case the surface of the perforated plate has considerably increased surface roughness

potentially causing increase in undesirable pressure loss when exposed to grazing flow conditions.

In [8] Randerberg proposed a double-perforated panel, consisting of two parallel, perforated plates separated by a small distance, typically 0.1-0.3mm. In such solution the main part of the energy dissipation takes place in the small gap between the plates. The practical implementation of this panel is limited by the high dependence on the plate distance strongly affecting the acoustic performance of the element. Based on a double layer structure, similar to the one proposed by Randerberg, the authors first presented the MGEs in [9]. The MGE is a patented acoustic solution [10] in which the energy dissipation of the acoustic waves is primarily originated by viscous friction that occurs inside a number of micro-paths. As shown in Figure 1, non-planar configurations of MGEs are possible.

While being fiber-less and recyclable material, the MGEs are still cost effective. The thickness, the weight, and the surface roughness can be designed to satisfy a wide range of applications.

In [9] a number of different MGEs have been tested and the acoustic behavior of the elements has been successfully modeled. An acoustic model, taking into account non-linear effects, has been proposed by adapting the existing models proposed by Maa and Allard for micro-perforated and micro-slit elements.

In [11] the first practical application of MGEs is presented. A number of such elements have been included to represent internal baffles in a novel design for exhaust system muffler. In this work, an extended investigation on the MGEs is presented, as a continuation of the initial investigations presented earlier (e.g in [9]). The effect of geometrical parameters on the element impedance is demonstrated based on 2-port tests and comparisons with simulated results.

The presence of grazing flow has also been accounted by measuring the impedance of a duct shaped MGE inserted centrally inside an expansion chamber.

DESCRIPTION OF MICRO-GROOVED ELEMENTS

The design of a typical planar MGE is described in details in [9]. In Figure 1, a tubular MGE is depicted. Such element is composed of at least two mated concentric layers presenting macroscopic slots and a number of micro-grooves at least on one of the contact surfaces. The slots made into one of the layers are not aligned with the ones present in the other layer coupled [11]. When an acoustic excitation is applied inside the inner duct, the acoustic waves pass through the slots of this duct - the inlet layer -, next they are forced to pass via the micro paths between the two layers and finally exhausting from the element thorough the slots of the outer duct.

The micro-paths are formed by the contact surface of one layer with the micro-grooves machined onto the contact surface of the other layer. In this way, a network of micro-

paths is created, where the dominating energy dissipation is expected to take place.

The presence of the micro-paths between the two contact surfaces represents the main difference between the MGEs and the panels introduced by Randerberg [9].

In addition, the performance of the panels presented in [9] is tremendously affected by the distance between the layers, which can vary due to external stresses acting on the panel.

Compared to the MPEs, the replacement of the micro-holes or the micro-slits of traditional MPEs with the micro-grooves simplifies the manufacturing process.

The constituting layers can be very thin, since the slots give only a small contribution to the dissipation. In this way a weight saving can be achieved and the external surfaces of the panel become smoother.

The two constituting layers can be coupled in several ways. In case of the tubular MPE, the external layer can be longitudinally composed of two parts clamped to the inner layer. Other possible solutions are the pressed mating and, in case of planar elements, point welding and riveting.

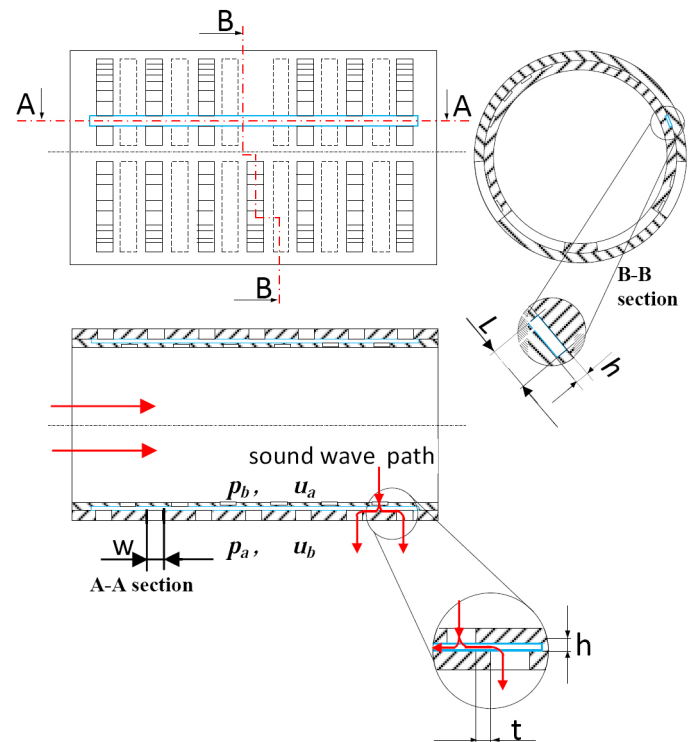


Figure 1. A scheme illustrating the principle and highlighting the dimensions determining the acoustical performance of a tubular MGE.

MODELING

The acoustic behavior of the MGEs in the absence of a grazing or bias flow has been modeled in [9] by adapting the existing models proposed by Maa and Allard for micro-perforated and micro-slit elements. Also the non-linear effects, due to the interaction of the sound wave with the

boundaries of the channels, have been taken into account in the model. In this paragraph, the model presented in [9] is summarized and extended for the grazing flow.

Let p_a and p_b represent the sound pressures upstream and downstream of the acoustic element and u_a and u_b the respective particle velocities (see Figure 1).

For acoustically thin samples, i.e. for samples with thickness much smaller than the wavelength, it is reasonable to assume a single particle velocity across the material, i.e. $u = u_a = u_b$ in Figure 1. This implies that the panel itself can be considered as an in line lumped element [12]. Accordingly,

$$p_a = p_b + Z_{tr}u \quad (1)$$

where Z_{tr} represents the transfer impedance, e.g. the ratio between acoustic pressure drop Δp through the material and the normal particle velocity.

Z_{tr} is usually referred to the impedance of air ($Z = \rho c$), where ρ is the density of air and c is the speed of sound, resulting in the so called normalized transfer impedance ($z_{tr} = Z_{tr}/Z$). Moreover, the normalized resistive and reactive part of the impedance (named resistance - r - and reactance - χ - respectively) are often treated separately, and normalized in respect to Z : $z_{tr} = r + j\chi$. In general it can be stated that, if Z_{tr-s} is the transfer impedance of a single constitutive element of a perforated sample (i.e. the impedance of a single micro-hole, or a single micro-slit or micro-path of the considered element), then the transfer impedance of the entire element becomes $Z_{tr} = Z_{tr-s}/\sigma$, where σ is the panel porosity.

As explained in [8], the acoustic impedance of the MGEs can be divided into the following four major contributions:

An internal part: which can be referred to the dissipation occurring in the micro-paths. Allard derived an expression for the transfer impedance related to the laminar flow travelling in a slit with infinitely large rectangular cross-section, by assuming negligible thermal conductivity [13]:

$$Z_{MSlit-i} = \frac{j\omega T}{\sigma c} \left[1 - \frac{\tanh(k\sqrt{j})}{k\sqrt{j}} \right]^{-1} \quad (2)$$

where ω is the angular frequency, T is the thickness of the plate, j is the imaginary unit, $k = h\sqrt{\omega/4\nu}$ is the so called shear wave number, Stokes number or perforation constant, ν is the kinematic air viscosity, h is the width of the slit, k is proportional to the ratio of the tube radius and viscous boundary layer thickness [1].

It has been found in [9] that the Allard formulation suits well to the case of micro-paths in MGEs as long as the thickness T of the plate is replaced by the length t of the micro-path and the ratio h/l between the width h and the depth l of the micro-grooves is less than 0.1 or larger than 10 and. An approximate formulation of the inner part has been proposed in [9] when h/l is between 0.1 and 10.

An external part: the axial dimension of the air column vibrating in the micro-paths is not equal to the micro-path length t (see Figure 1), plus the so called reactive end correction, depending on the cross-section of the micro-path and the surrounding geometries. The expression of the reactive end correction suggested in [9] for MGEs is the following:

$$\chi_{MGP-e} = \delta_{MGE-c} \omega / (\sigma c) \quad (3)$$

with

$$\delta_{MGE-e} = \delta_c \left(1 + \frac{1}{4} \right) \quad (4)$$

where $\delta_c = 8d/3\pi$ is the Ingard's expression for reactive end correction suitable for a small circular duct [14], with $d = 2\sqrt{A_s/\pi}$ being A_s the cross sectional area of the micro-path. The equation (4) fits to MGEs when the width L of the slits is in the order of 1-4mm, width w of the slots around 1mm, and $0.1 < h/L < 10$ (see Figure 1) [9]. In the present work it has been experimentally noticed that, for wider slits ($L > 4\text{mm}$), a better representation of the reactance can be obtained by using the expression proposed by Maa for the rectangular shaped slits [9] instead of the equations (3) and (4). A rectangular slit is equivalent to an ellipse by introducing the ellipticity factor:

$$\bar{e} = \sqrt{1 - \frac{h}{L}}, \quad (5)$$

where h and L are the width and length of the slit respectively. Once defined the function

$$F(\bar{e}) = \int_0^{\pi/2} \frac{d\theta}{(1 - \bar{e}^2 \cos^2 \theta)^{3/2}} \quad (6)$$

usually simplified as a series of terms, the reactive impedance for a panel with slit shaped aperture is then written as:

$$\chi_{MGP-e} = \varepsilon \omega \rho h F \quad (7)$$

where the coefficient ε is equal to 0.5 according to Maa's studies, and is approximately equal to 1.5 in case of MGE with wide slits ($L > 4\text{mm}$) and width of the slots $w \sim 1-2\text{ mm}$. Accounting for the end correction is necessary for both the real and imaginary parts of the transfer impedance. Ingard [14] estimated the resistive end correction of a circular duct as a function of the power dissipated by an oscillating motion of a fluid on a plane surface as:

$$R_s = \frac{1}{2} \sqrt{2\nu\rho\omega} \quad (8)$$

The end resistive correction for circular duct proposed by Ingard is $4R_s$ [14]. As a consequence, the generic expression for normalized external resistance for circular apertures becomes:

$$r_{MPE-e} = \frac{4R_s}{\sigma\rho c} \quad (9)$$

It should be noted that this expression has been found to fit well for MGEs.

A non-linear part: when the sound pressure level of the sound source is increased above a certain threshold, which mainly depends on the perforation geometry, the resistance increases and the reactance decreases, resulting in a non-linear (quadratic) relationship between the acoustic pressure amplitude and the particle velocity. The normalized non-linear term proposed by Ingard [16] can be expressed as:

$$r_{non-lin} = \frac{|u_h|}{\sigma c} \quad (10)$$

In the (10) $|u_h|$ is the absolute value or the peak particle velocity inside the apertures. The resistance is frequency independent. For the resistive part, this nonlinear term is added as an external term, whereas for the reactance Maa [3] suggested an empirical formula, which is multiplied with the end correction term:

$$\chi_{non-lin} = \chi_{MGP-e} \left(1 + \frac{|u_h|}{\sigma c} \right)^{-1} \quad (11)$$

The results above are valid for pure tone excitation and are suitable for the implementations to MGEs.

For broad-band excitation, the following non-linear term is proposed in [9]:

$$r_{non-lin-R} = \beta \frac{\tilde{u}}{\sigma c} \quad (12)$$

where the subscript R denotes the random noise excitation, $\tilde{u}$ is the total *rms* pressure value inside the apertures and the factor $\beta=0.1$ is a constant determined experimentally.

Grazing flow contribution: when a grazing flow is present (i.e. tangential flow over the perforated surface), the resistive part increases with the flow velocity, and the reactive part slightly decreases. In [22] Ingard proposed the following expression for the normalized grazing flow resistance:

$$r_{Graz} = \frac{K_g M_g}{\sigma} \quad (13)$$

where M_g is the Mach number of the grazing flow, and K_g is a coefficient equal to 0.15. Different values of K_g can be found in literature. In the present work it has been noticed that, for the MGEs tested, K_g is a function of M_g , varying from 0.15 to 0.90 in the conditions treated.

The reactive contribution due to the grazing flow is expressed in [17] as a decreasing coefficient for the external part contribution of the transfer reactance. As a consequence,

$$\chi_{MGP-Graz} = \chi_{MGP-e} F_\delta \quad (14)$$

with

$$F_\delta = \frac{F_{corr}}{(1 + (12.6 M_g)^3)} \quad (15)$$

Where $F_{corr}=1$ according to the theory summarized in [17]. In conclusion, the following expressions can be used for modeling the acoustic impedance of MGEs:

$$r_{MGE} = \text{Re} \left(\frac{j\omega t}{\sigma c} \left[1 - \frac{\tanh(k\sqrt{j})}{k\sqrt{j}} \right]^{-1} \right) + \frac{4R_s}{\sigma\rho c} + 0.1 \frac{\tilde{u}_{tot}}{\sigma c} + \frac{K_g M_g}{\sigma} \quad (16)$$

$$\chi_{MGE} = \text{Im} \left(\frac{j\omega t}{\sigma c} \left[1 - \frac{\tanh(k\sqrt{j})}{k\sqrt{j}} \right]^{-1} \right) + \frac{F_{corr}}{(1 + (12.6 M_g)^3)} \chi_{MGP-e} \left(1 + \frac{|u_h|}{\sigma c} \right)^{-1} \quad (17)$$

EXPERIMENTAL METHOD

A number of planar MPEs composed of different internal geometries and one cylindrical MPE have been tested by using the classical two port method [16] and a modified version of the impedance eduction technique [17]. Experiments have been performed in the acoustic laboratory of Tallinn University of Technology (TUT), which is equipped with a dedicated test-rig, described in detail in [18]. In Figure 2, a simplified schematic layout of the measurement setup is shown.

All tests performed with planar elements have been carried out in the absence of mean flow. The tubular element is exposed to mean flow. Two pressure transducer separations are used during the acoustic pressure measurements: $s_1=280\text{mm}$ and $s_2=35\text{mm}$. Considering the range $0.1\pi < k_w s_i < 0.8\pi$ (where k_w is the wavenumber and $i=1,2$) suggested in [19] for the two port experiments, the frequency region reckoned during the measurements varies from 60Hz to 3.9kHz at 20°C.