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Effect of noise generated by pressure control valves in natural gas distribution networks

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Abstract — *Piped natural gas is commonly distributed by regional gas distribution companies, which have pipelines that are responsible for distributing this fluid to final consumers. These meshes operate with maximum pressures in the order of 40 bar (4.0 MPa), however, there are other pressure levels involved, either for operational or safety reasons. The reduction of these working pressures is carried out in devices commonly called pressure reducing stations (EPRs), which have interconnected valves and accessories that enable the referred pressure reductions as necessary. These stations, therefore, include pressure control valves or simply pressure regulators, which produce aerodynamic noise when in operation due to their physical configuration, local flow conditions and their own functionality. This paper presents a proposal for the determination of aerodynamic noise present in pressure regulating valves of piped natural gas distribution networks, based on the resulting mathematical formulation, predicted and recommended in IEC 60534-8-4 "Prediction of noise generated by hydrodynamic flow" code. From these assumptions a computational code was developed in Matlab environment that allows the operations engineer to simulate and evaluate the noise level present in a certain operating condition, favoring his decision making.*

I. INTRODUCTION

Piped natural gas is commonly distributed through regional gas distribution companies, which own pipelines that are responsible for localized transportation and distribution of this fluid to end consumers, whether industrial, commercial, or residential, among others. This distribution meshes operate at lower pressures than transmission pipelines, reaching maximum pressures in the order of 40 bar (4.0 MPa), although there are usually still other pressure levels involved along the meshes, either for reasons of concern operational or safety.

The reduction of these working pressures is performed in devices commonly referred to as pressure reducing stations (EPRs), which feature a set of interconnected valves and accessories that enable said pressure reductions as required. These stations thus include pressure control valves of various types (axial and globe, for example), depending on the flow rates involved and the design configuration considered by each distribution company. Pressure control valves or simply pressure regulators produce aerodynamic noise when in operation due to their physical configuration, local flow conditions and their own functionality.

This noise has been, in many situations, a focal point of attention by regulatory and supervisory bodies, triggered in part by local legislation and current occupational health standards. Between 70 dBA and 85 dBA, the noise source becomes quite pronounced. However, long-term exposure to noise at this level is unlikely to damage human hearing. At 90 dBA and above, the noise source reaches a level where sustained exposure can begin to damage human hearing. For these reasons, guidelines from the Occupational and Safety Hazard Administration (OSHA) mandate an 90 dBA limit for exposure of up to eight hours, as reproduced in "Table 01".

Table.1: OSHA Exposure time limits for various noise levels (Lipták, 2006)

Duration of exposure (h)	Sound level (dBA)
8	90
4	95
2	100
1	105
0.5	110
0.25 or less	115 (max)

The present work presents, therefore, a revision corresponding to the mathematical formulation of the noise generated during the operation of pressure control valves (according to IEC 60534-8-4 "Prediction of noise generated by hydrodynamic flow"), as well as the development of a mathematical algorithm, developed in Matlab environment, which allows a quick assessment of this problem by the operations engineer responsible for a certain piped natural gas distribution network.

II. MAJOR SOURCES OF NOISE

2.1 – Mechanical vibration

Mechanical vibration is caused by the response of internal components within a valve to turbulent flow through the valve. The noise generated by this type of vibration presents a worrying feature, as resonance problems may arise in the valve, a fact that would certainly cause fatigue failures in the internal parts of the valve. In practice, however, noise from mechanical vibration is infrequent in control valves. It is noteworthy that there is currently no reliable method for predicting noise generated by mechanical vibration in control valves.

Mechanical vibration is generally below 100 dBA.

2.2 – Hydrodynamic noise

The Liquid flow noise, cavitation noise, and flashing noise can be generated by the flow of a liquid through a valve and piping system. Of the three noise sources, cavitation is the most serious because noise produced in this manner can be a sign that damage is occurring at some point in the valve or piping.

Hydrodynamic noise caused by liquid turbulence, cavitation, or flashing is generally below 110 dBA.

2.3 – Aerodynamic noise

Aerodynamic noise is a direct result of the conversion of the mechanical energy of the flow into acoustic energy as the fluid passes through the valve restriction. The proportionality of conversion is called acoustical efficiency and is related to valve pressure ratio and design. The aerodynamic noise is the most important component of the acoustic problem of a control valve, since it is generated by the pressure waves produced by the fluid turbulence or by other fluodynamic phenomena connected with supersonic waves. Aerodynamic noise can reach 150 dBA.

Aerodynamic noise prediction is based on the equations and nomenclature of the international standard for control valve noise prediction, IEC 60534-8-4.

III. AERODYNAMIC NOISE PREDICTION

The conversion of static pressure to kinetic energy at the vena contracta in the control valve creates high-velocity jets which can be subsonic, sonic or even supersonic. Turbulence and sonic shock waves create a noise spectrum with a characteristic peak frequency.

The standard IEC 60534-8-3 - consists of a mix of thermodynamic and aerodynamic theory and empirical information. The design of the method allows a noise prediction for a valve based on the measurable geometry of the valve and the service conditions applied to the valve (Emerson, 2017).

The method describes two different noise sources that can contribute to the overall noise generated by the valve: trim noise and valve outlet noise. The trim noise is dependent on the type of trim and its geometric features. The valve outlet noise is dependent on the valve outlet area, valve outlet Mach number and any expander downstream of the valve.

The flow regime for a particular valve is determined from inlet pressure, downstream pressure, fluid physical data, and valve pressure recovery factor. Five flow regimes are defined as (ANSI/ISA, 1991):

Regime I - Subsonic

Regime II - Sonic with turbulent flow mixing (recompression)

Regime III - No recompression but with flow shear mechanism

Regime IV - Shock cell turbulent flow interaction

Regime V - Constant acoustical efficiency (maximum noise)

The control valve aerodynamic noise prediction flowchart illustrated in “Fig. 1” illustrates the steps to be considered by the designer.

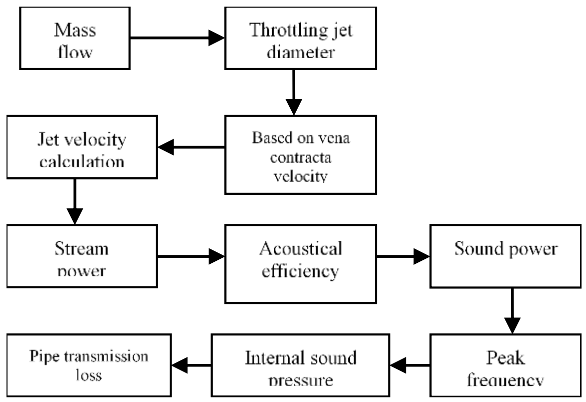

Fig. 1: Aerodynamic noise prediction flowchart on a control valve (Dresser, 2002 – adapted)

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3.1 – Throttling jet diameter

$$D_j = 4.6(10^{-3})F_d\sqrt{C_vF_L} \quad (1)$$

where C_V is the valve capacity and D_j is the equivalent jet diameter at the trim output, which is a typical constructive data for each trim type. It is directly proportional to the trim form factor F_d, whose typical values are easily found in the reference bibliographies (for a fully open circular flow orifice: $F_d = 1$). The most important of these parameters is the recovery coefficient F_L, which, at subsonic flow conditions, represents the energy fraction wasted inside the valve.

3.2 Flow regime

Regime I – Subsonic – this deals with all flow conditions in which the pressure drop is less than or just equal to the pressure drop that would produce sonic flow at the vena contracta. The flow is therefore subsonic and there is isentropic recompression.

$$p_1 > p_2 \geq p_{2C} \quad (2)$$

where p_{2C} is the downstream pressure, corresponding to the threshold of criticity.

$$p_{2C} = p_1 - F_L^{\,2}(p_1 - p_{VCC}) \quad (3)$$

and

$$p_{VCC} = p_1\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \quad (4)$$

where, p_{VCC} is absolute vena contracta pressure at critical flow conditions and γ is the ratio of specific heats.

Under this condition part of the mechanical energy existing in vena contracta is recovered as pressure energy downstream the vena contracta. The remaining energy is wasted by turbulence, thus changing into heat and noise.

Regime II – Sonic with turbulent flow mixing (recompression) - flow is sonic at the vena contracta.

$$p_{2C} > p_2 \geq p_{VCC} \quad (5)$$

Under such a condition the fluid speed in vena contracta reaches the sound speed and supersonic impact waves arise downstream. A loud noise is given out, due to the fact that the sound velocity is reached and other complex aerodynamic disturbances are generated.

Regime III – No recompression but with flow shear mechanism - flow is sonic at the vena contracta but supersonic after, resulting in shock waves. There is no isentropic recompression.

$$p_{VCC} > p_2 \geq p_{2B} \quad (6)$$

where p_{2B} is the valve outlet absolute pressure at break point.

$$p_{2B} = \frac{p_1}{\alpha}\left(\frac{1}{\gamma}\right)^{\frac{\gamma}{\gamma-1}} \quad (7)$$

and α is the recovery correction factor, given by:

$$\alpha = \frac{p_{VCC}}{p_{2C}} \quad (8)$$

Regime IV – Shock cell turbulent flow interaction - flow is sonic at the vena contracta but supersonic after. Shock interaction dominates noise. There is no isentropic recompression.

$$p_{2B} > p_2 \geq p_{2CE} \quad (9)$$

where p_{2CE} is the valve outlet absolute pressure where region of constant acoustical efficiency begins.

Regime V – Shock cell turbulent flow interaction - begins when downstream pressure drops to p_{2CE} and is where acoustical efficiency becomes constant.

$$p_{2CE} > p_2 \quad (10)$$

The velocity at the *vena contracta* is sonic. Further reductions in outlet pressure will not increase the noise level.

3.3 Jet velocity

$$U_{VC} = \sqrt{2\left(\frac{\gamma}{\gamma-1}\right)\left[1-\left(\frac{p_{VC}}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]\frac{p_1}{\rho_1}} \quad (11)$$

where ρ_1 is the upstream specific mass of gas.

3.4 Stream power

The noise of interest is generated by the valve in and downstream of the vena contracta. If the total power dissipated by throttling at the vena contracta can be calculated, then the fraction that is noise power can be determined. The noise from a freely expanding jet of air would be equal to some acoustical efficiency factor multiplied by the power of the jet W_m. Equation (12) is valid for Regime I while equation (13) is valid for Regime II, III, IV and V.

$$W_m = \frac{m_m (U_{VC})^2}{2}\ (watts), \quad (12)$$

$$W_m = \frac{m_m (c_{VCC})^2}{2}\ (watts), \quad (13)$$

where m_m is the mass flow and c_{vcc} is speed of sound in the vena contracta at critical flow conditions, given by:

$$c_{VCC} = \sqrt{\frac{\gamma R T_{VCC}}{M}}, \quad (14)$$

where R is the universal gas constant (8314 J/kmol.K), M is the molecular mass of gas and T_{VCC} is the vena contracta absolute temperature at critical flow conditions.

$$T_{VCC} = \frac{2T_1}{\gamma+1}, \quad (15)$$

3.5 Acoustical efficiency

The acoustic efficiency factor is not constant; it is approximately equal to 1.10^{-4} at *Mach* 1 (Singleton, 1999). It considers the flow conditions applied through the valve to determine the noise generation mechanism in the valve. For the five defined regimes, there is dependence on the relationship of vena pressure contracts and downstream pressure. For each of these regimes, an acoustic efficiency is defined and calculated. This acoustic efficiency establishes the fraction of the total energy flow. When designing a regulating valve, lower acoustic efficiency is desirable. For the 5 different regimes, consider:

$$\eta_1 = (10^{-4}) M_{VC}^{\ 3.6} \quad (16)$$

$$\eta_2 = \eta_3 = (10^{-4}) M_j^{\ 6.6 F_L^{\ 2}} \quad (17)$$

$$\eta_4 = \eta_5 = (10^{-4})\left(\frac{M_j^{\ 2}}{2}\right)\left(\sqrt{2}\right)^{6.6 F_L^{\ 2}} \quad (18)$$

where M_{VC} is the *Mach* number at the vena contracta, and M_j is the freely expanded jet *Mach* number.

$$M_{VC} = \frac{U_{VC}}{c_{VC}} \quad (19)$$

$$M_j = \sqrt{\left(\frac{2}{\gamma-1}\right)\left[\left(\frac{p_1}{\alpha p_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]} \quad (20)$$

3.6 Sound power

The sound power of a source is the total acoustic energy radiated by the source per unit of time. Given in watts.

$$W_{a1} = \eta_1 r_W W_m F_L^{\ 2} \quad (21)$$

$$W_{a2} = \eta_2 r_W W_{ms}\left(\frac{p_1 - p_2}{p_1 - p_{VCC}}\right) \quad (22)$$

$$W_{a3} = \eta_3 r_W W_{ms} \quad (23)$$

$$W_{a4} = \eta_4 r_W W_{ms} \quad (24)$$

$$W_{a5} = \eta_5 r_W W_{ms} \quad (25)$$

3.7 Peak frequency

Because the transmission loss through the pipe walls is affected by the frequency of the sound waves it is experiencing it necessary to calculate the peak frequency of the noise generated by the valve. The peak frequency is given in Hertz.

$$f_{p1} = \frac{0.2 U_{VC}}{D_j} \quad (26)$$

$$f_{p2} = f_{p3} = \frac{0.2 M_j c_{VCC}}{D_j} \quad (27)$$

$$f_{p4} = f_{p5} = \frac{0.35 c_{VCC}}{1.25 D_j \sqrt{M_j^2 - 1}} \quad (28)$$

3.8 Internal sound pressure

Sound power and sound pressure are customarily expressed using the logarithmic scale known as the decibel scale. This scale relates the quantity logarithmically to some standard reference. Thus, the internal sound pressure level, L_{pi}, referenced to 2×10^{-5} Pa for sound pressure is calculated in dB from the following:

$$L_{pi} = 10 log_{10} \left[\frac{3.2(10^9) W_a \rho_2 c_2}{D_i^2}\right] \quad (29)$$

where ρ_2 is the gas density and c_2 is the speed of sound, both at downstream conditions. D_i is the internal downstream pipe diameter.

$$\rho_2 = \rho_1 \left[\frac{p_2}{p_1}\right] \quad (30)$$

$$c_2 = \sqrt{\frac{\gamma R T_2}{M}} \quad (31)$$

3.9 Pipe transmission loss

The pipe vibration mode, for the objective of this prediction method is determined from a simplified model, considering the noise source peak frequency and the natural frequencies from the tube. Natural tube frequencies are functions of tube diameter, wall thickness and density. Transmission loss according to the model used by the IEC standard is based on the work of Fagerlund and Chow (Singleton, 1999).

Ring frequency (f_r) - as its name implies, this is the frequency at which a ring section of the pipe vibrates naturally circumferentially. Frequency that has a wavelength exactly equal to the circumference of the tube, favoring the production of a resonant stress around this circumference. Mathematically:

$$f_r = \frac{5000}{\pi D_i} \quad (32)$$

External coincidence frequency (f_g) - the frequency at which the external acoustic wave speed is equal to the velocity of a flexural wave is the pipe wall. To the speeds of sound of 5000 m/s and 343 m/s, in the steel and in the air, respectively (t = pipe wall thickness):

$$f_g = \frac{12.973}{t} \quad (33)$$

First internal coincidence frequency (f_o) - for frequencies higher than the cutoff frequency (wavelengths shorter) the acoustic pressure waves are able to travel in transverse direction rebounding off the pipe walls as they travel down the pipe. This frequency has the lowest natural frequency of the pipe wall. It produces a longitudinal flexural wave that spirals along the length of the pipe.

$$f_o = \left(\frac{f_r}{4}\right)\frac{c_2}{343} \quad (34)$$

Cutoff frequency (f_c) - frequency that is significant because at the cutoff frequency and below, the wavelengths are too long to reflect off the internal pipe wall, making them incapable of vibrating the pipe.

$$f_c = \frac{0.586 c_2}{D_i} \quad (35)$$

3.10 Transmission loss across the pipe wall

The only way in which noise within the downstream pipe can be experienced outside the pipe is through vibration of the pipe walls. If there is no vibration, no sound will be transmitted. Transmission loss through the pipe wall can be calculated by the equation:

$$TL = 10 log_{10}\left(7.6(10^{-7})\left(\frac{c_2}{t.f_p}\right)^2 \frac{G_x}{\left(\frac{\rho_2 c_2}{415 G_y}+1\right)}\left(\frac{p_a}{p_s}\right)\right) \quad (36)$$

where G_x and G_y are the frequency factors (according to "Table 2"), p_a is the atmospheric pressure outside pipe and p_s is the standard atmospheric pressure.

Table.2: Frequency factors (Lipták, 2006 - adapted)

$f_p < f_o$	$f_p \geq f_o$
$G_x = \left(\frac{f_o}{f_r}\right)^{2/3}\left(\frac{f_p}{f_o}\right)^4$ $G_y = \frac{f_o}{f_g} \Rightarrow f_o < f_g$ $G_y = 1 \Rightarrow f_o \geq f_g$	$G_x = \left(\frac{f_p}{f_r}\right)^{2/3} \Rightarrow f_p < f_r$ $G_x = 1 \Rightarrow f_p \geq f_r$ $G_y = \frac{f_p}{f_g} \Rightarrow f_p < f_g$ $G_y = 1 \Rightarrow f_p \geq f_g$

3.11 Downstream pipe velocity correction factor

Downstream pipe velocity correction factor can be calculated by the equation:

$$L_g = 16log_{10}\left(\frac{1}{1-M_2}\right) \quad (37)$$

where M_2 is the *Mach* number in downstream pipe.

$$M_2 = \frac{4m_m}{\pi D_i{}^2 \rho_2 c_2} \quad (38)$$

3.12Weighted sound pressure level 1 m from pipe wall

The recognized observation point for noise predictions is 1 m downstream from the valve and 1m from the pipe wall (the pipeline is considered as a line source). The sound pressure level is adjusted, with a 5 dB correction to account for all frequency peaks, for this location by:

$$L_{pAe,1m} = 5 + L_{pi} + TL + L_g - 10log_{10}\left(\frac{D_i+2t+2}{D_i+2t}\right) \quad (39)$$

IV. COMPUTATIONAL ROUTINE

The distributed natural gas comes from different production points, configuring variations in the average chemical composition of the gas, a condition that affects, even in small proportions, the calculations performed in the context of the present work. Thus, when proposing the corresponding computational routine, it was decided to allow the end user to adjust the partial percentages of the various independent components that might be part of the final chemical composition of the gas considered. As an example, the average chemical composition of the gas used in the reference calculations of the present work is reproduced in "Table 3". These values are compatible with the average chemical composition of natural gas from Bolivia.

The computational routine developed has the main objective of facilitating the effective evaluations and the necessary calculations, by the operations engineer involved. From simplified input data it is possible to obtain answers about the predominant flow rate type as well as the calculation of the noise generated by the pressure regulating valve.

Table.3: Average chemical composition of the natural gas considered

Component	Mole fraction %	Molar mass kg/kmol
CH_4	0.8901	16.043
C_2H_6	0.0593	30.069
C_3H_8	0.0185	44.096
n-C_4H_{10}	0.0042	58.123
i-C_4H_{10}	0.0031	58.123
n-C_5H_{12}	0.0011	72.151
i-C_5H_{12}	0.0008	72.151
n-C_6H_{14}	0.0008	86.178
N_2	0.0067	28.013
CO_2	0.0154	44.010
Natural gas	1.0000	18.374

In sequence, as an example, the result of a computational routine performed.

Program for Regulators Valves Noise Calculation - according International Standard IEC 60534-8-3

Preliminary data entry**********************

Chemical composition of natural gas analyzed:

CH_4: 89.01

C_2H_6: 5.93

C_3H_8: 1.85

nC_4H_{10}: 0.42

iC_4H_{10}: 0.31

nC_5H_{12}: 0.11

iC_5H_{12}: 0.08

nC_6H_{14}: 0.08

O_2: 0

N_2: 0.67

CO_2: 1.54

total percentage: 1.00

critical pressure of natural gas - P_C (MPa): 4.6316

critical temperature of natural gas - T_C (K): 204.4777

specific heat the constant pressure of natural gas - Cp: 2177.8405

specific heat at constant volume of natural gas - Cv: 1690.4426

ratio between specific heats of the natural gas - (Cp/Cv): 1.2883

molecular mass of the natural gas - M (kg/kmol): 18.3745

enter the value of F_d:0.31

enter the flow coefficient of the valve - C_V:152

specific mass of the fluid upstream - ρ_1 (kg/m^3):10.3642

jet diameter - D_j (m): 0.016679

Calculation of pressure and pressure rates

pressure of "vena contract - Pvc (MPa): 1.5123

pressure of "vena contract" at critical flow conditions - Pvcc (MPa): 1.3696

downstream pressure at the critical pressure drop (sonic flow) - P_{2C} (MPa): 1.5844

break point pressure (shocks turbulent) - P_{2B} (MPa): 0.93234

factor correction: 0.86444

downstream pressure (where acoustical efficiency becomes constant) - P_{2CE} (MPa): 0.13146

Subsonic flow; isentropic recompression; turbulent shear noise - Regime I

gas velocity for the "vena contracta" region - U_{VC} (m/s): 478.8963

stream power for the "vena contracta" region - W_m (Watts): 1043504.4265

absolute temperature for the "vena contracta" region - T_{CV} (K): 476.3826

speed of sound for the "vena contracta" region - c_{VC} (m/s): 526.9724

Mach number for the "vena contracta" region - M_{VC}: 0.90877

acoustical efficiency factor - η_1: 7.0865.10^{-05}

sound power - W_a (watts): 14.9745

peak frequency - f_p (Hz): 5742.604

Noise calculations*************************

downstream mass density - ρ_2 (kg/m^3): 7.0477

downstream sonic velocity - c_2 (m/s): 550.6218

Mach number at valve outlet - Mo: 0.28924

internal sound pressure level - L_{pi} (dB): 162.522

ring frequency - f_r (Hz): 15603.4258

external coincidence frequency - f_g (Hz): 2154.9834

first internal coincidence frequency - f_o (Hz): 6262.0888

cutoff frequency - f_c (Hz): 3163.376

frequency factor - G_x: 0.38479

frequency factor - G_y: 1

transmission loss across the pipe wall - TL: -51.4464

downstream pipe velocity correction factor - L_g: 2.3504

weighted sound pressure level 1m from pipe wall – L_{pAe_1m}: 105.7454

V. CONCLUSION

This paper presents a simplified and fast proposal for the determination of aerodynamic noise in pressure regulating valves used in piped natural gas distribution networks. Such a proposition, focused on the noise prediction standard IEC 60534-8-4, accomplished the difficult task of producing reasonably accurate answers to a more complex problem entirely by theoretical means. The possibility of variation in the input parameters of the problem also allows the professional involved, a more accurate comparative analysis from variable operational data.

It should also be noted that the IEC 60534-8-4 - Prediction of noise generated by hydrodynamic flow - is the most reliable method available at the moment.

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