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MODAL PROPAGATION ANGLES IN DUCTS
WITH SOFT WALLS AND THEIR CONNECTION
WITH SUPPRESSOR PERFORMANCE

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Abstract

The angles of propagation of the wave fronts associated with duct modes are derived for a cylindrical duct with soft walls (acoustic suppressors) and a uniform steady flow. The angle of propagation with respect to the local coordinate (angle of incidence on the wall) is shown to be a better correlating parameter for the optimum wall impedance of spinning nodes than the previously used mode cut-off ratio. Both the angle of incidence upon the duct wall and the propagation angle with respect to the duct axis are required to describe the attenuation of a propagating mode. Using the modal propagation angles, a geometric acoustics approach to suppressor acoustic performance was developed. Results from this approximate method were compared to exact modal propagation calculations to check the accuracy of the approximate method. The results are favorable except in the immediate vicinity of the modal optimum impedance where the approximate method yields about one-half of the exact maximum attenuation.

Introduction

The angles of propagation for the wave fronts making up an acoustic mode are derived for ducts with soft walls (acoustic liner). These are the angles which the velocity vector normal to the local wave front makes with the three coordinate axes. The angles associated with the group velocity vector are also derived. This effort is an extension of the work of Ref. 1 in which both approximate and exact wave propagation angles were derived for a cylindrical duct with uniform steady flow. In Ref. 1, only the hardwall results were given and the emphasis was on the relationship of the propagation angles in the duct and the farfield radiation pattern. This paper will consider ducts with soft walls and the relationship of the propagation angles to acoustic liner performance.

In Refs. 2 to 4, the optimum acoustic impedance and the maximum possible attenuation for a mode were shown to be intimately connected to and correlated by the mode cut-off ratio. The mode cut-off ratio is a somewhat abstract parameter, particularly for soft wall ducts, and an explanation of its effectiveness as a correlation parameter was that it was connected to the angle of incidence upon the liner wall. This paper formulates this connection and shows that angle of incidence is the more fundamental parameter and in fact correlates the optimum impedance values better than does the cut-off ratio.

The maximum possible attenuation for a mode was also shown to be approximately correlated by the cut-off ratio. However, the collapse of these attenuation curves was not as good as desired and some large errors could occur particularly near mode cut-off when using this correlation. A simple geometric approach similar to that of Ref. 5 was tried. This approach considers that the number of encounters that the wave has with the soft wall is determined by the wave angle with respect to the duct axis and by the duct size. The acoustic power remaining after each encounter with the wall is related to angle of incidence and the wall impedance properties. When these two concepts are combined in a simple model, the acoustic power attenuation for the mode can be calculated. The resulting equation is shown to be exact for a rectangular duct without Mach number and for only slightly soft walls. This approximate method is extended to include cylindrical ducts and uniform duct flow. Results from the approximate model are compared with the exact attenuation results for several conditions including acoustic impedances up to the optimum for both rectangular and cylindrical ducts with flow.

The purpose of this paper is to provide the foundation for a simple geometric acoustics approach which may be useful for complex suppressor geometries and multimodal noise sources for which exact methods are inefficient and extremely difficult.

Symbols

- $c$: speed of sound, or the magnitude of the wave velocity vector normal to a plane wave front, m/sec
- $c_p$, $c_s$: components of vector $c$, $c_p = x$, $c_s = r$, m/sec
- $c_v$: resultant velocity vector, m/sec
- $D$: duct diameter, m
- $f$: frequency, Hz
- $H$: rectangular duct height, m
- $I_p$: axial acoustic intensity leaving end of suppressor, N/m sec
- $I_T$: axial acoustic intensity entering suppressor, N/m sec
- $I_x$: axial acoustic intensity of reflected wave off soft wall, N/m sec
- $I_k$: axial acoustic intensity of incidence wave in soft wall, N/m sec
- $k$: combined radial-circumferential wave number ($\phi/2\pi$), m$^{-1}$
- $k_r$: radial wave number, m$^{-1}$
- $k_\phi$: circumferential wave number, m$^{-1}$
- $L$: suppressor length, m
- $M$: Mach number of uniform steady duct flow
- $n$: lining mode lobe number
- $P$: acoustic pressure, N/m$^2$
- $r$: radial coordinate, m
- $x$: duct radius, m
- $\theta$: angle of incidence, m$^{-1}$
Development of the Propagation Angles

The initial development of the expressions for the propagation angles can proceed exactly as in Ref. 1. The wave equation in a circular duct with a steady flow can be expressed as

\[
(1 - \mu^2) \frac{\partial^2 \phi}{\partial t^2} - \frac{2M \partial^2 \phi}{\partial r \partial t} + \frac{\partial^2 \phi}{\partial r^2} + \frac{1}{r} \frac{\partial \phi}{\partial r} + \frac{1}{r^2} \frac{\partial^2 \phi}{\partial \phi^2} = \frac{1}{c^2} \frac{\partial^2 \phi}{\partial t^2}
\]

(1)

where \( \mu \) is the number of lobes for the spinning mode (the transverse order), \( M \) is the axial wave number, \( k_x \) is the axial wave number, \( \alpha \) is the circular frequency, \( r_o \) is the outer wall radius, \( \alpha_{n,m} \) is the eigenvalue of the \( n,m \) mode, and \( J_m \) is the Bessel function of the first kind and order \( m \).

Wave Numbers and the Phase Velocity

Propagation Angles

For brevity, the \( m \) subscripts will be deleted from here on, and it will be understood that a single (but quite general) mode is being considered. When Eq. (2) is inserted into Eq. (1) the result is,

\[
\left( \frac{\alpha}{r_0} \right)^2 + \frac{k_r^2}{M^2} \frac{k_t^2}{k_t} = \left( k - M k_x \right)^2
\]

(3)

The first term in Eq. (3) is a combined radial-transverse wave number squared which could be denoted by,

\[
\frac{\alpha}{r_0} = k_{r0}
\]

(4)

This combination of wave numbers is the cause of the trouble in defining some of the propagation angles in cylindrical ducts.

The \( i \)th propagation angle can be shown (for hardwall ducts),\(^1\) to be given by,

\[
\cos \Psi_i = \frac{k_i}{\sqrt{\sum_{i=1}^{\infty} k_i^2}} = \frac{k_i}{k_t}
\]

(5)

where \( i = r, \phi, \) or \( x \), and \( \Psi_i \) is the angle between the normal to the wave front and the \( i \)th coordinate axis, and the denominator of Eq. (5) was obtained from Eq. (3). The angles \( \Psi_r, \Psi_\phi \) and \( \Psi_x \) can be seen in Fig. 1. The vector \( \vec{c} \) is normal to the local wavefront. The other angle set \( \Psi_{i0} \) and the resultant vector \( \vec{c}_R \) due to additional convective effects also shown in Fig. 1 will be discussed later.

Before proceeding further, one must recognize that, due to the soft wall boundary condition, the quantities \( \alpha \) and \( k_x \) in Eq. (3) are complex numbers. Only the real parts of the wave number are related to propagation wave numbers for use in Eq. (5) while the imaginary parts represent damping terms. The axial wave number can be rewritten as,

\[
k_x = k(\tau - ie)
\]

(6)

and this along with Eq. (3) yields,

\[
\alpha + i\tau = \frac{-1 + \sqrt{1 - (1 - \mu^2)(\frac{\alpha}{r_0})^2}}{1 - \mu^2}
\]

(7)

and thus,
\[
\tau = \frac{1}{\sqrt{2}} \left[ 1 - \left( 1 - \mu^2 \right)^2 \cos 2\varphi + \sqrt{1 + (1 - \mu^2) \left( \frac{R}{\pi\eta} \right)^2 - 2(1 - \mu^2) \left( \frac{R}{\pi\eta} \right)^2 \cos 2\varphi} \right]^{1/2}
\]

where \( R \) and \( \varphi \) are the amplitude and phase of the complex eigenvalue \( m \) and the frequency parameter is,

\[
\eta = \frac{\omega\rho_o}{\pi c} = \frac{10}{c}
\]

Equation (8) is obtained by solving for the imaginary part of Eq. (7).

To obtain the angle of incidence at the liner wall \( (\theta_0) \) in a simple form, the assumption is made that the combined radial-transverse complex wave number can be split as follows,

\[
k^2 = k^2_r + k^2_\theta = \left( \frac{\omega}{\tau_0} \right)^2 \cos^2 \theta + \left( \frac{\omega}{\tau_\eta} \right)^2 (\cos \theta)^2
\]

This assumption was used in Ref. 1 for hardwall ducts and was found to provide adequate results near the outer wall when compared to an exact solution using Hankel functions. The real part of \( k_r(k_r, \eta) \) for use in Eq. (5) can then be obtained as,

\[
k_r = \text{Real} \left[ \sqrt{\left( \frac{\omega}{\tau_0} \right)^2 - \left( \frac{\omega}{\tau_\eta} \right)^2} \right] = \frac{\mu}{\pi \eta} \text{Real} \left[ \frac{\omega}{\tau} \right]^{1/2}
\]

The propagation angles will be calculated only at the outer wall \( (r = r_0) \) where the approximations used here are reasonably accurate. Let,

\[
k_{r, \eta}(r_0) = \frac{\mu}{\pi \eta}
\]

thus

\[
\beta = \frac{1}{\sqrt{2}} \left( \frac{R^2 \cos 2\varphi - \mu^2 + \sqrt{R^4 - 2\mu^2 R^2 \cos 2\varphi + \mu^2}}{1 + \mu^2} \right)^{1/2}
\]

The circumferential wave number (at the outer wall) as seen from Eq. (10) is,

\[
k_\theta = \frac{m}{\tau_0} = \frac{\mu k}{\pi \eta}
\]

which is real unlike the \( k_r \) and \( k_\theta \).

The real parts of the wave numbers can now be used in Eq. (2) to obtain,

\[
\cos \theta_X = \frac{\tau}{\sqrt{\tau^2 + \left( \frac{\beta}{\pi \eta} \right)^2 + \left( \frac{m}{\pi \eta} \right)^2}}
\]

\[
\cos \theta_r = \frac{\beta}{\pi \eta \sqrt{\tau^2 + \left( \frac{\beta}{\pi \eta} \right)^2 + \left( \frac{m}{\pi \eta} \right)^2}}
\]

\[
\cos \theta_\theta = \frac{m}{\pi \eta \sqrt{\tau^2 + \left( \frac{\beta}{\pi \eta} \right)^2 + \left( \frac{m}{\pi \eta} \right)^2}}
\]

**Group Velocity Propagation Angles**

Equations (15) to (17) describe the orientation of the wave front at the outer wall of the circular duct and can be seen to include the duct Mach number through the propagation coefficient \( \tau \). There is an additional Mach number effect as shown in Fig. 1 which causes a drift in the axial velocity and produces a resultant vector \( (\mathbf{c}_r, \mathbf{c}_\theta) \) of propagation which is not normal to the wavefront. This vector is associated with the group velocity and the angles of propagation associated with \( \mathbf{c}_r \) and \( \mathbf{c}_\theta \) will be of use in a later discussion. These angles can be derived from the geometry shown in Fig. 1 using the fact that the vector \( \mathbf{c}_r \) alters only the \( x \) component of velocity but not \( c_r \) or \( c_\theta \). These angles can be expressed as,

\[
\cos \phi_X = \frac{\mu + \cos \varphi_X}{\sqrt{1 + \mu^2 + 2\mu \cos \varphi_X}}
\]

\[
\cos \phi_r = \frac{\cos \varphi_r}{\sqrt{1 + \mu^2 + 2\mu \cos \varphi_X}}
\]

\[
\cos \phi_\theta = \frac{\cos \varphi_\theta}{\sqrt{1 + \mu^2 + 2\mu \cos \varphi_X}}
\]

**Correlation of Optimum Impedance with Incidence Angle**

The angle of incidence on the soft wall (Eq. (16)) was used to correlate some of the optimum impedance calculations reported in Ref. 4. These calculations were made for a cylindrical duct with a 1/7th power boundary layer and a uniform core flow outside of the boundary layer. Standard modal solutions were used within the core flow and were coupled to the wall impedance by a Runge-Kutta integration through the boundary layer. These correlations are shown in Figs. 2 and 3. These calculations include several mode numbers \( (n) \) and the entire range of radial modes from well propagating to near cut-off. Comparison of Figs. 2 and 3 with Figs. 12 and 13 of Ref. 4 show that the angle of incidence correlates the modal optimum impedance somewhat better than the mode cut-off ratio did in Ref. 4. Using cut-off ratio the first radial mode for \( m = 3 \) showed some deviation from the correlation while \( m = 9 \) and 16 had quite pronounced deviations. This deviation does not appear when incidence angle is used. This improvement can be most easily explained by using the hardwall incidence angle given in Ref. 1 which was

\[
\cos \varphi_X \bigg|_{r_0} = \frac{\sqrt{1 - \mu^2} \sqrt{1 - (\frac{3}{\mu})^2}}{\sqrt{1 - (1 - \mu^2) \left( \frac{3}{\mu} \right)^2}}
\]

where the cut-off ratio \( \mu \) is given by
For higher radial modes ($a \gg m$) or for small $m$ the term $\sqrt{1 - (m/c)^2}$ approaches unity. Equation (21) then shows that for constant Mach number $\pi$ (as used in Figs. 2 and 3) the angle of incidence ($\Phi_T$) is a function only of cut-off ratio and either parameter is sufficient to correlate the optimum impedance. However, for large $m$, and low radial order $a = m$, the term $\sqrt{1 - (m/c)^2}$ in Eq. (21) influences the incidence angle and causes scatter in the calculation as shown in Ref. 4. At sufficiently high lobe numbers the splitting of the radial angle begins to occur and the mode cut-off ratio is still considered the resultant incidence angle.

An alternate correlation of one set of calculations ($e = 0.003$) in Fig. 5 is shown in Fig. 6. Here the resultant incidence angle ($\Phi_T$) is used rather than $\Phi_X$. Recall that $\Phi_T$ contains the axial velocity drift effect and is associated with the group velocity. Some of the scatter in the points nearer cut-off (small incidence angles) may have been reduced by using $\Phi_T$ and the numerical value of incidence angle indicates more of a normal incidence than does $\Phi_X$. The $\Phi_X$ seen to have more physical significance than the $\Phi_T$. For example in Ref. 1 it was found that $\Phi_X = 90^\circ$ at mode cut-off while $\Phi_T$ does not.

From the above discussion it is thus proposed that the angle of incidence of the wave fronts on the lined wall is a more fundamental and obviously more easily visualized parameter than the mode cut-off ratio for correlating liner optimum impedance. However, the mode cut-off ratio is still considered to be an extremely useful parameter for liner design purposes with the previous discussion serving to explain the reason for its effectiveness and pointing out its limitations.

### Rectangular Two Dimensional Duct, $N = 0$

Figure 7 shows a rectangular duct without flow with a transverse mode propagating and bouncing between the soft walls. The angle of incidence between the velocity vector normal to the wall and the $y$ coordinate is $\Phi_Y$ while the angle of the wave with respect to the duct axis is $\Phi_X$. The axial distance between encounters of the wave and the soft wall is,

$$\Delta x = \frac{H}{\tan \Phi_X}$$  \hspace{1cm} (23)

Where $H$ is the duct height. The total number of wall bounces in a duct of length $L$ is then

$$N = \frac{L}{\Delta x} = \frac{L}{H} \tan \Phi_X$$  \hspace{1cm} (24)

The ratio of axial flux after a wall bounce compared to the incident wave is given by

$$\frac{I_X^-}{I_X^+} = \left(1 + \frac{X}{\theta^2 + X^2}\right) \cos 2\Phi_Y - 2 \cos \Phi_Y + 1$$

$$\frac{I_X^+}{I_X^-} = \left(1 + \frac{X}{\theta^2 + X^2}\right) \cos 2\Phi_Y + 2 \cos \Phi_Y + 1$$  \hspace{1cm} (25)

Where $\theta$ and $X$ are the wall specific acoustic resistance and reactance. Note that Eq. (25) involves only the wall properties and the angle of incidence. Equation (25) was derived in the same way as the absorption coefficient is conventionally derived. A plane wave is assumed to be incident upon a plane absorber and the angle of reflection is equal to the angle of incidence. The absorption coefficient is $1 - I_X/I_X^+.

After the $N$ bounces in the duct the ratio of the final to initial axial acoustic intensity is

$$\frac{I_X}{I_X} = \left(\frac{I_X^-}{I_X^+}\right)^N$$  \hspace{1cm} (26)
Since only one mode is considered here the attenuation can be obtained from the intensity ratio as,

$$\Delta \alpha = 10 \log \frac{I_F}{I_L} + 10 \log \frac{I_0}{I_X}$$

(27)

Now consider the often studied problem of nearly hard walls which is quite a good approximation to the exact physical acoustics solutions except in the vicinity of the optimum impedance. With only slightly soft walls,

$$\frac{I_0}{I_X} = 1 - \frac{40}{\cos \varphi_y (\theta^2 + \chi^2)}$$

(28)

and

$$\frac{I_0}{I_X} = \frac{40}{\log \sin \psi_y (\theta^2 + \chi^2)}$$

(29)

Equations (28) and (29) were derived using Eq. (25) with the assumption that impedance is very large. To the first order of approximation, the axial propagation angle and the incidence angle for the \(j\)th transverse mode are given by

$$\tan \varphi_y = \frac{1}{\eta} \sqrt{1 - (\frac{1}{\eta})^2}$$

(30)

and

$$\cos \psi_y = \frac{1}{\eta}$$

(31)

Equations (15) and (16) were used with \(y\) replacing \(r\) and with \(m = 0\). If a three dimensional rectangular duct calculation is desired with a transversely spinning wave considered, \(m\) could be retained and the \(\psi\) coordinate would just replace the \(\varphi\) coordinate.

**Cylindrical Duct with Uniform Flow**

For a cylindrical duct with uniform steady flow the results of the previous section can be used directly as a first approximation. Of course \(\varphi_y\) must be used instead of \(\psi_y\) in Eq. (24). As shown in Ref. 1, the approximations leading to Eqs. (15) to (17) are valid only near the outer wall because of curvature effects in the cylindrical duct. Equation (24) represents the physics of reflection and absorption which occur at the wall and it should thus remain valid. However, the number of encounters of the wave with the wall as expressed by Eq. (32) involves the propagation process throughout the entire depth or radius of the duct. For a rectangular duct \(\psi_x\) does not change with distance from the wall and Eq. (33) is thus valid. For a cylindrical duct, as discussed in Ref. 1, \(\psi_x\) in a function of radius and is always equal to 90° at the duct centerline. Thus using the value of \(\psi_x\) at the wall, as done here, will underestimate the number of bounces somewhat, and the approximate calculation of attenuation will be less than the exact attenuation. Some integrated value of \(\psi_x\) must be used rather than the value at the wall, but this has not yet been accomplished.

**Rectangular Duct with Uniform Flow**

With a finite Mach number in the duct Eq. (27) remains valid but the number of encounters of the wave with the wall (Eq. (24)) and the ratio of the incident to reflected axial intensity (Eq. (25)) must be modified. The number of bounces must now include the axial propagation angle associated with the group velocity \(\psi_x\). Equation (24) thus becomes,

$$N = \frac{1}{H} \tan \psi_x$$

(33)

where Eq. (18) is used for \(\psi_x\).

The ratios of axial intensities were derived as in the previous section with the addition that uniform flow must be considered in the wave equation and the continuity of displacement must be used at the soft wall. The derivation proceeds exactly as in Ref. 7, except that the relation between the wave numbers and the propagation angle was incorrectly expressed in Ref. 7 (effect of \(M\) on angles ignored). The axial intensity ratio (incident to reflected) is,

$$\frac{I_0}{I_X} = \left[ (\theta^2 + \chi^2) \cos^2 \psi_y (1 + M \sin \psi_y)^2 \right]$$

$$- 20 \cos \psi_y (1 + M \sin \psi_y) + 1]$$

$$+ 20 \cos \psi_y (1 + M \sin \psi_y) + 1]$$

(34)

Note that Eq. (34) reduces to Eq. (25) when \(M = 0\).

**Comparison of Approximate and Exact Attenuations**

A series of exact single mode propagation calculations were performed as in Ref. 8. These calculations were made for both rectangular and cylindrical ducts. In each case a constant damping coefficient \(\delta\) was used and the propagation coefficient \(\psi_x\) was varied to produce a closed loop in the wall impedance plane (see Fig. 4, Ref. 8). Several points were used along this constant atten-
The ratio of the attenuation by the approximate calculation using the geometrical acoustics approach to that of the exact physical acoustics calculations are shown in Fig. 8 for zero steady flow. The abscissa represents the degree of attenuation used for the calculations with the value of unity representing the maximum possible attenuation for this case, the second symmetric mode. The symbols represent the average attenuation for the several points on a given exact damping contour while the bar represents the spread of these calculations. The averages are not unique, particularly where a large spread is present, since they depend upon the particular choice of points along the damping contour. Several observations can be made from Fig. 8. As damping is increased (far from the optimum impedance) the approximate and exact attenuations are seen to be in better agreement with very little scatter. This agrees with the comments made pertaining to the limiting Eq. (32) which states that the approximate equation is exact far from the optimum impedance. Up to about one-half of the maximum attenuation the average attenuation around a contour is very good although considerable scatter occurs. As the optimum impedance is approached the approximate calculation gives only about one-half the attenuation as the exact calculation. At first glance, the 50 percent error possible here both at intermediate and high damping may appear unacceptable. However, the upper half of the damping range represents a very small portion of the impedance plane which would be very hard to realize precisely in a real application. Furthermore, in a multimodal excitation situation, where approximate techniques would be most valuable, a given wall impedance would be near to only a small fraction of the modal optimum and errors involving only a few modes (assuming near equipartition of energy) would not seriously affect the final result. Also the apparently correct averaging, where considerable scatter occurs, would help the final multimodal result. The reason for the scatter observed will become more clear in a later discussion.

In Fig. 9 results are shown for the same mode as in Fig. 8 except that there is a steady inlet flow of $M = 0.4$. The results are much the same as in Fig. 8 with the low damping points showing agreement between approximate and exact attenuation techniques would be most valuable, a given wall impedance would be near to only a small fraction of the modal optimum and errors involving only a few modes (assuming near equipartition of energy) would not seriously affect the final result. Also the apparently correct averaging, where considerable scatter occurs, would help the final multimodal result. The reason for the scatter observed will become more clear in a later discussion.

To study the scattering of the approximate attenuation calculations, a complete scan over all values of propagation coefficient $\gamma$ was made along a constant damping coefficient contour of $\sigma = 0.02$. The approximate to exact attenuation ratio is shown in Fig. 10 along with the incidence angle $\Phi_y$ and the magnitude of the eigenvalue $R$. This scan involves several closed loops in the impedance plane and of course encompass several modes. In Fig. 10(a) the scatter in approximate attenuation is seen to be a smooth oscillatory function. The scatter is seen to be small for the higher modes (mode number noted near top) and increases for the lower modes. The value of damping used for this calculation could not be attained by the first mode. There is no error at the border of each mode and the average within a mode can be seen to be about unity in the attenuation ratio. If a cross plot were shown only for the lower modes (c) it would be seen that the scatter is cyclic with eigenvalue amplitude. The scatter seems to be least for the higher, near cut-off modes nearer normal incidence (small $\Phi_y$) and increase with angle of incidence. The generality of these results have not been tested.

### Cylindrical Duct

Figure 11 shows comparisons of approximate and exact attenuations for a cylindrical duct for the second radial of a seven lobed spinning mode. The scatter is seen to be extremely small when compared to Figs. 8 and 9. This is probably due to the high frequency ($\nu = 20$ versus 5) as compared to the rectangular duct results. The same roll-off at high damping is seen here as in the rectangular duct. A difference occurs in low damping where the asymptote being about 0.9 rather than unity as previously seen. This is due to the underestimation of the number of encounters of the wave with the wall due to using the axial propagation at the wall rather than an integrated value over a duct. Recall that axial propagation angle is constant over a rectangular duct but increases to 90 degrees at the center of a cylindrical duct.

### Concluding Remarks

The angle of incidence has been shown to be a more fundamental and more illuminating parameter than the mode cut-off ratio or the equivalent modal optimum impedances. The liner performance has been shown to involve both the angle of incidence and the axial angle of propagation along with the usual parameters of wall impedance and duct geometry. The use of these propagation angles in a ray acoustics propagation approach shows great promise although some additional effort is still needed to perfect this approach. If the ray acoustics approach can be perfected it would be an extremely powerful tool to solve problems in soft wall duct configurations which can not be presently handled except by costly numerical procedures. An example is refraction effects in variable area ducts with flow provided that the reflection effects are not dominant. Another important point is that the expression of modal properties in terms of propagation angles is nearly equivalent to an expression in terms of cut-off ratio. The same statements can also be made in terms of propagation angles were made in Refs. 2 to 4 in terms of cut-off ratio. Modes with nearly equivalent propagation angles will behave similarly in an acoustic liner. A group of modes with similar propagation angles can thus be treated as an entity without the need to distinguish between the individual modes. This is espe-
cially significant if a large number of propagating modes are available and carrying significant acoustic power. If a direct measurement of acoustic power in the duct can be made as a function of increments in the propagation angles, then a significant simplification will be obtained. It is possible that propagation angles may be easier to measure than cut-off ratios, although either one would be sufficient. This problem is currently being studied.

References


Figure 1. - Sketch of the propagation angles and vectors in the duct.

Figure 2. - Correlation of optimum resistance with wave incidence angle on the wall, \( \eta = 15, M_o = -0.4, \varepsilon = 0.05 \).
Figure 3. - Correlation of optimum reactance with wave incidence angle on the wall, \( \eta = 15, M_0 = -0.4, \varepsilon = 0.05 \).

Figure 4. - Effect of boundary layer thickness upon optimum resistance-incidence angle correlation, \( \eta = 15, M_0 = -0.04 \).
Figure 5. - Effect of boundary layer thickness upon optimum reactance-incidence angle correlation, \( \eta = 15, M_0 = -0.4 \).

Figure 6. - Correlation of optimum reactance with resultant incidence angle, \( \varepsilon = 0.005, \eta = 15, M_0 = -0.4 \).
Figure 7. - Sketch of wave front and propagation angles in a rectangular duct.

Figure 8. - Comparison of approximate and exact calculated attenuations over the complete attenuation range, rectangular duct, second symmetric mode, $M = 0$, $\eta = 5$. 
Figure 9. - Comparison of approximate and exact calculated attenuations over the complete attenuation range, rectangular duct, second symmetric mode, $M = -0.4$, $\eta = 5$. 
Figure 10. - Propagation calculations made over entire range of propagation coefficient, $\tau$, rectangular duct, $M = 0$, $\sigma = 0.02$, $\eta = 5$. 
Figure 11. - Comparison of approximate and exact calculated attenuations over the complete attenuation range, cylindrical duct, seven lobed spinning mode, second radial, $M = 0$, $\eta = 20$. 

![Graph showing the ratio of approximate to exact attenuations](image-url)