Damping Effect of Perforated Plates on the Acoustics of Annular combustors

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This paper aims at showing the influence of perforated plates on the acoustic modes in aeronautical gas turbines combustion chambers. The analytical model developed by Howe was implemented in a 3D acoustic Helmholtz solver to account for the effect of perforated plates. First, an analytic test case is used to validate the coding in the acoustic solver. Then, a computation of the acoustic modes in an actual industrial chamber is conducted, taking into account the perforated liners. Both longitudinal and azimuthal modes are studied.

Nomenclature

\( \bar{A}, \bar{B} \) NxN matrices
\( \bar{P} \) Acoustic pressure amplitude N vector
\( \bar{n} \) Unit normal vector
\( \delta \Omega \) Boundary of the computational domain
\( \gamma \) Polytropic coefficient
\( \hat{p} \) Spatial pressure fluctuation
\( \hat{u} \) Spatial velocity fluctuation
\( \Omega \) Computational domain
\( \omega \) Angular frequency \( \text{rad/s} \)
\( \rho \) Density \( \text{kg/m}^3 \)
\( a \) Aperture radius m
\( c \) Sound speed m/s
\( d \) Aperture distance m
\( f \) Mode frequency Hz
\( K_R \) Rayleigh conductivity 1/m
\( N \) Number of discretization nodes
\( St \) Strouhal number \( \frac{f a}{U} \)
\( t \) Time

\( U \) Bias flow speed m/s
\( U_{md} \) Speed of maximal damping m/s
\( U_{ref} \) Actual speed used in the industrial chamber m/s
\( Z \) Complex impedance

Subscripts

0 Time averaged quantity

Conventions

\( \nabla z \) Spatial gradient of \( z \)
i Imaginary complex number
\( \text{Im}(z) \) Imaginary part of the complex value \( z \)
\( \text{Re}(z) \) Real part of the complex value \( z \)

Superscripts

+ value upstream
- value downstream
' Fluctuating quantity
I. Introduction

In order to cut down pollutant emissions, industrial gas turbine combustion chambers are run on lean premixed regimes which on the other hand make them more subject to combustion instabilities. These instabilities result from an unfavorable coupling between the combustion process and the acoustic modes of the chamber. They might cause harmful damages to the combustion chamber and therefore need to be accurately predicted. Numerical simulations have proven to be efficient to predict combustion instabilities. However, it has recently been shown that even small changes in the geometry accounted for in simulations may have an impact on the acoustic modes of the combustion chambers, and thus on the prediction of combustion instabilities.

In aeronautical gas turbines, walls of recent combustion chambers are generally perforated with sub-millimeter holes for cooling purposes. It allows fresh air flowing in the casing (outside the combustion chamber) to enter the combustion chamber due to the pressure difference between the two sides of the walls. The jets issuing from the holes coalesce to form a protecting cooling film that isolates the wall from the hot gases flowing in the chamber. This technique is called full-coverage film cooling (FCFC).

In addition to modifying the dynamical and the thermal characteristics of the flow near the walls, FCFC may have an impact on the acoustics of the chamber. Indeed, perforated plates are known to have a damping effect on incident acoustic waves. The damping effect is generally enhanced by the presence of a bias flow through the plate, as encountered in FCFC. Acoustic waves force the shear layer between the jet and the main flow to break down and form vortex rings. This mechanism efficiently converts part of the acoustic energy into vortical energy. Vortices are in turn dissipated by viscosity without substantial sound production.

To perform predictive simulations of combustion instabilities, one should then either account for the presence of multiperforated walls in the chamber, or show that their effect can be neglected. To the authors’ knowledge, no study in the literature has properly addressed this question. Indeed, accounting for FCFC in simulations is a difficult task: due to the tiny diameter of the apertures, meshing the holes in numerical computations is clearly impossible.

It is thus necessary to use models to represent the macroscopic effect of perforated walls. This issue is encountered in computational fluid dynamics calculations, but also when resolving the Helmholtz equation to determine the acoustic modes of a chamber. This is the focus of the present paper. Howe proposed a model to represent the acoustic damping of perforated plates with bias flow. Howe’s model provides the acoustic impedance of a perforated plate and is therefore well adapted to Helmholtz solvers. To account for the plate thickness, which is neglected in Howe’s model, Jing and Sun proposed a modified version of Howe’s model. However, the use of the modified model does not provide significant difference on the configurations studied here. Efforts have also been made to account for the presence of the grazing flow, but no analytical model has yet been developed and thus can’t be implemented numerically.

In the present study, we will show how Howe’s model can be integrated in a Helmholtz solver to account for the presence of perforated walls in aeronautical combustion chambers. It will be applied to a helicopter industrial chamber. In section II.A, the Helmholtz solver AVSP is described. Howe’s model will be recalled in section II.B. Section III is dedicated to the analytic validation of the implementation of the model in an academic test case consisting in two coaxial cylinders separated by a perforated plate. A study of the influence of the parameters is also conducted in this section. Finally, computations are performed in a real industrial chamber, considering longitudinal modes in section IV.A and azimuthal modes in section IV.B.

II. Howe’s model in the Helmholtz solver AVSP

II.A. Description of the Helmholtz solver

Under the hypothesis of a non viscous, low Mach number mean flow and linear acoustics (which means the fluctuating quantities are small compared to the mean quantities), the wave equation can be written in a non reactive media as follows:

\[
\frac{1}{\gamma \rho_0} \frac{\partial^2 p'}{\partial t^2} - \nabla \left( \frac{1}{\rho_0} \nabla p' \right) = 0 \tag{1}
\]

Harmonic variations are supposed for \( p' \) and \( u' \), such that \( u' = Re(\hat{u}(x)e^{-i\omega t}) \) and \( p' = Re(\hat{p}(x)e^{-i\omega t}) \).
AVSP resolves the wave equation in the frequency domain, which amounts to resolving the system (Eq. (2)): the Helmholtz equation and the corresponding boundary condition.

\[
\begin{align*}
\nabla c^2 \nabla \hat{p} + \omega^2 \hat{p} &= 0 \text{ on } \Omega \\
cZ \nabla \hat{p} \cdot n - i\omega \hat{p} &= 0 \text{ on } \partial\Omega,
\end{align*}
\]

where \( Z \) is the complex impedance on the boundary of the domain.

After a decomposition in finite volumes, the problem takes the following matricial form:

\[
\bar{A} \bar{P} + \omega \bar{B}(\omega) \bar{P} + \omega^2 \bar{P} = 0,
\]

where \( \bar{A} \) is the matrix corresponding to the operator \( \nabla c^2 \nabla \hat{p} \), \( \bar{B} \) is the matrix corresponding to the boundary conditions and \( \bar{P} \) is the pressure vector. AVSP determines the eigenelements of Eq. (3) and so releases the field of harmonic acoustic pressure and the angular frequency.

II.B. Description of the model and numerical implementation

Let us consider an array of circular apertures of radius \( a \), with an aperture spacing \( d \), through which a mean flow parallel to the apertures axis goes at the speed \( U \). Under the excitation of an acoustic wave, Howe’s model can represent the reaction of this flow under the following hypotheses:

1. The acoustic excitation is at a low frequency, so that the wavelength is much larger than the orifices radius.
2. The flow has a high Reynolds number, the viscosity is then only dominant at the rims of the aperture leading to the shedding of vorticity.
3. The Mach number of the mean flow is low, so that the flow is incompressible in the vicinity of the aperture.
4. The plate is infinitely thin.
5. The aperture spacing is high compared to the aperture radius, so that the interaction between the apertures is negligible.

The reasoning is then done for an isolated hole.

The Rayleigh conductivity \( K_R \) of the aperture, relating the harmonic volumic flux \( \hat{Q} \) to the acoustic pressure jump across the plate, is defined by

\[
K_R = \frac{i\omega \rho \hat{Q}}{\hat{p}^+ - \hat{p}^-},
\]

where \( \rho \) is the mean density in the vicinity of the aperture, \( \omega \) is the pulsation of the acoustic perturbation and \( \hat{p}^+ \) and \( \hat{p}^- \) are the harmonic pressures upstream and downstream of the aperture respectively. We have

\[
\hat{Q} = a^2 \hat{u}^\pm,
\]

Figure 1. Array of circular apertures, of diameter \( 2a \) and aperture spacing \( d \), with a bias flow of speed \( U \).
where \( \hat{u}^\pm \) is the acoustic velocity on the plate, equal on both sides. Hence,

\[
K_R = \frac{i\omega \rho_0 d^2 \hat{u}^\pm}{\hat{p}^+ - \hat{p}^-}.
\]

(6)

Howe expresses the Rayleigh conductivity for a circular aperture in an infinitely thin plate as:

\[
K_R = 2a(\Gamma_R - i\Delta_R),
\]

(7)

where

\[
\Gamma_R - i\Delta_R = 1 + \frac{\pi}{2} I_1(St)e^{-St} - iK_1(St)sinh(St)\
St\left(\frac{\pi}{2} I_1(St)e^{-St} + iK_1(St)cosh(St)\right),
\]

with \( St = \frac{\omega a}{U} \). Using the momentum equation and Eq. (6), we obtain:

\[
\nabla \hat{p}.n = \frac{K_R d^2}{dz^2}[\hat{p}^+ - \hat{p}^-]
\]

(9)

Eq. (9) can then be used as a Neumann boundary condition in the Helmholtz solver. The normal pressure gradient on each multiperforated boundary is expressed as a function of \( a, d, U \) and \( \omega \). The dependence in \( \omega \) of \( Z \) implies that the matrix \( \bar{B} \) in Eq. (3) is a non linear function of the frequency. The non-linearity of the problem is handled by an iterative method. Each sub iteration considers that the matrix \( \bar{B} \) is constant and resolves a quadratic eigenvalue problem.\(^{17}\)

### III. Analytic validation of the coding

First, an academic test configuration, for which an analytic solution can be derived, is presented. We consider the geometry depicted in Fig. 2. It consists of two coaxial cylinders, the inner one being perforated. The outer radius is \( r_2 \). The perforated plate is located at \( r_1 \). \( r^-_1 \) and \( r^+_1 \) denote the upstream part of the plate and the downstream part respectively. Although the cylinder is 3D, the third dimension is considered small in regard to the others and so we will only consider radial and azimuthal modes, the longitudinal ones occurring at much higher frequencies.

### III.A. Determination of the analytical solution

![Figure 2. Academic configuration: cylinder of radius \( r_2 \), with a perforated plate at \( r_1 \).](image)

For this simple case, Eq. (2) takes the following form, \( \Omega \) denoting the interior of the domain and \( \delta \Omega \) the domain boundary.

\[
\begin{cases}
\Delta \hat{p} + k^2 \hat{p} = 0 \text{ on } \Omega \\
\nabla \hat{p}.n = 0 \text{ on } \delta \Omega.
\end{cases}
\]

(10)

Considering \( k^2 = k_r^2 + k_z^2 \), Eq. (10) can be cast in polar coordinates (Eq. (11)), with \( \hat{p} = R(\rho)\Theta(\theta)Z(z) \).\(^{18,20}\)

\[
\frac{1}{R} \frac{d^2}{d^2r} R + \frac{1}{rR} \frac{d}{dr} R + \frac{1}{r^2\Theta} \frac{d^2}{d^2\theta} \Theta + \frac{1}{Z} \frac{d^2}{dz} Z + (k_r^2 + k_z^2) = 0.
\]

(11)
Since we only consider the radial and azimuthal modes, we have

\[
\frac{1}{R} \frac{d^2}{dr^2} R + \frac{1}{R} \frac{d}{dr} R + \frac{1}{r^2} \left( \frac{1}{\Theta} \frac{d^2}{d\theta^2} \Theta + n_\theta^2 \right) + k_r^2 - \frac{n_\theta^2}{r^2} = 0. \tag{12}
\]

Under these conditions, the radial part of Eq. (11) can be reduced to a Bessel equation:

\[
\left( \frac{d^2}{dr^2} R + \frac{1}{r} \frac{d}{dr} R \right) + R \times \left( k_r^2 - \frac{n_\theta^2}{r^2} \right) = 0, \tag{13}
\]

whose general solution is of the form:

\[ R(r) = AJ_{n_\theta}(k_r r) + BN_{n_\theta}(k_r r), \tag{14} \]

where \( J_{n_\theta} \) and \( N_{n_\theta} \) are Bessel functions of the \( n_\theta \) order. In the domain \( r \leq r_1 \), the pressure can be written:

\[ R(r) = AJ_{n_\theta}(k_r r). \tag{15} \]

The Neumann function, which is singular in \( r = 0 \), is put aside. In the domain \( r_1^+ \leq r \leq r_2 \), solutions may be written:

\[ R(r) = CJ_{n_\theta}(k_r r) + DN_{n_\theta}(k_r r). \tag{16} \]

A null acoustic speed is imposed on the outer cylinder. Applying the condition \( \hat{u} = 0 \) in \( r = r_2 \), we obtain:

\[ CJ_{n_\theta}(k_r r_2) + DN_{n_\theta}(k_r r_2) = 0. \tag{17} \]

According to Eq. (6), jump conditions can also be written across the perforated plate:

\[ \hat{p}(r = r_1^+) - \hat{p}(r = r_1^-) = \frac{i \omega \rho d^2}{K_R} \hat{u}(r = r_1^-), \tag{18} \]

\[ \hat{p}(r = r_2^+) - \hat{p}(r = r_2^-) = -\frac{i \omega \rho d^2}{K_R} \hat{u}(r = r_2^-). \tag{19} \]

We then obtain the system

\[ [M][X] = 0, \]

where \( M \) is the matrix obtained by using Eq. (17), Eq. (18) and Eq. (19), and given by

\[
\begin{bmatrix}
0 & J'_{n_\theta}(k_r r_2) & N'_{n_\theta}(k_r r_2) \\
\frac{d^2}{K_R} k_r J_{n_\theta}(k_r r_2) + J_{n_\theta}(k_r r_1^-) & -J_{n_\theta}(k_r(r_1^+)) & N_{n_\theta}(k_r(r_1^+)) \\
J_{n_\theta}(k_r r_1^-) & -J_{n_\theta}(k_r(r_1^+)) + \frac{d^2 k_r}{K_R} J_{n_\theta}(k_r(r_1^+)) & -N_{n_\theta}(k_r(r_1^+)) + \frac{d^2 k_r}{K_R} N_{n_\theta}(k_r r_2)
\end{bmatrix},
\]

and \( X \) is the vector

\[
\begin{bmatrix}
A \\
B \\
C
\end{bmatrix}.
\]

Solving

\[ \det(M) = 0 \tag{20} \]

releases the eigenvalues of the configuration.

III.B. Results

Let us consider the following parameters for the multiperforated plate (MP): \( U = 5 \text{ m/s}, a = 3 \text{ mm} \) and \( d = 35 \text{ mm} \), \( r_1 = 0.2 \text{ m} \) and \( r_2 = 0.25 \text{ m} \). The configuration calculated in the Helmholtz solver AVSP\(^{17}\) contains a tetrahedric mesh of 1186 nodes shown on Fig. 2. The sound speed is uniform and equal to 347 m/s. The eigenfrequencies are gathered in Table 1 and compared to the analytical results obtained by solving Eq. (20). The first three modes are considered. AVSP provides a complex frequency \( \text{Re}(f) + i\text{Im}(f) \) where \( \text{Re}(f) \) is the frequency of the mode, and \( \text{Im}(f) \) is related to the amplification rate of the mode. With
the adopted convention \( p'(x,t) = Re(\hat{p}(x)e^{-i\omega t}) \), the ratio between the value of the pressure fluctuation at \( t = 0 \) and \( t = T \) is given by Eq. (21).

\[
\frac{|p'(x, t = T)|}{|p'(x, t = 0)|} = |e^{Im(\omega)T}|. \tag{21}
\]

Hence, the attenuation factor \( A \) in percent at a period \( T \) is given by \( 100[1 - |e^{2\pi Im(f)}|] \).

<table>
<thead>
<tr>
<th>AVSP results with MP</th>
<th>Analytics with MP</th>
</tr>
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<tbody>
<tr>
<td>( f ) ( \text{Re}(f) ) ( \text{Im}(f) ) ( \text{A}% ) ( \text{Re}(f) ) ( \text{Im}(f) ) ( \text{A}% )</td>
<td></td>
</tr>
<tr>
<td>( n_\theta = 1 ) 382.5 Hz -18.8 s(^{-1}) (26.7 %) 382.56 Hz -18.9 s(^{-1}) (27.8 %)</td>
<td></td>
</tr>
<tr>
<td>( n_\theta = 0 ) 534.1 Hz -97.5 s(^{-1}) (68.2 %) 533.21 Hz -97.5 s(^{-1}) (68.3 %)</td>
<td></td>
</tr>
<tr>
<td>( n_\theta = 2 ) 610.48 Hz -21.4 s(^{-1}) (19.8 %) 611.04 Hz -21.64 s(^{-1}) (19.9 %)</td>
<td></td>
</tr>
</tbody>
</table>

Table 1. Comparison of the eigenfrequencies between AVSP and analytics.

As expected, the eigenfrequencies have a negative imaginary part, which means the pressure fluctuation is damped. AVSP results are in good agreement with the theory. Figures 3 and 4 show the radial profiles of the real and imaginary parts of \( \hat{p} \) given by AVSP for the first and second modes. Analytical solutions are also given for comparison. Again, a good agreement is found. Note that the pressure jump across the perforated plate is visible on the real and imaginary parts of the harmonic pressure fluctuations, as well as on the isolines for the first azimuthal mode (Fig. 3).

Figure 3. First azimuthal mode: \( n_\theta = 1 \), \( \text{Re}(f) = 382.5 \text{ Hz} \), \( \text{Im}(f) = -18.8 \text{ s}^{-1} \). \( \text{Left: Re(\hat{p})}, \text{Right: Im(\hat{p})} \). Above: AVSP results. Below: Radial profiles of the real and imaginary part of \( \hat{p} \) along the radius represented by the arrow (comparison analytics/AVSP).

This comparison is conducted for a fixed bias flow speed (5 m/s), but Fig. 5 also shows good agreement between AVSP and analytical results of eigenfrequencies for various bias flow speeds.

Figure 6 shows the evolution of the resonant frequencies for the radial mode and the first azimuthal mode as a function of the bias flow speed. As the bias flow speed increases, the resonant frequency of both modes decreases. But in the case of the azimuthal mode, the resonant frequencies varies in a range from 380Hz to 260Hz, whereas the frequency of the radial mode varies a lot, it drops from 500Hz to 80Hz. The frequency of the radial mode hence becomes close to the one of the azimuthal mode and then is lower than it. Figure 7 displays the amplification rate of the radial and the first azimuthal mode as a function of the bias flow speed. In the case of the radial mode, the damping raises until reaching almost 100% and keeping this
Figure 4. Radial mode: $n_\theta = 0$, $\text{Re}(f) = 534.1$ Hz, $\text{Im}(f) = -97.5$ s$^{-1}$. Left: $\text{Re}(\hat{p})$. Right: $\text{Im}(\hat{p})$. Above: AVSP results. Below: Radial profiles of the real and imaginary part of $\hat{p}$ (comparison Analytics/AVSP).

Figure 5. Comparison between AVSP frequencies and analytics, for various speeds.

very high value of damping. On the contrary, the damping for the first azimuthal mode reaches a maximum at $U = 20$ m/s and then decreases.

The impact of the perforated plates is more important on radial modes than on azimuthal modes. Indeed, in this configuration, the pressure gradient for the radial mode across the perforated plate is more important than for an azimuthal mode. This can be seen on the structure of the modes on Fig. 4. For the radial mode, the perforated plate is located between a pressure node and a pressure antinode, which makes the pressure gradient important, and thus, according to Eq. (9), the perforated plate has a strong impact.
IV. Computations on an industrial helicopter chamber

Now that the coding in the Helmholtz solver has been validated on a simple test case, the purpose of this section is to run computations in an actual industrial chamber.

IV.A. Longitudinal modes

The combustion chamber studied here is part of an engine designed to equip a 6-ton helicopter, with a power of 900 kW at take off. It is built around a gas generator and the cold flow compressed by the two compressor stages enters the combustor chamber through the casing. A part of the air gets into the combustion chamber through the perforated plates in order to cool the walls. The chamber is a reverse-flow annular chamber fueled by 15 injectors. In this first section, the computational domain is limited here to a 24 degree sector. The computational domain includes the swirler, the casing and the chamber. The influence of the geometry on the acoustics of this chamber was previously studied and many computational fluid dynamics calculations were conducted but always neglecting the presence of perforated plates. The purpose of this section is to investigate the influence of the perforated liners on the acoustic modes. The locations of the MP are given in Fig. 8. The sound speed field is deduced from a previous LES computation (Fig. 9).

To assess the impact of MP in this chamber, we use a reference test case when walls are used in place of

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*a*Large Eddy Simulation.
the plates. Table 2 gives the first two eigenfrequencies for the computation with walls and MP. Once again, the modes are found damped, but only weakly for the first longitudinal mode \((L_1)\).

Figure 10 shows the spatial structure of the pressure fluctuation within the chamber. The pressure gradient across the perforated plates is more important for the second longitudinal mode \((L_2)\) than for \(L_1\). This explains why the damping is better for \(L_2\), as pointed to in section III.B. The dependency of the modes to the bias flow speed is now considered to see if the damping can be optimized. Figures 11 and 12 respectively show the evolution of the frequencies in the complex plan. The reference speed noted \(U_{\text{ref}}\) corresponds to the regime at which the chamber is actually run. Figures 11 and 12 respectively show the evolution of the frequencies in the complex plan as a function of \(U\), \(U\) ranging from 0 to \(U_{\text{ref}}\) for \(L_1\) and 0.07 \(U_{\text{ref}}\) for \(L_2\) to \(U_{\text{ref}}\).

Both modes have a similar behaviour. The resonant frequency increases when decreasing the bias flow speed, which was also observed on the cylinder. For both modes, a maximum damping speed \(U_{\text{md}}\) can be found. The value of this speed is \(U_{\text{md}}(L_1) = 0.1 U_{\text{ref}}\) for \(L_1\) and \(U_{\text{md}}(L_2) = 0.2 U_{\text{ref}}\) for \(L_2\). Beyond this speed, the damping of both modes decreases. The fact that \(U_{\text{ref}}\) is closer to \(U_{\text{md}}(L_2)\) than \(U_{\text{md}}(L_1)\) explains why the damping is more important for \(L_2\). The maximum damping reached by \(L_2\) is much more important (50%) than the one reached by \(L_1\) (26%).

<table>
<thead>
<tr>
<th></th>
<th>Walls</th>
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<th>MP</th>
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<tbody>
<tr>
<td></td>
<td>(Re(f))</td>
<td>(Im(f)) ((%)))</td>
<td>(Re(f))</td>
<td>(Im(f)) ((%)))</td>
</tr>
<tr>
<td></td>
<td>510 Hz</td>
<td>0 s(^{-1}) (0%)</td>
<td>511.4 Hz</td>
<td>-4.5 s(^{-1}) (5.4%)</td>
</tr>
<tr>
<td></td>
<td>1153.7 Hz</td>
<td>0 s(^{-1}) (0%)</td>
<td>1163.7 Hz</td>
<td>-48.9 s(^{-1}) (23.2%)</td>
</tr>
</tbody>
</table>

Table 2. Comparison of the first two eigenfrequencies corresponding to the first two longitudinal modes, with walls, and with MP.
IV.B. Azimuthal mode

The full annular chamber is now considered to search for azimuthal modes. The full annular mesh contains 466485 nodes and was obtained by duplicating the mesh of the sector fourteen times. The initial field of sound speed is constructed by duplicating the one obtained from a LES sector computation. The frequencies of the longitudinal modes released by the full annular computation are the same as the frequencies obtained with a one-sector run. The second azimuthal mode is here studied. Figure 13 shows the spatial structure of $Re(\hat{p})$ from both sides of the chamber, for the actual regime $U_{\text{ref}}$. Figure 14 shows the evolution of the frequency in the complex plan, $U$ ranging from $U_{\text{ref}}$ to $U = 0.01 \ U_{\text{ref}}$. The plot has the same tendency as for the first two longitudinal modes: as $U$ decreases, the resonant frequency increases. The imaginary part of the frequency reaches a maximum at $U_{md} = 0.2 \ U_{\text{ref}}$, which is the same speed as $U_{md}(L_2)$. The damping reached at this speed is maximum and has a value of 16%. Afterwards, the damping decreases.
V. Conclusion

A model for the acoustic behaviour of perforated plates was implemented in a Helmholtz solver. This tool allows to take into account MP in computations of the acoustics of a chamber. A damping is predicted by the code, as expected with perforated plates, but this damping strongly depends on the parameters of the MP and also on the location of the plate. If the plate happens to be between a pressure antinode and node, the fact that the acoustic pressure gradient is important makes the damping stronger.

MP are designed to cool walls of combustion chambers and are hence not designed to have the best acoustic damping. However, a mode can be strongly damped if its structure has an important pressure gradient across the plate. If not the case, the parameters of the plate can be changed, in order to improve the damping. Indeed, there exists a speed for which the damping is maximum. It was found that with the geometrical parameters of the perforated plates of the helicopter chamber considered here, and the flow rate injected into the perforations, the speed appears to be far above the maximum absorption speed in the case of the first longitudinal mode. The tool developed here can be used to make optimization of modes damping, by helping choosing the best location in the first place, and then to find the best parameters which optimize the damping.

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References


