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Noise control for a dipole sound source using micro-perforated panel housing integrated with a Herschel–Quincke tube

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Abstract

This study presents a passive noise control approach for directly suppressing sound radiation from an axial-flow fan, which involves micro-perforated panels (MPPs) backed by cavities and integrated with a hollow tube with the characteristics of a Herschel–Quincke (HQ) tube (MPPHQ housing). Noise suppression is mainly achieved by sound cancellation between the sound fields from the fan with a dipole nature and sound radiation from a vibrating panel via vibro-acoustic coupling. In addition, noise mitigation is supplemented by sound absorption in micro-perforations and acoustic interference at the hollow tube boundaries. A two-dimensional theoretical model is established to investigate the HQ segment housing a sound source. Results show that the HQ segment enhances the sound suppression performance in the passband region of the MPP housing device, and thus, a widened effective working band is obtained. Optimization is conducted to identify the optimal parameters for the MPPHQ housing. The proposed method has the potential to effectively control ducted-fan noise and to enhance the quality of products with a ducted-fan system.

1 Introduction

Noise control for an axial fan in a finite duct with a low aspect ratio remains a technical challenge that has many applications, which range from small fans in domestic products, driving fans in building ventilation systems, to large turbo-fans used in aircrafts. There are various types of passive and active noise suppression approaches for a long duct with a high aspect ratio. Porous materials have been widely used to line the duct wall and absorb noise in central air conditioning and ventilation systems [1, 2]. But the hygiene issue due to the accumulation of dust inside the porous materials has received more and more attentions; and the performance of duct lining is usually poor without sufficient length. Moreover, an expansion chamber [1] or multiple chambers with perforated tubes [3] have often been implemented in vehicle exhaust systems to abate noise at the desirable frequency range. However, the devices employed for controlling noise in the low frequency region are usually very bulky and significant pressure loss occurs in the flow duct due to sudden expansion and blockage of the flow by the internal tubes. Multiple chambers with micro-perforated elements as well as different alignments and configurations or multiple layers of micro-perforated panels (MPPs) have also been developed to improve the performance [4-9].

Both duct linings and expansion type mufflers mainly target the sound field that is away from the sound source. In other words, they attenuate the noise along the sound propagating path. In practice, due to space limitations or the finite duct length, the silencer is frequently installed close to the sound source such as a fan. Thus, alternative techniques have been proposed to allow another acoustic path for a dipole sound source to undergo sound cancellation, such as an enclosure with an opening [10] or an expansion chamber housing a fan [11]. Recently, a membrane housing device [12] was introduced for suppressing subsonic axial fan noise at the source position based on the interaction between membrane vibration and the sound fields in the duct and cavities, thereby achieving effective sound cancellation. This device is attractive but it is still bulky and special apparatus is required to apply tensile force on the membrane. To address these problems, the membrane can be replaced by an MPP to control the first two blade passing frequencies (BPFs) in the low and medium frequency ranges, thereby effectively achieving sound cancellation and sound absorption due to the MPP [13]. The device becomes an expansion chamber housing device with the covering MPP (hereinafter, MPP housing). However, when the chamber length is any odd multiple of half the wavelength, a passband exists where there is no noise reduction effect. The present study aims to develop a broadband passive noise control device with optimal design to deal with the sound radiation from an axial fan at the source position. In this device, a hollow tube with the characteristics of a Herschel–Quincke tube (HQ tube; see [14-16]) is integrated with the MPP housing (MPPHQ). The addition of a HQ tube to the MPP housing device is expected to widen the stopband but without significantly influencing the geometry of the overall device. Therefore, the objectives of the current study are: (1) to widen the stopband of the insertion loss by using an HQ tube integrated into the MPP housing device to control the dipole sound source; (2) to optimize the device by evaluating the bandwidth in the low and middle frequency ranges; (3) to establish a numerical model and provide a thorough understanding of the noise suppression mechanism obtained by integrating an HQ tube and MPP housing device to control the dipole sound source.

The remainder of this paper is organized as follows. Section 2 outlines the numerical model of the MPPHQ device and explains its noise suppression performance. Section 3 focuses on the analytical model of the HQ housing a dipole sound source at the centre under the assumption of plane wave propagation, as well as describing the noise suppression mechanisms. Section 4 presents the experimental validation of the theoretical model as well as the sound quality evaluation for the device. Main conclusions are given in Section 5.

2 Performance of MPPHQ controlling a dipole source

To fully describe the vibro-acoustic coupling among the acoustics field in duct, HQ tube backing cavities of MPP and panel vibration, the commercial finite element software COMSOL Multiphysics® was employed due to its strong multi-physics capability. Subsequently, the pressure and sound intensity field can be calculated and visualized.

2.1 Numerical model of a MPPHQ

The two-dimensional (2D) finite element model of the hybrid MPPHQ device for controlling a dipole source shown in Fig. 1 comprises four parts: the main duct, two backed cavities, MPP, and hollow tubes. The main duct has a middle segment for the MPP housing device with length L_c^*, two small segments for the hollow tube, and upstream and downstream sides with lengths of L_{up}^* and L_{dn}^*, respectively, and the same duct height of h^*. The cavity is rectangular in shape with height h_c^* and length L_c^* parallel to the duct, and the inner surfaces are covered by a perforated panel with two fixed ends. Immediately outside the cavity wall, a hollow tube with non-uniform cross section is employed with side width W^* and upper width H^*. Thus, the MPPHQ device is formed from the original MPP housing device [13] and the newly introduced hollow tube. A dipole source is located at the mid-point of the MPPHQ: $x^* = 0.5\ L_c^*$, $y^* = 0.5\ h^*$. It should be noted that it is difficult to install both the compact MPP housing device and the hollow tube on one surface of the duct. Therefore, in the experimental validation process based on a real three-dimensional (3D) situation, the hollow tube component was moved to the other two side surfaces of the square duct. In the numerical model, the MPP housing and HQ segment are placed on the same surface, and the 3D model can be simplified as a 2D model in order to reduce the use of computational resources and the simulation time. In addition, the predicted results can be validated based on the experiment in a convenient manner with fairly high accuracy. All of the variables considered are dimensionless values. The process of converting dimensional to dimensionless values is achieved as follows, where all of the dimensional variables are normalized based on three basic quantities comprising the air density ρ_0^*, speed of sound c_0^*, and duct height h^*:

$$\begin{gathered} H=\frac{H^*}{h^*},\ W=\frac{W^*}{h^*},\ h_c=\frac{h_c^*}{h^*},\ L=\frac{L^*}{h^*} \\ f=\frac{f^*h^*}{c_0^*},\ \omega=\frac{\omega^*h^*}{c_0^*},\ k_0=2\pi f,\ m=\frac{m^*}{\rho_0^*h^*} \quad , \\ B=\frac{B^*}{\rho_0^*\left(c_0^*\right)^2\left(h^*\right)^3},\ F=\frac{F^*}{h^*\rho_0^*\left(c_0^*\right)^2},\ p=\frac{p^*}{\rho_0^*\left(c_0^*\right)^2} \end{gathered} \tag{1}$$

where m is the plate-to-air mass ratio, B is the dimensionless bending stiffness of the plate, F is the concentrated force exerted on the fluid by the dipole source, and p is the sound pressure.

The acoustic field in the main duct with the dipole sound source is governed by the following equation.

$$(1-M^2)\frac{\partial^2 p}{\partial x^2}+\frac{\partial^2 p}{\partial y^2}-\frac{2M}{c_0}\frac{\partial^2 p}{\partial x\partial t}-\frac{1}{c_0^2}\frac{\partial^2 p}{\partial t^2}=\frac{\partial F}{\partial x} \tag{2}$$

The vibration of the MPP at $y = 0$ is governed by

$$m\frac{\partial^2\eta}{\partial t^2}+B\nabla^4\eta+\Delta p=0 \quad , \tag{3}$$

where B is the bending stiffness, m is the plate mass per unit area, η is the displacement of the plate, and Δp is the sound pressure difference across the interface at $y = 0$, $\Delta p = p|_{y=0+} - p|_{y=0-}$.

The effect of the MPP is implemented as

$$\bar{v} = \left(1 - \sigma \cdot imag(Z)/Z\right) i\omega\eta + \frac{p_{y=0-} - p_{y=0+}}{\rho_0 c_0 Z} \tag{4}$$

and

$$Z = \left(\frac{32\mu t}{\sigma\rho_0 c_0 d^2}\left[\left(1+\frac{K^2}{32}\right)^{1/2} + \frac{\sqrt{2}}{32}K\frac{d}{t}\right] + \frac{2\alpha R_s}{\sigma\rho_0 c_0} + \frac{|u_h|}{\sigma c_0} + \frac{0.15M}{\sigma} \right) + i\left(\frac{\omega t}{\sigma c_0}\left[1+\left(1+\frac{K^2}{32}\right)^{-1/2} + 0.85\frac{d}{t}\right] + \frac{0.85 d\omega F_\delta}{\sigma c_0 (1+|u_h|/\sigma c_0)} \right), \tag{5}$$

where $\bar{v}$ is the average velocity on the surface of the MPP, Z is the acoustic impedance of the MPP [17], v_0 is the particle velocity of the fluid in the holes, M is the Mach number, $K = d\sqrt{\omega\rho_0 / 4\mu}$, $R_s = 0.5\sqrt{2\mu\omega\rho_0}$, d is the hole diameter, t is the panel thickness, σ is the perforated ratio, μ is the coefficient of viscosity, α is a factor equal to 2 for holes with rounded edges, u_h is the absolute value or the peak particle velocity inside the holes, and $F_\delta = (1+(12.6M)^3)^{-1}$.

For frequencies below the first cut-on frequency of the duct, the upstream and downstream outlets of the duct with the radiation impedance are governed by

$$\frac{\partial p}{\partial n} + \frac{Z_{open}{}^{-1} ikp}{(1-M)} = 0 \quad and \quad \frac{\partial p}{\partial n} + \frac{Z_{open}{}^{-1} ikp}{(1+M)} = 0 \ . \tag{6}$$

where $\vec{n}$ is the local outward normal direction and Z_{open} is the so-called radiation impedance of the tube termination with a correction for flow effect. For the unflanged duct with equivalent radius a, the impedance at the opening can be set as follows [18, 19].

$$Z_{open} = \frac{(Z_0+1)+\sigma(Z_0-1)}{(Z_0+1)-\sigma(Z_0-1)},\ \sigma = \left(\frac{1-M}{1+M}\right)^{1.33}, Z_0 = \frac{1}{4}(ka)^2 + jk\delta_0, \delta_0 = 0.6133a \tag{7}$$

The other walls of the duct and cavity are assumed to be acoustically rigid.

The acoustic field can be computed inside the device and the sound power at the two ends of the duct can be obtained. The sound insertion loss (*IL*) can be expressed as:

$$IL = 10\log_{10}\frac{\int |p_t|^2\, ds\Big|_{original}}{\int |p_t|^2\, ds\Big|_{silenced}}\ , \tag{8}$$

where p_t is the sound pressure at the duct outlet and s is the cross-sectional area of the duct. The subscript 'original' denotes the straight duct condition without inserting the silencer and 'silenced' indicates the insertion of the silencer.

2.2 Optimization of the MPPHQ device

The performance of this device is characterized by the widest stopband that can be achieved for *IL*. The traditional pure axial fan without any accessories mainly generates noise in the first and second BPFs so the target is to reduce these two BPFs. The two stopbands are defined as $f \in [f_1, f_2]$ in the low frequency range and $f \in [f_3, f_4]$ in the medium frequency range, where *IL* is above a criterion level IL_{cr} = 5 dB, which is the maximum *IL* of the expansion chamber with an expansion ratio of 2. The hollow tube segments enhance the *IL* performance in the passband at around $f = 0.167$ for the MPP housing when the properties of MPP are d^* = 0.1 mm, t^*=1 mm, and $\sigma^* = 2.7\%$ [17]. Two parameters were varied in the hollow tube section, i.e. the side width *W* and upper width *H* of the hollow tube. For each parameter, a rough step size was used in the preliminary stage to obtain a general prediction of the performance. Further precise calculations were conducted until favourable *IL* performance was obtained by using a minor step size in a more accurate analysis. Finally, a mesh size of 0.1 was used for both the duct and hollow tube section, where a total of 3,348 triangular elements were used for the fluid domain. A mesh size of 0.01 was used for the MPP with 600 mesh elements.

The side width W of the hollow tube section was fixed as 0.1 and the upper width H varied from 0.1 to 0.25 in order to examine its effects on the *IL* capacity of the proposed model. The numerical results are presented in Fig. 2. Figs 2(a-d) show the *IL* spectra for H = 0.1, 0.15, 0.2, and 0.25, respectively. The closed circles in the dashed line denoted by f_1 and f_2 represent the lower limit and upper limit of the effective frequency band (within the range of $IL > IL_{cr}$) for the MPPHQ device, respectively. The *IL* performance of the MPPHQ device was better than that of the MPP housing device in terms of the stopband, which was in the frequency range of f = 0.15 to f = 0.2. There was a slight drop in *IL* in the medium frequency range of $f \in [0.37, 0.43]$, but the *IL* in this frequency range was still larger than IL_{cr}, and thus, the stopband was maintained. In addition, as the upper width H increased, the upper limit of the effective frequency f_2 shifted towards lower frequencies. The upper limit f_2 shifted from 0.23 to 0.18 when H increased from 0.1 to 0.25. Thus, to facilitate convenient applications in various types of ducted-fan products with different rotational speed and variable frequency BPF noise, the stopband (the effective frequency band) should be as wide as possible. Fig. 2 suggests that the upper width H of the hollow tube should be fairly small in order to meet the wide band requirement, which also confirms with the requirement for a silencer with compact geometry. However, the trough point at f = 0.15 was still slightly lower than IL_{cr} between the two peaks due to the effect of the chamber length (L_c = 3.4). In order to maintain the performance of the MPP housing device and improve the passband region, the criterion for the *IL* level was relaxed to 3 dB. Therefore, during the optimization process, the cost functions were defined as follows: 1) the IL output in the trough region should be around f = 0.15 larger than 3 dB; and 2) selecting the widest frequency ratio for the effective band $f_r = f_2/f_1$. Fig. 2(e) shows the contour of the ratio of the frequency limit f_2/f_1 as a function of the tube geometry with side width W and upper width H. The frequency ratio varied in the range from 2.5 to 7 when the side width W and upper width H varied from 0.1 to 1. The shaded region with the widest stopband (f_2/f_1 = 6.1) was obtained when the side width W and upper width H were about 0.15 and 0.1, respectively. Using this geometry, the sound suppressing capacity of the MPP housing device were maintained and the original passband was greatly improved while still keeping a compact geometry, where the total length and height of the side-branch were 3.5 and 0.7, respectively, with L = 3 and H_c = 0.5 in the MPP cavity part, and a wall thickness of 0.1. IL was improved for the silencing device by introducing a shallow tube. We compared the performance in terms of IL for the MPPHQ and the MPP housing device with equivalent geometry, i.e. L = 3.5 and H_c = 0.7. The top row in Fig. 3 shows the different geometry configurations, including the optimal MPP housing device with L = 3 and H_c = 0.5, the newly designed MPPHQ with L = 3.5 and H_c = 0.7, and the equivalent geometry MPP housing device with L = 3.5 and H_c = 0.7. The second row compares IL for these different devices. Both the MPPHQ and equivalent MPP housing device with a slightly longer MPP and cavity could contribute to good IL performance in the passband $f \in [0.15\ 0.25]$, which existed in the MPP housing device where L = 3 and H_c = 0.5. However, the decrease in IL due to the side-branch length for the equivalent geometry MPP housing device was worse than that for the MPPHQ at around f = 0.14. This phenomenon was more obvious at the medium frequency of f = 0.42, and thus, the MPP housing device with equivalent geometry yielded a very shallow effective medium frequency band $f \in [0.26, 0.37]$ and the MPPHQ contributed to a wider stopband [0.27 0.46]. This motivated us to design the MPPHQ because it was possible to improve the passband while retaining the optimal MPP housing device performance by slightly altering the geometrical size of the device.

3 Performance of the HQ tube

3.1 Theoretical model of HQ housing a dipole with plane wave approximation

In order to understand the enhancement of *IL* in the passband of the MPP housing device due to the addition of a shallow tube or narrow HQ tube, we consider the interaction between the propagating waves and radiating waves from the dipole source in the pure HQ tube housing. The geometry of a non-uniform HQ tube housing a sound source with the characteristic of a dipole at the centre is shown in Fig. 4, where the HQ-tube housing is formed by two side-branch HQ segments with side width W and upper width H, which are connected to the main duct with length L and cross-sectional area h. The dipole sound source is located at the centre and it radiates sound waves P_{s-} and P_{s+} to two sides of the main duct, where '$_-$' and '$_+$' denote the forward and backward directions, respectively. When these waves encounter the junctions denoted as (1) and (2), several pairs of travelling waves are formed, where the sound waves in the middle of the main duct are denoted as P_{m-} and P_{m+}, the sound waves in the upstream and downstream region of the duct are P_{u-}/P_{u+} and P_{d-}/P_{d+}, respectively, and the sound waves travelling in the two side-branches are P_{b1-} /P_{b1+} (upper branch) and P_{b2-}/P_{b2+} (lower branch). L_1 and L_2 are the duct length for the middle main house section and the centre-to-centre interface length of the non-uniform tube segment, respectively.

With a dipole source located at the centre, the waves radiating to the left and right sides of the device are

$$P_{s-} = \frac{F}{h} e^{-ikx}, \;\; P_{s+} = \frac{-F}{h} e^{ikx} \;, \tag{9}$$

where F is the concentrated force exerted on the fluid by the dipole source.

From the pressure continuity at junctions (1) and (2), we can obtain

$$P_{u-} + P_{u+} = P_{b1-} + P_{b1+} + P_{b2-} + P_{b2+} = P_{m-} + P_{m+} + \frac{-F}{h} \mathrm{e}^{-0.5ikL_1} \tag{10}$$

$$P_{d-} + P_{d+} = P_{b1-} e^{-ikL_2} + P_{b1+} e^{ikL_2} + P_{b2-} e^{-ikL_2} + P_{b2+} e^{ikL_2} = P_{m-} e^{-ikL_1} + P_{m+} e^{ikL_1} + \frac{F}{h} \mathrm{e}^{-0.5ikL_1} \;, \tag{11}$$

respectively. By considering the conservation of the volumetric flow at junctions (1) and (2) inside the tubes, we can obtain the following expressions.

$$h(P_{u+} - P_{u-}) = W(P_{b1+} - P_{b1-} + P_{b2+} - P_{b2-}) + h[P_{m+} - P_{m-} + \frac{-F}{h} \mathrm{e}^{-0.5ikL_1})] \tag{12}$$

$$h(P_{d-} - P_{d+}) = W(P_{b1-} e^{-ikL_2} - P_{b1+} e^{ikL_2} + P_{b2-} e^{-ikL_2} - P_{b2+} e^{ikL_2}) + h(P_{m-} e^{-ikL_1} - P_{m+} e^{ikL_1} + \frac{F}{h} \mathrm{e}^{-0.5ikL_1}) \tag{13}$$

The termination properties provide another two equations and subsequently all the the pressure components can be determined. With anechoic ends, there are no reflections from the two outlets, so $P_{u-} = 0$ and $P_{d+} = 0$. Therefore, Eqs. (9)–(13) can be solved and the two outgoing waves are as follows.

$$\begin{aligned} P_{u+} &= \frac{2F \cos(kL_1/2)\left(\cos kL_2 - 1\right)}{h\left(1 - \cos kL_2\right)\left(1 + \cos kL_1\right) + W \sin(kL_1)\sin(kL_2) + i\sin(kL_1)(1 - \cos kL_2)} \\ P_{d-} &= \frac{-2F \cos(kL_1/2)\left(\cos kL_2 - 1\right)}{h\left(1 - \cos kL_2\right)\left(1 + \cos kL_1\right) + W \sin(kL_1)\sin(kL_2) + i\sin(kL_1)(1 - \cos kL_2)} \end{aligned} \tag{14}$$

IL is obtained for the housing device based on the ratio of the acoustical power of the sound source in a straight duct and the acoustical power at the two ends of the silencing device with the same sound source:

$$IL = 10 * \log_{10}(\frac{W_0}{W_1}) \ . \tag{15}$$

W_0 is defined as the total sound power radiated by the source in a rigid, uniform duct with a cross-sectional area of *h.* W_1 is defined as the total sound power at the two ends of the device.

$$W_0 = \frac{h}{2}(|P_{s-}|^2 + |P_{s+}|^2), \quad W_1 = \frac{h}{2}(|P_{d-}|^2 + |P_{u+}|^2) \tag{16}$$

3.2 Analysis of an HQ tube housing a dipole sound source

The analytical model given above was established based on the plane wave approximation to help understand the physics related to the path difference and sound cancellation. For the HQ housing a dipole sound source at the centre, certain restrictions exist for the approximation method because it is not a pure plane wave case. Fig. 5 compares the performance in terms of *IL* for the analytical and FEM results without considering the MPP, pressure field, and intensity field distribution. The outer length (L_c) and height (H_c) of the HQ segment were fixed at 3 and 0.5, respectively. As shown in Fig. 5(1a), when both the side width *W* and upper width *H* of the tube were fairly small at 0.1, there was no visible difference between the two *IL* spectrum curves at frequencies below $f = 0.5$. When the tube side width *W* remained a constant but the upper width *H* increased to 0.3, as shown in Fig. 5(2a), the plane wave approximation method was also suitable. However, when the tube side width *W* increased to 0.5, the plane wave approximation method was only applicable below $f = 0.25$ for any tube heights, as shown in Figs 5(3a) and 5(4a). A possible explanation for this difference is that as the side width *W* increased, the inner tube walls were too close to the dipole sound source and intensive coupling occurred. This mechanism can be observed based on the pressure field and intensity field distribution, which are shown in the second and third rows, respectively. Fig. 5(1b) shows that the sound pressure distribution in the main duct and the HQ segment was of a plane wave type when *W*=0.1 and *H*=0.1. On the other hand, Fig. 5(2b) shows that the sound waves in the main duct near the entrance and exit of the HQ tube did not take the form of a plane wave due to the coupling between the sound source radiation and the nearby HQ openings when *W* is increased to 0.5. The sound intensity vectors ($\vec{I} = \frac{1}{2}\mathrm{Re}(p\vec{v}^{*})$) are also shown in the third row for the two different configurations where the 'normalized' vector fields were applied, which means that the lengths of the intensity vectors were drawn in the same in order to emphasize the directional nature of the intensity field. For the tube segment with a greater side width *W* and upper width *H*, strong coupling was found near the inner wall of the tube, as shown in Fig. 5(2c). Moreover, the peaks in IL spectrum can be explained by path difference method when the HQ segment had a narrow tube width and height as shown in Fig. 1(a). The dipole sound waves radiated to the left- and right-hand sides separately, and part of the sound energy that radiated to the left-hand-side went through the path of the HQ tube segment and then met the sound energy propagating to the right-hand side from the dipole source at the trailing edge of the HQ tube. When the path difference was a multiple of the wavelength such that $L_4 - L_3 = n\lambda$ (see Fig. 4) for any integer of *n*, *IL* was at the peak. However, when the side width *W* of the tube segment was equal to or larger than 0.3, the path difference was unable to explain the *IL* peaks. By contrast, we found that *IL* was also at the peak when $L_4 + L_3 = ((2n+1)/2)\lambda$ and $L_4 - L_3 \neq n\lambda$. The occurrence of the peak contributed by the HQ tube compensated for the deficiency in the passband of the MPP housing device.

3.3 Mean flow effect of MPPHQ housing

In real applications, the proposed device can be used to control fan noise, and thus, the aerodynamic effect should be considered. For simplicity, the current study considers the mean flow and uniform acoustic pressure excitation from the dipole source on the MPP. Fig. 6 compares the performance of the MPPHQ housing with and without the mean flow. In the presence of the mean flow, the levels of the second peak (f = 0.14) decreased slightly and smoothed out. The third peak at f = 0.21, which was mainly attributed to the HQ tube, shifted to slightly higher frequencies due to the shorter path difference in the case of the mean flow. The result was due to the convective effect on the sound speed towards the downstream direction accelerated the sound propagation, thereby implying lengthening of the travelling distance. The deceleration effect imposed towards the upstream direction shortened the travelling distance. The fourth peak decreased because of the increase in the acoustic resistance with the flow rate, which enhanced the damping on the system but weakened the sound cancellation capability.

4 Experimental study

Experimental evaluations were conducted to validate the numerical simulation and to examine the performance of the HQ and MPPHQ housing device. A 3D isometric view and photograph of the experimental test setup are shown in Figs 7(a) and 7(b), respectively. The main duct had a cross-sectional area of 100 mm × 100 mm and a length of 620 mm. Two sides of the main duct were installed with side-branch cavities with a length of 300 mm and height of 50 mm, which were covered with the MPP (MPP properties: B = 0.0032, m = 0.98, σ = 2.7, d^* = 0.5 mm, and t^* = 1 mm). Another pair of side-branch cavities (380 mm × 100 mm × 60 mm) inserted in a relatively small rigid block (320 mm × 100 mm × 50 mm) were installed with the hollow tube segments. There was a gap with a width of 30 mm between the rigid block and the outside rigid wall. The gap could be adjusted in the range from 5 mm to 30 mm as required. The duct and rigid block were made of 10-mm thick acrylic plate, which was considered to be acoustically rigid. The experiments were performed in an anechoic chamber with a cut-off frequency of 80 Hz. The background noise level in the anechoic chamber was about 20 dB, and thus, it could eliminate most of the noise from the outside environment that might affect the accuracy of the measurements. The experimental setup is illustrated in Fig. 7(c). A small loudspeaker with a diameter of 40 mm was utilized to simulate the dipole sound source. The loudspeaker was held as a cantilever by a rigid rod and it was located at the centre of the duct in the vertical direction. The radiated noise was measured at the outlet of the duct at an angle of 45 degrees from the centre line. Four 1/2-in. microphones (type B&K 4189) were used, which were supported by a Nexus conditioning amplifier (type B&K 2693). The loudspeaker was driven by a function generator (Hioki 7075) via an audio power amplifier (Crest LA 1201) with white noise. The signals from the microphones were acquired via an analogue-to-digital converter (NI 9234 & 9162) with MATLAB. Finally, the sound pressures were measured directly at the exits of the upstream and downstream ducts. In reality, the radiation from the loudspeaker was not a pure dipole because the reflected sound was also scattered by the loudspeaker surface and the junctions between the interface at the rigid wall and MPP. The sound radiation from the noise source was considered to comprise one dipole with an anti-phase relationship and one monopole with an in-phase relationship. In addition, the measurements at one point at the upstream outlet were contaminated by the sound pressure at the downstream outlet. Therefore, the dipole component was extracted from the real noise source during signal processing for the theoretical validation [13].

4.1 Experimental results obtained for the HQ and MPPHQ housing a dipole sound source

Fig. 8(a) compares the experimental and numerical results obtained for the HQ housing device for a speaker with dipole characteristics that generated white noise at all frequencies. It shows that the experimental IL spectrum (circle) matched the numerical predictions (dashed line) quite well, although there were some deviations in IL. Fig. 8(b) compares the IL spectra obtained from the experiment and the numerical prediction for the MPPHQ housing device, where they were generally in quite good agreement. According to the experimental results, the MPPHQ could only achieve 3 dB for IL instead of the predicted value of 5 dB in the frequency range from 500 Hz to 800 Hz, but the results showed that the MPPHQ housing device could reduce the sound radiation from the speaker in a practical manner and the function of the shallow tube was demonstrated effectively. In order to examine the feasibility of the MPPHQ housing device, it was compared with another possible housing silencing device for a dipole sound source. Fig. 8(c) compares the performance in terms of IL for the hybrid MPPHQ housing device, MPP housing device and expansion chamber housing a dipole source. Fig. 8(c) shows that the MPPHQ housing device (dash-dotted line with circle) generally performed better than the MPP housing device (solid line with circle) according to the experimental results. If only the MPP housing device was used, IL was negative or zero in the frequency range from 500 Hz to 800 Hz. After adding a shallow hollow tube segment, the performance in terms of IL was enhanced in the low frequency range by about 3 dB. In addition, there was an improvement in the medium frequency range from 1300 Hz to 1600 Hz. The lower limit frequency in the experiment was 150 Hz due to the limitations on the capacity of the small loudspeaker that acted as a dipole source, and the upper limit was 1700 Hz, which was the cut-on frequency of the duct.

4.2 Performance of the MPPHQ tube housing an axial fan

In order to examine the performance of the MPPHQ device housing an axial fan, a commercially-available fan was installed at the centre of the MPPHQ device. The fan has seven rotor blades and seven downstream struts, as shown in Fig. 9(a). The specified operational speed ranges from 2500 to 3950 rpm and the first BPF was 290 Hz to 460 Hz. A thin plastic membrane with a thickness of 0.01 mm and density of 920 kg/m^3 covers the opening to avoid bias flow. Fig. 9(b) compares the sound pressure level measured at the outlet of the MPPHQ housing device and MPP housing device. The experimental results showed that the MPPHQ housing device (dash-dotted line) generally performed better than the MPP housing device (dashed line). For the MPP housing device housing the axial fan, the noise reductions were about 11 dB and 10 dB at the first (470 Hz) and second (940 Hz) BPFs, respectively. For the MPPHQ housing device, the noise reductions were about 20 dB and 21 dB at the first and second BPFs, respectively. The noise reduction at the first BPF was similar for both devices but the MPPHQ housing could reduce the noise in the vicinity of all the peaks so that a wider bandwidth noise suppression can be achieved. Fig. 9(c) compares the sound pressure level (SPL) spectrum of MPPHQ with and without membrane covering HQ opening for axial fan noise measurement. The overall performance of the MPPHQ housing device with and without membrane remains almost the same especially the BPFs, although slight variation can be observed at the non-peak region of the SPL spectrum. Note that the measured flow speed is low in the experiment (Mach number: M=0.013). The potential whistling problem associated with the opening of the HQ tube was not observed during the experimental test in the present flow situation. A thin acoustically transparent perforated panel (e.g., perforation ratio=30%, orifice diameter =1~2 mm) may be used to cover the opening to avoid the whistling or shift its frequency to insensitive regions.

5 Conclusions

In this study, a hybrid passive noise control device, MPPHQ, is proposed for directly suppressing sound radiation from an axial-flow fan through vibro-acosutic coupling, micro-perforations absorption and HQ interference. To understand the noise suppression mechanism of the HQ tube, a 2D theoretical model of an HQ housing a dipole sound source was established using the approximate plane wave approach. The theoretical results were validated by both FEM simulations and experiments. The HQ segment was introduced to MPP housing device to improve the sound attenuation performance. The optimization process was conducted based on the tube width and height. Experiments were conducted to investigate the sound attenuation performance for both the HQ segment and the MPPHQ housing device. The main conclusions are summarized as follows.

1. For the HQ housing a dipole, the *IL* peak could be achieved when the path difference between the sound radiation from the dipole through the HQ tube segment was a multiple of the full wavelength under the condition that the HQ segment had a narrow tube width and height. When the upper width *H* of the HQ segment increased to 0.3, the *IL* peak could be achieved provided that the sum of their paths was an odd multiple of the half wavelength.

2. When HQ segment with proper size is added on the MPP housing device, the original *IL* trough of MPP housing can be elevated so that a broadening stopband can be achieved. Besides, the HQ tube occupied a small space and it did not significantly influence the original performance of the MPP housing device in other frequency ranges.

3. The theoretical and experimental results showed that the proposed MPPHQ housing device could work effectively over a wider bandwidth than the MPP housing device. In addition, the MPPHQ housing device could effectively suppress the sound radiation of the first and second blade passage frequencies which are dominant noise of the axial fan.

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Figure List