IEC 60534-8-4

Edition 3.0 2015-09

INTERNATIONAL STANDARD

NORME INTERNATIONALE

Industrial-process control valves –
Part 8-4: Noise considerations – Prediction of noise generated by hydrodynamic flow

Vannes de régulation des processus industriels –
Partie 8-4: Considérations sur le bruit – Prévisions du bruit généré par un écoulement hydrodynamique

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Part 8-4: Noise considerations – Prediction of noise generated by hydrodynamic flow

FOREWORD

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International Standard IEC 60534-8-4 has been prepared by subcommittee 65B: Measurement and control devices , of IEC technical committee 65: Industrial-process measurement, control and automation.

This third edition cancels and replaces the second edition published 2005. This edition constitutes a technical revision.

This edition includes the following significant technical changes with respect to the previous edition:

a) Hydrodynamic noise is predicted as a function of frequency.
b) Elimination of the acoustic power ratio

The text of this standard is based on the following documents:

FDIS	Report on voting
65B/1005/FDIS	65B/1017/RVD

Full information on the voting for the approval of this standard can be found in the report on voting indicated in the above table.

This publication has been drafted in accordance with the ISO/IEC Directives, Part 2.

A list of all parts in the IEC 60534 series, published under the general title *Industrial-process control valves*, can be found on the IEC website.

The committee has decided that the contents of this publication will remain unchanged until the stability date indicated on the IEC web site under "http://webstore.iec.ch" in the data related to the specific publication. At this date, the publication will be

- reconfirmed,
- withdrawn,
- replaced by a revised edition, or
- amended.

INTRODUCTION

It is valuable to predict the noise levels that will be generated by valves. Safety requirements, such as the occupational health standards require that human exposure to noise be limited. There is also data indicating that noise levels above certain levels could lead to pipe failure or affect associated equipment. See IEC 60534-8-3. Earlier hydrodynamic noise standards relied on manufacturer test data and were neither generic nor as complete as desired. The method can be used with all conventional control valve styles including globe, butterfly, cage type, eccentric rotary, and modified ball valves.

A valve restricts flow by converting pressure energy into turbulence, heat and mechanical pressure waves in the fluid contained within the valve body and piping. A small portion of this mechanical vibration is converted into acoustical energy. Most of the noise is retained within the piping system with only a small portion passing through the pipe wall downstream of the valve. Calculation of the mechanical energy involved is straightforward. The difficulties arise from determining first the acoustic efficiency of the mechanical energy to noise conversion and then the noise attenuation caused by the pipe wall.

This part of IEC 60534 considers only noise generated by normal turbulence and liquid cavitation. It does not consider any noise that might be generated by mechanical vibrations, flashing conditions, unstable flow patterns, or unpredictable behaviour. In the typical installation, very little noise travels through the wall of the control valve body. The noise predicted is that which would be measured at the standard measuring point of 1 m downstream of the valve and 1 m away from the outer surface of the pipe in an acoustic free field. Ideal straight piping is assumed. Since an acoustic free field is seldom encountered in industrial installations, this prediction cannot guarantee actual results in the field.

This prediction method has been validated with test results based on water covering a majority of control valve types, in the DN 15 to DN 300 size range, at inlet pressures up to 15 bar. However, some types of low noise valves may not be covered. This method is considered accurate within ± 5 dB(A), for most cases, if based on tested values of x_{FZ} using the method from IEC 60534-8-2. The applicability of this method for fluids other than water is not known at this time.

INDUSTRIAL-PROCESS CONTROL VALVES –

Part 8-4: Noise considerations – Prediction of noise generated by hydrodynamic flow

1 Scope

This part of IEC 60534 establishes a method to predict the noise generated in a control valve by liquid flow and the resulting noise level measured downstream of the valve and outside of the pipe. The noise may be generated both by normal turbulence and by liquid cavitation in the valve. Parts of the method are based on fundamental principles of acoustics, fluid mechanics, and mechanics. The method is validated by test data.

2 Normative references

The following documents, in whole or in part, are normatively referenced in this document and are indispensable for its application. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

IEC 60534-1, *Industrial-process control valves – Part 1: Control valve terminology and general considerations*

IEC 60534-2-3, *Industrial-process control valves – Part 2-3: Flow capacity – Test procedures*

IEC 60534-8-2, *Industrial-process control valves – Part 8-2: Noise considerations – Laboratory measurement of noise generated by hydrodynamic flow through control valves*

IEC 60534-8-3, *Industrial-process control valves – Part 8-3: Noise considerations – Control valve aerodynamic noise prediction method*

3 Terms and definitions

For the purpose of this document, all of the terms and definitions given in IEC 60534 series and the following apply:

3.1
acoustical efficiency η
ratio of the stream power converted into sound power propagating downstream to the stream power of the mass flow

3.2
fluted vane butterfly valve
butterfly valve which has flutes (grooves) on the face(s) of the disk. These flutes are intended to shape the flow stream without altering the seating line or seating surface

3.3
independent flow passage
flow passage where the exiting flow is not affected by the exiting flow from adjacent flow passages

3.4
peak frequency f_p
frequency at which the internal sound pressure is maximum

3.5
valve style modifier F_d
ratio of the hydraulic diameter of a single flow passage to the diameter of a circular orifice, the area of which is equivalent to the sum of areas of all identical flow passages at a given travel

4 Symbols

Symbol	Description	Unit
$A(f)$	Frequency dependent A-weighting value	dBA (ref P_o)
A_η	Valve correction factor for acoustic efficiency (see Table 2)	Dimensionless
c_1	Speed of sound in liquid	m/s
c_a	Speed of sound in air at standard conditions = 343	m/s
c_S	Speed of sound in pipe (for steel pipe 5 000)	m/s
C	Flow coefficient (K_v and C_v)	Various (see IEC 60534-1)
C_R	Flow coefficient (K_v and C_v) at rated travel	Various (see IEC 60534-1)
C_i	Flow coefficents of n stages (i=1…n) in a multistage valve (K_v and C_v)	Various (see IEC 60534-1)
C_n	Flow coefficent of last stage in a multistage valve (K_v and C_v)	Various (see IEC 60534-1)
D_i	Internal pipe diameter	m
D_j	Jet diameter	m
D	Nominal valve size	m
d_H	Multihole trim hole diameter	m
d_o	Seat or orifice diameter	m
F_{cav}	Frequency distribution function (cavitating)	Dimensionless
F_d	Valve style modifier	Dimensionless
F_L	Liquid pressure recovery factor of a valve without attached fittings	Dimensionless
F_{Ln}	Liquid pressure recovery factor of the last throttling stage	Dimensionless
F_{turb}	Frequency distribution function (turbulent)	Dimensionless
f	Frequency	Hz
f_c	Cutoff frequency	Hz
f_{ji}	Octave band frequency	Hz
f_r	Ring frequency	Hz
$f_{p,turb}$	Internal peak sound frequency (turbulent)	Hz
$f_{p,cav}$	Internal peak sound frequency (cavitating)	Hz
K_c	Differential pressure ratio of incipient choked flow (approximately in the range of F_L^3 to F_L^2)	Dimensionless
$L_{pe,1m}$	External sound pressure level 1 m from pipe wall	dB (ref P_o)

Symbol	Description	Unit
$L_{pAe,1m}$	A-weighted external sound pressure level 1 m from pipe wall	dBA (ref P_o)
$L_{pAe,1m,i}$	A-weighted external sound pressure level 1 m from pipe wall of stage i (number i from 1…n) in multistage valve with n stages	dBA (ref P_o)
L_{pi}	Internal sound pressure level at pipe wall	dB (ref P_o)
$\dot{m}$	Mass flow rate	kg/s
n	Number of stages in multistage trim	Dimensionless
N	Numerical constants (see Table 1)	Various
N_o	Number of independent and identical flow passages in valve trim or throttling stage	Dimensionless
P_a	Reference pressure = 1×10^5	Pa
P_o	Reference sound pressure = 2×10^{-5}	Pa
p_1	Valve inlet absolute pressure	Pa
p_2	Valve outlet absolute pressure	Pa
$p_{1,i}$	Inlet absolute pressure of stage i (number i from 1…n) in multistage valve with n stages	Pa
$p_{2,i}$	Outlet absolute pressure of stage i (number i from 1…n) in multistage valve with n stages	Pa
p_v	Vapour pressure of liquid	Pa
Δp	Pressure differential	Pa
Δp_c	Pressure differential for U_{vc} calculation	Pa
St_p	Strouhal number for peak frequency calculation	Dimensionless
t_S	Pipe wall thickness	m
TL	Transmission loss	dB
TL_{fr}	Transmission loss at ring frequency f_r	dB
U_{vc}	*Vena contracta* velocity	m/s
W_a	Sound power of noise created by valve flow which propagates downstream	W
W_m	Mechanical stream power	W
x_F	Differential pressure ratio	Dimensionless
x_{Fz}	Differential pressure ratio of incipient cavitation noise with inlet pressure of 6×10^5 Pa	Dimensionless
x_{Fzp1}	Differential pressure ratio corrected for inlet pressure	Dimensionless
η_{turb}	Acoustic efficiency factor (turbulent)	Dimensionless
η_{cav}	Acoustic efficiency factor (cavitating)	Dimensionless
η_s	Acoustic efficiency factor of pipe wall	Dimensionless
ρ_1	Density of liquid	kg/m^3
ρ_a	Density of air = 1,293	kg/m^3
ρ_S	Density of pipe material (= 7 800 for steel)	kg/m^3

Table 1 – Numerical constants *N*

Constant	Flow coefficent	
	K_v	C_v
N_{14}	$4,9 \times 10^{-3}$	$4,6 \times 10^{-3}$
N_{34}	1	1,17

5 Preliminary calculations

5.1 Pressures and pressure ratios

There are several pressures and pressure ratios needed in the noise prediction procedure. They are given below.

The differential pressure ratio x_F for liquids depends on the pressure difference p_1-p_2 and the difference of the inlet pressure p_1 and the vapour pressure p_v.

$$x_F = \frac{p_1 - p_2}{p_1 - p_v} \tag{1}$$

The differential pressure for beginning choked flow is approximately $F_L{}^2(p_1 - p_v)$. Some calculations are based on the following pressure differential:

$$\Delta p_c = \textit{lower of } (p_1 - p_2) \textit{ or } F_L{}^2 (p_1 - p_v) \tag{2}$$

For low differential pressure ratios, the noise is mainly generated by turbulence. If x_F exceeds $x_{z\ F,p1}$ cavitation noise overlays the turbulent noise. At $x_F = 1$, cavitation noise has a second minimum and for $x_F > 1$, in the flashing region, there is a very gradual increase in sound level as x_F increases above $x_F = 1$.

5.2 Characteristic presssure ratio x_{Fz}

The valve specific characteristic pressure ratio x_{Fz} can be measured with dependency on the valve travel according to IEC 60534-8-2. It should not be confused with K_c, the value at which choked flow caused by cavitation starts. It identifies the pressure ratio at which the cavitation is acoustically detected. The value of x_{Fz} depends on the valve and closure member type and the specific flow capacity.

Alternatively, the value of x_{Fz} can be estimated from equations (3), (4), and (5). Calculations of hydrodynamic noise based on equation (3), (4) and (5) can create uncertainties as illustrated in Annex A. Figures 4 to 9 include typical curves of x_{Fz} for different control valve types. Both equation (3a) and Figures 4 to 9 are based on an inlet pressure of 6×10^5 Pa. If a different inlet pressure is required, then the x_{Fz} value shall be corrected using equation (5).

$$x_{Fz} = \frac{0{,}90}{\sqrt{1 + 3\, F_d \sqrt{\dfrac{C}{N_{34}\, F_L}}}} \quad \text{for valve types except multihole trims} \tag{3}$$

$$x_{Fz} = \frac{1}{\sqrt{4{,}5 + 1650 \dfrac{N_0\, d_H{}^2}{F_L}}} \quad \text{for multihole trims} \tag{4}$$

NOTE N_{34} is a numerical constant, the values of which account for the specific flow coefficient (K_v or C_v) used.

When x_{Fz} is obtained by testing at an inlet pressure of 6×10^5 Pa, then the tested value shall be corrected for the actual inlet pressure using the following equation and using x_{Fzp1} in place of x_{Fz}:

$$x_{Fzp1} = x_{Fz} \left(\frac{6 \times 10^5}{p_1} \right)^{0,125} \quad (5)$$

5.3 Valve style modifier F_d

The valve style modifier depends on the valve and closure member type and on the flow coefficient C (see IEC 60534-2-3).

5.4 Jet diameter D_j

The jet diameter D_j can be predicted as in IEC 60534-8-3 per the following equation:

$$D_j = N_{14} F_d \sqrt{C F_L} \quad (6)$$

5.5 Jet velocity

The *vena contracta* flow velocity, used in calculating the mechanical power, is determined as follows:

$$U_{vc} = \frac{1}{F_L} \sqrt{\frac{2 \Delta p_c}{\rho_L}} \quad (7)$$

5.6 Mechanical power W_m

The mechanical energy dissipated in the valve orifice is determined from the following equation:

$$W_m = \frac{\dot{m} U_{vc}^2 F_L^2}{2} \quad (8)$$

6 Noise predictions

6.1 Internal sound pressure calculation

The portion of the mechanical power W_m from 5.6 converted to valve internal noise and radiated into the downstream pipe is a function of the acoustic efficiency η.

For turbulent conditions defined here where ($x_F \leq x_{Fzp1}$):

$$W_a = \eta_{turb} W_m \quad (9)$$

For cavitating conditions defined here where ($x_{Fzp1} < x_F < 1$):

$$W_a = (\eta_{turb} + \eta_{cav}) W_m \quad (10)$$

For turbulent flow due to the relatively low fluid velocity U_{vc} the valve is considered a monopole source with an acoustical efficiency of approximately 10^{-4} at U_{vc} = c_1 *(see* reference [1][1]). The acoustic efficiency factor for turbulent flow is calculated as follows using A_η from Table 2:

$$\eta_{\text{turb}} = 10^{A_\eta} \left(\frac{U_{\text{vc}}}{c_1} \right) \tag{11}$$

Table 2 – Typical values of A_η

Valve or fitting	A_η
Globe, parabolic plug	–4,6
Globe, V-port plug	–4,6
Globe, ported cage design	–4,6
Globe, multihole drilled plug or cage	–4,6
Butterfly, eccentric	–4,3
Butterfly, swing-through (centered shaft), to 70°	–4,3
Butterfly, fluted vane, to 70°	–4,3
Butterfly, 60° flat disk	–4,3
Eccentric rotary plug	–4,6
Segmented ball 90°	–4,6
Drilled hole plate fixed resistance	–4,6
Expanders	–4,0

Additional noise is produced as cavitation begins. Cavitation is the second part of a two-part process. Vapour bubbles develop when the pressure at a point is lower than the vapour pressure of the fluid at that point. This occurs at the *vena contracta* or point of maximum velocity and minimum pressure in the valve. The second part of this process is the collapse of these vapour bubbles as the fluid pressure rises above the vapour pressure as the vapour leaves the point of minimum pressure. The energy which created the bubbles is returned to the flowing fluid in the form of a high intensity jet as the bubble collapses. This can cause noise and serious damage. The process of cavitation, the energies involved, the reasons that water is one of the most destructive liquids, and why some other liquids cause less damage is part of current hydraulic research.

Reference [3] includes a mathematical model for the sound power of a cavitating jet. The calculation noise prediction model includes the fact that cavitation occurs in a turbulent flow field because at any point the static pressure varies randomly with time and that there is the probability that at some instant the pressure falls below the threshold pressure (i.e. nearly the vapour pressure). They define the average duration of a pressure minimum with values lower than the threshold pressure. This depends on the peak frequency of turbulent noise. Together with a constant velocity bubble-growth model, the radius of the most-frequently occurring cavitation bubbles can be estimated. After these bubbles have grown to a certain size, they collapse in the collapse time, which determines the peak frequency of the cavitation noise.

In the cavitation region ($x_{\text{Fzp1}} \leq x_{\text{F}} \leq 1$), this modified theoretical model (see reference [2]) for cavitating jets combined with many test results for validation leads to the following acoustical efficiency factor equation.

$$\eta_{\text{cav}} = 0{,}32\,\eta_{\text{turb}} \sqrt{\frac{p_1 - p_2}{\Delta p_{\text{c}}} \frac{1}{x_{\text{Fzp1}}}}\, \exp(5 x_{\text{Fzp1}}) \left(\frac{1 - x_{\text{Fzp1}}}{1 - x_{\text{F}}} \right)^{0{,}5} \left(\frac{x_{\text{F}}}{x_{\text{Fzp1}}} \right)^{5} (x_{\text{F}} - x_{\text{Fzp1}})^{1{,}5} \tag{12}$$

[1] Numbers in square brackets refer to the bibliography.

where “exp(x)” represents the constant e raised to the power of the object x.

The internal sound pressure level L_{pi} is calculated as follows:

$$L_{pi} = 10\log_{10}\left(\frac{3{,}2\ x\ 10^{9}\ W_a\ \rho_1\ c_1}{D_i^2}\right) \quad (13)$$

where the appropriate value for W_a is from equation (9) or (10), depending on whether turbulent or cavitating flow, is used.

Using equations (14) and (15), the internal sound pressure level can be predicted at each third octave center frequency, f_i, as given in Table 3.

For turbulent conditions ($x_F \le x_{Fzp1}$):

$$L_{pi}(f_i) = L_{pi} + F_{turb}(f_i) \quad (14)$$

For cavitating conditions ($x_{Fzp1} < x_F < 1$):

$$L_{pi}(f_i) = L_{pi} + 10\log_{10}\left(\frac{\eta_{turb}}{\eta_{turb}+\eta_{cav}}10^{0,1F_{turb}(f_i)} + \frac{\eta_{cav}}{\eta_{turb}+\eta_{cav}}10^{0,1F_{cav}(f_i)}\right) \quad (15)$$

$$F_{turb}(f_i) = -8 - 10\log_{10}\left[\frac{1}{4}\left(\frac{f_i}{f_{p,turb}}\right)^{3} + \left(\frac{f_i}{f_{p,turb}}\right)^{-1}\right] \quad (16)$$

$$F_{cav}(f_i) = -9 - 10\log_{10}\left[\frac{1}{4}\left(\frac{f_i}{f_{p,cav}}\right)^{1,5} + \left(\frac{f_i}{f_{p,cav}}\right)^{-1,5}\right] \quad (17)$$

Table 3 – Indexed third octave center frequencies and "A" weighting factors

Index, i	Third octave center frequency f_i (Hz)	"A" weighting factor $\Delta L_A(f_i)$ (dB)	Index, i	Third octave center frequency f_i (Hz)	"A" weighting factor $\Delta L_A(f_i)$ (dB)
1	12,5	–63,4	18	630	–1,9
2	16*	–56,7	19	800	–0,8
3	20	–50,5	20	1 000*	0
4	25	–44,7	21	1 250	0,6
5	31,5*	–39,4	22	1 600	1,0
6	40	–34,6	23	2 000*	1,2
7	50	–30,2	24	2 500	1,3
8	63*	–26,2	25	3 150	1,2
9	80	–22,5	26	4 000*	1,0
10	100	–19,1	27	5 000	0,5
11	125*	–16,1	28	6 300	–0,1
12	160	–13,4	29	8 000*	–1,1
13	200	–10,9	30	10 000	–2,5
14	250*	–8,6	31	12 500	–4,3
15	315	–6,6	32	16 000*	–6,6
16	400	–4,8	33	20 000	–9,3
17	500*	–3,2			

* Octave center frequencies (octave center frequencies could be used in place of third octave center frequencies, but of course the corresponding index numbers would be changed. If octave bands are used, the constant 8 in equation (12) should be replaced by 3 and the constant 9 in equation (13) should be replaced by 4.)

The peak frequencies are different for turbulent and cavitating flow. The turbulent peak frequency can be calculated as in IEC 60534-8-3 as follows:

$$f_{p,turb} = St_p \frac{U_{vc}}{D_j} \tag{18}$$

$$St_p = \frac{0{,}036\, F_L^2\, C\, Fd^{0,75}}{N_{34}\, x_{Fzp1}^{1,5}\, D\; d_0} \left(\frac{1}{p_1 - p_v}\right)^{0,57} \tag{19}$$

The following equation determines the peak frequency in the cavitation region [2,3,8].

$$f_{p,cav} = 6\, f_{p,turb} \left(\frac{1 - x_F}{1 - x_{Fzp1}}\right)^2 \left(\frac{x_{Fzp1}}{x_F}\right)^{2,5} \tag{20}$$

6.2 Transmission loss

As in IEC 60534-8-3 for aerodynamic flow, the following frequencies are needed to calculate the transmission loss.