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A study of the sound transmission mechanisms of a finite thickness opening without or with an acoustic seal

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ABSTRACT

This work is concerned with the transmission of sound through an opening in a wall of finite thickness where the opening may be filled with an acoustic seal represented by a pair of mass layers. Unlike past work where efforts were made on either developing a prediction model or investigating the influence of opening parameters on the sound transmission performance, the work presented here aims at studying the coupling mechanism of the system by considering the fluid-loaded opening as a resonant system. This treatment allows a better understanding of the system property at both resonance and off-resonance frequencies. It is found that, unlike that for the opening of vanishing thickness, the transmission loss for an opening of finite thickness exhibits obvious oscillation with maxima and minima in the frequency spectrum. The latter are associated with the resonance of the fluid-loaded opening. At resonance, the opening extracts the acoustic energy from a region much larger than its physical dimension, causing more energy to transmit through the opening. As a result, negative value of transmission loss occurs. At frequencies off the resonance, the way in which energy flows into the opening is found to be dependent upon the reactance of the wave impedance of the fluid-loaded system. For a sealed opening, the reactance of the wave impedance is greatly increased due to the large surface density of the mass layers of the acoustic seal, resulting in less energy injected into and thereby transmitted through the opening. The underlying physics leading to the improvement in sound insulation is sought by using a wave impedance expression from which equivalent system representation is derived in different frequency ranges.

1. Introduction

Sound transmission through openings has a variety of engineering applications, from door leaks, ventilation ducts to sound barriers [1–4]. For an opening of small cross-sectional area with negligible thickness, research activity in the past on sound radiated by monopole source and piston has laid a foundation for studying the transmission mechanism and the influence of the opening on the transmission characteristics. For example, the acoustic field generated by the motion of the air particles in the plane of the opening subject to an acoustic excitation can be assumed to be equivalent to that produced by a massless piston vibrating transversely to its plane. The assumption simplifies the process in dealing with the sound transmission problem and holds for the acoustic wavelength much larger than the dimension of the opening [5,6].

Of more practical relevance is the sound transmission through openings in walls of finite thickness. These openings may exist due to manufacturing deficiency or for functional purposes. Typical examples include constructional cracks in building walls, leaks in vehicle cabin structures and gaps in conjunctions between wall elements. The sound insulation performance of the walls in these situations is greatly constrained by the additional paths created by the openings through which the sound transmitted could make a major contribution to the noise level in the receiving field.

The pioneer work on studying the influence of the opening can be traced back to Gomperts [7], who deduced a simple formula to predict the transmission of sound through circular and slit-shaped openings. Experiments demonstrated the reliability of the model at the off-resonance frequencies. Around the resonance frequencies, the model failed in rendering good prediction accuracy as the viscosity effect of the opening was ignored. The model was then improved [8] by combining with the viscosity formula developed by Ingerslev and Nielsen [9], yielding better accuracy for ka up to 0.5 (k being the wavenumber and a the radius for circular openings or the breadth for the slits). Wilson and Soroka [10] developed a model for a circular opening that extended the validity of the model to a greater range. In a subsequent study, Sauter and Soroka [11] found that the difference in transmission between a circular

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opening and a rectangular opening was marginal and it was concluded that the model can be applied to a rectangular opening by replacing the radius a with an equivalent radius of the opening cross section. A survey made by Morfey [12] of the low-frequency acoustic properties of circular, rectangular and elliptic openings demonstrated also the weak dependence of the acoustic response on the opening shape. Oldham and Zhao [13] examined the models of Gomperts and Soroka using the sound intensity technique with the results showing that the models were valid over a large frequency range.

In parallel to the development of a model for predicting the sound transmission through an opening, efforts were also made to study the influence of opening parameters on the sound transmission property. Uris et al. [2] studied the influence of the length and depth of a slit on the sound reduction index of a lightweight partition. Later, they studied the relative position of the slits on the partition, showing a strong dependence of the sound insulation upon the position of the slit on the partition [14].

Despite the considerable efforts dedicated to the relevant field, there remain questions to which proper answers should be given for better understanding the mechanism underlying the sound transmission process. For example, it is known that the transmission curve would exhibit an oscillatory feature with maxima and minima due to resonances of the fluid-loaded opening. Negative values of transmission loss may occur around the resonance frequencies. This phenomenon, which apparently violates the power conservation, may lead to a misunderstanding of the transmission characteristics of the system and a comprehensive explanation to this, however, appears to be lacking in the literature. This paper will fill this gap by resorting to a simplified system model and the relation between the opening parameters and the sound transmission will be investigated in terms of the wave impedance of the coupled system. The advantage of qualifying the coupled system, which consists of the opening and the fluid acting upon its two ends, as the wave impedance, is that the system characteristics could be examined independently without the constraint of external excitation, thus allowing the model to be coupled to a more complicated acoustic field [15]. The developed model will then be employed to study the sound insulation characteristics of an opening filled with an acoustic seal. Analysis shows that the acoustic property of the opening is greatly changed due to the presence of the acoustic seal and the system will exhibit different vibration behaviors at resonance and off-resonance frequencies.

This paper is structured as follows: The development of a general model that is able to calculate the sound transmission through a circular opening, of small cross section, without and with an acoustic seal is described in Section 2; Section 3 presents a study of the sound transmission through an empty opening. An explanation to the occurrence of the negative transmission loss is given and the dependence of the transmission upon the wave impedance of the coupled system is discussed; Sound transmission through an opening filled with an acoustic seal is investigated in Section 4 and approximate wave impedance formulas are obtained to facilitate the understanding of the system behavior in different frequency regions; Conclusions are drawn in Section 5 to close the paper.

2. Model development

The system under investigation consists of a circular opening flush mounted in an infinite baffle of finite thickness that separates an air media into two domains. This arrangement corresponds to an opening positioned at a place far away from a corner, or an edge, of a wall where the reflection and/or diffraction generated may make substantial influence on the radiation impedance of the opening. Inside the opening, an acoustic seal is inserted, occupying a space from $x = d_2$ to $x = d_2 + d_3$. The dynamic property of the acoustic seal may be characterized by two mass layers, with surface densities m_{23} ($x = d_2$) and m_{34} ($x = d_2 + d_3$), vibrating in a piston-like motion at its two ends. This is a reasonable approximation for the vibration in the low-frequency range where the diameter of the opening is much smaller than the acoustic wavelength [16]. The vibration of the air at the ends of the opening, i.e. $x = 0$ and $x = d_2 + d_3 + d_4$, is also assumed to exhibit piston-like motion at low frequencies. For the convenience of the modeling, two virtual mass layers ($m_{12} = m_{45} = 0$) are defined at the two ends. The four mass layers divide the whole acoustic domain into five subdomains. A dual-digit subscript is used to indicate a quantity located at the interface between a pair of domains. The time harmonic factor $e^{j\omega t}$ is understood and omitted hereafter. The cross section of the configuration is shown schematically in Fig. 1. The symmetry of configuration enables the modeling process to be implemented in a two-dimensional coordinate system with the origin set at the center of the entry of the opening. By replacing the radius with an equivalent radius of the rectangular cross section, the model to be developed is also applicable to sound transmission through a rectangular opening of small aspect ratio [10].

2.1. Pressure fields

A plane wave $p_{i,1}$ incident upon the opening at an angle of θ_i with respect to the normal to the wall surface could be described as

$$p_{i,1} = P_{in} e^{-j(k_x x - k_y y)}, \tag{1}$$

where $k_x = k_0 \cos\theta_i$, $k_y = k_0 \sin\theta_i$ and $k_0 = \omega/c_0$, with ω and c_0 being, respectively, the angular frequency and the speed of air, and P_{in} is the pressure amplitude of the incident wave. The resultant pressure p_1 in the incident field (domain 1) could be determined from the superposition of the blocked pressure p_{blk} and the radiated pressure

$$p_1 = p_{blk} + p_{rad,1} = p_{i,1} + p_{r,1} + p_{rad,1}, \tag{2}$$

where $p_{r,1}$ is the reflected pressure in the absent of the opening

$$p_{r,1} = P_{in} e^{-j(-k_x x - k_y y)}, \tag{3}$$

and $p_{rad,1}$ is the pressure radiated backward to the incident field by the air piston at the end ($x = 0$) of the opening. For a vibrating piston

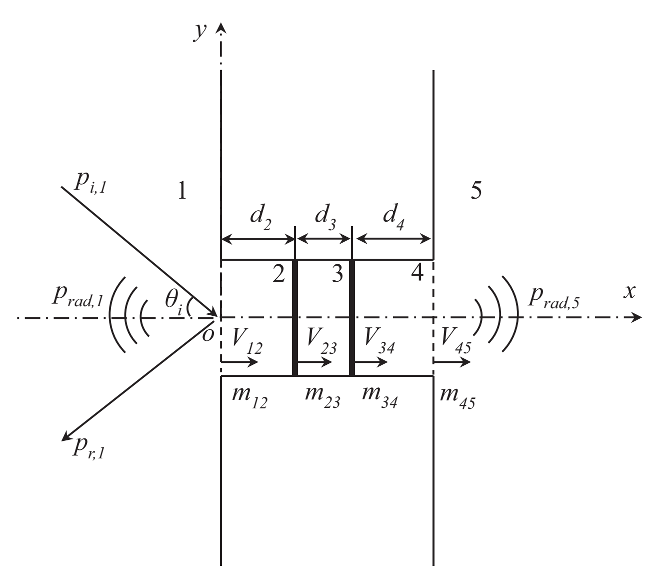

Fig. 1. A schematic representation of the cross section of the system under investigation.