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NOISE CONTROL FOR A BETTER ENVIRONMENT

Study on Theoretical Calculation of Soundproofing Air Vents

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In this study, a design of an air vent with the sound insulation ability is proposed. The vent is designed in form of a rectangular cavity which is composed of one inlet and several outlets. The volume of the rectangular chamber and the areas of the inlet and outlet openings must be designed large enough for air permeability efficiency. Nevertheless, this will lead to a generation of sound pressure component of plane wave and higher-order acoustic modes inside the cavity, resulting in the decrease of soundproofing effect due to frequent occurrence of resonance. In this study, the mechanism of generation of both sound pressure components and the prevention method are clarified by theoretical calculation.

Keywords: Sound propagation, Resonance, Higher-order mode
I-INCE Classification of Subject Number: 37

1. INTRODUCTION

Road traffic noise is an increasing problem for urban residents in Southeast Asian developing countries during recent years[1][4]. However, no noise control measures have proven effective. Since most of houses in these countries have multiple vents for natural ventilation to adapt the hot and humid climate, the residents' daily activities are frequently disturbed by road traffic noise which propagates from outdoor into indoor through openings[5]. To solve this problem, in this study, we propose a design of an air vent with the sound insulation ability. The vent is designed in form of a rectangular cavity which is composed of one inlet and several outlets. The volume of the rectangular chamber and the areas of the inlet and outlet openings must be designed large enough for air permeability efficiency. Nevertheless, this will lead to a generation of sound pressure component of plane wave and higher-order acoustic modes inside the cavity, resulting in the decrease of soundproofing effect due to frequent occurrence of

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resonance. To add a soundproofing featuring, the soundproof vent hole unit installed in dwelling walls as shown in Fig. 1 is proposed in this work. Road noise propagates from the outside through the holes into the wall, then passes through the large cavity and propagates into the room. In order to install in the wall, it is considered that the rectangular cubic is reasonable as the unit shape. In this study, the mechanism of generation of both sound pressure components and the prevention method are clarified by theoretical calculation and experimental.

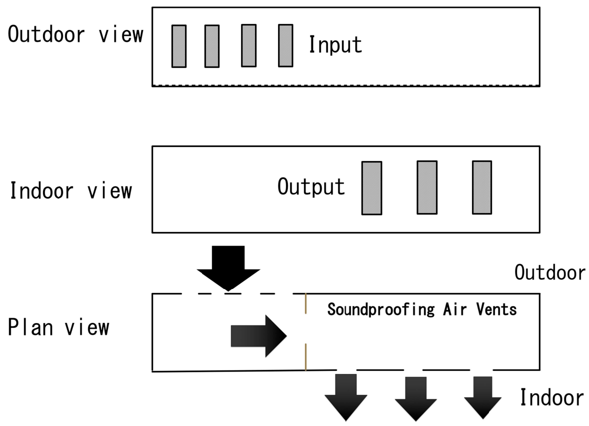

Fig. 1 A proposed air vent

2. METHOD OF ANALYSIS

Model of analysis is shown in Fig. 2. The rectangular cubic having a cross sectional $a \times b$ and length d which has the input having a cross sectional $(a_{i2} - a_{i1})$. $\times(b_{i2} - b_{i1})$. The outputs which has an opening area $(a_{o2} - a_{o1}) \times (d_{o2} - d_{o1})$ is located on the cavity and its center is at the l position from the input.

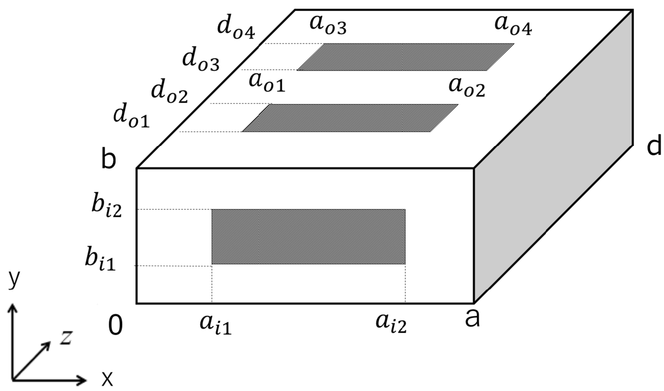

Fig. 2 Model of analysis

The velocity potential Φ is given by

$$\phi = (Ae^{\mu z} + Be^{-\mu z})\{C\sin(\alpha x) + D\cos(\alpha x)\}\left\{E\sin(\sqrt{s^2 - \alpha^2}\, y) + F\cos(\sqrt{s^2 - \alpha^2}\, y)\right\} \quad (1)$$

Here, $j^2 = -1$, A, B, C, D, E and F are arbitrary constants determinable from the boundary conditions, other symbols are constants.

Let $V_x = -\partial\phi / \partial x$, $V_y = -\partial\phi / \partial y$ and $V_z = -\partial\phi / \partial z$ are the velocity component in the x, y and z directions, respectively. We assume the walls of the cavity to be perfectly rigid therefore the boundary conditions may be expressed as

[1] at $x = 0$, $V_x = -\partial\phi / \partial x = 0$ (2)

[2] at $x = a$, $V_x = -\partial\phi / \partial x = 0$ (3)

[3] at $y = 0$, $V_y = -\partial\phi / \partial y = 0$ (4)

[4] at $y = b$, $V_y = -\partial\phi / \partial y = V_{o1} F_{o1}(x, y) + V_{o2} F_{o2}(x, y)$ (5)

[5] at $z = 0$, $V_z = -\partial\phi / \partial z = V_i \, F_i(x, y)$ (6)

[6] at $z = d$, $V_z = -\partial\phi / \partial z = 0$ (7)

where V_i and V_o are the driving velocity at the input and output, $F_i(x, y) = 1$ at the input and $F_i(x, y) = 0$ elsewhere, $F_o(x, y) = 1$ at the output and $F_o(x, y) = 0$ elsewhere.

In order to find ϕ, let ϕ_a be the solution of Eq.(1) obtained for the following boundary conditions:

[1] at $x = 0$, $V_x = -\partial\phi_a / \partial x = 0$ (8)

[2] at $x = a$, $V_x = -\partial\phi_a / \partial x = 0$ (9)

[3] at $y = 0$, $V_y = -\partial\phi_a / \partial y = 0$ (10)

[4] at $y = b$, $V_y = -\partial\phi_a / \partial y = 0$ (11)

[5] at $z = 0$, $V_z = -\partial\phi_a / \partial z = V_i \, F_i(x, y)$ (12)

[6] at $z = d$, $V_z = -\partial\phi_a / \partial z = 0$ (13)

and let ϕ_b be the solution of Eq.(1) obtained for the following boundary conditions:

[1] at $x = 0$, $V_x = -\partial\phi_b / \partial x = 0$ (14)

[2] at $x = a$, $V_x = -\partial\phi_b / \partial x = 0$ (15)

[3] at $y = 0$, $V_y = -\partial\phi_b / \partial y = 0$ (16)

[4] at $y = b$, $V_y = -\partial\phi_b / \partial y = V_{o1} F_{o1}(x, y) + V_{o2} F_{o2}(x, y)$ (17)

[5] at $z = 0$, $V_z = -\partial\phi_b / \partial z = 0$ (18)

[6] at $z = d$, $V_z = -\partial\phi_a / \partial z = 0$ (19)

then ϕ can be obtained as $\phi = \phi_a + \phi_b$. From the above boundary conditions ϕ can be determined and the sound pressure at the output side becomes

$$P = jk\rho c \left(\frac{4U_i}{S_{ab}} \frac{\cos k(z-d)}{k \sin kd} + \sum_{\bullet}^{\bullet} \sum_{\bullet}^{\bullet} \left[\frac{4V_i}{S_{ab}} \frac{\cosh\{\mu_{m,n}(z-d)\}}{\mu_{m,n} \sinh(\mu_{m,n} d)} I_{m,n} \cos(n\pi) \right.\right.$$

$$+\frac{4V_o}{S_{ad}}\cosh\left\{\frac{n\pi}{d}(\mathrm{z-d})\right\}\left(O_{1m,n}+O_{2m,n}\right)\frac{1}{\beta_{m,n}\tan\left(\beta_{m,n}\,\mathrm{b}\right)}\Bigg]\cos\left(\frac{m\pi}{a}x\right)\Bigg) \tag{20}$$

where k is wave-number, other symbols are defined by

$$O_{1m,n}=V_{o1}\int_{a_{o1}}^{a_{o2}}\cos\left(\frac{m\pi}{a}x\right)dx\int_{d_{o1}}^{d_{o2}}\cosh\left\{\frac{n\pi}{d}(\mathrm{z-d})\right\}dz \tag{21}$$

$$O_{2m,n}=V_{o2}\int_{a_{o3}}^{a_{o4}}\cos\left(\frac{m\pi}{a}x\right)dx\int_{d_{o3}}^{d_{o4}}\cosh\left\{\frac{n\pi}{d}(\mathrm{z-d})\right\}dz \tag{22}$$

$$I_{m,n}=\int_{a_{i1}}^{a_{i2}}\cos\left(\frac{m\pi}{a}x\right)\mathrm{dx}\int_{b_{i1}}^{b_{i2}}\cos\left(\frac{n\pi}{b}y\right)\mathrm{dy} \tag{23}$$

$$\mu_{m,n}=\sqrt{\left(\frac{m\pi}{a}\right)^2+\left(\frac{n\pi}{b}\right)^2-k^2} \tag{24}$$

$$\beta_{m,n}=\sqrt{k^2+\left(\frac{n\pi}{d}\right)^2-\left(\frac{m\pi}{a}\right)^2} \tag{25}$$

3. RESULS AND DISCUSSION

In order to examine the correctness of Eq. (25) an experiment was conducted[6]. An rectangular cubic was processed with an aluminum alloy having a thickness of 2 mm. Dimension of the rectangular cubic considered in this work are a=0.402m, b=0.16m and d=0.60m, respectively. When the output is located at $l=d$, the plane wave sound pressure component becomes

$$P=jk\rho c\frac{4U_i}{S_{ab}}\frac{1}{k\sin kd} \tag{26}$$

and the higher-order modes sound pressure components become

$$P=jk\rho c\sum\sum\left[\frac{4V_i}{S_{ab}}\frac{I_{m,n}\cos(n\pi)}{\mu_{m,n}\sinh\left(\mu_{m,n}d\right)}+\frac{4V_o}{S_{ad}}\frac{O_{1m,n}+O_{2m,n}}{\beta_{m,n}\tan\left(\beta_{m,n}\,\mathrm{b}\right)}\right]\cos\left(\frac{m\pi}{a}x\right) \tag{27}$$

The resonance will occur at the frequency when those denominators become zero. Similarly, When the output is located at $l=d/2$, the plane wave and the higher-order modes sound pressure component becomes

$$P=jk\rho c\frac{4U_i}{S_{ab}}\frac{1}{k\sin(kd/2)} \tag{28}$$

$$P=jk\rho c\sum\sum\left[\frac{4V_i}{S_{ab}}\frac{I_{m,n}\cos(n\pi)}{\mu_{m,n}\sinh\left(\mu_{m,n}d/2\right)}+\frac{4V_o}{S_{ad}}\frac{O_{1m,n}+O_{2m,n}}{\beta_{m,n}\tan\left(\beta_{m,n}\,\mathrm{b}\right)}\right]\cos\left(\frac{m\pi}{a}x\right) \tag{29}$$

Comparision between experimental results and calculated results are shown in Fig. 3 and Fig. 4. The upper is the experimental results and the lower indicates the calculate value of the 1st, the 2nd and 3rd of Eq. (25).

4. CONCLUSIONS

This work deals with a rectangular cubic which the output is mounted on the face that perpendicular to the input face. The distribution of sound pressure is calculated by solving the wave equations assuming that the loss can be neglected. The results obtained show that theoretical values related to the distribution of sound pressure inside the rectangular cubic and resonance frequencies agree well with the experimental values. A future direction of this study will be to determine the output area and position where the output sound pressure becomes the smallest by using the obtained calculation result.

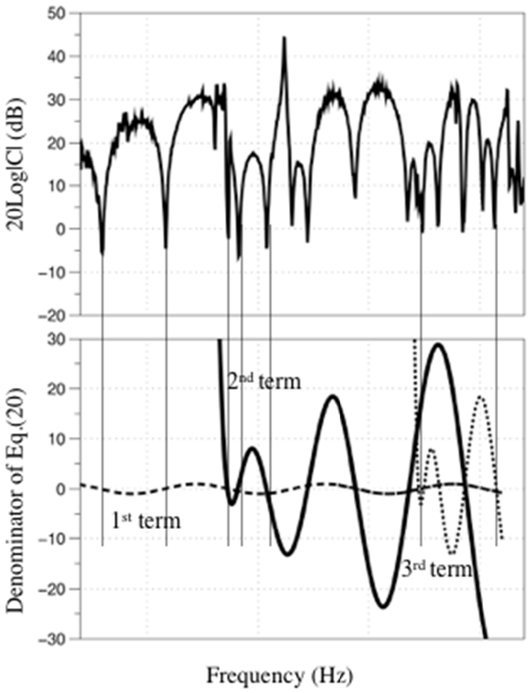

Fig. 3 Experimental and calculation results when the output is located at *l=d.*

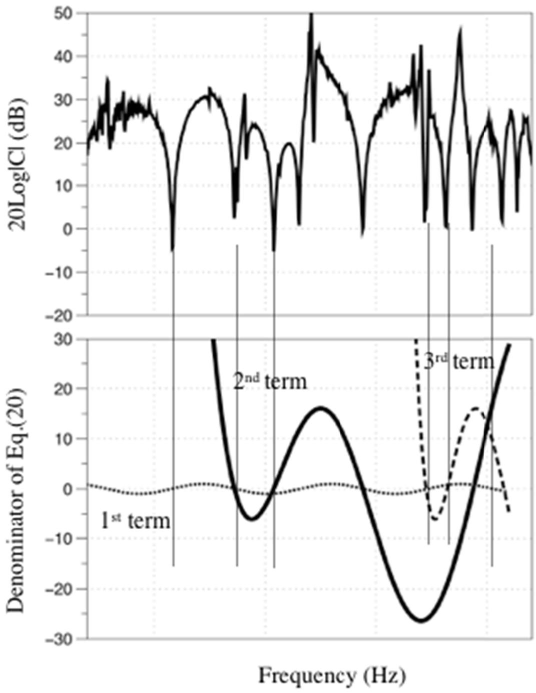

Fig. 4 Experimental and calculation results when the output is located at *l=d/2*

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