

Micro-Grooved Elements - A Novel Solution for Noise Control

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ABSTRACT

The goal of this paper is to present a novel type of advanced acoustic material - micro grooved element (MGE) - which is designed for noise control in a wide range of applications. MGEs have been proved to offer a respectable alternative for the existing micro-perforated elements (MPEs), while being cost effective and causing low pressure loss. These elements have been found to be suitable for substitution of fibrous materials, typically present in silencer units.

Currently, the cost of the MPEs is relatively high due to the technological complexity of manufacturing process. On the other hand, cheaper solutions of MPEs, based on irregularly shaped micro-apertures, potentially cause higher pressure loss due to surface roughness.

The key concept of the MGEs is the use of micro-grooves forming acoustic channels, instead of the micro-holes of MPEs, which the sound wave has to pass. This allows to replace the laser perforation process, used to manufacture the MPEs with circular cross section, with less time consuming and more cost effective alternatives.

The acoustical performance of the MGEs has been modeled by adapting the theoretical models provided by Allard and Maa for rectangular ducts, circular duct and non-linearities.

The transfer impedance and the absorption coefficient of a number of MGEs and of several types of MPEs have been experimentally measured and compared by using the classical two-port method. Additionally the non-linear behavior of such elements has been experimentally investigated by varying the excitation sound pressure level during the tests.

CITATION: Auriemma, F., Rammal, H. and Lavrentjev, J., "Micro-Grooved Elements - A Novel Solution for Noise Control," *SAE Int. J. Mater. Manf.* 6(3):2013, doi:10.4271/2013-01-1941.

INTRODUCTION

The problem of sound propagation in a narrow, uniform, circular duct has been first studied in the late 1897 by Reyleigh [1], and the approximation for short tubes has been provided in 1926 by Crandall [2]. Nevertheless, the idea of micro-perforated elements (MPEs) was first introduced in '60s by Maa [3,4,5,6]. 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 (~ 400 Pa 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 hole radius [6].

As a matter of facts, common perforated panels, having larger than millimeter apertures, have been used for noise control since the beginning of the previous century. Since the acoustic resistance provided by such apertures is usually not enough to achieve the desired sound absorption, porous

materials have often been utilized in combination with common perforates.

Even today, most of the porous materials, like fibers and foams, are not renewable and their performance deteriorates over time. Disintegrating and being blown out of the silencer casings by the passing pressure waves, the porous materials can also radiate small particles that pollute the surrounding media [7]. As the described issues are not tolerable in a number of silencer applications, e.g. in aero engine inlets, novel alternative solutions are of interest.

Environmental issues have recently redirected the interest of researchers towards absorptive elements where porous components are excluded. In this scenario, one of the most attractive solution is represented by the MPEs, which have a simple structure and well predictable absorption characteristics. The panels may be made of wide range of materials, e.g. cardboard, plastic, plywood and sheet metal. The panels can be planar (noise cancellation walls and barriers), cylindrical (integrated with exhaust and inlet system tubes) or formed to any desired shape. The MPEs are often utilized in room acoustics [8], but recently also in automotive and aerospace field. Lately, Allam proposed a new type of

straight flow muffler based on these elements [9], and Kabral developed micro-perforated elements for motorcycle applications [10-11].

In addition to the micro-circular holes, also micro-rectangular slits can be utilized in such elements [12, 13]. In both cases a laser beam treatment is usually used to achieve the perforations. Since a number of micro-apertures are required to provide an adequate porosity, the manufacturing process is typically expensive and time consuming. Accordingly, the traditional MPEs with circular holes are often considered too expensive for commercial applications [13].

However, lower-cost MP absorbers are available [14] being manufactured by punching a metal sheet with a properly shaped matrix. This process produces small irregular slits into the sheet, which tend to generate viscous losses when being exposed to flowing fluid. However, due to the punching, the plate surface presents a surface roughness which potentially can cause undesirable pressure loss when a grazing flow is present. Moreover, due to the irregular geometry, the acoustic performance of such kind of panels is less predictable during the design phase [13,15].

In order to improve the performance of MPEs, Randerberg presented a study on the effects of horn shaped micro-holes [16]. The inner part of each horn has dimensions approximately equivalent to traditional MPEs, but the proposed shape increases the surface area of the opening and offers a better impedance match to the incoming wave. As a consequence, the micro-horn absorber has a potential as an enhanced MPE. Obviously the manufacturing process is even more complex and, at the moment, the horn shaped elements are rarely utilized.

In [16] Randerberg also proposed a double-perforated panel, consisting of two parallel, perforated plates separated by a small distance, typically 0.1-0.3mm. The perforations are circular (up to 1mm diameter) and additionally a slit shaped version has been proposed. The main part of the energy dissipation takes place in the small gap between the plates, which constitutes the main conceptual difference with the traditional MPEs. Therefore, a strict control of the plate spacing is required. The measured absorption coefficients exhibited slightly better performance compared to the traditional MPEs. Nevertheless, the practical use of this panel is limited by the high dependence on the plate distance affecting the acoustic performance of the element.

In this paper, the micro-grooved elements (MGEs) are presented. Similarly to the other micro-perforated materials, the MGEs are typically recyclable, noncombustible and environmentally friendly acoustic materials. In addition, this type of absorber is conceptually more cost effective compared to most of the MPE alternatives. Moreover the thickness and the weight of the elements can be easily controlled, as well as the roughness of the external surfaces, resulting in potentially lower pressure losses.

The MGEs are patented acoustic solutions [17] in which the energy dissipation of the acoustic waves is primarily originated by viscous friction that occurs inside a number of micro-paths.

Similarly to the panels presented by Randerberg and described above, the MGEs are also multi-layered elements. However, the main structural difference is that there is no need for accurate (sub-millimeter) fine tuning and preserving of the distance between the plates, since the surfaces of layers are simply mated to each other.

In this paper, several configurations of MGEs have been tested by using the well-known 2-port method [18-19] and the results have been compared to the typical MPEs. Also a mathematical model, suitable for predicting the performance of MGEs, has been developed, by adapting the existing models (Allard [12], Maa [3]) for micro-slits and micro circular apertures for the micro path in the MGEs.

DEVELOPMENT OF MICRO-GROOVED ELEMENTS

A MGE (see Figure 1a) is composed of at least two overlying layers (see Figure 1b and 1c) presenting macroscopic slots and a number of micro-grooves at least on one of the contact surfaces (see Figure 1b) [17]. When the two constituting layers are mated to form the MGE, the slots manufactured at one of the layer are not aligned with the ones present at the other one. In this way the sound waves enter the element via the slots of the first layer - inlet - (see Figure 2). Consequently, they are forced to travel through the micro paths inside the MGE. 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 energy dissipation is primarily expected to take place. Eventually the sound waves get out the element via the slots present at the second later - outlet -.

Technically, the novelty of the MGEs is the replacement of the micro-holes or the micro-slits of traditional MPEs with the micro-grooves which are easier to manufacture, especially in case of sub-millimeter dimensions. Moreover the macro-slots presented on the element do not require any special technology for production.

Although such elements require at least two mated layers, the materials can be very thin, since the slots give only a small contribution to the dissipation. In this way an extra weight can be saved, if needed, and the external surfaces of the panel become smoother.

In order to keep the two constituting layers coupled, several technological methods can be implemented, for example by welding, riveting or clamping.

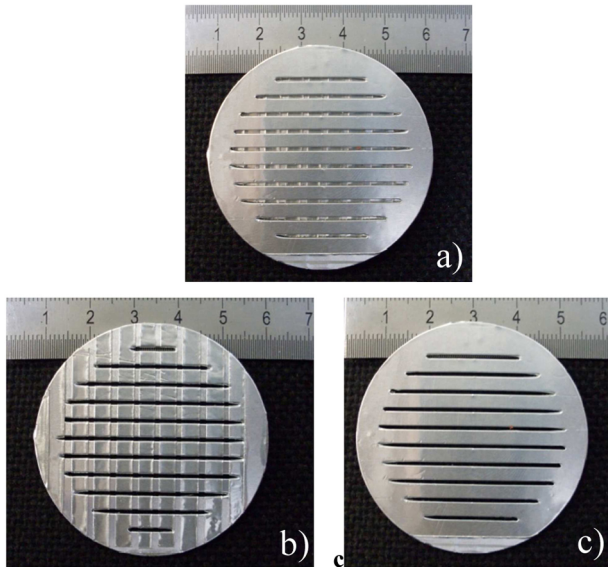


Figure 1. Specimen of micro-grooved element (MGE). a) complete element; b) outlet layer; c) inlet layer.

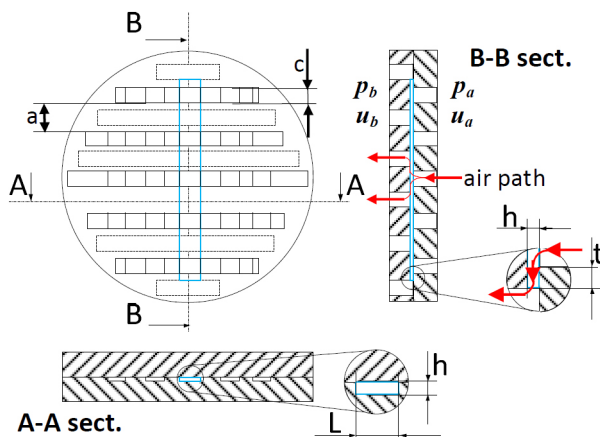


Figure 2. Scheme representing the functioning and the key dimensions of a MGE.

EXPERIMENTS

Two different samples of MPEs and a number of different MGEs have been tested by using the classical two port method [18,19]. Experiments have been performed in the acoustic laboratory of Tallinn University of Technology (TTU), which is equipped with a dedicated test-rig, described in details in [20-21]. In Figure 3, a simplified schematic layout of the measurement setup is depicted.

All tests performed have been carried out in the absence of 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 [22] for the two port experiments, the frequency region reckoned during the measurements varies from 60Hz to 3.9kHz at 20°C.

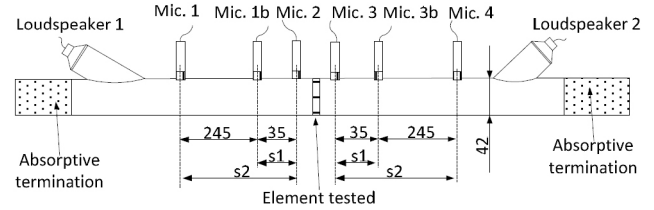


Figure 3. Schematization of the measurement setup.

Determination of Transfer Impedance and Absorption Coefficient

The transfer impedance of each sample and the absorption coefficients of the same samples, coupled with a 25mm back cavity, have been carried out during the experiments. The transfer impedance, also known as separation impedance [13], is believably the most suitable parameter for the characterization of the performance of permeable materials.

Let p_a and p_b represent the respective sound pressures upstream and downstream of the acoustic element and u_a and u_b the respective particle velocities (see Figure 2). 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. $u = u_a = u_b$. This implies that the panel itself can be considered as an in line lumped element [18] whose circuit analogy can be represented as in Figure 4.

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.

The normalized transfer impedance,

$$z_{tr} = r + j\chi \quad (2)$$

Is equal to $Z_{tr}/(\rho c)$, where ρ is the air density and c is the speed of sound.

In general it can be said 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-groove of the considered element), then the transfer impedance of the entire element is $Z_{tr} = Z_{tr-s} / \sigma$, where σ is the panel porosity.

A number of methods for the determination of the transfer impedance can be found in literature, and are summarized in [13].

In this work, the 2 port data of the samples have been experimentally determined in order to carry out the associated transfer impedance.

In particular the transfer matrix **T** of samples [18], has been utilized. It relates the acoustic pressures p_1 and p_2 with

the volume velocities q_a and q_b (where $q_a = u_a * A$, $q_b = u_b * A$ and A is the sample area) according to the following relationship:

$$\begin{pmatrix} p_b \\ q_b \end{pmatrix} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{pmatrix} p_a \\ q_a \end{pmatrix} \quad (3)$$

It can be easily shown that, under the assumption of constant particle velocity across the sample, $T_{11}=1$, $T_{21}=0$, $T_{22}=1$ and $T_{12}=(p_1-p_2)/q$, where $q=q_a=q_b$. By multiplying T_{12} by the sample area A , the transfer impedance is computed.

Once the transfer impedance is known, the absorption coefficient (α) of the considered element, can be easily carried out [12].

In particular, when a thin acoustic element is coupled with a back cavity, the transfer impedance of the panel and the cavity impedance (Z_{cav}) form a series impedance (Z_{tot}):

$$Z_{tot} = Z_{tr} + Z_{cav} \quad (4)$$

The cavity impedance can be expressed as [12]:

$$Z_{cav} = -j \cot(w_l l), \quad (5)$$

where w_l is the wavelength and l is the cavity length.

As a consequence, once the reflection coefficient (R) has been computed, for example by using the equation

$$R = \frac{Z_{tr} - 1}{Z_{tr} + 1}, \quad (6)$$

the absorption coefficient of the coupled system can be obtained:

$$\alpha = 1 - |R|^2 \quad (7)$$

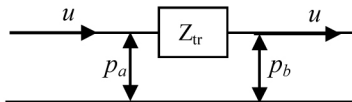


Figure 4. Equivalent acoustic circuits of a thin acoustic element.

Determination of the Particle Velocity

The 2-port data have additionally been used to carry out the amplitude of the particle velocity u inside the panel. In [15] the procedure is described in details and only briefly summarized here.

Once the transfer function $H_i = p_i/e$ between the acoustic pressure p_i measured by the microphone i and the loudspeaker electric signal e is known, it is possible to

compute the transfer function $H_{ev} = u_d/e$ of the particle velocity at the panel inlet cross-section by using the classical wave decomposition technique [19]. Subsequently, the amplitude of the particle velocity can be obtained via the relation

$$|u_a| = \left| \frac{u_a}{e} \right| \cdot |e| = |H_{ev}| \cdot |\sqrt{S_{ee}}|, \quad (8)$$

where S_{ee} is the auto spectrum of the voltage.

Thereafter, the obtained value must be divided by the microphone calibration factor K [V/Pa] for the level correction. As a consequence, the expression for the particle velocity becomes:

$$|u_a| = \sqrt{2} \frac{|H_{ev}| \sqrt{S_{ee}}}{K} \quad (9)$$

Hereby, the *rms* particle velocity is:

$$\tilde{u}_a = \frac{|H_{ev}| \sqrt{S_{ee}}}{K}. \quad (10)$$

It should be noted here that when performing the determination of impedance using pure tone excitation, the peak particle velocity should be used, whereas with random noise the *total rms* value is more suitable:

$$\tilde{u}_{a-tot} = \frac{\sqrt{\sum |H_{ev}|^2 S_{ee}}}{K} \quad (11)$$

Now the particle velocity inside the holes, slits or grooves can be calculated as follow:

$$\tilde{u} = \tilde{u}_{a-tot} / \sigma \quad (12)$$

MODELING

Regarding the acoustic modeling of such acoustic panels, one of the main advantages of the MPEs is the predictability of the acoustic performance by the implementation of existing mathematical models. In this paper an attempt has been made to adapt the models developed for MPEs by Maa and Allard for the prediction of MGPs performance. In the following sections the modeling of the panels has been treated by dividing the process into internal and external part.

Internal Part

Maa considered the MPEs as a lattice of short narrow tubes, separated by distances much larger than their diameters though small compared to the wavelength of the impinging sound wave [3]. His formulation is based on Crandall's