

# 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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ICS 17.140.20; 23.060.40; 25.040.40

ISBN 978-2-8322-2879-1

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**INDUSTRIAL-PROCESS CONTROL VALVES –****Part 8-4: Noise considerations –  
Prediction of noise generated by hydrodynamic flow**

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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

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## 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  $\pm 5$  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 $\eta$**

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 $P_0$ )
$A_n$	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 coefficients of $n$ stages ( $i=1\dots n$ ) in a multistage valve ( $K_v$ and $C_v$ )	Various (see IEC 60534-1)
$C_n$	Flow coefficient 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 $P_0$ )

Symbol	Description	Unit
$L_{pAe,1m}$	A-weighted external sound pressure level 1 m from pipe wall	dBA (ref $P_0$ )
$L_{pAe,1m,i}$	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 $P_0$ )
$L_{pi}$	Internal sound pressure level at pipe wall	dB (ref $P_0$ )
$\dot{m}$	Mass flow rate	kg/s
$n$	Number of stages in multistage trim	Dimensionless
$N$	Numerical constants (see Table 1)	Various
$N_0$	Number of independent and identical flow passages in valve trim or throttling stage	Dimensionless
$P_a$	Reference pressure = $1 \times 10^5$	Pa
$P_0$	Reference sound pressure = $2 \times 10^{-5}$	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
$\Delta p$	Pressure differential	Pa
$\Delta p_c$	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
$U_{vc}$	<i>Vena contracta</i> 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 $6 \times 10^5$ Pa	Dimensionless
$x_{Fzp1}$	Differential pressure ratio corrected for inlet pressure	Dimensionless
$\eta_{turb}$	Acoustic efficiency factor (turbulent)	Dimensionless
$\eta_{cav}$	Acoustic efficiency factor (cavitating)	Dimensionless
$\eta_s$	Acoustic efficiency factor of pipe wall	Dimensionless
$\rho_l$	Density of liquid	kg/m <sup>3</sup>
$\rho_a$	Density of air = 1,293	kg/m <sup>3</sup>
$\rho_s$	Density of pipe material (= 7 800 for steel)	kg/m <sup>3</sup>

**Table 1 – Numerical constants  $N$** 

Constant	Flow coefficient	
	$K_v$	$C_v$
$N_{14}$	$4,9 \times 10^{-3}$	$4,6 \times 10^{-3}$
$N_{34}$	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  $p_v$ .

$$x_F = \frac{p_1 - p_2}{p_1 - p_v} \quad (1)$$

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:

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

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 pressure 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  $6 \times 10^5$  Pa. If a different inlet pressure is required, then the  $x_{Fz}$  value shall be corrected using equation (5).

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

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

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  $6 \times 10^5$  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:

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

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### 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  $\eta$ .

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]<sup>1</sup>). The acoustic efficiency factor for turbulent flow is calculated as follows using  $A_\eta$  from Table 2:

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

**Table 2 – Typical values of  $A_\eta$**

Valve or fitting	$A_\eta$
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 ( $x_{Fzp1} \leq x_F \leq 1$ ), 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.

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

<sup>1</sup> 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:

$$L_{pi} = 10 \log_{10} \left( \frac{3,2 \times 10^9 W_a \rho_1 c_1}{D_1^2} \right) \quad (13)$$

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 ( $x_F \leq x_{Fzp1}$ ):

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

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

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

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

<https://standards.iteh.ai/catalog/standards/sist/bbfa131a-42bf-437c-b005-1c117a7f514d/iec-60534-8-4-2015>

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

**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:

$$f_{p,turb} = St_p \frac{U_{vc}}{D_j} \quad (18)$$

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

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

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

## 6.2 Transmission loss

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