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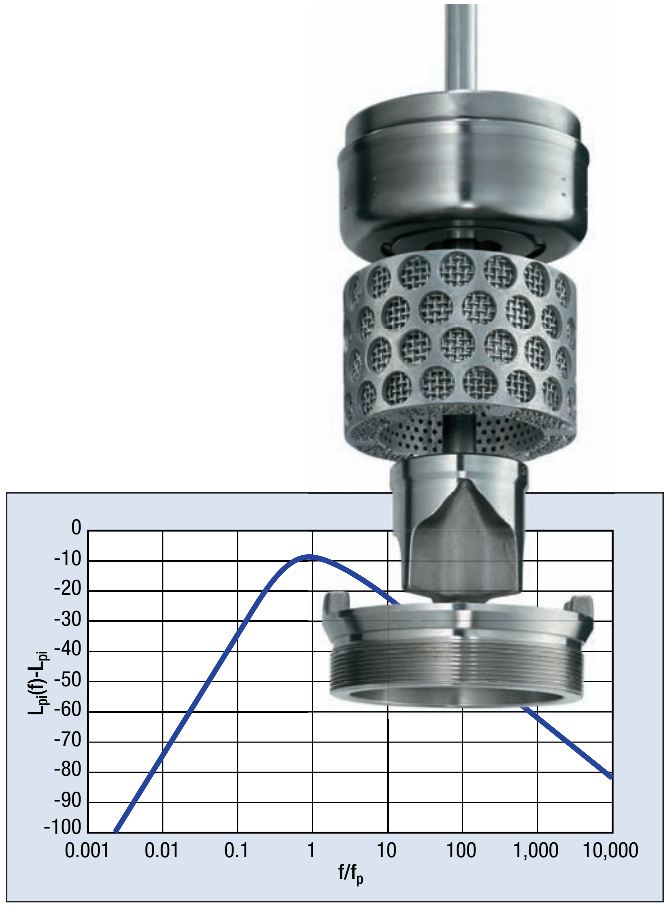

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Improvement of IEC 60534-8-3 standard for noise prediction in control valves

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The current IEC 60534-8-3 standard for predicting aerodynamic noise in control valves regulating the flow of gases and vapors is being revised by Working Group 9 of the IEC Subcommittee 65B. The tests described in reference 1 established that the present standard can lead to considerable inaccuracies concerning the predicted sound pressure level. A similar conclusion was drawn at previous meetings of the above mentioned working group and at a seminar held by Valve World magazine.[2] In this article, possible improvements are described.

The current IEC 60534-8-3 [3]

The currently valid version of this standard is mainly based on publications written by Lighthill[4] and Reethof.[5] The standard contains the following stages:

- Calculating the mechanical stream power, W_m
- Calculating the acoustical efficiency factor, η, for five flow regimes
- Determining the acoustic power ratio, r_W, (ratio between sound power in the downstream pipe and inside the valve)
- Calculating the peak frequency, f_p , for internal noise
- Calculating the internal sound pressure level, L_{pi}, in the downstream pipe
- Calculating the transmission loss, TL
- Calculating the sound pressure level, $L_{pAe,1m}$, one meter away from the pipe on the outlet side.

Extensive laboratory measurements on standard valves[1] show the following pattern of accuracy achieved using the currently valid method (Fig. 1):

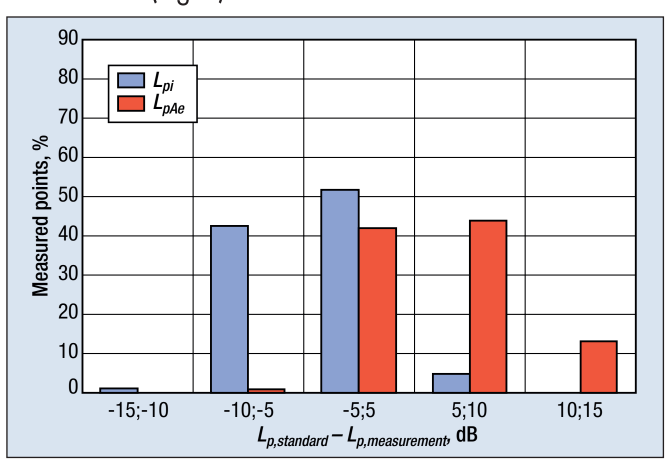

Fig. 1: Accuracy achieved using the current IEC 60534-8-3 standard.[3]

Just 42 % of all the 3,000 measured points of $L_{pAe,1m}$ are within the error band of ±5 dB, whereas 43 % are within the error band of 5 dB to 10 dB. Therefore, the calculated sound pressure level is often overpredicted by 5 dB to 10 dB.

While 51 % of all calculated values for L_{pi} are within a ±5 dB tolerance band, 42 % are found to be in the –10 dB to –5 dB band and are, therefore, underpredicted.

This situation, which has been confirmed by other manufacturers and users, is unsatisfactory and needs to be improved.

Potential for IEC 60534-8-3 improvement

Internal noise in the pipe. Basically, the noise transmitted to the downstream pipe is proportional to W_m. The proportionality factor is η, which depends on the pressure ratio or differential pressure ratio, x, the valve F_L factor and the isentropic exponent, γ. An additional factor of influence is the flow regime type (I to V). The following equations are not consistent with those in the standard, but have the same background and lead to identical results. They are used since they provide better clarity.

The stream power of mass flow depends on the mass flow rate, ṁ, and the velocity at the vena contracta, U_{vc}, i.e., at the narrowest point between the plug and seat:

$$W_m = \frac{\dot{m}\left(U_{vc}\right)^2}{2} \tag{1}$$

The maximum value possible for W_m is W_{ms} since the stream velocity cannot exceed the speed of sound at the vena contracta, c_{vc}:

$$W_{ms} = \frac{\dot{m}\left(c_{vc}\right)^2}{2} \tag{2}$$

The internal sound power, W_a, calculated using η (in the current IEC: $\eta_{IEC,current} \cdot r_W$) is:

$$W_a = \eta W_m \tag{3}$$

R	Condition	Mach number, *M*	Acoustical efficiency factor, η	f_p
I	$x \le x_C$	$\sqrt{\left(\frac{2}{\gamma-1}\right)\left[\left(1-\frac{x}{F_L^2}\right)^{(1-\gamma)/\gamma}-1\right]}$	$\left(1\times10^{A_\eta}\right)F_L^2M^{3.6}$	$\frac{St_pMc_{vc}}{D_j}$
II	$x_C < x \le x_{vcc}$	$\sqrt{\frac{2}{\gamma-1}\left[\left(\frac{1}{\alpha(1-x)}\right)^{(\gamma-1)/\gamma}-1\right]}$	$\left(1\times10^{A_\eta}\right)\frac{x}{x_{vcc}}M^{6.6F_L^2}$	$\frac{St_pMc_{vcc}}{D_j}$
III	$x_{vcc} < x \le x_B$	$\sqrt{\frac{2}{\gamma-1}\left[\left(\frac{1}{\alpha(1-x)}\right)^{(\gamma-1)/\gamma}-1\right]}$	$\left(1\times10^{A_\eta}\right)M^{6.6F_L^2}$	$\frac{St_pMc_{vcc}}{D_j}$
IV	$x_B < x \le x_{CE}$	$\sqrt{\frac{2}{\gamma-1}\left[\left(\frac{1}{\alpha(1-x)}\right)^{(\gamma-1)/\gamma}-1\right]}$	$\left(1\times10^{A_\eta}\right)\frac{M^2}{2}\left(\sqrt{2}\right)^{6.6F_L^2}$	$\frac{1.4St_pc_{vcc}}{D_j\sqrt{M^2-1}}$
V	$x > x_{CE}$	$\sqrt{\frac{2}{\gamma-1}\left[(22)^{(\gamma-1)/\gamma}-1\right]}$	$\left(1\times10^{A_\eta}\right)\frac{M^2}{2}\left(\sqrt{2}\right)^{6.6F_L^2}$	$\frac{1.4St_pc_{vcc}}{D_j\sqrt{M^2-1}}$

Table 1: Regime-specific equations for the Mach number, acoustical efficiency factor and peak frequency

L_{pi} at the downstream pipe (inside diameter, D_i) for reflection-free acoustic conditions downstream of the valve is determined as:

$$L_{pi} = 10\log_{10}\left[\frac{\left(3.2\times10^9\right)W_a\,\rho_2 c_2}{D_i^2}\right] \tag{4}$$

The flow regime types are defined by varying jet shapes in the area upstream and downstream of the vena contracta. These jets change their shape when certain differential pressure ratios are exceeded.

In regimes II to IV, higher Mach numbers, M, arise downstream of the vena contracta.
Yet, M at the vena contracta itself remains unchanged at 1. Table 1 contains the conditions for the five flow regimes and their effects on M and η.

In regime I, the flow is subsonic. The sound generation has the character of a dipole jet. The highest Mach number is reached at the vena contracta, not exceeding Mach 1 at the maximum. Downstream of the vena contracta, the jet expands, leading to partial pressure recovery (hence the F_L factor).

In regime II, sonic and supersonic flows exist together, which means that strongly turbulent flow and shock cell structure dominate. Pressure recovery drops until the top limit of regime II is reached.

In regime III, the rise in pressure is nonisentropic. The flow is supersonic and shear turbulence predominates.

	New		Presently used		
Valve type	**Aη**	**St_p**	**r_W**	**Aη**	**St_p**
Globe valve, parabolic plug	– 4.0	0.23	0.25	– 4.6	0.2
Globe valve, V-port plug	– 4.0	0.23	0.25	– 4.6	0.2
Globe valve, cage design	– 3.6	0.1	0.25	– 4.6	0.2
Globe valve, multihole drilled plug	– 4.8	0.1	0.25	– 4.6	0.2
Butterfly valve	– 4.1	0.19	0.5	– 4.3	0.2
Rotary plug valve	– 3.6	0.18	0.25	– 4.6	0.2
Segmented ball valve, 90°	– 3.6	0.18	0.25	– 4.6	0.2
Drilled-hole plate fixed resistance	– 4.8	0.1	0.25	– 4.6	0.2

Table 2: Aη and St_p as a function of the valve type

In regime IV, the shock cells disappear and a Mach disk forms. The dominant mechanism is the interaction between shock cells and turbulence.

The acoustical efficiency remains constant in regime V.
The differential pressure ratios that determine the flow regimes (see also Table 1) are defined as:

$$x = 1 - \frac{p_2}{p_1} \tag{5a}$$

$$x_{vcc} = 1 - \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)} \tag{5b}$$

$$x_C = F_L^2 \left(1 - x_{vcc}\right) \tag{5c}$$

$$\alpha = \frac{1 - x_{vcc}}{1 - x_C} \tag{5d}$$

$$x_B = 1 - \frac{1}{\alpha}\left(\frac{1}{\gamma}\right)^{\gamma/(\gamma-1)} \tag{5e}$$

$$x_{CE} = 1 - \frac{1}{22\,\alpha} \tag{5f}$$

Acoustical efficiency factor and introduction of the level coefficient, A_η

To adapt the prediction to the measured data, the coefficient A_η was introduced for η to take into account that the degree of the acoustical efficiency depends on the valve type (globe, butterfly, ball, rotary plug) and on the valve's size and design (valve position, ratio between seat size and nominal size, plug type, etc.).

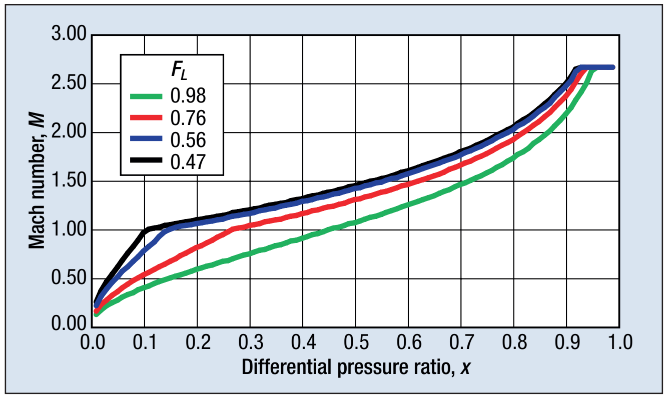

Fig. 2: Mach number, M, as in Table 1 as a function of x and for various F_L values.

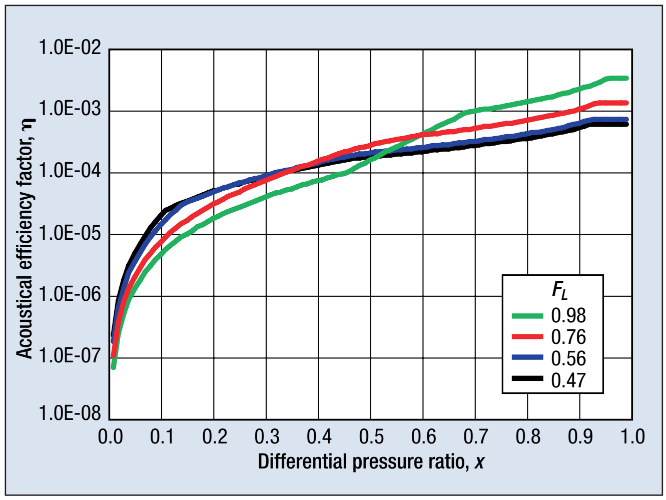

Fig. 3: η as in Table 1 as a function of x and for various F_L values (Aη = –4).

The current IEC 60534-8-3 has fixed the acoustical efficiency factor at –4. However, it refers to the acoustical efficiency inside the valve. The transmission to the internal sound in the downstream pipe is calculated using r_W ($\eta_{IEC,current} \cdot r_W$ decisive).
For the internal sound pressure level measurements as described in reference 6, L_{pi} is measured. Basically, this r_W could be determined from these measurements if the internal acoustical efficiency η_{IEC} is considered to be theoretical. A better comparison is achieved by introducing a η related to the pipe right from the beginning since only this factor can be determined in valve measurements (in the sense of $\eta_{IEC,current} \cdot r_W$).

In the currently valid IEC 60534-8-3 standard, A_η, therefore, corresponds to the following term:

$$A_{\eta,IEC,current} = -4 + 10\log_{10}(r_W) \tag{6}$$

The factor r_W assumes the values 0.25, 0.5 and 1 in the current standard. As a result, A_η assumes the values −4.6, −4.3 and −4.0 (see also Table 2).

The internal sound measurements illustrated in Fig. 1 were used to optimize the A_η value as a typical value for individual valve types. In an actual series of measurements, A_η can of course also be specified depending on the valve plug position or differential pressure ratio. The results using fixed values are listed in Table 2, which produces a clear improvement (see section on accuracy improvement).

Frequency distribution and introduction of the variable Strouhal number, St_p.
The internal noise has a dominating frequency range. A measure for this is the theoretical f_p, which cannot easily be validated though in measured data with one-third octave or octave band analyses. It is more suitable to sort measured data for various differential pressure ratios and then determine the frequency distribution function, $L_{pi}(f)–L_{pi}$, which will be further described.

The peak frequency depends on the equivalent jet diameter, D_j, as well as on M (see Table 1) of the jet, the flow regime type and St_p. The peak frequency behaves differently with a supersonic flow in the higher regimes and when shock cells occur.

$$D_j = 0.0046 F_d \sqrt{C_v \; F_L} \qquad (7)$$

The valve style modifier, F_d, which is also used to calculate the C_v coefficient[7] and predict the hydrodynamic noise in valves[8] (typical values can be found in reference 7), has a considerable influence on D_j and hence, on f_p. The speed of sound at c_{vc} also depends on the differential pressure ratio, but cannot fall below c_{vcc}.

$$c_{vc} = \sqrt{\gamma \frac{p_1}{\rho_1} \left(1 - \frac{x}{F_L^{\,2}}\right)^{(\gamma-1)/\gamma}} \qquad (8)$$

$$c_{vcc} = \sqrt{\frac{2\gamma}{\gamma+1} \frac{p_1}{\rho_1}} \qquad (9)$$

A range from 0.1 to 0.3 for St_p is given in literature.
Therefore, a theoretical mean value of 0.2 was assumed in the current IEC 60534-8-3 standard.

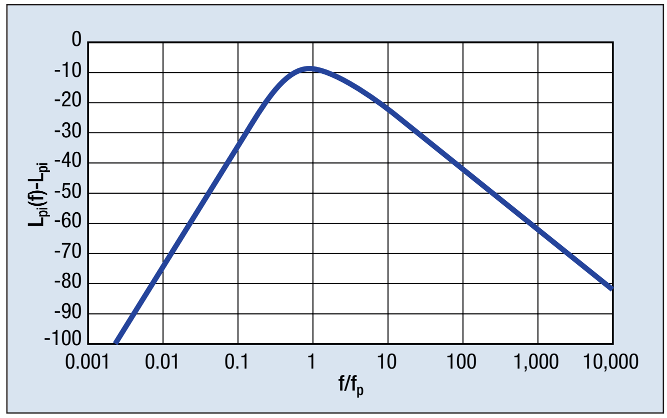

Fig. 4: Function $L_{pi}(f)–L_{pi}$ as a function of f/f_p as per Eq. 10.

Similar to A_η, this value is based though on the valve type. The internal sound measurements in Fig. 1 were used to optimize the St_p value depending on the valve type. The results are listed in Table 2.

Investigations carried out by Reethof[5] have shown that the frequency-dependent internal sound pressure level, $L_{pi}(f_i)$, can be calculated as:

$$L_{pi}(f_i) = L_{pi} - c - 10\,\log\left\{\left[1+\left(\frac{f_i}{2f_p}\right)^2\right]\left[1+\left(\frac{f_p}{2f_i}\right)^4\right]\right\} \qquad (10)$$

The constant, c, is 8 for one-third octave center frequencies and 3 for octave center frequencies.
Figs. 2 and 3 show how M and η are related for Aη = −4, for example. Fig. 4 shows the frequency distribution function from Eq. 10.

External noise. The pipe absorbs noise and only lets a proportion of the internal noise escape to the atmosphere. A measure for this is TL, which depends on the frequency. Other influencing variables include the pipe wall thickness, t_p, the density, ρ_2, and the speed of sound, c_2, of the compressible medium downstream of the valve. The values in the IEC refer to steel.
The current IEC standard defines the transmission loss as follows, however, without the correction term, ΔTL. Nevertheless, the experimental data from Fig. 1 showed that this correction term, which depends on the nominal size, is necessary especially for small nominal sizes.

$$TL(f) = 10\log_{10}\left[\left(7.6\times10^7\right)\left(\frac{c_2}{t_p f}\right)^2 \frac{G_x(f)}{\frac{\rho_2\, c_2}{415 G_y(f)}+1}\right] - \Delta TL \qquad (11a)$$

Fig. 5: Low-noise valve with V-port plug and flow divider.

$$\Delta TL = \left\langle \begin{array}{ll} 0 & \text{for } D > 0.15 \\ \dfrac{16}{(1{,}000D - 46)^{0.36}} & \text{for } 0.05 \le D \le 0.15 \\ 9.7 & \text{for } D < 0.05 \end{array} \right\rangle \quad (11b)$$

The ring frequency, f_r ,and the coincidence pipe frequencies, f_o, and f_g, which are included in it, are determined as:

$$f_r = \frac{5{,}000}{\pi D_i} \quad (11c)$$

$$f_o = \frac{f_r}{4}\left(\frac{c_2}{343}\right) \quad (11d)$$

$$f_g = \frac{\sqrt{3}\,(343)^2}{\pi\, t_p (5{,}000)} \quad (11e)$$

ΔTL equals 0 in the current IEC standard.

The values for $G_x(f)$ and $G_y(f)$ are listed in Table 3.

$L_{pe,1m}(f_i)$ one meter away from the pipe downstream of the valve is calculated as:

$$L_{pe,1m}(f_i) = L_{pi}(f_i) + TL(f_i) - 10\log\left(\frac{D_i + 2t_p + 2}{D_I + 2t_p}\right) \quad (12)$$

$L_{pAe,1m}$ is found by adding up all individual sound pressure levels with the corresponding A-weighting in a nonlogarithmic scale:

$$L_{pAe,1m} = 10\log_{10}\left(\sum_{i=1}^{N} 10^{\frac{Lpe,1m(f_i)+W_{fi}}{10}}\right) \quad (13)$$

where:

$i_{1...N}$ = One-third octave band center frequency index for the following frequencies (12.5, 16, 20, 25, 31.5, 40, 50, 63, 80, 100, 125, 160, 200, 250, 315, 400, 500, 630, 800, 1,000, 1,250, 1,600, 2,000, 2,500, 3,150, 4,000, 5,000, 6,300, 8,000, 10,000, 12,500, 16,000, 20,000 Hz)

W_{fi} = A-weighting factor for the one-third octave band center frequency, fi (–63.4, –56.7, –50.5, –44.7, –39.4, –34.6, –30.2, –26.2, –22.5, –19.1, –16.1, –13.4, –10.9, –8.6, –6.6, –4.8, –3.2, –1.9, –0.8, 0, 0.6, 1, 1.2, 1.3, 1.2, 1, 0.5, –0.1, –1.1, –2.5, –4.3, –6.6, –9.3).

$f < f_o$	$f \ge f_o$
$G_x = \left(\frac{f_o}{f_r}\right)^{2/3}\left(\frac{f}{f_o}\right)^4$	$G_x(f) = \left(\frac{f}{f_r}\right)^{1/2}$ for $f < f_r$ $G_x = 1$ for $f \ge f_r$
$G_y(f) = \left(\frac{f_o}{f_g}\right)$ for $f_o < f_g$ $G_y(f) = 1$ for $f_o \ge f_g$	$G_y(f) = \left(\frac{f}{f_g}\right)$ for $f < f_g$ $G_y(f) = 1$ for $f \ge f_g$

Table 3: Values for $G_x(f)$ and $G_y(f)$

R	Condition	Mach number, M	Acoustical efficiency factor, η	f_p
I	$x \le x_C$	$M = \sqrt{\frac{\log_{10}(1-x)}{\log_{10}(1-x_{cr})}} \le \sqrt{\frac{\gamma+1}{\gamma-1}}$	$(1\times10^{A_\eta})1.176 x_T M^{3,6}$	$\frac{St_p M c_{vc}}{D_j}$
II	$x_C < x \le x_{vcc}$	$M = \sqrt{\frac{\log_{10}(1-x)}{\log_{10}(1-x_{cr})}} \le \sqrt{\frac{\gamma+1}{\gamma-1}}$	$(1\times10^{A_\eta})\frac{x}{x_{vcc}} M^{7.764 x_T}$	$\frac{St_p M c_{vcc}}{D_j}$
III	$x_{vcc} < x \le x_B$	$M = \sqrt{\frac{\log_{10}(1-x)}{\log_{10}(1-x_{cr})}} \le \sqrt{\frac{\gamma+1}{\gamma-1}}$	$(1\times10^{A_\eta}) M^{7.764 x_T}$	$\frac{St_p M c_{vcc}}{D_j}$
IV	$x_B < x \le x_{CE}$	$M = \sqrt{\frac{\log_{10}(1-x)}{\log_{10}(1-x_{cr})}} \le \sqrt{\frac{\gamma+1}{\gamma-1}}$	$(1\times10^{A_\eta}) M^{7.764 x_T}$	$\frac{1.8 St_p M c_{vcc}\sqrt{\alpha(1-x)}}{D_j}$
V	$x > x_{CE}$	$M = \sqrt{\frac{\log_{10}(1-x)}{\log_{10}(1-x_{cr})}} \le \sqrt{\frac{\gamma+1}{\gamma\ \ 1}}$	$(1\times10^{A_\eta}) M^{7.764\, x_T}$	$\frac{1.8 St_p M c_{vcc}\sqrt{\alpha(1-x)}}{D_j}$

Table 4: Regime-dependent equations for the Mach number, acoustical efficiency factor and peak frequency applying the new calculation method for the Mach number according to Eqs. 16a and b

The current standard takes a different approach. The transmission loss is calculated only at f_p, and the influence of the other frequencies and A-weighting is represented by the 5-dB constant:

$$L_{pAe,1m} = L_{pi} + 5 + TL(f_p) - 10\log_{10}\left[\frac{D_i + 2t_p + 2}{D_i + 2t_p}\right] \quad (14)$$

This constant (5 dB) is not correct for frequency profiles different from the one in Eq. 10 and in limit ranges of the ring or peak frequencies. A more accurate approach is, therefore, provided by using Eqs. 12 and 13.

Until now, the noise generated inside the valve was examined. At high valve outlet velocities greater than 0.3 Mach, however, other sources of noise act downstream of the valve, which arise when the flow breaks off and the shock cells shift, for example, into the pipe expansion. These effects, which, in principal, are correct, are taken into account in the current standard. In the new standard, this means a higher internal sound pressure level, L_{pis}, coupled with a different frequency spectrum, $L_{piR}(f_i)$, which is superimposed as follows:

$$L_{piS}(f_i) = 10\log_{10}\left(10^{L_{pi}(f)/10} + 10^{L_{pir}(f)/10}\right) \quad (15)$$

The other equations to calculate $L_{piR}(f_i)$ are not specified in this article because the focus lies with valve noise for Mach numbers at the valve outlet below 0.3. This is achieved by carefully sizing the valve to prevent such critical conditions from occurring.

Procedures to be taken with special valves. The current IEC 60534-8-3 standard deals with determining the pressure ratios in multistage valves, for example, at the last stage before the outlet and then the noise prediction for just this last stage. The influence of the other stages is taken into account using a correction factor, which depends on the quantity of stages.

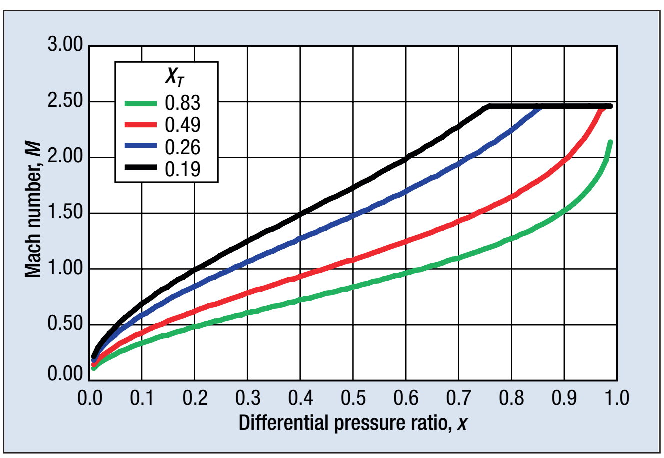

Fig. 6: Mach number M as in Table 5 as a function of x and for various x_T values

Valve type	New A_η	S_{tp}
Globe valve, parabolic plug	−3.5	0.23
Globe valve, V-port plug	−3.5	0.23
Globe valve, cage design	−3.25	0.1
Globe valve, multihole drilled plug	−4.55	0.12
Butterfly valve	−3.55	0.19
Rotary plug valve	−3.35	0.18
Segmented ball valve, 90°	−3.35	0.18
Drilled-hole plate fixed resistance	−4.55	0.12

Table 5: A_η and St_p dependent on the valve type based on the new Mach number in Table 4

However, there are some valve types not included in the standard, for example low-noise valves with flow dividers (Fig. 5). In this case, the following procedure is helpful:

Using the series of data measured for the internal noise as per IEC 60534-8-1,[6] L_{pi} and $L_{pi}(f_i)$ are measured in relation to x. The Mach number at the valve outlet should be smaller than 0.3 to ensure that additional noise sources are reduced to a minimum. The following variables can be derived from this:

- The experimentally determined η as a function of x
- Possibly another frequency profile function, $L_{pi}(f_i) - L_{pi}$ (and maybe also a new value for St_p).

Using this special information, partly without the theoretical equations, the prediction for each operational case can then be performed.

Different method to calculate the Mach number. Calculating M in Table 1 in regime I arises from the energy conversion in the valve, though based on incompressible media and, therefore, also from the F_L value. The C_v coefficient equations in the IEC standard are based on the work performed by DeFillipis.[9] In this case, they work with the more correct x_T value. The German VDMA standard[10] applies the equations by DeFillipis, resulting in the following for M:

$$M = \sqrt{\frac{\log_{10}(1-x)}{\log_{10}(1-x_{cr})}} \leq \sqrt{\frac{\gamma+1}{\gamma-1}} \quad (16a)$$

$$x_{cr} = 1 - \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1} \times \frac{xT\gamma \times 0.442/1.4}{(0.31+0.122 \cdot \gamma)^2}} \quad (16b)$$

Fig. 6 shows this Mach number as a function of x and x_T.

According to DeFillipis, x_T is approximately equal to 0,85 · F_L^2, meaning that, to be strictly correct, all equations can only be described as a function x_T. In principle, the number of flow regimes could then be reduced as well (as IV and V are identical). This, however, requires new values for A_η and St_p to be used, which are listed in Table 5.

Accuracy improvement as a result of the new approaches. The modifications described (Eqs. 10–14 and Tables 1 and 2) in the sections on internal and external noise of the present IEC 60534-8-3 standard lead to a considerable improvement in accuracy, as demonstrated in Fig. 7.

After applying the improvements, 86 % of all 3,000 measured points for L_{pi} lie within the error band of ±5 dB. In contrast, just 7 % are in the error bands between –10 dB and –5 dB and between 5 dB and 10 dB.

In the case of $L_{pAe,1m}$, 79 % of all calculated values are within the ±5 dB tolerance band, just 17 % lie in the band between 5 dB and 10 dB.

Figs. 8 and 9 plot the curves of actual examples of measured and predicted noise data. It is also clearly visible that the accuracy for internal and external noise is much higher.

These approaches were presented to the other members of Working Group 9 of the IEC Subcommittee 65B, who are currently integrating them into a new draft to revise IEC 60534-8-3.

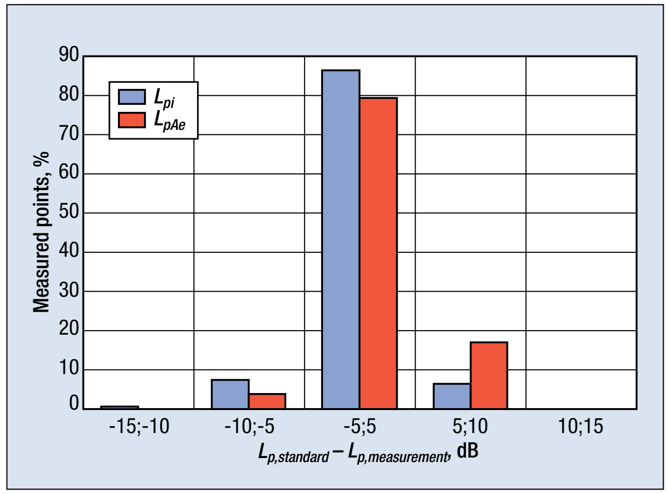

Fig. 7: Accuracy of the IEC 60534-8-3 standard with improvements presented in the section on potential for improvement (Table 1, Table 2, ΔTL from Eq. 11).

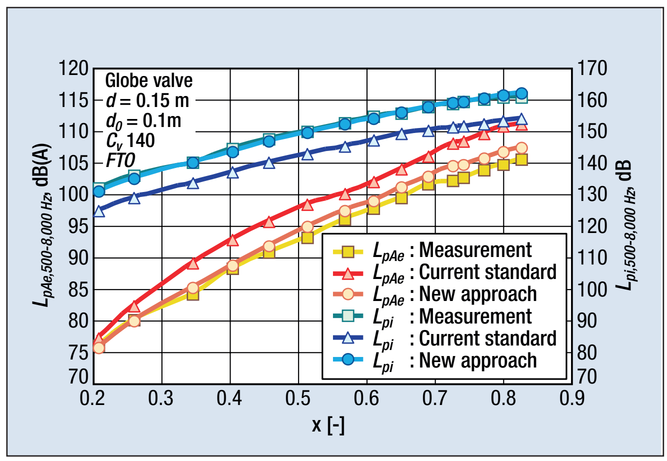

Fig. 8: Example of a globe valve (measurement with air).

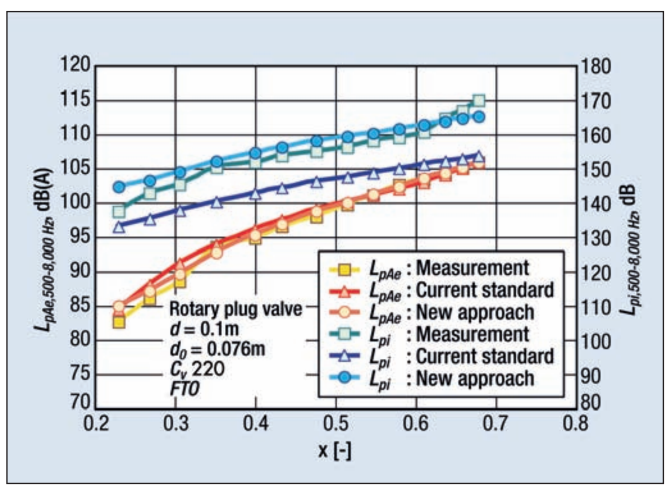

Fig. 9: Example of a rotary plug valve (measurement with air).

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