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MEASUREMENT OF VALVE NOISE SIGNATURES

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1. INTRODUCTION

This paper describes measurements of the noise signature of a set of representative high-pressure flow regulating valves with an analysis of the noise producing mechanism for choked conditions. Valves and their piping provide significant noise sources in many installations such as power stations, chemical plants, oil refineries and gas regulator stations. It is common experience that the noise propagates almost without attenuation along extensive lengths of pipe while the perceived noise is generally that transmitted through the pipe, which acts as a distributed source. Prediction of the noise from a given installation requires a quantitative knowledge of the behaviour of the valve as a source as well as an understanding of the transmission loss through a pipe including any effect of interactions between the associated pipework and the valve. The problem of predicting pipe transmission loss has not yet been solved and since most measurements on valve noise have been made on complete installations[1], quantitative information on the acoustic behaviour of the valve itself cannot be extracted from these measurements.

Since the acoustic behaviour of valves or the pipe transmission loss cannot be predicted on the basis of existing knowledge[1], a systematic study of valve noise was undertaken to, in part, remedy this deficiency. To simplify the problem, measurements were made of the noise signature when the valves were discharged into a reverberant chamber, thus avoiding uncertainties due to our inability to estimate adequately the pipe transmission loss[2]. This eliminates the major complication in the evaluation of the results of previous tests on valves in representative installations but care was now necessary to ensure that the measurements did define the valve noise signature under normal operating conditions. Eight series of tests were run on seven different valves over a wide range of pressure drops and valve openings with the valve discharging into the reverberation chamber, which was at atmospheric pressure. Further tests were run at constant pressure ratio for a range of flow densities. Check and calibration tests were necessary on both rig and room while some preliminary measurements were made of pipe transmission loss.

2. VALVE NOISE MEASUREMENTS

To establish the noise signature for a given valve we are interested in both the overall source strength and its spectral distribution. Since the noise is generated aerodynamically, we can assume that, for a given geometry, the source strength will be proportional to the gas density ρ and the characteristic flow velocity V raised to some high power n, noting also that the geometry of a given valve will vary as its effective aperture or throat area changes. Thus the sound intensity I can be expressed as

$$I = K \rho \; V^n \tag{1}$$

where n usually lies between 6 and 9. If the valve is choked, and for most regulator applications this is the case, then the characteristic flow velocity, which was found to be the fully expanded velocity, will depend only on the pressure ratio across the valve while the mass flux will depend only on the throat area and the upstream pressure. The mass flux is significant since its value governs the effective Mach number of the expanding flow immediately downstream of the throat. This expanding pipe section acts as a complex acoustic horn and it was found that, particularly at partial valve openings, the local Mach number has a pronounced effect on the shape of the spectrum. Thus two sets of measurements are required, one covering velocity changes, and the other changes in valve opening (or valve geometry).

Initial investigations included the calibration of the reverberant room, including the installed pipework, measurement and minimisation of flow noise from the external pipework and rig control systems. Such background noise was always more than ten, and generally more than twenty, dB below the valve noise for frequencies above 200 Hz. Some tests were necessary to establish a sufficient and suitable pipework termination was fitted downstream of the valve under test so that flow conditions were equivalent to those in a normal installation. There was, however, some inevitable radiation loss at the open end of the pipe at low frequencies. This was minimised by fitting a large flange there, and partly compensated by a gain in reverberation time found at the low frequencies.

2.1. Test conditions

The noise signature of eight different nominally 50 mm valves were measured for pressure ratios (that is ratio of upstream to downstream absolute pressure p_1/p_2) varying between 1.3 and 5 (choking occurs at a pressure ratio of 1.89). Two of the valves had axial annular flow, two had axial flow with an offset rotating plug and one was a gate valve. The remainder were two twin and one single ported mushroom valve, all with cross flow through the port. The flow capacity and hence the fully open throat area of all the regulator valves was around 2000 mm^2, about half that of the gate valve. The air flow velocity through the valve was not measured but a characteristic velocity was calculated from the measurements of p_1/p_2 and the gas temperature assuming complete adiabatic expansion. Each valve was tested at from six to ten different valve openings over its operating range. Although some difficulties were experienced in maintaining the valve throat area constant in all cases, the measurements did confirm that, as expected, once it was choked the mass flow was directly proportional to the upstream pressure.

2.2. Variation of source strength with flow density

If the noise is generated aerodynamically according to equation (1) then intensity should be proportional to the flow density at constant flow velocity or pressure ratio. Since it was not possible to pressurise the reverberant room, the discharge pipe from the valve under test was extended to a second valve outside the room. This involved a further calibration of the chamber for the noise induced by the new external control valve and taking due account of transmission loss through the discharge pipe to the room when evaluating the measurements. Transmission loss was also determined independently and is reported elsewhere[3].

Measurements at a constant pressure ratio of two with downstream pressure increasing from two to seven atmospheres established that the noise transmitted to the room increased by 6 dB for each doubling of pressure, that is, was proportional to the square of the flow density. The transmitted sound power will be proportional to the square of the acoustic pressure difference across the pipe walls and since the transmission loss is large (more than 20 dB) we can neglect the pressure fluctuations in the chamber when calculating this difference. The mean square acoustic pressure fluctuations in the pipe p^2 will be, from (1),

$$p^2 = \rho a_o I = K \rho^2 a_o V^n \qquad (2)$$

where a_o is the speed of sound. This is in good agreement with the measurements so it supports the assumption that the valve noise is generated aerodynamically.

2.3. Measurements of maximum valve noise power

Space does not permit a detailed presentation here of all the valve noise measurements for the wide range of operating conditions that were investigated. For all the valves the noise power always increased with increase in flow velocity (i.e. pressure ratio) and in valve port area (i.e. mass flow). When the valves were not choked ($p_1/p_2 < 1.89$) there were large differences between the noise output of the individual valves which were less obvious once choking occurred. As a general rule, however, the noise power output was proportional to the energy dissipation in the valve, the greatest output being obtained with the valve approaching full delivery with the highest pressure ratio.

Once the valves were choked, some 40 runs at fixed valve opening indicate that, with few exceptions, the noise power varies as the fully expanded velocity to the 8^{th} power. This is typical of turbulent mixing noise (acoustic quadrupoles) and suggests that this forms the major components of the source as it does for jet noise. Assuming this is so, the total noise power will be given by

$$W = k \; \rho V^8 A / a_o^5 \qquad (3)$$

where A is the cross section area. The flow kinetic energy per second will be $\frac{1}{2}\rho V^3 A$, so the factor $2k(V/a_o)^5$ represents the fraction of this that appears as acoustic power. The valve noise power output measured near full delivery lay, with the exception of one valve, within a narrow band of $\pm$ 1 dB, as indicated in Fig. 1. Part of the scatter of $\pm$ 2 dB shown is due to small differences in mass flow. The maximum level of 137 dBA when $p_2/p_1 = 5$ represents a value of 10^{-4} for the factor 2k above, which suggests that the maximum noise output of all but one of the valves can be predicted to within 1 dB by equation 3, once they are choked.

2.4. Valve noise signature.

The variation of valve noise power with reduction of valve opening at constant pressure ratio is plotted in Fig. 2. This represents one mode of operation in normal service. The results show that the acoustic behaviour now differs substantially with valves of different design and examination of the spectra confirms this. When they are fully open the noise spectra of all the valves are the same shape, being smooth and double peaked. The existence of two peaks seems due to the way the flow expands first through the valve throat and then through a conical diffuser connecting the 50 mm valve to the 100 mm discharge piping. The frequency of the peaks corresponds to an equivalent Strouhal number at these two positions. At partial flows, the spectra all exhibit several narrowband peaks and troughs. There is some correspondence between the spectral distributions obtained at different partial valve openings but at the same mass flow, suggesting that the acoustic characteristics of the flow passages downstream of the throat are responsible for this complex behaviour, which at present has not been analysed.

3. CONCLUSIONS

We have seen that satisfactory measurement of valve noise signature can be made using the techniques described here. The measurements show that the noise is generated aerodynamically and the source strength is directly proportional to flow density. Furthermore, the maximum noise of choked regulator valves occurs when they are fully open while its level can be closely predicted by a simple universal relation, equation (3). At fractional valve openings both the level and spectral distribution are significantly affected by the valve geometry, although the noise power is less than the predictions of equation (3). When the valves are not choked the noise generation appears to be strongly dominated by valve geometry.

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REFERENCES

1. Sharpe, J.M. et al. 'Noise abatement at gas pipe-line installations', published by American Gas Assoc. 1960, Vol. II and Vol. III.
2. Schuder, C.B. 'Coping with valve control noise', Chem.Eng. 1970.
3. Small, D.J. 'Noise of high pressure gas regulator valves', Proc. 7th I.C.A., Budapest, 1971, paper 20N15, pp. 345-348.

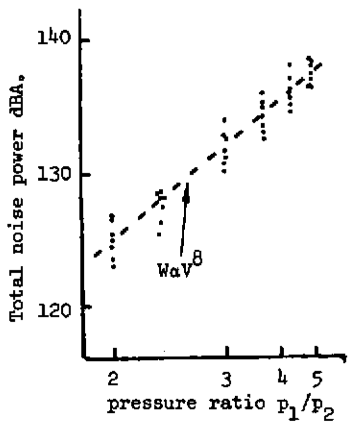

Fig. 1.

Noise power at high flows dBA Ref. 10^{-12} watts.

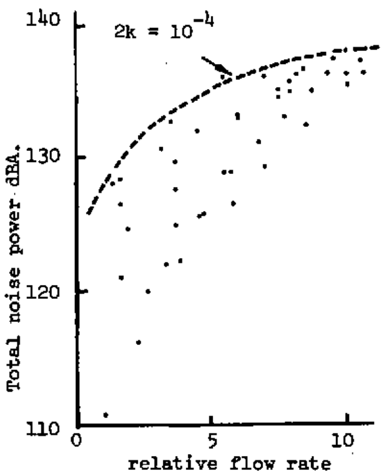

Fig. 2.

Noise power at part flow.

$p_2/p_1 = 5$; dBA ref. 10^{-12} watts.