On Circular Pipe Wall Vibratory Response Excited by Internal Acoustic Fields

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Internal acoustic fields in pipes as generated by control valves, for example, interact with the pipe walls such that very large drops in transmission loss are experienced at a series of discrete frequencies, the coincidence frequencies. This paper develops the analysis for the calculation of the pipe vibration amplitude at the coincidence frequencies so that the radiation external to the pipe can be estimated. External acoustic loading and material damping are considered. The analysis was applied to experiments which were performed on a somewhat idealized test fixture. Combining the results of the analysis with the test results permitted the estimation of the material damping for the particular material used. Results show that structural damping for the low carbon steel varies from 0.04 at high frequency to a high of 0.20 at the lowest frequency which agrees fairly well with the results from other research on flat steel plates. The rather large values of the estimated damping coefficients may well be caused by the visco-elastic damping materials used at the ends of the pipe's tee section.

Introduction

Research on control valve and regulator noise and its reduction has been in progress at The Pennsylvania State University's Noise Control Laboratory for some seven years with support from the National Science Foundation and several control valve manufacturers [1 through 13]. The noise generated downstream of the throttling element of control valves forms a propagating acoustic field in the downstream piping which in turn interacts with the pipe wall vibratory modes. The pipe wall vibrations result in the development of an acoustic field outside of the pipe which radiates into the surrounding space in a manner which depends on the pipe geometry and the external acoustic environment. The acoustic levels of this external field can become very high for large valves and high pressure dops, resulting in acoustic environments in violation of the OSHA noise rules.

The valve manufacturers have developed empirically based noise prediction techniques which are acceptably precise for the valve sizes tested but are not sufficiently reliable for the larger valve sizes or new configurations.

The research at The Pennsylvania State University has dealt with the various aspects of the valve noise prediction problem as reported in the references cited. The topic of this paper is the transmission of the noise through the pipe walls and the radiation into the surrounding medium. Work along similar lines has been reported by Bull and Norton [14] and Fagerlund [15] who uses a statistical energy approach.

The thrust of the work reported here is to start from fundamental principles recognizing that the transmission of the acoustic energy from inside the pipe is at a maximum at the coincidence conditions between the pipe's internal acoustic field and the pipe wall vibratory modes as developed at The Pennsylvania State University by Walter [11,21,22]. Both the radiation loading due to the external acoustic field and the pipe's material damping are considered in the formulation. The results reported are but one of several steps necessary in the development of a fundamentals based valve noise prediction technique which is the thrust of our research.

Analytical Model of the Acoustic Field—Pipe Vibratory Mode Coincidence Conditions and the Resulting Radiation

The development of this analytical model of the pipe wall motion resulting from an internal acoustic field is presented in five parts:

1 A mathematical and physical description of the wave propagation inside of the pipe.

2 A mathematical and physical description of the wave propagation outside of the pipe.

3 Mathematical development for the acoustic loading of the external pipe wall.

4 Analytical description of the pipe wall response in the presence of the internal damping of the pipe material and the external acoustic loading.

5 Solution method of the simultaneous set of partial differential equations for a specific ideal case and comparison with experimental data.

1 A Mathematical and Physical Description of the Wave Propagation Inside of the Pipe. In setting up the wave equation in cylindrical coordinates, we have assumed that linear acoustic conditions will exist. The assumption is open

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to question since acoustic intensities in excess of 130 dB exist downstream of typical conventional valves. Actually, only the (0,0) plane wave mode will experience progressive wave steepening, whereas the dispersive nature of all the other modes will result in the retention of the original spectral content.

A second assumption is that we are dealing with anechoically terminated, thus effectively infinitely long, constant cross section circular pipes.

From the general wave equation in cylindrical coordinates with uniform axial flow, equation (1)

$$\frac{\partial^2 p}{\partial r^2} + \frac{1}{r}\frac{\partial p}{\partial r} + \frac{1}{r^2}\frac{\partial^2 p}{\partial \theta^2} + (1-\mathrm{M}^2)\frac{\partial^2 p}{\partial x^2} = \frac{1}{C^2}\frac{\partial^2 p}{\partial t^2} + \frac{2\mathrm{M}}{C}\frac{\partial^2 p}{\partial x \partial t} \tag{1}$$

where

p is the complex pressure
r is the radial coordinate
θ is the circumferential coordinate
x is the axial coordinate
C is the speed of sound
t is time
M is the axial convection Mach number.

Through the well-known separation of variable approach, we find the general solution in the form of equation (2).

$$P_{mn} = \bar{A}_{mn} Z_m(k_{mn} r) e^{j(\omega t + k_x x + m\theta)} \tag{2}$$

where

k_{mn} is the radial wave number for circumferential mode m and radial node n,
k_x is the axial wave number.

For forward wave propagation inside the pipe, the term Z_{mn} is in the form of Bessel functions of the first kind of order m. Thus, equation (2) becomes equation (3).

$$P_{mn} = \bar{A}_{mn} J_m(k_{mn} r) e^{j(\omega t - k_x x + m\theta)} \tag{3}$$

We shall assume that we are dealing with a hard-walled duct so that the acoustic velocity at the internal wall must be zero. (Note that we have shown in an earlier study [11] that the pipe wall vibrations resulting from the acoustic field do not significantly affect the internal acoustic field.) This radial boundary condition is given by the solution of equation (4).

$$\frac{\partial}{\partial r} J_m(k_{mn} r)/_{r=R} = 0 \tag{4a}$$

or

$$k_{mn} J'_m(k_{mn} R) = 0 \tag{4b}$$

If we let γ_{mn} be the eigenvalues that satisfy $J'_m(\gamma_{mn}) = 0$, we have

$$k_{mn} = \frac{\gamma_{mn}}{R} \tag{5}$$

We can now set up the dispersion relationship, relating the axial wave number k_x to the frequency ω and the parameter k_{mn} so that

$$k_x = \frac{-M\left(\frac{\omega}{c}\right) \pm \sqrt{\left(\frac{\omega}{c}\right)^2 - (1-\mathrm{M}^2)k_{mn}^2}}{1-\mathrm{M}^2} \tag{6}$$

The subscripts m and n refer to the circumferential and radial order of the higher-order acoustic modes that will "spiral" down the duct. This concept is well known and has been explained in prominent text books [16,17,18] and many articles in the literature. Three conditions are of interest.

Case I: If $\omega/c < (1 - \mathrm{M}^2)^{1/2} k_{mn}$, k_x is complex so that from equation (3), the particular mode will decay with x.

Case II: If $\omega/c = k_{mn}$, so that k_x becomes zero, the pressure field inside the pipe is a function of r, θ, and t, and is constant in the x direction. Thus, the field spins circumferentially and does not propagate. This is commonly called the cut-off condition.

Case III: If $\omega/c > (1 - \mathrm{M}^2)^{1/2} k_{mn}$, so that k_x is real, we find that the pressure field propagates unattenuated down the pipe with a velocity, termed the group velocity, C_g, which is equal to $\partial\omega/\partial k_x$ and spins with an angular velocity $(d\theta/dt)_{x=\text{const}} = \pm\ \omega/m$. The phase velocity $(dx/dt)_{\theta=\text{const}}$ becomes ω/k_x.

We have, therefore, established an expression for the acoustic pressure at the pipe wall for values of x and θ as a function of frequency ω for the various (m,n) modes.

2 A Mathematical and Physical Description of the Wave Propagation Outside of the Pipe. The solution to equation (1) is again of the form (2), but

$$Z_m(k_{mn} r) = H_m^{(2)}(k_{mn} r) \tag{7}$$

and the pressure for the m,n th mode becomes

$$P_{mn} = \bar{A}_{mn} H_m^{(2)}(k_{mn} r) e^{j(\omega t - k_x x \pm m\theta)} \tag{8}$$

We shall be dealing with far field conditions only so that $k_{mn} r >> m$, so that

$$H_m^{(2)}(k_{mn} r) \approx \sqrt{\frac{2}{\pi k_{mn} r}}\, e^{j\left(-k_{mn} r + \frac{2m+1}{4}\pi\right)} \tag{9}$$

Thus

$$P_{mn} = \bar{A}_{mn}\sqrt{\frac{2}{\pi k_{mn} r}}\, e^{j\left(-k_{mn} r + \frac{2m+1}{4}\pi + \omega t - k_x x \pm m\theta\right)} \tag{10}$$

Pressure surfaces of constant phase then are given by

$$r \approx \frac{1}{k_{mn}}\left[k_x x - \frac{(2m+1)\pi}{4} - \omega t\right] \tag{11}$$

Since $m/k_{mn} << 1$, equation (11) represents spiraling cones around the (x) axis and the angle (δ) with the x axis is given by

$$\tan^{-1}\sqrt{\frac{k^2}{k_{mn}^2} - 1}.$$

Interestingly, this angle (δ) also is the angle between the normal to the pipe surface and the direction of propagation. Thus

$$k_{mn} = k\cos\theta,\ k_x = k\sin\theta$$

so that constant k_{mn} represents the projection of wave number $\bar{k}$ in the r direction and k_x represents its projection in the x direction.

3 The Damping Effect (Acoustic Loading) of the Pipe Due to the Radiation Outside of the Pipe. The pipe wall as a result of the internal acoustic field, which in turn is caused by the throttled flow through the valve orifice, will vibrate with a complicated vibrational pattern with the radial velocity a function of both circumferential angle θ and axial distance x. Since the air velocity on the outside surface of the pipe wall is equal to the wall velocity, the air mass will be set into motion. The reaction of this acoustic radiation will increase the effective mass of the pipe wall and will also add to the damping because of the phase changes that take place as a result of compressibility effects.

From a study of the radiation from a radially vibrating sphere (in its pulsating mode only), we can see that for the case of the sphere's radius a much smaller than the wave length λ ($a >> \lambda$), the additional mass of the air (about three times the volume of the sphere) dominates the vibration with the damping effect being negligible. On the other hand, for high frequencies ($ka << 1$), the effective mass increment becomes very small and the effective radiation damping becomes significant, meaning that energy dissipation by the surrounding air becomes an important factor [19].

The analysis for the pipe is far more complicated because of the higher order vibrational bending modes that are excited and which are of concern to us. Yet the simplified analysis for the pulsating sphere does shed some light on what we might expect.

The pipe wall vibration velocity V_{mn} in the m,n^{th} mode will be given by equation (12)

$$V_{mn} = V_0 \cos k_x x \cos m\theta \, e^{i\omega t} \tag{12}$$

The general potential field of the acoustics generated by the pipe wall vibration outide of the pipe is given by equation (13)

$$\phi_{mn} = \bar{B}_{mn} H_m{}^{(2)}(k_{mn} r)\, e^{\,j(\omega t - k_x x + m\theta)} \tag{13}$$

The air velocity at the wall has to be equal to the wall velocity—a boundary condition—so that

$$V_{mn} = -\left.\frac{\partial \phi}{\partial r}\right|_{r=R} \tag{14}$$

Thus, equation (13) becomes

$$\phi_{mn} = \bar{B}_{mn} H_m{}^{(2)}(k_{mn} r) \cos k_x x \cos m\theta \, e^{i\omega t} \tag{15}$$

and

$$V_{mn} = -\bar{B}_{mn} k_{mn} H_m{}^{(2)\prime}(k_{mn} R) \tag{16}$$

The sound radiated from a vibrating cylinder is determined by its radiation impedance $\bar{Z}_{mn}$ which is defined by (17)

$$\bar{Z}_{mn} = \left(\frac{P_{mn}}{V_{mn}}\right)_{r=R} = -\frac{i\rho C k H_m{}^{(2)}(k_{mn} R)}{k_{mn} H_m{}^{(2)\prime}(k_{mn} R)} \tag{17}$$

By definition, the Hankel function is defined in terms of Bessel and Neumann functions as shown in equation (18)

$$H_{mn}{}^{(2)}(k_{mn} R) = J_m(k_{mn} R) - iN_m(k_{mn} R) \tag{18}$$

so that the radiation impedance Z_{mn} can be expressed as equation (19)

$$Z_{mn} = \frac{2\rho C k R}{\pi k_{mn}{}^2 R^2 \{[J'_{mn}(k_{mn} R)]^2 + [N'_m(k_{mn} R)]^2\}} - i\frac{\rho C k R [J_m(k_{mn} R) J'_m(k_{mn} R) + N_m(k_{mn} R) N'_m(k_{mn} R)]}{k_{mn} R \{[J'_{mn}(k_{mn} R)]^2 + [N'_m(k_{mn} R)]^2\}} \tag{19}$$

or in abbreviated form

$$Z_{mn} = X + iY$$

The acoustic pressure exerted on the pipe wall is then given by equation (20)

$$P_{mn} = \bar{Z}_{mn} \cdot V_{mn} = (X_{mn} + iY_{mn})\, V_{mn} \tag{20}$$

4 Analytical Description of the Pipe Wall Response in the Presence of the Pipe Wall Material Damping and the External Acoustic Loading. The derivation of the elastic response of an infinitely long, thin cylindrical shell to internal and external pressure fields was based on the formulation given by E. M. Frymoyer [20] and further developed by Walter [11].

The equations of motion are derived by applying Hamilton's minimum total energy principle. The Flügge thin shell approximations are used and allow the expansion of the displacement vector $\mathbf{u}_i$ into a Taylor series in the thickness coordinate. The assumptions listed next reflect this approach.

The coordinate system for the equations is given in Fig. 1.

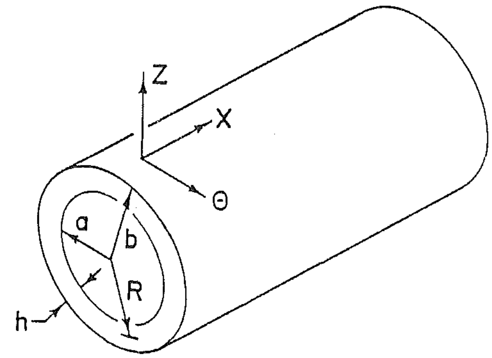

Fig. 1 Coordinate system for the shell

(a) The thickness of the pipe shell is small compared with other dimensions.

(b) Strians and displacements are sufficiently small so that quantities of second and higher order magnitude may be neglected compared with the first order terms.

(c) The transverse normal stress is small compared with the other normal stresses and may be neglected.

(d) Normals to the undeformed middle surface remain straight and normal to the deformed middle surface, and suffer no extension.

(e) The material is linearly elastic, homogeneous, and isotropic.

(f) Shear deformations and rotary inertia effects are neglected.

The three dynamic equilibrium equations are as follows:

$$\frac{\partial^2 u}{\partial x^2} + \frac{(1-\upsilon)}{2R2}\left[1 + \frac{h^2}{12R^2}\right]\frac{\partial^2 u}{\partial \theta^2} - \frac{\rho}{E'}\frac{\partial^2 u}{\partial t^2} + \frac{(1+\upsilon)}{2R}\frac{\partial^2 v}{\partial x \partial \theta} + \frac{1}{R}\left[\upsilon\frac{\partial w}{\partial x} - \frac{h^2}{12}\frac{\partial^3 w}{\partial x^3}\right] + \frac{(1-\upsilon)}{24R^3}h^2\frac{\partial^3 w}{\partial x \partial \theta^2} = 0 \tag{21}$$

$$\frac{(1+\upsilon)}{2R}\frac{\partial^2 u}{\partial x \partial \theta} + \frac{(1-\upsilon)}{2}\left[1 + \frac{h^2}{4R^2}\right]\frac{\partial^2 v}{\partial x^2} + \frac{1}{R^2}\frac{\partial^2 v}{\partial \theta^2} - \frac{\rho}{E'}\frac{\partial^2 v}{\partial t^2} + \frac{1}{R^2}\frac{\partial w}{\partial \theta} - \frac{(3-\upsilon)}{24}\frac{h^2}{R^2}\frac{\partial^3 w}{\partial x^2 \partial \theta} = 0 \tag{22}$$

$$\frac{1}{R}\left[\upsilon\frac{\partial u}{\partial x} - \frac{h^2}{12}\frac{\partial^3 u}{\partial x^3}\right] + \frac{h^2}{24}\frac{(1-\upsilon)}{R^3}\frac{\partial^3 u}{\partial x \partial \theta^2} + \frac{1}{R^2}\frac{\partial v}{\partial \theta} - \frac{h^2}{24R^2}(3-\upsilon)\frac{\partial^3 v}{\partial x^2 \partial \theta} + \frac{w}{R^2}\left[1 + \frac{h^2}{12R^2}\right] + \frac{h^2}{6R^4}\frac{\partial^2 w}{\partial \theta^2} + \frac{h^2}{12}\frac{\partial^4 w}{\partial x^4} + \frac{h^2}{6R^2}\frac{\partial^4 w}{\partial x^2 \partial \theta^2} + \frac{h^2}{12R^4}\frac{\partial^4 w}{\partial \theta^4} + \frac{\rho}{E'}\frac{\partial^2 w}{\partial t^2} = \frac{P_{mn}}{E'h} - \frac{\bar{Z}_{mn}\dfrac{\partial w}{\partial t}}{E'h} \tag{23}$$

where u, v and w are the deflections in the x, θ, and r directions, respectively.

The following assumed displacement field terms are submitted into the equations:

$$u = B_1 \cos k_x x \cos m\theta \tag{24}$$

$$v = B_1 \delta_1 \sin k_x x \sin m\theta \tag{25}$$

$$w = B_1 \delta_2 \sin k_x x \cos m\theta \tag{26}$$

The indicated derivatives are taken and the following dimensionless parameters are formed:

$$\alpha = hk_x \text{ dimensionless axial wave number} \quad (27)$$

$$\beta = \frac{h}{R} \text{ dimensionless shell thickness} \quad (28)$$

$$\Omega = \omega h \sqrt{\frac{\rho(1-v^2)}{E}} \text{ dimensionless frequency} \quad (29)$$

The internal damping of the material is incorporated by using a complex modulus of elasticity E

$$\text{so that } E' = \frac{E(1+i\eta)}{1-v^2} \quad (30)$$

Equations (21), (22) and (23) take on the form

$$\left\{-\alpha^2 - \frac{1-v}{2}\left(1+\frac{1}{12}\beta^2\right)m^2\beta^2 + \frac{\Omega^2}{1+i\eta}\right\} + \left\{\frac{1+v}{2}m\alpha\beta\right\}\delta_1$$

$$+\left\{\alpha\beta\left(v+\frac{1}{12}\alpha^2\right) - \frac{1-v}{24}m^2\alpha\beta^3\right\}\delta_2 = 0 \quad (31)$$

$$\left\{\frac{1+v}{2}m\alpha\beta\right\} + \left\{-\frac{1-v}{2}\left(1+\frac{1}{4}\beta^2\right)\alpha^2 - m^2\beta^2 + \frac{\Omega^2}{1+i\eta}\right\}\delta_1$$

$$+\left\{-m\beta^2 - \frac{3-v}{24}m\alpha^2\beta^2\right\}\delta_2 = 0 \quad (32)$$

$$\left\{\alpha\beta\left(-v-\frac{\alpha^2}{12}\right) + \frac{1-v}{24}m^2\alpha\beta^3\right\} + \left\{m\beta^2 + \frac{3-v}{24}m\alpha^2\beta^2\right\}\delta_1$$

$$+\left\{\beta^2\left[1+\frac{1}{12}\beta^2(m^2-1)^2\right]\right.$$

$$+\frac{1}{12}\alpha^2(\alpha^2+2m^2\beta^2) - \frac{\Omega^2}{1+i\eta}$$

$$\left.+Z_{mn}\frac{i\Omega\sqrt{1-v^2}}{(1+i\eta)\sqrt{\rho E}}\right\}\delta_2 = \frac{P_{mn}h(1-v^2)}{B_1E(1+i\eta)} \quad (33)$$

The three simultaneous equations are of the form

$$C_1B_1 + C_2(B_1\delta_1) + C_3(B_1\delta_2) = 0 \quad (34)$$

$$C_4B_1 + C_5(B_1\delta_1) + C_6(B_1\delta_2) = 0 \quad (35)$$

$$C_7B_1 + C_8(B_1\delta_2) + C_9(B_1\delta_2) = A \quad (36)$$

The solution for the radial displacement amplitude $B_1\delta_2$ is then given by the following matrix equation:

$$B_1\delta_2 = \frac{A\begin{vmatrix} C_1 & C_2 \\ C_4 & C_5 \end{vmatrix}}{\begin{vmatrix} C_1 & C_2 & C_3 \\ C_4 & C_5 & C_6 \\ C_7 & C_8 & C_9 \end{vmatrix}} \quad (37)$$

which is equal to

$$B_1\delta_2 = \frac{A(C_1C_5 - C_2C_4)}{C_1C_5C_9 + C_2C_6C_7 + C_4C_8C_3 - C_3C_5C_7 - C_2C_4C_9 - C_1C_6C_8} \quad (38)$$

We have in equation (38) the solution for the radial pipe wall motion amplitude at the coincidence frequency for the $(m,n)^{\text{th}}$ mode in the form of

$$w_{mn} = |B_1\delta_2| \text{sub } k_x x \cos m\theta \quad (39)$$

5 Solution Method for the Simultaneous Set of Partial Differential Equations. An analytical trial and error approach was used to search for values of the material damping factor η for the several modes for which we had reliable experimental data and comparing these values with published data. These comparison were based on experiments which included sound source simulating the higher order acoustic modes generated by a typical control valve as the noise source pipe wall acceleration measurements are at the specific coincidence frequencies for modes m,n.

We first determined the values for our steel pipe of h, R, v, E, ρ.

We next calculated the coincidence frequencies for every mode m,n of interest using the relationships for the intersection of the dispersion characteristics for the internal acoustic field and the like mode pipe wall bending mode (see [11, 20, 21]).

Next, we calculated the values of C_1 through C_9 as defined by equations (34–36) as well as $\bar{Z}_{mn}$ and A. These values determined, we can calculate $B_1\delta_2$ from equation (38) for every mode m,n of interest.

The dynamic response of the pipe as a function of x, θ, and t is then given by equation (40)

$$w = B_1\delta_2 \sin k_x x \cos m\theta\, e^{i\omega t} \quad (40)$$

and the potential acoustic field outside of the pipe by equation (4)

$$\phi_{mn} = B_{mn}H_m^{(2)}(k_{mn}r) \sin k_x x \cos m\theta\, e^{i\omega t} \quad (41)$$

From the dynamic boundary conditions at the pipe wall, we have

$$\frac{\partial w}{\partial t} = -\left.\frac{\partial \phi_{mn}}{\partial r}\right|_{r=R} \quad (42)$$

and B_{mn} can be calculated from equation (43)

$$B_{mn} = -\frac{i\omega B_1\delta_2}{k_{mn}H_m^{(2)\prime}(k_{mn}R)} \quad (43)$$

In order to proceed further, we must have reliable values of the internal damping coefficient η as defined in equation (30) and seen to occur in the equations for the pipe wall response, equations (31–33). The literature indicates that the structural damping of steels is somewhat dependent on the geometry of the structure, varies with the composition of the steels, and is also a function of frequency. The order of magnitude of η is given by 10^{-2}.

We then proceed to substitute the measured values of the pipe wall velocities and internal pressures at the wall in order to estimate the damping factor for each coincidence condition at the respective m,n modes. The results of this process are shown in Table 1.

We see from Fig. 2 that η tends to decrease with increasing frequency which agrees generally with findings from studies on flat plates (23).

The published literature cites values of the damping factor of metals ranging from 10^{-3} to 10^{-1} with 10^{-2} being typical for steels. The somewhat higher values obtained in our study may well be due to the damping contributed by the short sections of viscoelastic material covering used at the ends of the long pipe as described in [11].

We are now in a position to determine the relationship for the acoustic pressure in a free field at some distance r from the center of a pipe which is excited by an internal acoustic field resulting from such sources as valves, fans or others.

Thus, the acoustic pressure, p_{mn}, at a distance r from the pipe wall is given by

$$p_{mn} = \rho\frac{\partial \phi_{mn}}{\partial t} \text{ so that} \quad (44)$$

$$p_{mn} = \frac{\rho\omega^2 B_1\delta_2}{k_{mn}H_m^{(2)\prime}(k_{mn}R)} H_m^{(2)}(k_{mn}r)\sin k_x x \cos m\theta e^{i\omega t} \quad (45)$$

At the coincidence frequency ($\omega = \omega_{mn}$) of the m,n mode and a distance r, the acoustic pressure becomes

$$p_{mn} = \frac{\rho\omega^2{}_{mn} B_1 \delta_2 H_m{}^{(2)}(k_{mn} r)}{k_{mn} H_m{}^{(2)\prime}(k_{mn} R)} \sin k_x x \cos m\theta\, e^{i\omega t}$$

Conclusions

A method has been developed to calculate the acoustic field strength outside of an infinitely long straight pipe which is subjected to an internal acoustic field generated by any of many possible compressible fluid flow noise sources. As a result of earlier studies, we had established that the predominance of the acoustic energy is transmitted through the pipe wall at the coincidence frequencies between the acoustic field and the like mode pipe wall bending vibrations. We note, however, that by far most of the internal field's acoustic energy is propagated down the pipe.

We also note that at the coincidence frequencies for the modes studied the pipe wall motion and thus the external radiation is controlled by the damping factor of this thick-walled but realistic pipe material. The damping factor ranged from 0.04 to 0.2 for this low carbon steel 8.28 cm internal diameter pipe of 0.305 cm thickness.

Several important assumptions which we made must be kept in mind when attempting to apply these results or techniques to situations which differ markedly from our conditions.

1 We assumed that the internal pipe wall vibrations do not affect the internal acoustic field. The assumption is valid for metallic pipes and gaseous fluids and breaks down if liquids are used in either metallic or nonmetallic pipes as then our assumption of decoupled fields is no longer valid. Walter [11] proved this point and established limits.

2 The pipe motion equations are based on small displacements and thin pipe wall theory.

3 The material is purely elastic, homogeneous and isotropic.

4 The acoustic source inside the pipe is assumed to be devoid of any marked peaks in its spectrum.

5 The solution is dominated by the transmission at the coincidence frequencies.

The work reported in this paper is a portion of a larger effort to develop a fundamentals based valve noise prediction technique.

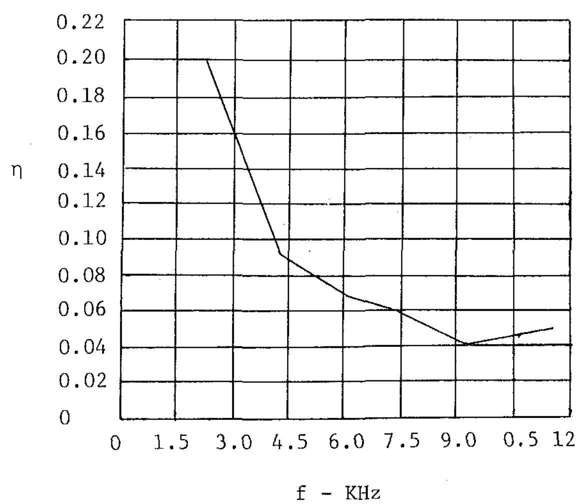

Fig. 2 Plot of damping factor versus frequency for low carbon steel pipe of radius 4.14 cm and 0.305 cm wall thickness

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Table 1

Mode (m,n)	Measured velocity m/sec	Calculated velocity m/sec	Damping factor η	Coincidence frequency	Measured wall pressure dB
(1,1)	1.02×10^{-4}	2.0×10^{-4}	0.20	2545	120
(1,2)	3.02×10^{-5}	4.25×10^{-5}	0.06	7261	106
(2,1)	7.96×10^{-5}	15.1×10^{-5}	0.09	4336	112
(2,2)	5.34×10^{-5}	6.89×10^{-5}	0.04	9353	104
(3,1)	7.54×10^{-5}	11.3×10^{-5}	0.07	6026	109
(3,2)	8.02×10^{-5}	12.0×10^{-5}	0.05	11,400	112

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