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# Effect of Grazing Flow on the Acoustic Impedance of an Orifice

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**A linearized potential flow model is developed to study the effect of grazing flow on the acoustic behavior of an orifice. In accordance with the previous flow visualization and measurements, this model uses the particle velocity continuity boundary condition rather than the widely used displacement to match the flowfields separated by the shear layer over the orifice. An experiment is also carried out to validate the present theory in the case of circular or rectangular orifices. The theory agrees well with the experiment. In addition, further comparison is made between the present and two existing impedance models. One major objective of the present investigation is to make a tentative judgement on the two boundary conditions for a sound wave transmitting through a turbulent boundary layer over an orifice. It is found that, compared to the particle displacement continuity boundary condition, the use of the particle velocity continuity boundary condition results in a much better agreement between the theoretical predictions and the experimental data. Finally, there is a discussion on the influence of plate thickness on orifice impedance in the presence of grazing flow.**

## Nomenclature

$A$ = $L/2$
$B$ = characteristic orifice length, equal to $R$ for a circular orifice or $A$ for a rectangular orifice
$c_0$ = sound speed
$H$ = shape factor of the wall boundary layer
$i$ = $\sqrt{-1}$
$k$ = sound wave number, $\omega/c_0$
$L$ = length of the side of a rectangular orifice parallel to grazing flow
$M_G$ = freestream Mach number of grazing flow
$N$ = number of the orifices in the test perforated plates
$p$ = sound pressure
$p_+$ = sound pressure in the far distance of the region where $x_3 > 0$
$p_-$ = sound pressure in the far distance of the region where $x_3 < 0$
$Q$ = fluctuating volume flux through an orifice
$R$ = radius of a circular orifice
$r$ = normalized specific acoustic resistance of an orifice
$S$ = transverse area of an orifice
$Sr$ = Strouhal number, $\omega R/U_G$ for a circular orifice or $\omega A/U_G$ for a rectangular orifice
$T$ = plate thickness
$t$ = time
$U_G$ = freestream velocity of grazing flow
$W$ = length of the side of a rectangular orifice transverse to grazing flow
$w$ = fluctuating velocity normal to the upper surface of an orifice
$x$ = normalized specific acoustic reactance of an orifice
$\boldsymbol{x}$ = coordinates of the observer point, $(x_1, x_2, x_3)$
$\boldsymbol{y}$ = coordinates of the source point, $(y_1, y_2, y_3)$
$z$ = normalized specific acoustic impedance of an orifice, $r + ix$
$\delta^*$ = displacement thickness of the wall boundary layer
$\theta$ = momentum thickness of the wall boundary layer
$\rho_0$ = air density
$\sigma$ = open area ratio of a perforated plate
$\omega$ = angular frequency of the incident sound

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## I. Introduction

A PERFORATED liner is extensively used to reduce noise in a flow duct and to suppress combustion instabilities in the afterburner of a jet engine or the combustor of a rocket engine. In particular, a kind of liner with adjustable impedance that has potential applications in the active control of duct noise and unstable flow in turbomachines has received some attention since the 1970s.[1–6] In all of these applications, grazing flow over the surface of a perforated liner would always be present. It has been well known that grazing flow changes markedly the acoustic impedance of a perforated liner, thereby affecting the noise attenuation in the flow duct. A better understanding of this so-called grazing flow effect is essential for the design of high-efficiency acoustic liners.

During the past decades, considerable research has been carried out to investigate the effect of grazing flow on the acoustic impedance of a perforated liner. It has been well established from previous studies[7–14] that the acoustic resistance of a perforated plate (or an orifice) is almost linearly increased and the acoustic reactance is slightly decreased with an increase of the grazing flow velocity. The visualization and measurements of the flow details in the vicinity of an orifice reveal that vortices are generated from the orifice leading edge under the acoustic excitation and convected downstream by the mean grazing flow.[15,16] It has been concluded that strong interaction occurs between the vortical and acoustic flow at an orifice in the presence of grazing flow, which accounts for the change of the orifice acoustic impedance.

The mechanism of sound–vortex interaction has been applied with some success to predict the effect of bias flow on the acoustic impedance of an orifice, such as the work of Bechert,[17] Howe,[18] Hughes and Dowling,[19] and Jing and Sun.[20] When grazing flow is present, most of the orifice impedance models either result from experimental data[8,12,14] or rely on empirical parameters such as the discharge coefficient[9] or the effective orifice area.[13] So far, few purely theoretical predictions of this grazing flow effect have been attempted in light of the mechanism of sound–vortex interaction. Rice[21] modeled the flow above a circular orifice such that a spherically symmetrical perturbation was impressed on the steady grazing flow. Rice's model predicts that the resistance is directly proportional to grazing flow speed, whereas the orifice reactance is not a function of grazing flow speed. This later conclusion is contrary to the widely held experimental result that the reactance decreases slightly. Ronneberger[22] has made an attempt to describe the orifice flow in terms of wavelike disturbances of a thin shear layer over the orifice. Although the theory and the experiment are qualitatively consistent, they agree poorly with each other in numerical quantities. Howe et al.[23] presented a linearized potential model to study the effect of grazing flow on the acoustic impedance of an orifice in an infinitely thin plate. In this model, the shear layer over the orifice was simplified as a vortex sheet, and the two flow regions separated by the vortex sheet were matched by the continuity of sound pressure and particle displacement. In addition, the Kutta condition was applied to make sure that the vortex sheet leaves tangentially from the leading edge of the orifice. A similar approach had previously been adopted by Kaji et al.[24] to model the radiation impedance of a tube opening subjected to a grazing flow. The Howe et al.[23] method has further been extended to include the influences of orifice shape[25] and plate thickness.[26] To the authors' knowledge, the vortex sheet models of Refs. 23, 25, and 26 for the grazing flow effect have not been fully verified by experiment.

In the present investigation, we first use the two-microphone method developed by Dean[27] to measure the acoustic impedance of several selected perforated plates subjected to grazing flow. The orifices in the test perforated plates are either circular or rectangular. When the measured data are compared with the theoretical results based on the models of Refs. 23 and 26, it is found that there exist large discrepancies between them. Through a careful examination, we deduce that the differences between the theories and the experiment might result from the use of the Miles[28]–Ribner[29] particle displacement continuity boundary condition in these models.

As we know, the particle displacement continuity boundary condition for a sound wave transmitting through a shear layer results from the assumption of a clearly defined flow interface consisting of a continuous impermeable vortex sheet. However, in practice the flow passing an orifice is usually highly turbulent, as it is in the present experiment, thus has a thick shear layer relative to the transverse dimension of the orifice. In this situation, a clearly defined flow interface over the orifice could not be expected to exist. In fact, the flow visualization carried out by Nelson et al.[15] shows that discrete vortices are formed at the leading edge of a resonator orifice, then convected downstream with their size growing and finally impinged on the trailing edge. Nelson et al. have also measured the fluctuating velocity near the resonator orifice. Their measurements showed that there were large-amplitude changes and a phase change of approximately 180 deg for the streamwise component while the variations of both the amplitude and the phase of the vertical component were small between the upper and lower regions of the shear layer. Obviously this result indicates that, for instantaneous fluctuating velocity, the streamwise component is discontinuous, whereas the vertical component is continuous across the shear layer. To explain the experiment, Nelson et al.[30] further set up a model to simulate the flow/acoustic interaction that occurred at a resonator orifice, in which the vortical flow is modeled by means of an array of discrete vortices. Good agreement between the theory and the experiment was obtained. As expected, the continuity of the vertical fluctuating velocity component at the orifice opening was also demonstrated in the computation. In the situation of high grazing flow Mach number, as encountered at a jet engine liner, the basic flow patterns near a resonator orifice given by Nelson et al.[15] were reproduced in the experiment of Worraker and Halliwell.[16] Moreover, they have also evaluated several existing theoretical models and concluded that the approach taken by Nelson et al.[30] lent itself most readily to explaining their experimental results. In conclusion, the detailed theoretical and experimental studies of Nelson et al.[15,30] and Worraker and Halliwell[16] have provided evidence to support the use of the particle velocity continuity boundary condition rather than the displacement when sound transmission through a turbulent shear layer over an orifice is considered.

In this paper, a linearized potential flow model is developed on the mechanism of sound–vortex interaction to investigate the influence of grazing flow on the acoustic behaviors of an orifice. In this model, the particle velocity continuity boundary condition is used to match the acoustic fields on the two sides of the grazing flow shear layer, and the Kutta condition is applied on the orifice leading edge to account for viscous effect. The details of the present flow model will be described in the next section. Savkar[31] has previously proposed computing sound transmission through a turbulent shear layer by the use of the particle velocity continuity boundary condition rather than the displacement. Unfortunately, for the problem of the radiation of spinning acoustic modes from a round duct, which has been examined in Ref. 31, the marginal variation that resulted from using the two different boundary conditions does not allow a critical judgement of the boundary condition. By contrast, in the present investigation it is found that the use of the particle velocity match results in a great improvement of the theoretical predictions for both the acoustic resistance and reactance of an orifice subjected to highly turbulent grazing flow. In this sense, the present investigation supports the use of the particle velocity continuity boundary condition rather than the displacement.

In most practical applications, the plate thickness is comparable to the orifice transverse dimension. Therefore, it is essential to incorporate the influence of plate thickness into an impedance prediction scheme. The influence of plate thickness in the presence of grazing flow has previously been treated by Howe,[26] but the method is applicable only for a rectangular orifice. Different from the study of Ref. 26, the present model can be applied to the case of an orifice of arbitrary shape with the influence of plate thickness being taken into account.

## II. Basic Equations

As shown in Fig. 1, there is an incompressible mean flow over the upper opening of a square-edged orifice in a rigid plate of small, but finite, thickness. As a general case, the orifice is of arbitrary shape. A low-frequency harmonic sound wave incident on the plate produces a uniform pressure difference $\Delta p = (p_+ - p_-)\exp(-i\omega t)$ between the two sides of the plate in the vicinity of the orifice. The unsteady flow through the orifice forced by this pressure difference will be considered.

An approach similar to that of Howe[26] is adopted to model the unsteady flow. However, different from that of Howe,[26] the present model uses the particle velocity continuity boundary condition rather than the displacement to match the sound fields separated by the shear layer over the upper orifice opening. This boundary condition implies that the vertical component of the fluctuating velocity is continuous while the streamwise component is discontinuous across the shear layer. According to the discussion in the Introduction, the use of the particle velocity continuity boundary condition is consistent with the previous flow visualization and measurements.[15,16] To simplify the problem, we further hypothesize that the shear layer is thin for low-frequency sound propagation, and so the mean flow is taken to be uniform in the two regions above and below the upper orifice opening at velocities of $U_G$ and zero, respectively. To account for viscous effect, the Kutta condition is applied requiring the vertical component of the fluctuating velocity to be zero at the leading edge of the orifice. The application of the Kutta condition is supported by previous experiments.[15,16,32] The model just described is equivalent to assuming that infinitely small vortices are shed at the leading edge of the orifice, convected downstream at constant streamwise velocity, and annihilated at the trailing edge of the orifice. This assumption has ruled out the instabilities of the shed vortices. According to Nelson et al.,[30] this is justified by experiment, where the presence of the solid boundaries allows a stable train of vortices to form that do not interact with each other within any appreciable distance of the orifice. Note that a similar model of vortex shedding and convection has been employed by Howe[33] to study the acoustic impedance of a slit connected to a circular cylindrical

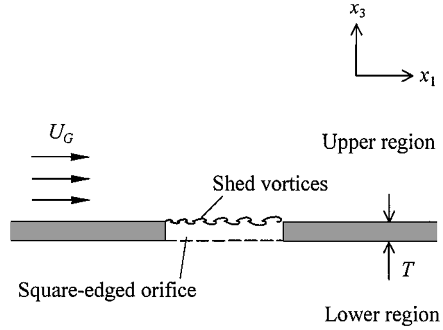


**Fig. 1 Uniform grazing flow of velocity $U_G$ past a squared-edged orifice of arbitrary shape in a rigid plate of small, but finite, thickness.**

cavity, and generally good agreement between the theoretical results and the experimental data was reported.

The formulation of the present model is given as follows. As shown in Fig. 1, a rectangular coordinate system $(x_1, x_2, x_3)$ is introduced. The plane $x_3 = 0$ of the coordinate system is taken to coincide with the midplane of the plate, the $x_1$ axis is parallel to the mean flow, and the coordinate origin is at the geometrical center of the orifice. On the condition that the plate thickness is far smaller compared to the sound wavelength, it is assumed that the velocity of the fluid confined in the orifice only has a vertical component, and its value is independent of $x_3$. Let $w(x_1, x_2)$ be the fluctuating velocity of the fluid particles in the orifice. In accordance with the particle velocity continuity boundary condition, the vertical component of the fluctuating velocity of fluid particles just above the upper orifice opening and below the lower orifice opening is equal to $w(x_1, x_2)$. In the approximations of low-frequency incident sound and small plate thickness, the sound pressure above and below the plate can be formulated as

$$p = p_+ - i\rho_0\left(\omega + iU_G\frac{\partial}{\partial x_1}\right) \times \int_S \frac{w(y_1, y_2)}{2\pi|\boldsymbol{x} - \boldsymbol{y}|}\,\mathrm{d}y_1\,\mathrm{d}y_2 \qquad \left(x_3 > +\frac{1}{2}T\right)$$

$$= p_- + i\rho_0\omega\int_S \frac{w(y_1, y_2)}{2\pi|\boldsymbol{x} - \boldsymbol{y}|}\,\mathrm{d}y_1\,\mathrm{d}y_2 \qquad \left(x_3 < -\frac{1}{2}T\right) \tag{1}$$

The motion of the fluid in the orifice is governed by the following equation:

$$-\rho_0 T\frac{\mathrm{d}w}{\mathrm{d}t} = i\rho_0\omega T w = [p] \qquad \left(|x_3| < \frac{1}{2}T, \sqrt{x_1^2 + x_2^2} < R\right) \tag{2}$$

where $[p]$ is the fluctuating pressure difference between the upper and lower orifice opening. By the use of the pressure continuity condition, the following differential equation can be deduced from Eqs. (1) and (2):

$$\left(2i\omega - U_G\frac{\partial}{\partial x_1}\right)\int_S \frac{w(y_1, y_2)\,\mathrm{d}y_1\,\mathrm{d}y_2}{2\pi\sqrt{(x_1 - y_1)^2 + (x_2 - y_2)^2}} + i\omega T w = \frac{p_+ - p_-}{\rho_0} \tag{3}$$

Integrate Eq. (3) with respect to $x_1$, and introduce the following nondimensionalization:

$$\boldsymbol{X} = \boldsymbol{x}/B, \qquad \boldsymbol{Y} = \boldsymbol{y}/B, \qquad Sr = \omega B/U_G$$

$$Z = i\rho_0 Sr U_G w/\pi(p_+ - p_-)$$

Then we obtain the following integral equation:

$$\int_S \frac{Z\,\mathrm{d}Y_1\,\mathrm{d}Y_2}{\sqrt{(X_1 - Y_1)^2 + (X_2 - Y_2)^2}} - 2i\pi Sr(T/B)\int_{\mathrm{LE}}^{X_1} Ze^{2iSr(X_1 - Y_1)}\,\mathrm{d}Y_1 = 1 + \alpha e^{2iSrX_1} \tag{4}$$

where LE is the leading edge of the orifice and $\alpha$ is the unknown function of $X_2$ that must be determined to ensure that the Kutta condition is satisfied at the LE of the orifice.

Following Grace et al.[25] we solve Eq. (4) numerically to obtain the acoustic impedance of the orifice. A Cartesian grid covering the upper orifice opening is used to discretize Eq. (4). On each grid cell $Z(Y_1, Y_2)$ is assumed to be constant, but Green's function is analytically integrated. The Kutta condition is applied by imposing $Z = 0$ on the first grid cell in each grid row of constant $Y_2$. The discretization of Eq. (4) leads to a set of linear equations that can be readily solved to obtain the values of $\alpha$ and $Z$. The fluctuating volume flux through the orifice is given by

$$Q = \int_S w(x_1, x_2)\,\mathrm{d}x_1\,\mathrm{d}x_2 \tag{5}$$

Then the acoustic impedance of the orifice can be obtained from its definition as

$$z = \frac{p_+ - p_-}{\rho_0 c u_{\mathrm{av}}} \tag{6}$$

where the average velocity through the orifice is given by $u_{\mathrm{av}} = Q/S$.

In Ref. 26, to solve the governing equation based on the particle displacement match, the assumption has to be made that the motion of the vortex sheet over the orifice does not vary along the transverse direction of grazing flow. Therefore the method is applicable only for a rectangular orifice with large value of $W/L$. It is obvious from the preceding basic equations that the present model has no such limitation. For a circular orifice, the normalized specific acoustic impedance given by Eq. (6) is the function of three nondimensional parameters: grazing flow Mach number $M$, Helmholtz number $kR$, and normalized plate thickness $T/R$. In the case of a rectangular orifice, four nondimensional parameters are needed: grazing flow Mach number $M$, Helmholtz number $kA$, normalized plate thickness $T/A$, and normalized orifice width $W/A$.

## III. Experiment

Dean[27] has reviewed the experimental techniques for the measurement of the acoustic impedance of a perforated plate subjected to grazing flow. Among the methods, the two-microphone method has found wide application because of its simplicity and reliability. Moreover, one advantage of the two-microphone method over the others is that its measurement arrangement is more similar to conditions where a perforated plate is practically applied. The present experimental setup is shown in Fig. 2. Grazing flow is introduced through a square-section wind tunnel of internal width 120.0 mm (4.724 in.). A pitot-static tube 3 mm (0.118 in.) in diameter is used to measure the grazing flow speed. A test perforated plate is flush mounted in the wall of the wind tunnel and is backed by a small cylindrical cavity. The depth and inner diameter of the cavity are 150 and 35 mm (5.906 and 1.378 in.), respectively. A 50-W electrical driver is mounted opposite to the test perforated plate. In the present experiment, a harmonic sound source is used. Two B&K 4133 $\frac{1}{2}$-in. microphones are used to detect the sound pressures, one is located at the bottom of the cavity and the other is flush mounted in the wind tunnel wall at a distance of 30 mm (1.181 in.) from the axis of the cylindrical cavity. Through an A/D converter, the microphone signals are input into a computer for processing. According to Dean,[27] the acoustic impedance of a sample can be calculated from the amplitudes of and the phase difference between the sound

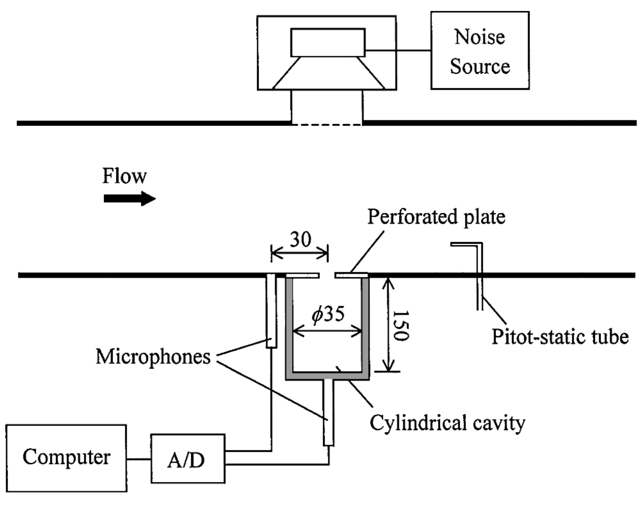


**Fig. 2 Experimental setup; lengths are in millimeters.**

**Table 1 Geometrical parameters of the test perforated plates**

| Number[a] | $R(W \times L)$, mm (in.) | $T$, mm (in.) | $N$ | $\sigma$, % |
|---|---|---|---|---|
| 1 | 1.5 (0.0591) | 2.0 (0.0787) | 4 | 2.94 |
| 2 | 2.25 (0.0886) | 2.0 (0.0787) | 1 | 1.65 |
| 3 | 3.5 (0.138) | 0.5 (0.0197) | 1 | 4.00 |
| 4 | 3.5 (0.138) | 2.0 (0.0787) | 1 | 4.00 |
| 5 | 6.0 × 2.0 (0.236×0.0787) | 1.0 (0.0394) | 1 | 1.25 |

[a]Orifices in perforated plates 1–4 are circular, and that in no. 5 is rectangular.

pressures measured by the two microphones. The geometric parameters of the five test perforated plates in the present experiment are shown in Table 1. The forcing sound pressure level is kept sufficiently low so that the measured acoustic impedance is not affected by a nonlinear effect.

## IV. Results and Discussion

### A. Comparison Between Present and Existing Models

In this subsection, the present theory is compared with two existing models, which have been widely used for impedance prediction in practical applications. The empirical model of Ref. 8, which has been concluded from systematic experiments, is as follows:

$$l_{\text{eff}} = T + 0.375[1.7R(1 - 0.7\sqrt{\sigma})]$$
$$r = \left(0.123/c_0C_f^2\right)(U_G + 1.5w_A) \tag{7}$$

where the discharge coefficient $C_f = 0.61$ and $w_A$ is the amplitude of the fluctuating particle velocity in the orifice, which is negligible compared to $U_G$ when the nonlinear effect is absent. In Ref. 21, Rice gave a theoretical impedance model of a circular orifice as

$$z = (M_G/2) - i(kR/2) \tag{8}$$

Note that this impedance model is obtained by solving linearized governing equations outside the right hemisphere covering the upper opening of the orifice above the plate. Thus, as discussed by Rice, the total reactance would be determined by adding the proportions of reactance provided by the fluid within this hemisphere and within the orifice itself to the end correction, $-\frac{1}{2}kR$, given by Eq. (8). In addition, we think it is also necessary to include the proportion of reactance provided by the fluid within the half-space below the plate, and the value is expected to be $-0.85kR$ because the steady flow velocity is zero in this region. Therefore, assuming that all of the proportions of reactance due to the fluid within the right hemisphere covering the upper opening of the orifice is blown away by grazing flow, the total orifice reactance that replaces the imaginary part of Eq. (8) is given as

$$x = -k(T + 1.35R) \tag{9}$$

The present experimental and theoretical results, and the calculations of the two impedance models are presented for comparison in Figs. 3 and 4. Because the test perforated plates in Table 1 have small open area ratio ($\sigma < 5\%$), the normalized specific acoustic impedance of the perforated plates is calculated from the simple relation $z/\sigma$. We can see that, in Figs. 3 and 4, the experimental data agree well with the empirical results of Eq. (7) for the acoustic resistance over the range of the grazing flow speed in the present study, whereas generally good agreement is also obtained for the acoustic reactance at high grazing flow Mach number (approximately higher than 0.05). In fact, it is obvious that the empirical effective orifice length in Eq. (7) was obtained from the experiments carried out at high grazing flow speed. This result of comparison makes us believe that the present experiment is reliable.

Let us see the variation of the acoustic resistance. Figures 3 and 4 show that the acoustic resistance given by the present model increases slowly or even decreases within a small range of grazing flow speed that is near to zero but increases linearly with the increasing grazing flow speed beyond this range. Therefore in terms of the variation of the acoustic resistance given by the present model there are two different regions, the sinking region and the linearly increasing region, as named in this paper. In Figs. 3 and 4, it can be seen that the agreement between the present model and Rice's model is very good in the linearly increasing region. We can also see that, for the acoustic resistance, the calculations of the present and Rice's[21] models are somewhat larger than those of the empirical model of Ref. 8, but the agreement between them is generally good.

For the acoustic reactance, the calculations of Rice's model and the empirical model of Ref. 8 do not change with grazing flow speed. However, both the previous and present experiments show that the acoustic reactance decreases from its linear value as grazing flow speed increases and tends to be roughly constant when grazing flow speed is high. By comparison, the present model gives generally good predictions of the variation of the acoustic reactance at low grazing flow speed, as shown in Figs. 3 and 4. Figures 3 and 4 show that the present model and the empirical model of Ref. 8 agree with each other very well for the acoustic reactance at high grazing flow speed. However, it can be seen from Figs. 3 and 4 that, at comparatively high grazing flow Mach number (about larger than 0.1), the present model's predictions for the acoustic reactance is not as satisfactory as for the acoustic resistance. We take a view similar to Melling's[34] that only if more flow details in the vicinity of the orifice are known could more accurate calculation of the variation of the acoustic reactance be made. The accurate prediction of the acoustic reactance in the presence of grazing flow is still a problem of great challenge.

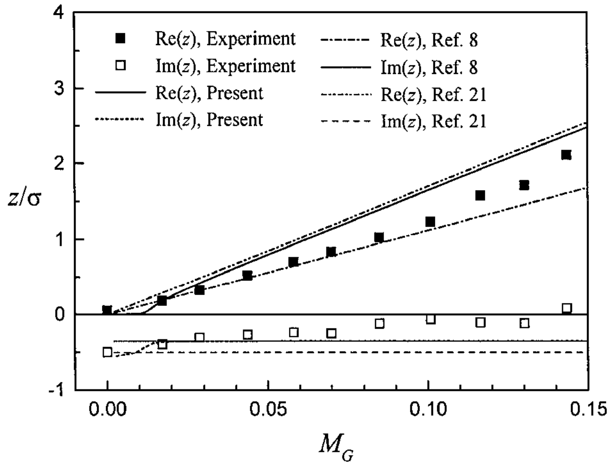


**Fig. 3 Comparison between present model, existing models, and experiment; normalized specific acoustic impedance plotted as a function of the grazing flow Mach number for perforated plate 1 at 200.0 Hz; $T/R = 1.33$ and $kR = 5.54 \times 10^{-3}$.**

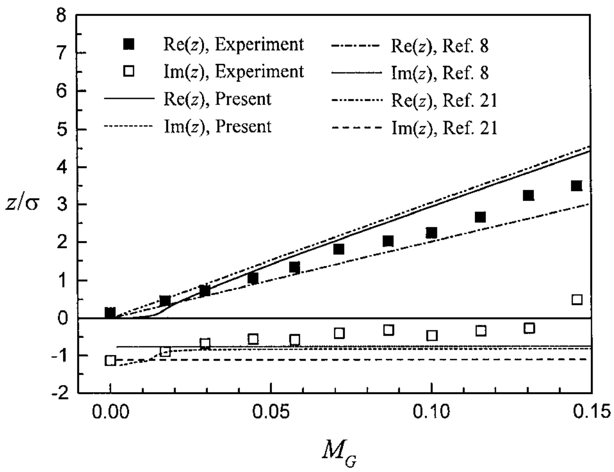


**Fig. 4 Comparison between present theory and experiment and existing models; normalized specific acoustic impedance plotted as a function of the grazing flow Mach number for perforated plate 2 at 200.0 Hz; $T/R = 0.89$ and $kR = 8.31 \times 10^{-3}$.**

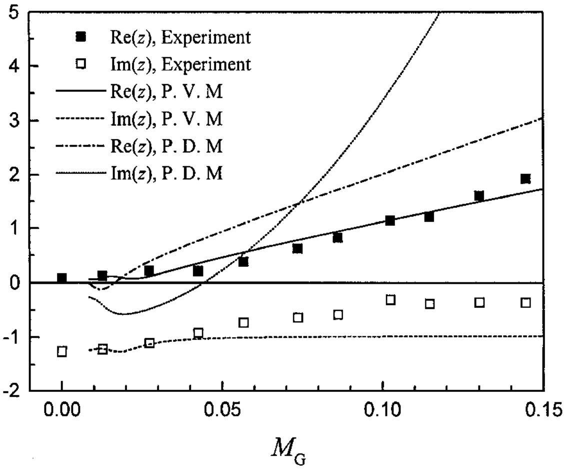


**Fig. 5 Comparison between the calculations based on PVM and PDM in the case of a circular orifice; normalized specific acoustic impedance plotted as a function of the grazing flow Mach number for perforated plate 3 at 500.0 Hz; $T/R = 0.14$ and $kR = 3.23 \times 10^{-2}$.**

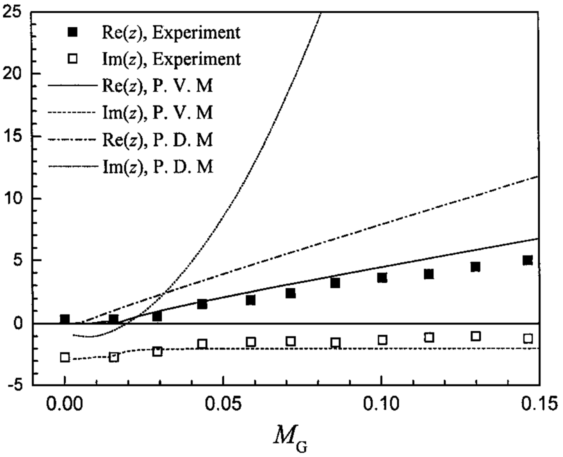


**Fig. 6 Comparison between the calculations based on PVM and PDM in the case of a rectangular orifice; normalized specific acoustic impedance plotted as a function of the grazing flow Mach number for perforated plate 5 at 500.0 Hz; $kA = 9.24 \times 10^{-3}$, $T/A = 1.0$, and $W/A = 1.0$.**

### B. Comparison Between the Two Boundary Conditions

In Figs. 5 and 6, the present theoretical results are compared with those obtained from the models of Refs. 24 and 26, which are based on particle displacement match. For comparison purposes, the present experimental results are also presented. For simplicity, the short forms of the two boundary conditions, particle velocity match (PVM) and particle displacement match (PDM), are used in this paper. For perforated plate 3, the ratio between the plate thickness and the orifice radius is 0.14, and so we consider it to be thin. Figure 5 shows that there are large discrepancies between the theoretical results based on PDM and the experimental data, especially for the acoustic reactance. The computed acoustic reactance based on PDM increases too fast compared to the experimental results as the grazing flow speed is increased. Also, it does not approach the classical value $-1.7kR/\sigma$ as the grazing flow speed approaches zero. By comparison, the present calculations using PVM are in generally good agreement with the experimental data. Similar results are obtained for perforated plate 5 in which the orifice is rectangular, as shown in Fig. 6.

As discussed by Savker,[31] the basis of the PDM is the assumption of a clearly defined flow interface. However, it is hard to expect this assumption to be valid in the present experiment and most engineering applications. Figure 7 shows the velocity distribution of the wind tunnel wall boundary layer. The velocities have been normalized by $u_e$, which is the flow speed at the wind tunnel center. When $u_e = 10.0$ and 50.0 m/s (393.7 and 1968.5 in./s), the velocity

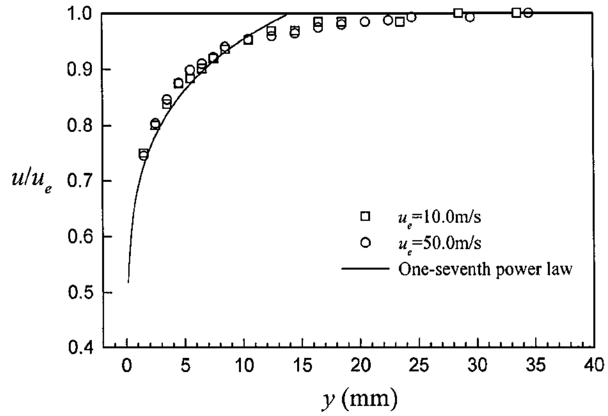


**Fig. 7 Wall boundary layer: flow velocity as a function of the distance from the wall.**

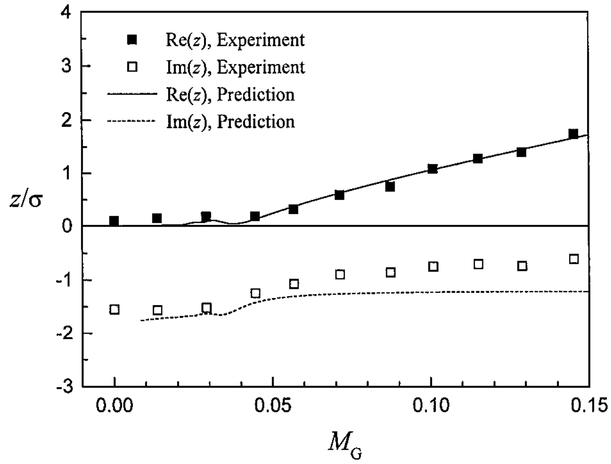


**Fig. 8 Normalized specific acoustic impedance plotted as a function of the grazing flow Mach number for perforated plate 4 at 500.0 Hz; $T/R = 0.57$ and $kR = 3.23 \times 10^{-2}$.**

distributions in the wall boundary layer are almost the same with $(\delta^*, \theta, H)$ being correspondingly equal to (2.33, 1.37, 1.70) and (2.45, 1.49, 1.64), and agree well with the well-known one-seventh order law for a turbulent boundary layer. According to the data in Fig. 7, the boundary-layer thickness is about 30.0 mm (1.181 in.); thus, it is much larger than the diameter of the orifices in all test perforated plates. In the case of such an ill-defined, highly turbulent flow, there would be little reason to use the PDM. It is evident from the comparisons made in Figs. 5 and 6 that the present investigation favors the PVM rather than the displacement.

### C. Influence of Finite Plate Thickness in the Presence of Grazing Flow

In this subsection, the present model is employed to study the influence of finite plate thickness on the orifice impedance in the presence of grazing flow. Recall that, in Sec. IV.A, the variation of the acoustic resistance predicted by the present model is divided into two different regions, sinking and linearly increasing. The acoustic resistance increases slowly or even decreases with the increase of grazing flow speed in the sinking region, whereas it goes up in proportion to grazing flow speed in the linearly increasing region. The sinking region for the variation of acoustic resistance is further demonstrated in Fig. 8. We can also see this sinking region from the experimental results of Kompenhans and Ronneberger.[11] It is believed that the presence of this sinking region is due to the influence of finite plate thickness. The present theory indicates that, when the Helmholtz number is constant, an increase in the normalized plate thickness results in the increase of the upper grazing flow speed limit of this sinking region where the acoustic resistance begins to increase linearly, as shown in Fig. 9. Figure 10 shows that, when the normalized plate thickness keeps constant, increasing the Helmholtz number produces the same results. With respect to the variation of

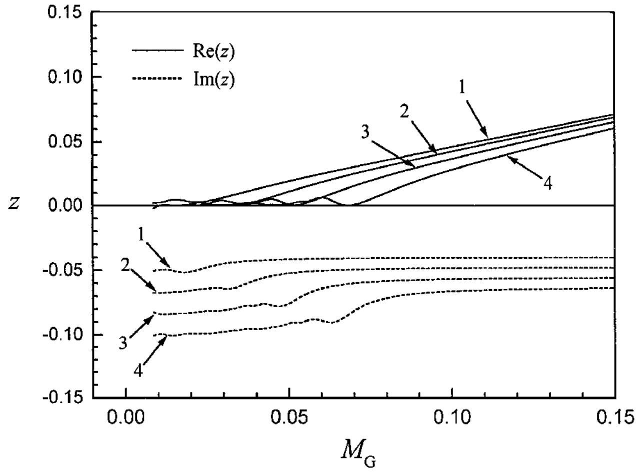


**Fig. 9 Acoustic impedance of a circular orifice plotted as a function of the grazing flow Mach number for different normalized plate thickness when $KR = 0.032$: 1) $T/R = 0.01$, 2) $T/R = 0.5$, 3) $T/R = 1.0$, and 4) $T/R = 1.5$.**

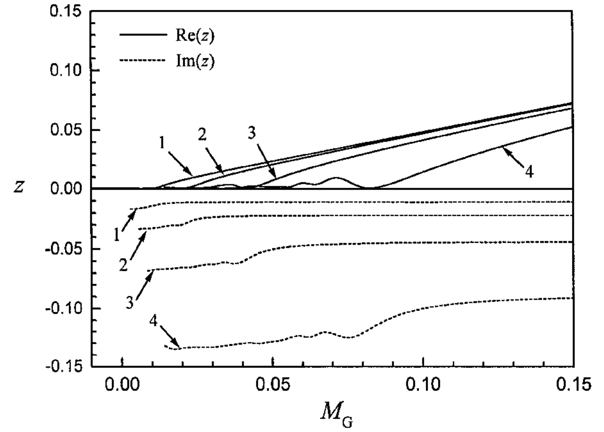


**Fig. 10 Acoustic impedance of a circular orifice plotted as a function of the grazing flow Mach number for different Helmholtz numbers when $T/R = 1.0$: 1) $kR = 6.5 \times 10^{-3}$, 2) $kR = 1.3 \times 10^{-2}$, 3) $kR = 2.6 \times 10^{-2}$, and 4) $kR = 1.3 \times 10^{-2}$.**

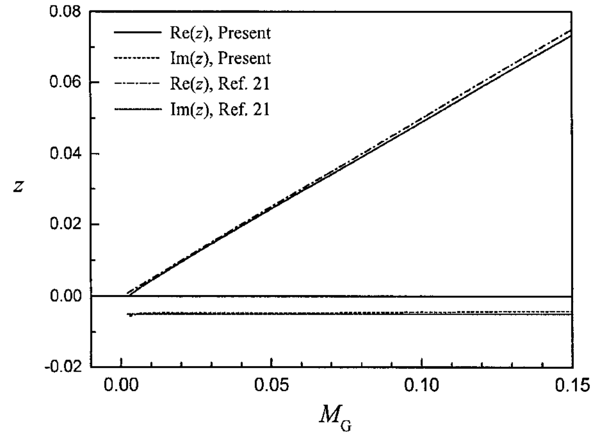


**Fig. 11 Comparison between the present model and that of Rice[21] when the normalized plate thickness and Helmholtz number are small; $T/R = 0.01$ and $kR = 3.7 \times 10^{-3}$.**

the acoustic reactance, it is demonstrated by the present theory that the increase of the plate thickness leads to the increase of the absolute value of the acoustic reactance (Fig. 9). Figure 9 also reveals a phenomenon of interest that, when the plate thickness is constant, the value of grazing flow speed at which the acoustic reactance becomes unchanged coincides with the upper grazing flow speed limit of the sinking region of the acoustic resistance. This coincidence can also be seen from the present experimental results in Figs. 3–6 and 8.

Note that Rice's[21] model does not include the influence of plate thickness, and so it predicts that the acoustic resistance linearly increases over the entire range of grazing flow speed. In Fig. 11, the present model is compared with Rice's model when both the normalized plate thickness and the Helmholtz number are very small. It is found that the two models are in surprisingly good agreement both for resistance and reactance, although they have adopted completely different approaches.

## V. Conclusions

It is well known that the presence of grazing flow has marked effect on the acoustic impedance of an orifice. In the present paper, this grazing flow effect has been theoretically and experimentally investigated.

Generally good agreement is obtained between the present theory and experiment. Further, the present theory and experiment are compared with two existing impedance models (Lewis and Garrison's[8] empirical model and Rice's[21] theoretical model). For acoustic resistance, they are in generally good agreement. With respect to the acoustic reactance, the calculations of both the empirical model and Rice's do not change with grazing flow speed. As we know, this result is contrary to experiment. By comparison, the present model gives a good prediction of the variation of the acoustic reactance at low grazing flow Mach number. However, at comparatively high grazing flow Mach number (about larger than 0.1 in the present study), the present model's predictions for the acoustic reactance is not as satisfactory as for the acoustic resistance. Only if more flow details in the vicinity of the orifice were known, could a more accurate calculation of the variation of the acoustic reactance be made.

The present investigation demonstrates that, in terms of the variation of the acoustic resistance, there are two different regions, sinking and linearly increasing. The acoustic resistance increases slowly or even decreases with the increase of grazing flow speed in the sinking region, whereas it goes up in proportion to grazing flow speed in the linearly increasing region. It is further shown that the presence of this sinking region is due to the influence of finite plate thickness. When the Helmholtz number is constant, an increase in the normalized plate thickness results in the increase of the upper grazing flow speed limit of this sinking region where the acoustic resistance begins to linearly increase. For the acoustic reactance, increasing plate thickness leads to the increase of its absolute value. It is also found that, in the limit situation of very small normalized plate thickness and Helmholtz number, the present and Rice's[21] models are in surprisingly good agreement both for resistance and reactance, although they have adopted completely different approaches.

It is very important to study what boundary condition is valid for the acoustic motion transmitting through a turbulent shear layer over an orifice. In the present investigation it is found that, compared with the particle displacement continuity boundary condition, the use of the particle velocity continuity boundary condition results in a great improvement of the theoretical predictions both for the acoustic resistance and reactance. Therefore, the present investigation indicates that it is appropriate to use the particle velocity continuity boundary condition rather than the displacement. This conclusion is only tentative because of the limitations of the present investigation. It is obvious that both further theoretical and experimental studies are needed to clarify this problem of great interest.

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