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# Noise Generation by Fluid Flow Through Pipes

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Much of the mid- and high-frequency noise in process plants derives from the flow of fluids through valves, orifices, and other pipework-located discontinuities. The myriad of sources involved generates, within the plant, a sound field that is diffuse and uniform. In this paper the author develops a tentative parametric formulation by which the generalized fluid pipe source can be described and also presents an analytical basis on which the in-plant and community environmental criteria can be related to the sound generating propensity of the individual source.

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## INTRODUCTION

The subject of fluid dynamic noise concerns the generation of sound through the motion of fluids (gases and liquids). In past years, considerable effort has been expended in identifying and characterizing the mechanisms of fluid dynamic noise in fluid machines such as gas turbine (jet) engines, propellers and fans, pumps, compressors, and so forth. On the other hand, comparatively little work has been directed toward the development of suitable descriptions of flow-induced noise in passive (as opposed to active) systems such as fluid-filled ducts and pipes. This paper represents a tentative effort to identify and characterize sources of flow noise in this latter type of system.

Noise generation by fluid flow through pipes can represent a dominant source of sound in process plants, such as refineries and petrochemical plants. In such plants, large amounts of fluids are transported through pipes over considerable distances, often at very high pressures and velocities. The pipework incorporates many control valves, metering orifices, branch points, bends, and other pressure dropping discontinuities. Each of these flow discontinuities can form a significant noise source; in combination they can control the plant noise environment—particularly in the mid- and high-frequency part of the spectrum.

This dominance by pipework sources has been observed in a number of process plants. We have also observed that the number and spatial extent of these in-pipe sources give rise to a plant acoustical environment which is very nearly diffuse, i.e., sound energy is apparently incident from every angle, and the energy density throughout the plant is very nearly uniform. It is thus quite difficult and sometimes impossible to identify and measure individual sources in a functional plant. Such a situation represents, obviously, an almost intractable noise control problem unless these sources are carefully considered in the plant design stages. Since most pipe system sources cannot be effectively covered by a "vendor specification," it is necessary that means of estimating the noise generated by in-pipe sources be developed.

This paper is presented in two parts:

In the first part, we develop a tentative parametric formulation by which the generalized fluid pipe source can be described. On the basis of this formulation, we discuss possible means for noise control. Our treatment is limited to systems carrying flow in the low- and mid-subsonic range.

In the second part, we discuss some aspects of the measurement and specification of piping system sources. In particular, we domonstrate the interdependence that occurs between the community and in-plant environmental levels for the refinery-like situation. We also comment upon the problems and limitations of laboratory measurements.

## NOMENCLATURE

$c$ = velocity of sound
$D$ = pipe diameter
$\tilde{F}$ = fluctuating force
$f$ = frequency
$h$ = pipe wall thickness
$K$ = pressure loss factor
$L$ = effective source length
$N$ = number of sources
$\langle p^2 \rangle$ = mean-square sound pressure
$\Delta p$ = total pressure drop
$q$ = dynamic pressure
$R$ = radius of plant
$S$ = plant-community distance
$U$ = flow velocity
$W$, $w$ = sound power
$\delta$ = characteristic dimension
$\rho$ = density
$\tau$ = transmission coefficient
$\epsilon$ = blockage fraction

Subscripts

$c$ = community
$f$ = fluid
$p$ = plant

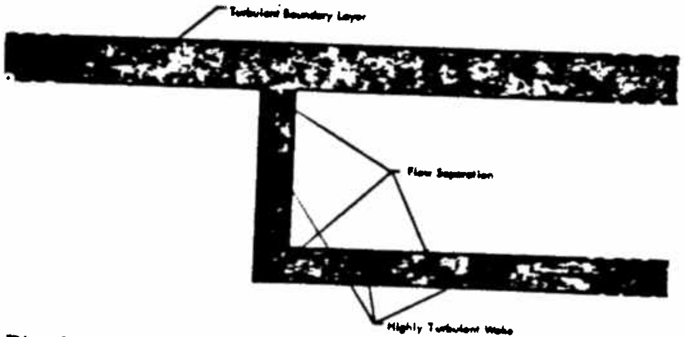

Fig.1 The regions of sound generation in fluid

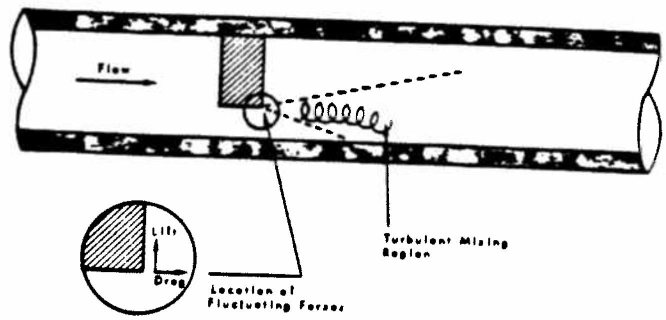

Fig.2 Generalized concept of pipe component

The treatments in this paper are by no means exhaustive; the ideas are tentative and incomplete. Hopefully, however, the contents may provide some guidance to those contemplating work in the area of piping system noise. Perhaps, also, some discussion may be provoked.

## PART I — THE MECHANISMS OF NOISE GENERATION

We wish to develop a parametric understanding of the noise generation and propagation mechanisms for the typical pipe components shown in Fig.1. In doing so, we shall limit our attention entirely to systems in which the fluid velocities are subsonic. Such conditions have wide practical application and are currently more tractable than the situation in which choked flow occurs.

As shown in Fig.1, at bends and branch points the flow will tend to "separate" from the pipe wall with the resultant formation of a turbulent wake. At major discontinuities, like valves and orifices, very intense turbulent flows may be generated; in some instances, the wake of one discontinuity may impinge on another discontinuity further downstream. Such surface/flow interactions are a source of system pressure drop and, we hypothesize, are basic to the subsonic flow noise generating mechanisms.

Let us set the stage for our discussions by reviewing briefly the fundamental mechanisms of fluid dynamic noise generation. These shall be described with reference to the generalized concept of a noise generating pipe component shown in Fig.2.

The most fundamental mechanism—but probably the least common in practice—is termed the fluid dynamic "monopole." It occurs when the total flow passing through a fluid system periodically fluctuates. The fluid system pulsates, like a piston, and radiates sound quite efficiently. A potential, but not yet proven, example of an aerodynamic monopole source is the afterburner of a jet engine in cases where the burning process is unsteady.

The next mechanism is termed the fluid dynamic "dipole" source. This source results from the impacting of turbulent flow against a surface, or the shedding of turbulence by a surface. In each case, the presence of turbulence introduces a fluctuating component to the "drag" and "lift" forces exerted by the flow on the surface. These fluctuating forces react on the fluid to generate sound. It should be carefully understood that the mechanism does not require physical motion of the surface; an absolutely rigid surface will radiate very considerable dipole sound.

The specific case of a surface within the confines of a fluid-filled pipe provides the appropriate conditions for dipole noise generation. This mechanism is thus of much concern in this paper. The application is, however, somewhat special for two reasons. First of all, if we use the notation of Fig.2, which defines "lift" at right angles to the pipe axis, then the presence of the pipe wall effectively suppresses the radiation from the "lift" dipole (such an occurrence is not too hard to imagine). Second, as we shall show shortly, the presence of the pipe reinforces the radiation from the "drag" dipole in a quite interesting way.

There are many examples of fluid dynamic dipole noise in real life: fan noise is almost entirely dipole in origin; so too are the vocalized sibilants "S" and "Sh"; the whistling of wind through wires is an example of a "well-ordered" (discrete frequency) aerodynamic dipole.

The final mechanism of interest is the fluid dynamic quadrupole. The quadrupole source arises from turbulent mixing of fluid. It does not require solid surfaces. Quadrupole radiation might be expected from the turbulent wake downstream of the flow constriction in Fig.2. The aerodynamic quadrupole mechanism is a major source of the noise of turbojet engines; the sources arising from the mixing of the jet exhaust with the surrounding atmosphere.

## Parameters

The basis for the parametric development given in the following is a recent study of the noise generated by flow "spoilers" in a jet pipe (1, 2).[1] This study showed quite clearly that when the subsonic pipe flow is interrupted by solid surfaces, the dominant source mechanism is the fluid dynamic dipole and that this source can be described in terms of the mean flow parameters (specifically the pressure drop) and the system geometry.

In the free-field environment, the radiation from a fluid dynamic dipole source is given by

$$W \sim \tilde{F}^2 f^2/\rho_f c_f^3 \tag{1}$$

where $\tilde{F}$ is the rms fluctuating force, $f$ is the characteristic frequency of fluctuations, and $\rho_f$ and $c_f$ are the fluid density and velocity of sound, respectively. When the source is enclosed within the confines of a pipe whose diameter is less than the acoustic wavelength, it can be shown (3, 4) that the free field power relation is modified by a term $(c/fD)^2$ to give

$$W \sim \tilde{F}^2/\rho_f c_f D^2 \tag{2}$$

Since the characteristic frequency of noise generation by a pipe discontinuity will lie, generally, below that frequency for which the wavelength is greater than the pipe diameter, equation (2) is probably more relevant than equation (1) in developing our arguments.

Expressing the fluctuating drag force ($\tilde{F}$) in terms of the flow and geometric parameters,[2] equation (2) becomes

$$W \sim \Delta p^2 \varepsilon^2 D^2/\rho_f c_f \tag{3}$$

where $\Delta p$ is the total pressure head loss across the discontinuity, and $\varepsilon$ is the fraction of the pipe area blocked by the obstruction. The characteristic frequency of noise generation is given by

$$f \sim (\sqrt{\Delta p/\rho_f c_f})/\delta \tag{4}$$

where $\delta$ is the characteristic dimension of the obstruction—this establishing the scale of the eddies shed by the obstruction.

If we now suppose that the sound power generated at the flow discontinuity propagates away from the source as a plane acoustic wave in the contained fluid, we may write the mean square sound pressure within the pipe as

$$\langle p^2 \rangle \sim \Delta p^2 \varepsilon^2 \tag{5}$$

This sound field will propagate principally in a downstream direction since the sound sources are more particularly involved with downstream surface interactions, and since upstream propagation is inhibited by the high flow velocity and geometric blockage offered to the sound field by the flow constriction. The sound field will also decay with distance as a result of loss mechanisms at the wall of the pipe. In general, the rate of pressure decay in the propagating wave will be proportional to the pipe diameter, and we can associate the sound power radiated through the pipe wall with an effective radiating length that is proportional to the diameter.[3] If we further define the transmission coefficient $\tau$ of the pipe wall, we may write the total sound power radiated to the outside environment in the form

$$W_{rad} \sim \Delta p^2 \varepsilon^2 D^2 \tau/\rho_f c_f \tag{6}$$

The transmission coefficient is best considered in terms of a characteristic frequency of the pipe—the ring frequency—the frequency at which the circumference of the pipe equals one wavelength for compressional waves. For most pipe

[1] Underlined numbers in parentheses designate References at end of paper.

[2] A fundamental assumption in the development is that the steady and fluctuating components bear a constant relationship to each other, although, of course, the fluctuating flow component may lie several orders of magnitude below the steady component.

[3] We assume here that the effective radiating length is a function only of the pipe diameter—that it is independent of frequency and pipe wall thickness. In fact, this condition is very nearly achieved if the pressure decay is controlled by viscous and thermal losses at the pipe wall (see Beranek, "Acoustic Measurements," page 72, published by Wiley). If, however, the pressure decay is controlled by acoustic transmission through the pipe wall, then a very different and interesting situation arises. It can be shown that the total source power generated within the pipe is effectively radiated through the pipe wall, the only influence of the pipe thickness (transmissibility) being to change the effective length of pipe over which the source power is distributed.

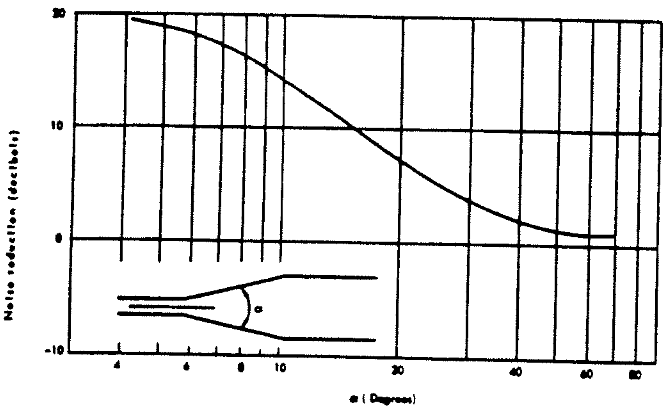


Fig.3 Area transition noise—influence of angle

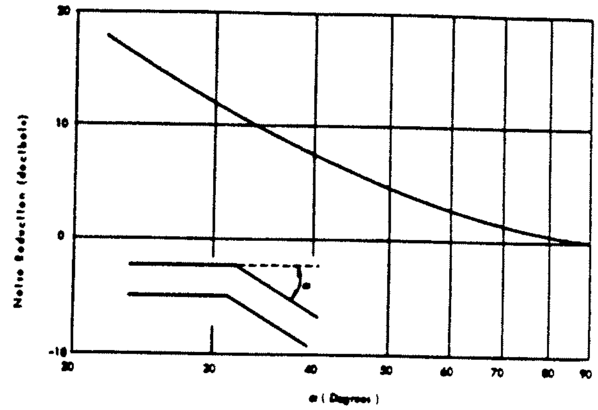


Fig.4 Bend Noise—influence of bend angle

materials and sizes found in process plants, the frequencies of interest will lie well below the ring frequency. This is indeed fortunate since below the ring frequency, we may assume the transmission coefficient to be approximately independent of the frequency (5). For a given material, it will have a value proportional to the ratio D/h, where h is the pipe wall thickness. Thus, we may now write the total sound power radiated into the outside environment as

$$W_{rad} \sim \Delta p^2 \varepsilon^2 D^3/h \rho_f c_f \qquad (7)$$

This form of the expression is suitable for a discussion of noise control techniques. It also suggests those parameters that must be studied in an investigation of pipe noise sources.

## Control

It is clear from equation (7) that the pressure drop appearing across a discontinuity is quite important in determining the noise output. Important, too, is the diameter of the pipe, although this is hardly an independent parameter if the pipeline is to carry the required mass-flow of fluid. It is pertinent to note that little is to be gained by increasing the thickness of the pipe wall, perhaps nothing if the postulate in footnote[3] is true.

For such elements as bends and branch points, the effective blockage factor $\varepsilon$ tends to unity, and the pressure drop can be written in the usual form

$$\Delta p = Kq \qquad (8)$$

where K is the pressure loss factor, and $q(= {}^1/_2\, \rho U^2)$ is the dynamic pressure of the flow. In these terms, equation (7) becomes, assuming constant fluid properties,

$$W_{rad} \sim K^2 U^4 D^3/h \qquad (9)$$

Further, if the mass flow through the pipe is to be constant, we have

$$W_{rad} \sim K^2 U^{2.5}/h \qquad (10)$$

The effective noise control that can be achieved by reducing the pipe flow velocity is, therefore, controlled by an exponent of 2.5 (7.5 dB per velocity doubling).

Many elements in fluid piping can be designed to reduce the pressure loss factor K. The reductions available by redesign of two typical configurations are shown in Figs.3 and 4. It should be noted that the extent of noise reduction obtained by design changes may be limited by noise generated in other parts of the pipe system. Thus, only a limited part of the theoretical noise reduction given in these figures might be obtained in reality.

For more conventional discontinuities, like valves and metering orifices, the value of $\varepsilon$ will lie between zero and unity. For adjustable discontinuities (valves), $\varepsilon$ will increase with blockage.

It should be noted that equation (7) relates to total noise radiated by the discontinuity and that most parametric changes will be accompanied by a change in the characteristic frequency of noise generation as given by equation (4). Further, it must be expected that when constricted velocities in discontinuities become very high (approaching sonic conditions), turbulent mixing noise of quadrupole character may begin to play a significant role. The parametric studies described in the foregoing are limited entirely to an assumed dominance by surface interaction sources having a dipole characteristic.

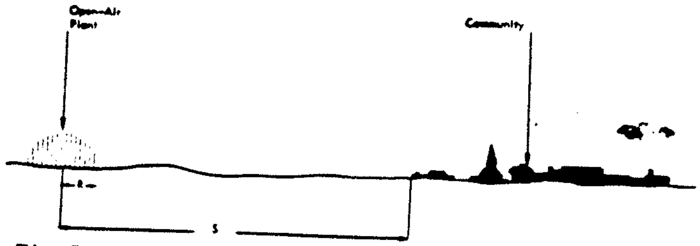


Fig.5 Hypothetical plant/community configuration

### Further Mechanisms

In deriving equation (6), we have postulated a purely fluid dynamic acoustic source which radiates sound power within the pipe; this sound power then passes to the outside environment via the acoustic transmission coefficient of the pipe wall. The possibility must not be neglected, however, that the intense turbulence generated at a pipe discontinuity might drive the pipe wall directly and so cause sound to be radiated to the outside environment without the intermediate step of the in-pipe acoustic field. Also, the fluctuating lift forces (which have been discounted as sources of fluid-borne acoustic energy) might bodily shake the pipe and thereby radiate sound energy directly into the outside environment.

In general, the degree of matching between the scale of convecting turbulence and wavelength of flexural waves in the pipe wall will be very poor. Thus, we would not expect significant acoustic radiation from the pipe wall via the first mechanism.

The second mechanism could be especially significant for the case in which the pipe is fastened to a surface (building wall) which can radiate efficiently; particularly if the fastening is rigid and is located at, or close to, the constriction position. This mechanism, however, should not be significant when the pipe is free from radiating surfaces. This is generally the case in process plants.

## PART II — MEASUREMENT AND CRITERIA

There is clearly a need for laboratory studies of the noise generated by fluid flow through pipes. There is also a need for acceptable standards by which the noise of pipe components—especially "vendor" components like valves—can be measured and specified in a meaningful way. Finally, methods must be developed whereby, individually and collectively, fluid flow sources can be incorporated satisfactorily into the environmental criteria for plants. Some tentative thoughts along these lines are presented in the following.

### Environmental Criteria

It is our experience that in certain types of process plant, the in-plant noise environment is controlled by pipework sources. The number and spatial extent of these sources is such that the in-plant noise environment is very nearly diffuse and is quite uniform except at positions close to major sources.

The concept of a diffuse uniform sound field applied to an array of outdoor noise sources is, to the best of our knowledge, quite unique. It does, however, suggest a number of analytical approaches whereby we may relate the number and power of individual sources with the general in-plant environment and also the environment generated in surrounding communities.[4]

Consider the open-air plant configuration shown in Fig.5. The plant is located a distance S from a community. The noise sources within the plant are derived from fluid flow in pipes and are assumed more-or-less evenly distributed through a hemispherical volume of radius R. The average source has an acoustic power w distributed over a length L of pipework. The total number of sources is N.

First of all, in such an environment the range of influence of the direct field of individual sources will be quite limited. Specifically, the distance (r) from the average source at which the direct field will equal the diffuse pressure field can be shown to be of order

$$r = R^2/4NL \qquad (11)$$

Thus, for the condition of 1000 sources in a plant of radius 500 ft, with an effective source radiation length of 20 ft (20 dia for 1-ft-dia pipework), the limit of direct field influence will be about 3 ft. In such a situation, it is the diffuse field generated by many sources, not the direct field of the single source, which dominates the plant environment.

The mean-square diffuse sound pressure in the plant $\langle p_p^2 \rangle$ is related to the total plant sound power by the relation

$$N \cdot w = \langle p_p^2 \rangle 2\pi R^2/4\rho c \qquad (12)$$

when $\rho$ and c are the atmospheric density and velocity of sound, respectively.

Finally, we can relate the mean-square sound pressure in the community with the in-plant en-

[4] The approaches used in this text derive from reverberant room theory. Atmospheric absorption effects and shielding effects are neglected.

vironmental pressure (neglecting atmospheric absorption).

$$\langle p_c^2 \rangle = \langle p_p^2 \rangle R^2/4S^2 \quad (13)$$

Clearly, the design criteria for in-plant and community environments are not independent.

## Laboratory Studies

The parametric studies presented in the first part of this paper provide a tentative basis for correlating the results of field and laboratory studies of pipework component noise. Such studies are required if progress is to be made in predicting and quieting pipe system designs.

Already manufacturers of control valves are actively pursuing laboratory studies of their products (6). In the next few years, we may expect the noise of valves to be catalogued and quieter designs developed.

A major problem in undertaking meaningful laboratory studies lies in the accurate measurement of the radiated sound power. In the foregoing paragraphs, we have tried to stress the importance of this parameter. The length of pipe (downstream from the noise generating component) from which sound radiation may occur is considerable, and the physical experiment must be long enough to contain this. Perhaps a feasible laboratory configuration would be to coil the pipe (rather like a tuba) and evaluate the total effective source in a calibrated reverberant room.

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