STUDY ON THE ACOUSTIC RESONATORS OF SLANTED PERFORATION WITH GRAZING FLOW

Wenwen Tian, Lin Du, Xiaoyu Wang, Dakun Sun, Xiaodong Jing and Xiaofeng Sun
Beihang University, Fluid and Acoustic Engineering Laboratory, Beijing, China
email: lindu@buaa.edu.cn

Abstract
Previous studies have shown that flow has a great influence on the acoustic impedance of liner. Those studies are largely based on sharp edged perforations with an axis normal to the wall. But for slanted-perforation resonator, which has an axis at different angles with respect to the direction of the grazing flow, has not receive much attention in the literature. Therefore, the influence of flows on the acoustic response of the slanted-perforation resonator is a problem of interest. This paper will solve the linearized Navier Stokes equations to analyze this problem. Besides, in order to investigate the influence of the opening direction on the acoustic characteristics of the system, two kinds of slit openings are presented in this paper. It is clearly show that the grazing flow has a significant impact on the impedance of such a slanted-perforation resonator.

Key words: sound absorption, slanted-perforation resonator, grazing flow

1. Introduction

In the study of duct acoustics, the sound absorption is one of the important topic. According to the difference of absorption principle, absorbing body can be divided into two categories which are the porous materials and the resonance sound absorption structure. Compared with porous materials, resonance structure can effectively put energy into heat energy consumption by the resonator characteristics. Here we consider larger perforated resonator, which consists of perforated plates backed with cavity structures. At present, as the liner can effectively reduce the noise, it is widely used in aviation and navigation. Previous studies have shown that grazing flow has a great influence on the acoustic impedance and makes a difference to the sound absorption performance of perforated plates. In engineering applications, perforation liner will inevitably be affected by the grazing flow. When the fluid flow through the surface of perforated plate, the impedance of perforation will change. In this case vortex shedding is the main reason caused dissipation [1]. Actually, Vortex Sound Theory [2,3] predicts that shed vortices do not only absorb sound, the acoustic filed would be amplified or attenuated when the vorticity interacts with the acoustic filed. This depends on the relative residence time of hydrodynamic disturbance over the perforated resonator to the acoustic period, which can be described using the energy formula proposed by Howe [4] at low Mach numbers. When the acoustic power \( P = -\rho_0 \int \langle (\vec{\omega} \times \vec{v}) \cdot \vec{u}' \rangle dV \) is positive, there is a net sound produced, where \( \rho_0 \) is the mean fluid density, \( \vec{\omega} = \nabla \times \vec{v} \) is the vorticity, \( \vec{v} \) is the impressible flow velocity, \( \vec{u}' \) is the acoustic particle velocity and the brackets \( \langle ... \rangle \) denotes time averaging. Using the above approximation, the occurrence of whistling can be explained very well.

In addition, the perforation geometry has a significant influence on the aeroacoustics response of wall perforations. The effect of a grazing flow on the impedance of wall perforations is experimentally investigated by means of a multi-microphone impedance tube setup. In order to obtain some
experimental data which is useful for optimizing liner, measurements are carried out for perforations with different shapes [5,6]. In acoustic liners for aircraft engines one often uses circular sharp edged perforations with an axis normal to the wall. Most of the existing literatures are mainly based on this type of perforation. But in combustion chamber liners (film cooling liners), wall perforations at 30° angle relative to the direction of the flow are often used. These liners are used with the dual purpose of providing acoustic damping and film cooling of the walls. However, the current study is not enough. It is interesting to explore the application of a wall perforation which is slanted with a grazing flow.

In the previous studies [7,8] the frequency-domain model is applied to study the acoustic damping performance of resonators. The paper [8] investigated the acoustic properties of generic resonators, and showed that the one single hole slice simplification of complete resonators can produce correct results for the acoustic filed and predict the transmission loss very well. Based on the work of Du et al. [8], we will use this simplified modeling to study complete resonator with slanted perforations in the form of Helmholtz-resonator.

2. Numerical method and configuration of interest

When the turbulent eddy viscosity is added to the linearized Navier-Stokes equations, a significant improvement can be achieved in the comparison between calculations and experimental data [9,10]. To study the interaction of flow and sound, the 3D linearized Navier-Stokes equations (LNSEs) are presented briefly in this section. The LNSEs in the frequency domain are defined as

\[ \text{i} \omega \rho' + \nabla \cdot \left( \mu_0 u_0 + \rho_1 u' \right) = M , \]

\[ \rho_0 \left( \text{i} \omega u' + (u' \cdot \nabla) u_0 + (u_0 \cdot \nabla) u' \right) - \nabla \sigma = F , \]

\[ \sigma = -p' I + \mu (\nabla u' + (\nabla u')^T) + (\mu_0 - \frac{2}{3} \mu_1) (\nabla \cdot u') I . \]

Where \( p' \), \( u' \) and \( \rho' \) are the fluctuating components of density, velocity and pressure. \( \rho_0 \), \( u_0 \) and \( p_0 \) are the mean flow components of density, velocity and pressure, which are obtained by solving an incompressible RANS along with a \( k-\varepsilon \) turbulence model in the present work. \( I \) is the unit matrix. Furthermore, the total viscosity \( \mu = \mu + \mu_t \) includes the dynamic viscosity \( \mu \) and eddy viscosity \( \mu_t \), and the second term is determined by the RANS solution. Similar to previous studies [10], the relation between pressure and density can be regarded as isentropic. This assumption is valid for subsonic gas flows at high Reynolds numbers. It then follows that the pressure and density perturbations are related as

\[ \frac{\partial p'}{\partial x} = c_0^2 \frac{\partial \rho'}{\partial x} , \frac{\partial p'}{\partial y} = c_0^2 \frac{\partial \rho'}{\partial y} . \]

Here \( c_0 \) is the local adiabatic speed of sound. The terms \( M \) and \( F \) represent acoustic sources, and the term \( \omega = 2\pi f \) is the angular frequency. Besides, the perturbation quantities are assumed to have a harmonic time dependence, so a frequency domain approach is hence adopted. The above set of equations are implemented in the commercial finite element method code. In order to investigate the difference of slanted-perforation resonator, here gives the schematic of Helmholtz-resonator and three perforation geometries, see Figure 1.
We define the first geometry of 30 degree to the downstream as downstream-30, the second geometry of no tilt as vertical-up, and the third one of 30 degree to the upstream as upstream-30. Assuming no circumferential mode exists in the frequency range of concern, the paper is performed on one hole slice for reducing the computational cost, as shown in Figure 2. The slice configuration is downstream-30 resonator in Fig.2.

The sound waves are excited by the source term $\mathbf{F}$ (force amplitude along duct axis) in the momentum equation, and the source location is at the duct in the region $-1m < x < -0.98m$. Relevant physical data for the studied resonators is summarized in Table 1. Only plane waves prevail in the frequency range 500-2500Hz outside the region of the resonators.

<table>
<thead>
<tr>
<th>Table 1: Geometrical details and flow conditions.</th>
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<tbody>
<tr>
<td>Diameter of Duct (D)</td>
</tr>
<tr>
<td>Number of holes</td>
</tr>
<tr>
<td>Hole diameter (d)</td>
</tr>
<tr>
<td>Height of holes (h)</td>
</tr>
<tr>
<td>Mach number (grazing)</td>
</tr>
<tr>
<td>Diameter of chamber</td>
</tr>
<tr>
<td>The length of computational domain</td>
</tr>
<tr>
<td>The length of upstream domain</td>
</tr>
</tbody>
</table>
3. Boundary conditions and mesh size

In the RANS simulations, the no slip boundary conditions are applied on the duct and resonator walls. The side faces of slice adopted the slip boundary conditions. For LNSEs simulations, adiabatic and slip conditions are applied at the duct walls. To avoid numerical errors caused by the reflections at the in- and out-flow boundaries, non-reflecting boundary conditions are applied.

The mesh used for flow calculations and acoustics simulation must ensure convergence. Before applying the numerical model to conduct parametric studies, mesh independence study is conducted. Figures 3 and 4 illustrate the comparison of turbulent dynamic viscosity and the comparison of velocity varied with the mesh size. It can be seen that the velocity is almost independent on the increase of the mesh size. Actually, this paper used a finer grid for all areas, which is not totally necessary for the whole computational area but can get a more accurate result. Here gives the RANS mesh properties used in this paper, see Table 2. For the LNSE converged results, it can be seen that the transmission loss is a constant when the minimum size varying from 0.01mm to 0.1mm from Figure 5. The Fig.6 shows the overview of the mesh around the upstream-30 resonator. In fact, the finer grid located around the resonator with 0.5-3mm, and coarse mesh used in regions of only plane wave propagation with 1-2.5mm.

![Figure 3: Comparison of turbulent dynamic viscosity varied with the mesh size](image1)

![Figure 4: Comparison of velocity in x direction varied with the mesh size](image2)
Table 2: Mesh properties for RNAS.

<table>
<thead>
<tr>
<th>Mesh</th>
<th>Number</th>
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<tbody>
<tr>
<td>Triangular elements</td>
<td>26980</td>
</tr>
<tr>
<td>Quadrilateral elements</td>
<td>78923</td>
</tr>
<tr>
<td>Total number of elements</td>
<td>105903</td>
</tr>
<tr>
<td>Total number of freedom degree</td>
<td>132186</td>
</tr>
<tr>
<td>Max element size</td>
<td>D/71 (D refers to the duct diameter)</td>
</tr>
<tr>
<td>Min element size</td>
<td>D/285 (D refers to the duct diameter)</td>
</tr>
<tr>
<td>Max element growth</td>
<td>1.3</td>
</tr>
</tbody>
</table>

Figure 5: The transmission loss varied with the mesh size.

Figure 6: Proportional overview of the mesh around the upstream-30 resonator.

It is well known that the flow structure in the orifice should be solved in order to capture the phenomenon of vortex sound interaction. In the present paper, three Mach numbers of grazing flow are investigated 0.11, 0.15 and 0.22. The three configurations of resonators subjected to inflow \( M_s = 0.11 \) are shown in Figures 7 and 8. It can be see that these three kinds of resonators have different flow structure under grazing flow. Based on the solutions of flow structures, the acoustic velocity could be solved by LNSE method. Figures 9 and 10 are the acoustic velocity contours when the mean flow velocity is at the case of \( M_s = 0.15 \).
Figure 7: Radial velocity (Mach number=0.11).

Figure 8: Axial velocity (Mach number =0.11).

Figure 9: Radial acoustic velocity (Mach number=0.15).

Figure 10: Axial acoustic velocity (Mach number =0.15).

4. Results

We first get the flow distribution based the RANS solution. Then based on the result of the base flow mentioned above, the transmission loss can be predicted by solving the LNSEs. The comparisons of transmission loss for the three different Mach numbers (0.11, 0.15, 0.22) and resonators (downstream-30, vertical-up, upstream-30) are be shown in Figure 11 and Figure 12. To study the influence of base flow on the sound field, the transmission loss $TL$ is calculated in absence of grazing flow as reference.

The effect of the mean grazing flow Mach number on the acoustic damping performance of $TL$ is studied first. As shown in Figure 11, the peak of transmission loss is positive and tends to decrease with increasing Mach number for downstream-30 and vertical-up. While for upstream-30, It is interesting to note that the peak value of transmission loss of upstream-30 at $M_g = 0$ is positive.
while the other three are negative when there exists grazing flow, which indicates potentiality of whistling at the three Mach numbers \((M_g = 0.11, 0.15, 0.22)\).

As seen from the Figure 12 the transmission loss of upstream-30 is obviously different from the downstream-30 and vertical-up. For the upstream-30, a strong negative value of transmission loss appear at a certain point of frequency at \(M_g = 0.11\). But at the same Mach number 0.11, the results for the vertical-up and downstream-30 resonators are positive at the frequency from 500 Hz to 2500 Hz. And this phenomenon exists all the way at the other two Mach numbers. As mentioned above, some negative values appear at the \(M_g = 0.11, 0.15, 0.22\) for upstream-30. In this aspect, the upstream-30 resonator seems to be unstable on damping performance. Besides, From the Fig.12, the peak of downstream-30 is slightly higher than that of vertical-up at all three Mach numbers while the trend for the other two resonators are very much the same for the transmission change in value.

![Figure 11: Downstream transmission loss of resonators at different inflow Mach numbers.](image1)

![Figure 12: Downstream transmission loss of three different resonators.](image2)

5. Conclusion

In this paper, numerical study of acoustic damping performance of slanted-perforation resonator is conducted. To gain insight on the effects of the mean flow Mach number and the geometry of slanted-perforation resonator structure on the acoustic damping performance, parametric studies are conducted by using a 3D frequency-domain model. The transmission loss \(TL\) is be used to characterize the sound damping effect. For the downstream slanted- and vertical-up resonators, the values of transmission loss are all positive at \(M_g = 0.11, 0.15, 0.22\). Actually, this result agreed with other studies [5,6] and can be explained by Vortex Sound Theory [2,3]. However, the value of transmission losses for upstream-30 resonator are negative at \(M_g = 0.11, 0.15, 0.22\). It is an interesting phenomenon, and vortex-sound interaction created at resonator openings is probably
responsible for it. In further study, we will focus on the analysis of power ratio and the details of flow structure, which would be of help to work on this issue.

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REFERENCE