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AERO DYNAMIC NOISE SOURCES IN INDUSTRY SESSION: University of Loughborough

THE CALCULATION AND CONTROL OF NOISE FROM PRESSURE VENTING SYSTEMS

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1 INTRODUCTION

Vent system noise arises from two main sources:

a "pressure let down" noise due to high velocity flow at points where sonic or near sonic conditions occur, and

b turbulence noise set up at the point where the vented gas mixes with the atmosphere:

These are basic phenomena and, whilst they have similar origins in a mathematical sense, they produce very different sorts of noise in that the former is a line source and the latter a point source. The following analysis shows how the resultant noise levels can be calculated for a range of gases or mixtures of gases after having first calculated the sound power output. A description of some methods of controlling the two types of noise is given: these are based upon techniques developed in ICI which have been proved by direct use.

2 BASIC EQUATIONS

The sound power produced by both a let down or a vent can be calculated from the following equation which has been deduced from published data:

$$L_W = 60\log_{10} V_j + 20\log_{10} d + 20\log_{10}\frac{\rho_j}{\rho_o} - 4 \qquad \text{(1)}$$

L_W = sound power in dBW re 10^{-12} W

V_j = velocity at let down orifice or vent exit, m/s

d = orifice or vent diameter, m

ρ_j = fluid density at the orifice or vent exit, kg/m^3

ρ_o = density of fluid mixed with jet from valve or vent orifice, kg/m^3

The frequency distribution for both types of noise is given in a non-dimensional form in Fig 1; the frequency at which the spectrum peaks is given by:

$$f_o = 0{\cdot}13\,\frac{V_j}{d} \quad \text{Hz} \qquad \text{(2)}$$

Having determined the value of f_o the sound power in each of the preferred octave bands (63,125 ... 8000 Hz) can be derived from Fig 1 by application of the correction to L_W calculated from equation (1).

Experimental evidence that is available from our own resources suggests that, while the frequency distribution is often rather flatter than that shown in Fig 1, it is sufficiently practical to be used. In addition, experimental evidence supports the correctness of the sound power equation within an accuracy of $\pm$ 4 dB.

Having derived the sound power, the sound pressure levels can be calculated by the methods outlined in the following sections.

3 PRESSURE LET DOWN NOISE

This is very often the major source of noise in vent systems and examination of equation (1) shows that this can be expected since the velocity may be sonic and the actual jet density will often be much larger than the fluid density after let down. For a sonic condition:

$$V_j = \sqrt{2\left(\frac{k}{k+1}\right) g \; \frac{R}{M} \; T_1} \qquad \text{m/s} \; \dots\dots\dots\dots\dots\dots \; (3)$$

$$\rho_j = \left(\frac{2}{k+1}\right)^{1/k-1} \times \frac{p_1 M}{R T_1} \qquad \text{kg/m}^3 \; \dots\dots\dots\dots\dots\dots \; (4)$$

p_1 = source pressure, kg/m^2

T_1 = source temperature, ^oK

R = universal gas constant = 8.315 J/K mol

M = molecular weight

k = isentropic index

In some vent systems the sonic condition occurs at one or more points other than the let down valve and thus there are two or more points of large scale noise production. This noise is radiated from the vent pipe between each of the sonic let down points: the final sonic let down noise is radiated from the end of the vent system. In many systems the final sonic let down is at the end of the vent system and it produces a sonic turbulent mixing noise. The fluid density after let down (ρ_0) depends on the geometry of the vent system and the pipe roughness factor: if the sonic let down is straight to atmosphere then the density of air should be used (1.2 kg/m^3). The calculation of sound pressure level at points outside the vent pipework depends upon:

a pressure let down noise energy transmitted down the pipe, which itself is a function of pipe geometry and the degree of attenuation due to normal flow

b noise generated by turbulent flow downstream of the valve: this in itself can be considerable if high velocities exist. Pipe turbulent flow noise can be a significant source of plant noise and has been the subject of separate study.

c attenuation of the pipe wall and any external lagging

For vent lines up to 50 metres long it can be assumed as a first approximation that the internal sound pressure level in the pipe is uniform and given by the following equation:

$$L_{p_I} = L_W + 10 \log_{10} \frac{\rho_0 \, a_0}{A_p} - 26 \qquad \text{dB} \; \dots\dots\dots\dots\dots\dots \; (5)$$

where A_p = pipe cross section, m^2

a_0 = sonic velocity in fluid downstream of valve, m/s

ρ_0 = density of fluid downstream of valve, kg/m^3

The wall transmission loss (T.L.) can be calculated using the well known equation:

$$T.L. = 15 \log_{10} mf - 31 \qquad \text{dB} \; \dots\dots\dots\dots\dots\dots \; (6)$$

The external sound pressure level, L_{pE} is calculated from

$$L_{pE} = L_w + 10\log_{10}\frac{\rho_o\, a_o}{A_p} - 15\log_{10} mf + 5 \quad \text{dB} \ldots\ldots\ldots\ldots (7)$$

At positions not exceeding 5 m from the pipe, calculations must be made assuming a line source if the local sound pressure level L_p is to be derived.

$$L_p = L_{pE} - 10\log_{10}\frac{2X}{d} - 1 \quad \text{dB} \ldots\ldots\ldots\ldots (8)$$

where X = distance of point of calculation from pipe, m.
d = pipe diameter, m.
m = mass per unit area kg/m^2
f = frequency, Hz

For systems with a subsequent sonic flow condition L_p is calculated from a consideration of the reverberation between the two sonic points. It is necessary to allow for wall transmission loss and attenuation due to distance. This yields the following approximate equation:

$$L_p = L_w - 10\log_{10}(\pi d l) - 15\log_{10} mf - 10\log_{10}\frac{2X}{d} + 51 \quad \text{dB} \ldots (9)$$

where l = distance between sonic points, m

With short vent systems most of the acoustic let down energy is transmitted straight to atmosphere and apparently appears as a turbulent mixing noise; this is not an unusual case and care must be taken when defining the true source of the noise.

4 TURBULENT MIXING NOISE

This is a simpler case to deal with as the noise source can usually be treated as a point source; the value of ρ_o to be used in calculating L_W is that for air (1.2 kg/m^3). The sound pressure level (L_p) at a point distance X metres from the source is calculated from the equation:

$$L_p = 60\log_{10} V_j + 20\log_{10}\frac{\rho_j}{1.2} + 20\log_{10} d - 20\log_{10} X - 12 \quad \text{dB} \ldots (10)$$

The spectral distribution can be deduced from Fig 1, after calculating f_0. For large values of X due allowance should be made for air attenuation.

Important features of the above equation are the control exercised by velocity, density, distance, and diameter on noise level; of these velocity and density are conditioned by diameter and proper selection of the correct diameter is a precondition of minimising noise generation.

5 USE OF CALCULATED VALUES OF Lp

Sound pressure levels calculated from the equations in sections 3 and 4 must be assessed against the desired values of sound pressure level at the position under consideration to determine the need to apply the noise control measures outlined in the following sections. The effectiveness of these measures is dependent upon their being properly sized and a knowledge of the physical properties of the fluids being handled.

6 CONTROL OF PRESSURE LET DOWN NOISE

Most of the noise is generated local to the valve by highly turbulent flow and apart from the obvious use of absorption silencers there is clearly a part to be played by arranging the flow system in the most orderly manner. Papers on "silent trim" valves in which the flow is controlled by multiple orifice or multiple stepped valve trims have recently been produced (Ref 1 and 2). These are little better than the control arrangements described in Fig 2.

An example of the performance of such an arrangement is illustrated in Fig 3. It is interesting to note that they have been applied to gaseous systems whose rate varied between 800 kg/hr and 120,000 kg/hr. The achieved attenuations have varied between 20 and 30 dB and this has been adequate for most purposes.

There is a need to control the fluid velocities on the low pressure side of pressure reduction systems or there may be considerable noise generated by turbulent flow downstream of the valve; only too many systems seen in practice have received scant attention to this aspect of their noise producing properties.

For such systems one can acoustically treat the pipe walls and fit a silencer at the atmospheric end of the vent; this may be the only practical method of minimising noise from HP vents.

HP pressure let down noise is best controlled by:

a avoiding long lines after the pressure let down

b fitting the let down valve directly on the silencer inlet; the silencer exit should be dimensioned to avoid velocities which can promote turbulent mixing noise.

Examples of designs developed to satisfy these requirements are outlined in Fig 4; these were devised to overcome reliability problems with commercially available silencers due to "sonic" damage, high temperatures and the need to flare some of the gases at the silencer exit. Experience has shown them to have a long life if correctly fabricated. Of the two types the "Ring Packed" type has been widely used for difficult conditions and attenuations in excess of 40 dB have been achieved as illustrated in Fig 5. The "inverted bucket" type has, in general, been employed on smaller vents; the largest system using these vents up to 103,000 kg/hr of flammable gas from a 400 psig system and it has been satisfactory in service.

7 CONTROL OF MIXING NOISE

The essence of this is in the reduction of the efflux velocity to a level well below Mach 1 thus giving the added benefit of reducing the jet density to a minimum. As previously mentioned HP vent systems require suitable silencers to overcome the noise set up by high density sonic jets but they also provide a means of achieving the required low exit velocity and density. Occasionally there are duties, e.g. venting pressurised CO_2 systems, which demand an ability to handle temperatures in which "dry ice" can be formed thus providing a possible means of plugging small orifices; for such duties conventional methods have been employed in conjunction with piccolo silencers and an example of this is shown in Fig 6.

ACKNOWLEDGEMENTS

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REFERENCES

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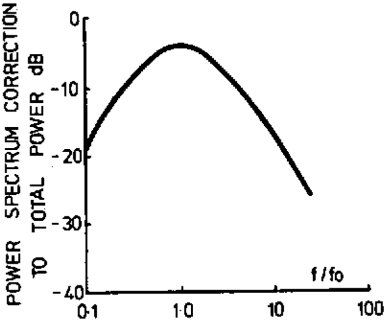


FIG. 1 VENT NOISE SPECTRUM

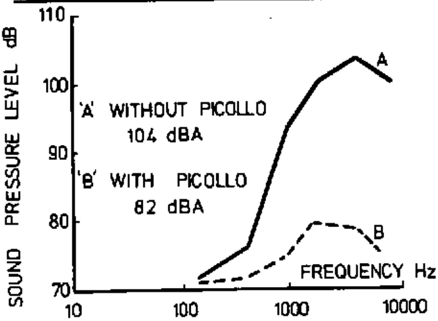
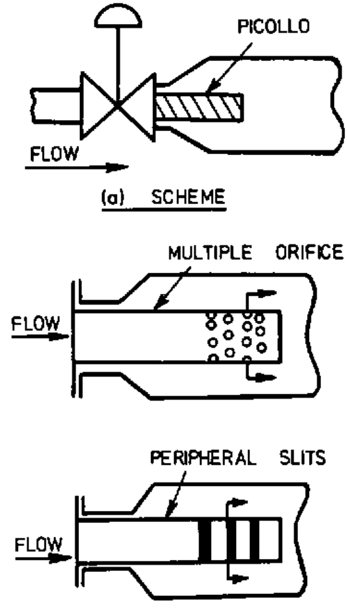
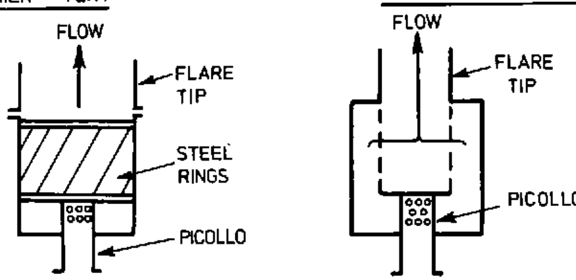


FIG. 3 PICOLLO PERFORMANCE HUMIDRIER VENT



(b) PICOLLO TYPES

FIG. 2 CONTROL METHODS FOR LET DOWN VALVE NOISE.



(a) RING PACKED

(b) INVERTED BUCKET

FIG. 4 SILENCERS FOR H.P VENTS

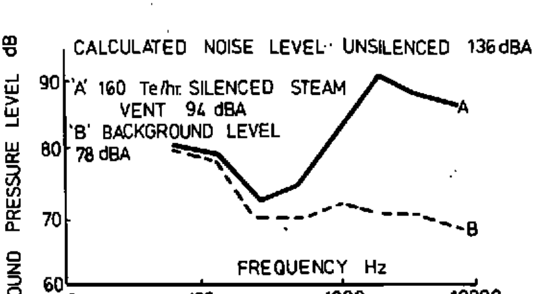


FIG. 5 RING PACKED SILENCER PERFORMANCE NOISE MEASURED 80ft. FROM 900psig STEAM VENT

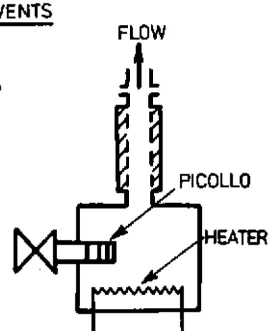


FIG. 6 CO_2 VENT SYSTEM