NUMERICAL STUDIES OF THE ACOUSTIC IMPEDANCE OF MICRO-PERFORATED PANELS UNDER GRAZING FLOW

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Micro-perforated panels (MPPs) have been widely used for broadband noise absorption in various engineering applications. When a MPP is exposed to a grazing flow, literature shows that existing acoustic impedance formulae based on different flow parameters give very different results. Therefore, there is a need for investigating the issue and finding more relevant parameters for the reliable acoustic impedance predictions. The issue is technically challenging because of the flow complexity near the hole of the panel. In the present work, 3D URANS CFD simulations are carried out to investigate the acoustic behaviour of a MPP hole with a backing space under grazing flow. Based on one-cell simulations and numerical experiments, the porosity of the panel is then introduced, leading to a new acoustic resistance formula. The proposed formula uses the velocity gradient in the viscous sublayer over the duct wall as the new parameter, and is found to be applicable at a Mach number up to roughly 0.25, within a certain Reynolds number range and under the linear acoustic excitation regime. The accuracy and the improvement of the model are demonstrated through comparisons with some existing ones as well as with the experimental data reported in the literature.

Keywords: Micro-perforated panels, acoustic impedance prediction, grazing flow

1. Introduction

Micro-perforated Panels (MPPs), recognized as the next generation sound absorption material, have been widely used in various noise control applications. In many cases, the MPPs are exposed to a grazing flow. Literature [1] shows that the existing acoustic impedance prediction models, derived from theoretical [2-6] or semi-theoretical models, [7, 8] as well as empirical models [9-18] give different prediction results. Therefore, there is still no universally accepted model that can be used for the acoustic impedance prediction of MPPs with a grazing flow.

This paper revisits this important issue, with an aim at establishing a suitable impedance prediction model for Micro-perforated panels with fully developed turbulent grazing flow under linear acoustic excitation regime, when the acoustic impedance of MPPs is independent of sound pressure level. Computational methods are used to achieve this goal.

2. Computational methods

2.1 Model built-up

Figure 1 shows the three dimensional computational model built according to the geometry used in reference [1]. The model was built based on the assumptions that there are no interactions between the holes of the Micro-perforated panels. The computational model includes a square duct with the
height of 24mm, inside which a Helmholtz resonator unit, composed of a MPP hole and its backing cavity is flush-mounted along the duct wall. Figure 1b gives the detailed geometrical information.

![Flow](image)

(a) 3D view

![Normal wall](image)

(b) 2D view

Figure 1: Geometry of the computational model.

Table 1 lists the MPPs being investigated here. Panel 1 is the same one which was used in reference [1].

<table>
<thead>
<tr>
<th>Number</th>
<th>Orifice diameter d (mm)</th>
<th>Panel thickness t (mm)</th>
<th>t/d</th>
<th>Perforated ratio δ</th>
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<tr>
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<td>1.7</td>
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<tr>
<td>3</td>
<td>0.3</td>
<td>0.3</td>
<td>1</td>
<td>1.39%</td>
</tr>
</tbody>
</table>

2.2 Computational setup

Fluent is used as the computational tool. The computational method is 3D URANS. The boundary conditions for all wall surfaces are solid wall with no slip. The inlet surface of the channel uses mass flow rate boundary condition. For the outlet surface of the channel, the pressure release boundary condition is used, by setting it to be one atmospheric pressure.

Both the stable mean flow field and the aero-acoustic coupling simulation use the pressure-velocity coupling scheme PISO (Pressure-Implicit with Splitting of Operators). The second order scheme is chosen for both the spatial discretization and time integration. $k-\varepsilon$ turbulence model is used as the
turbulent model. The flow medium is air with the density satisfying the ideal gas law. The time step sizes for the stable mean flow simulation and the aero-acoustic coupling simulation are chosen to be 5e-6 s and 5e-7 s respectively, verified to satisfy the required calculation accuracy. Extensive verifications and validation of the simulation methods using basic configurations were carried out to ascertain the correctness of the modelling (not shown here for brevity).

3. Acoustic impedance prediction and CFD model validation

The normalized acoustic impedance of an MPP hole is defined as:

\[ Z_{\text{hole}} = \frac{1}{\rho C} \frac{p_{\text{in}} - p_{\text{out}}}{u}. \]  

(1)

where \( p_{\text{in}} \) and \( p_{\text{out}} \) are the space-averaged acoustic pressure at the inlet and outlet surface of the MPP hole, respectively; and \( \bar{u} \) is the space-average acoustic velocity normal to the hole section.

The normalized acoustic impedance of the MPP can then be obtained through the porosity of the panel \( \delta \) as:

\[ Z_{\text{MPP}} = \frac{Z_{\text{hole}}}{\delta}. \]  

(2)

Figure 2 compares the acoustic impedance of the MPP obtained by CFD predictions and experimental measurements in reference [1]. It can be seen that the CFD simulation results agree well with the experimental data, especially for the acoustic resistance.

![Figure 2: Comparison of the normalized acoustic impedance of panel 1 between CFD and experiments.](image)

4. Impedance prediction model of MPPs with grazing turbulent flow

4.1 Relationship between the velocity gradient in the viscous sublayer over the duct wall and the acoustic impedance of MPPs

Consider the case of a linear shear flow passing over a plane wall with a circular hole, generating the Stokes flow near the hole. The distributions of the velocity and pressure near the hole is shown to be determined by the velocity gradient of the linear shear flow [19]. It is straightforward to surmise
that the velocity gradient in the viscous sublayer influences the acoustic impedance of Micro-perforated panels with grazing turbulent flow.

Numerical experiments are carried out to find the relationship between the velocity gradient in the viscous sublayer over duct wall, denoted by $G$, and the acoustic impedance of MPPs under a grazing turbulent flow. Focus is put only on the acoustic resistance.

The normalized acoustic resistance of MPPs under grazing turbulent flow, $R_{\text{flow}}$, is assumed to take the following form:

$$R_{\text{flow}} = R_m + \theta.$$  \hspace{1cm} (3)

where $R_m$ stands for the acoustic resistance in the hole under no flow condition; and $\theta$ a correction terms to be determined.

Both $\theta$ and $G$ are non-dimensionalized in the form $\theta C_{fd}$ and $G_{t} \delta_{fd}$ respectively to collapse the CFD calculation data for acoustic resistance of the MPPs. Here, $f$ is the acoustic excitation frequency; $C$ the speed of the sound; $t$ and $d$ the thickness and diameter of the hole respectively.

Figure 3 shows the CFD calculation data for acoustic resistance of the panel 1. It can be seen from the figure that a straight line fits the computational data very well. Therefore the relationship between $\theta C_{fd}$ and $G_{t} \delta_{fd}$ is found to be reasonably linear, and the acoustic resistance of MPPs under grazing flow is believed to be related to the velocity gradient in the viscous sublayer region. The same linear relationship can also be found from the computational data of panel 2 and 3 (not given here).

![Figure 3: Relationship between $\theta$ and $G$ for panel 1. $|V_a| = 0.025 m/s$](image)

### 4.2 Resistance model of the MPPs with grazing turbulent flow

Curve-fitting the calculated resistance data, the expression used to calculate the normalized acoustic resistance of the MPPs under grazing turbulent flow is proposed as follows:

$$R_{\text{flow}} = R_m + \frac{0.0139}{\delta} \left[ 2.558 \left( \frac{t}{d} \right)^{-3.236} + 1.129 \right] \frac{G_{t} \delta}{C} \left[ \frac{98.49 - 167.7 \left( \frac{t}{d} \right)^{-2.195}}{C} \right] \frac{f_{d}}{C}. \hspace{1cm} (4)$$
where $R_m = \frac{32 \mu t}{d} \sqrt{1 + \frac{K^2}{32}}$, $K = d \sqrt{4 \nu}$, $t$ is the panel thickness, $d$ is the diameter of the hole, $\delta$ is the panel porosity, $f$ is the frequency of the acoustic excitation, $\omega$ is the angular frequency, $\mu$ and $\nu$ are the dynamic and kinematic viscosity of the air respectively, $C$ is the speed of sound. The velocity gradient in the viscous sublayer over the duct wall $G$ can be calculated from the following equations:

$$ G = \frac{U^2}{\nu}, \quad (5) $$

$$ U_r = \frac{\tau_w}{\rho}. \quad (6) $$

$$ \tau_w = \frac{\rho U^2 \lambda}{8}. \quad (7) $$

$$ \lambda = \frac{0.178}{Re^{1/3}}. \quad (8) $$

where $U_r$ is the friction velocity, $\tau_w$ is the wall shear stress, $U$ is the free-stream velocity, $\lambda$ is the Darcy friction factor, $\rho$ is the air density, $\nu$ is the kinematic viscosity of the air, the Reynolds number $Re = \frac{hU}{\nu}$, with $h$ being the height of the square duct.

Figure 4 shows a comparison of the normalized acoustic resistance between the results predicted by the proposed model and the experimental data [1] for panel 1. The comparisons clearly show that the normalized acoustic resistance predicted by the proposed expression not only capture the trend but also agree well with the experimental data.

![Normalized acoustic resistance comparison](image)

Figure 4: Normalized acoustic resistance comparisons between proposed model and experimental data for panel 1. $f = 3.15 KHz$, $|V_o| = 0.025 m/s$.

Figure 5 compares the acoustic resistance of the panel 1 predicted by the proposed model with those given by Kirby and Cummings’ model [16] and Allam and Abom’s model [13]. Compared with Kirby and Cummings’ model [16], the present model seems to work better, especially at low
March number range before 0.1. Allam and Abom’s model [13] fails to capture the nonlinearly increasing trend of the acoustic resistance. The model proposed in this work seems to follow the experimental results quite well within the entire range.

$$\frac{\partial}{\partial x} (\rho u^2 v) = -\frac{1}{\rho} \frac{\partial p}{\partial x}$$

5. Conclusions

In the present work, the acoustic behaviours of micro-perforated panels with fully developed grazing turbulent flow are investigated through numerical simulations.

A new formula is proposed to predict the acoustic resistance of MPPs with grazing turbulent flow, applicable at a Mach number up to roughly 0.25, within a certain Reynolds number range and under the linear acoustic excitation regime. The model uses the velocity gradient in the viscous sublayer over the duct wall as the new flow parameter. The accuracy of the model as compared with the existing ones are demonstrated through comparisons with the information provided in the open literature. It is shown that the present model agrees well with the experimental data and outperforms other existing models in terms of both the prediction accuracy and application range.

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