72AE7

BRITISH ACOUSTICAL SOCIETY - SPRING MEETING
5th-7th April, 1972 - University of Loughborough.

AERODYNAMIC NOISE SOURCES IN INDUSTRY.

CONTROL VALVE NOISE: CAUSE & CURE

A. NELSON & P. SMAGGHE, MASONEILAN LIMITED.

NOMENCLATURE

C_f	Critical-flow factor of control valve, a ratio
C_v	Flow coefficient, gpm./(psi)$^{\frac{1}{2}}$(for water, 60°F.)
f	Frequency, cycles/sec. (Hz.)
g	Gravitational constant, ft./sec.^{2}
P	Sound Pressure, lb./sq.ft.
p	Static Pressure, psia
r	Radial distance to noise source, ft.
Sg	Gas Property correction factor, dB
SPL	Sound-pressure level, db (Ref, 2 x 10^{-4} microbar)
V	Velocity, ft./sec.
W	Power, ft.-lb./sec.
X	Fraction of mechanical power conversion (p_1-p_2/0.47 p_1) limit to 1.
η	Acoustical efficiency, a ratio
ρ	Mean density lb./cu.ft.

Subscripts

a	=	Acoustical
i	=	At vena contracta
m	=	Mechanical
o	=	downstream
s	=	Sonic
v	=	Vapor
1	=	Inlet
2	=	Outlet

CONTROL VALVE NOISE

Noise generated by control valves normally takes one of three separate and distinct forms

1) Noise produced by mechanical vibration of the valve trim
2) Noise produced by cavitating liquids
3) Noise caused by aerodynamic throttling

It is quite important that these three noise sources be understood so far as their generating mechanism is concerned. Only then can their effective improvement or cure be made. Luckily, mechanical vibration noise seldom happens simultaneously with cavitation and aerodynamic noise. However, if this happens the cure for one is usually the cure for the other.

1. NOISE PRODUCED BY MECHANICAL VIBRATION

Two mechanisms are involved: The first is mechanical vibration, induced by pulsation and eddying of the fluid passing through the valve. The frequency is usually low i.e. between 50 and 500 Hz. However, if this turbulence induced vibration of the valve trim approaches the natural frequency of the plug-stem combination, then we have the second mechanism - resonance. This resonance, occuring at frequencies between 2000 and 7000 Hz is most harmful since it can lead to fatigue failure of the valve stem or guide post. It can also hammer away solid stainless steel parts by large amounts.

One can regard this as a good noise since it warns operators that a mechanical failure is in the offing and some action is needed.

The phenomenon has become less common since the introduction of top guided; single seat valves since they generally have less clearance in the guide bushings also, the lower weight of a single seat plug increases the natural frequency of the trim making it less susceptible to fluid induced vibration.

Possible cures for this type of noise include reduction of guide clearances and increase in stem size (a 40% increase in stem dia doubles the undamped natural frequency of the valve trim). Another attempt to cure this problem can be made by changing the flow or pressure conditions to which the valve is subjected. Quite often, changing the valve around in the line to give a reversal of flow direction sufficiently alters the flow pattern to shift inducing frequencies away from the trim excitation range.

2. NOISE PRODUCED BY CAVITATING LIQUIDS

Cavitation noise should never be heard in a well designed process plant. Hardly anything destroys a valve trim as surely as a cavitating liquid. With the intoduction of special valve trims having very little pressure recovery and special valves having multiple velocity head loss trim, there is seldom an excuse to have cavitation in a throttling valve (except, perhaps, for some rather large valve sizes where no anticavitation trim is available as yet).

With the present availability of good engineering data it is possible to predict quite accurately whether or not a selected valve will cavitate under a given process condition. One such equation introduced by Masoneilan International in 1962 allows the prediction of a critical pressure drop at which a given valve will cavitate i.e.

$$\Delta p_{crit} = C_f^2 \, (p_1 - p_v) \qquad - (1)$$

If the operating pressure drop in the plant exceeds Δp crit, then the valve is cavitating. If that be the case, one should solve for the required Critical Flow Factor Cf, then select a valve from the manufacturers catalogue that has a Cf factor equal to or higher than the one calculated by the following equation:

$$Cf = \sqrt{(p_1 - p_2/(p_1 - p_v)} \qquad - (2)$$

Even though cavitation can be avoided in almost all cases there is still interest in the prediction of cavitating noise. Our laboratory investigations indicate noise to be a function of: the amount of decrease in downstream pressure beyond the pressure that causes incipient cavitation, and the difference between downstream pressure and vapour pressure. The peak in cavitation noise can be expected where these two variables are nearly equal i.e. the noise decreases as the difference between p_2 (actual and p_2 (incipient cavitation) approaches zero, and if the difference between the outlet pressure and the vapour pressure approaches zero. The latter is understandable since the process of cavitation is converted into a process of flashing.

To satisfy demands for an empirical equation to predict cavitation noise we suggest:

$$SPL = 10 \log (CvCf) + 8 \log (p_2(crit) - p_2) + 20 \log (p_2 - p_v) + 33 \text{ dbA where } p_2 (crit) = p_1 - C_f^2 (p_1 - p_v) \qquad - (3)$$

Note this equation is only reasonably accurate for water and when using Schedule 40 piping downstream.

3. AERODYNAMIC NOISE

This is the most important form of acoustical annoyance so far as control valves are concerned. Aerodynamic noise is a byproduct of the reconversion of kinetic energy through turbulance into heat downstream of the throttling orifice. There are two basic contributory factors. One is the terminating shock front of a supersonic jet generating from the vena contracta of the valve orifice (at higher than critical pressure drop). The second comes from the general turbulence of the fluid boundary and is effective above, as well as below, choked flow in the valve orifice.

Unfortunately, there is no way to avoid aerodynamic noise, since we have not as yet invented a valve that can reduce pressure without causing turbulence. However, the degree of noise generation can be affected by various parameters.

The sound pressure measured in the proximity of a throttling control valve is a result of pressure waves in the atmosphere. The acoustical power that generates these pressure waves is created by the supersonic shock front in a jet and by turbulent boundary layers within the valve. It is directly related to the mechanical energy converted in the valve. This then makes the SPL value a direct function of mass flow or Cv since the latter is an expression of flow capacity.

Commencing with an expression for the mechanical power converted in a valve:

$$W_m = (\rho i \, v^3 \pi \, (2.3 \times 10^{-4}) CvCf)/8g \quad \text{ft.lb/sec} \qquad - (4)$$

Multiplying by an acoustical efficiency factor yields the acoustical power.

$$Wa = Wm \times \eta \qquad - (5)$$

Conversion into sound pressure can be made as follows:

$$P = (Wa \rho o\ Vs/4\pi r^2 g)^{\frac{1}{2}} \quad \text{lb/sq.ft} \quad - (6)$$

Taking into account a transmission loss and a correction factor for other gases gives the final equation for the sound pressure level at 3ft from the valve outlet and pipe wall :

$$SPL = 10 \log_{10} (X \eta 10'' \ CvCf\ p_1\ p_2) - L_T + Sg - (7)$$

Comparison between calculated and measured SPL's has shown a large measure of agreement on a variety of valves in a range of service conditions.

4. WHAT TO DO ABOUT AERODYNAMIC NOISE

An important requirement in noise control is to keep the valve outlet velocity below a certain limit, depending on the type and size of valve, in order to prevent the occurance of a secondary noise source that might be even worse than that produced by the valve itself. This is particularly important with valves having special "low noise" trim.

If the calculated SPL exceeds the limit by 5 to 10 db the following simple cures are applicable.

1) Increase the pipe wall thickness downstream (doubling the thickness reduces SPL by 5 db).
2) Use acoustical insulation downstream (can reduce SPL by 5 to 10 db)

If the valve noise is more than 10 db above the limit a downstream in-line silencer can attenuate between 10 and 20 db depending on the frequency range.

Another approach recommended by Masoneilan is the use of noise reducing expansion plates downstream of a valve. These absorb some of the pressure reduction over the whole system. Thus, pressure ratio across the valve is kept below critical. The use of multiple plates is recommended where overall pressure ratio is more than 10:1.

Comment has often been made that if a valve is operating in a remote area the noise problem can be disregarded. This is a dangerous simplication since noise is simply audible vibration. The vibration can cause considerable mechanical damage to gauges, valve and pipe mounted instruments. Pipeline bolting at 2500 ANSI rating has been known to loosen under severe vibration.

Elimination of vibration is an important aspect that should not be overlooked. Money spent for safety and for the reduction of maintenance costs, is money well spent.