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Sound Pressure Slopes of Turbulent and Cavitating Liquids, and a Method to Predict Such Levels

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This paper presents an author-created, unique method called the ABC method to predict sound pressure of turbulent and cavitating liquids according to ISA Standard 75.17 and is measured 1 m from the outer wall of a downstream pipe, attached to the outlet of a valve [Baumann, 2023, "Method to Establish Sound and Acceleration Levels of High Pressure Reducing Valves," ASME Open J. Eng. 2, p. 001000-1]. The purpose of this paper is to explain how such sound pressure levels are estimated. It is accomplished by adding the results of equations in sub-sectors A, B, and C, all in dBA. This gives the sound pressure of a given valve handling water and installed in a Schedule 40 steel pipe. Additional modifiers are shown to modify the ABC method results for other pipes or fluids. The method is open and entirely based on known fluid-mechanic and acoustic laws such as Newton's and Lighthill's. It is claimed that such methods will give prediction results superior to those expected using the current International Electrical Commission (IEC) Standard 60534-8-4 [Baumann, 1970, "On the Prediction of Aerodynamically Created Sound Pressure Levels of Control Valves," ASME Paper WA/FE—28, Presented at the Annual ASME Winter Meeting, November]. A table is shown presenting test data which are compared to ABC method calculated values. Additional graphical information supporting such great prediction accuracy is also shown. [DOI: 10.1115/1.4065127]

Keywords: cavitation sound, control valve sound

1 Introduction and Historical Background

The need to predict sound levels of control valves have become more urgent with the introduction of Federal laws restricting sound levels in a working place to defined levels. Users of control valves appealed to valve manufacturers to solve this problem. The response of manufacturers, here and in Europe, centered initially on aerodynamic sound since laboratory tests would be performed easier with air [1]. One of the earlier results of this effort was my prediction method [2] published in 1970. Other valve manufacturers followed with empirical methods based on limited test data and lacked fundamental considerations.

In order to consolidate these separate methods, an ISA committee (International Society for Automation) wrote the first sound level prediction standard, S75.17 [3], published in 1989. On the international level, the IEC followed my co-authored Standard 60534-8-3 on aerodynamic noise prediction [4]. Finally, I wrote an ABC method based on an ASME research paper on aerodynamic valve noise [5].

An interest in hydrodynamic sound level prediction only started around 1970 by a German manufacturer [1] spurred on by commercial interests (they manufactured valves for hot water distribution). Their effort produced a German VDMA (VereiI Deutcher Maschine und Anlagen) standard, which later morphed into IEC Standard 60534-8-4 [6]. This standard too was based on limited test data and lacked fundamental hydrodynamic and acoustic considerations. I took part, in my capacity as the US delegate, in the formulation of this standard. At this time, my efforts to add more scientific equations failed, and I was out-voted. Other early sources of cavitation treatment are by Stiles [7] in 1961 and Franklin and McMillan [8] in 1984.

Users of control valves, knowing the sound level of a prospective valve, could either accept their choice or look for valves having anti-cavitation valve trims [9].

The subject of hydrodynamic sound started to interest me about 55 years ago when I observed test data of control valves handling cavitating water. The resultant observations showed sound pressure slope lines at higher frequencies which looked very different from the typical turbulent created sound pressure slopes at lower frequencies. Analyzing the upper slopes, I found a relationship of sound relating to the sixth power of jet velocity or 60 log (given pressure drop ratios). This, of course, is a slope typical for freely expanding gas jets, as described by Lighthill [10]. Later, in 1968, I expanded Lighthill's theories to adapt fluid-mechanical requirements of control valves producing aerodynamic sound [2,11] after including what the author found, *that cavitation of liquids is an aerodynamic phenomenon*, whose sound pressure slope, after combining with the fluid-mechanical caused turbulence sound, results in the sound pressure one experiences outside a pipe attached to a valve.

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In retrospect, one must not be surprised that water vapor, a gas, imploding at the speed of sound should fit into the laws of aero-acoustics. What might be surprising is that the typical pipe exterior sound pressure level is the summation of the cavitation gas pressure slopes and the fluid-dynamic-based turbulent sound levels.

2 An Introduction of the ABC Sound Level Estimating Method

This unique ABC method is introduced to predict expected sound pressure for all types and sizes of valves. The accuracy of this method can be supported by a selection of identified test data in which such data are compared to calculated values. Adherence to acceptable physical and acoustic laws together with proper use of such laws in the following equations will also support validation.

The ABC method can best be described as a summation of sound level slopes of involved parameters when plotted against a common dominator. A special section will explain the foundation and the functional aspects of the ABC method.

3 Proposed Equations for Liquid Generated Sound Pressures Levels

$$\text{SpLAe} = \text{A} + \text{B} + \text{C} - \text{rw} + \text{modifiers, dB(A)} \quad (1)$$

The following input variables are used in the following equation: The nominal pipe size, D; the wall thickness of the pipe $t_p = 8$ log D; the relative flow capacity of the valve $= C_v/D^2$; the pressure recovery factor FL; the Xfp, the coefficient of incipient cavitation; the Xy factor; and their individual weighting on the overall sound level. The accuracy of my method depends on precise input data such as C_v, FL, and Xfz. Such should only be obtained from the tested valve itself. General catalog data are not permissible if prediction accuracy is desired. The following is a suggested method to accurately predict sound levels of valves under given flow conditions (Sound pressure level exterior of the pipe and on the A weighted scale (SpLAe).

3.1 Input Needed. Nominal pipe diameter downstream of the valve (D) in inch ($D = d/25.4$ if d is in mm), $P1$ in psia (kPa $\times$ 0.145), $P2$ in psia (kPa $\times$ 0.145), flow coefficient C_v (at operating conditions), pressure recovery coefficient without expander, FL (at given valve travel), coefficient of incipient cavitation Xfz, liquid vapor pressure P_v, valve type modifier, rw (three for globe valves).

Initial calculations (note, all results are given in dB(A)):

$$\text{Pressure ratio } X = (P1 - P2)/(P1 - P_v) \quad (2)$$

$$Xy = X \text{ at maximum cavitation} = (1 + Xfz)/2 \quad (3)$$

3.2 Calculation Steps and Explanations of Variables. $A = 8\log(D) + 12\log(C_v/D^2) + 4$, dB(A). The effect of the valve diameter with Schedule 40 pipe wall thickness is measured to be proportional to 8 log(D). While the influence of diameter on sound is proportional to (D), the ratio between pipe wall sickness and pipe diameter decreases with an increase in pipe size at a rate of $D^{0.8}$, hence 8 log(D). Likewise, the relationship between flow capacity of the valve C_v/D^2 is given as dB(A) $= 12 \log (C_v/D^2)$. C_v in gallon per minute water provides the mass flow required for sound power calculations. The factor 4 accounts for various employed conversion factors.

$$B = 35\log(P1) - 35 \text{ for US} \quad \text{or}$$
$$35\log(1/0.145 \times P1) - 35 \text{ if } P \text{ is in kPa} \quad (4)$$

Upon analyzing the conventional long-hand sound level equation (1), it was found that $P1$ appears 3.5 times. Thus, we have dB(A) $= P1^{3.5}$ or 35 log ($P1$) $-$ 35 (-35 is required to account for "o" in the dB scale).

C is the effect valve orifice fluid velocity has on sound level. A better understanding of Eqs. (5), (6), and (7) is shown in Fig. 2. Here the pressure ratio factor, X, is given in place of Uvc since Uvc is proportional to $(P1 - P2)^{0.5}$, and $X = (P1 - P2)/(P1 - P_v)$.

$$C_1 = 30\log(10X), \ X \text{ maximum} = \text{FL}^2 \quad (5)$$

This equation gives a turbulent sound level in dB(A) at 1 m. Turbulence exists throughout sub-equations $C2$ and $C3$.

C_2 = Increasing cavitation sound pressure exists once X exceeds Xfz.

$$\text{If } X \text{is larger than } Xfz \text{but below } Xy, \text{ then } C_2 = 60\log(X/Xfz) \quad (6)$$

otherwise, use 0 (zero). Note, ζ is embedded in the C_2 equation.

C_3 = Decreasing cavitation sound pressure slope. This happens once X exceeds Xy. If X is larger than Xy, then $C_3 = -120 \log(X/Xy)$, otherwise use 0 (zero).

$$Xy = (Xfz + 1)/2 \quad (7)$$

$$C = C_1 + C_2 + C_3, \text{ in dB(A). Note, } C_3 \text{ is usually negative} \quad (8)$$

Final external sound level, $\text{LpAe}_{1\text{m}} = \text{A} + \text{B} + \text{C} - \text{rw} + \text{modifiers}$, in dB(A) at 1 m.

3.3 Modifiers. For pipe schedule: Schedule 40 = 0, Schedule 80 = -3 dB, Schedule 160 = -7 dB.

For liquids other than water, add 20 log $(62.5/l_{\text{id.}})$ + 25 log(4750/$\text{Sp}_{\text{liq.}}$), where $\rho_{\text{liq.}}$ = density of chosen liquid (lbs/ft^3) and $\text{Sp}_{\text{liq.}}$ = speed of sound of chosen liquid (ft/s).

For pipe materials other than steel, add 20 log(7800/density chosen material). Density in kg/m^3.

3.4 Example. An example could help understand the proposed method.

A 4 in. (100 mm) V-port globe valve in a 4 in. pipe has the following data: $C_v = 93.5$, $P1 = 100$ psia, $P2 = 30$ psia, FL = 0.9, $Xfz = 0.28$, $C_v/D^2 = 5.2$, $P_v = 0.4$ psia, $Xy = (1 + 0.28)/2 = 0.64$, and $X = 0.7$.

A = 8 log(4) + 12 log(5.2) + 4 = **17.4 dB(A)**;
B = 35 log(100) − 35 = **35 dB(A)**;
$\mathbf{C_1}$ = 30 log(10X), since X is less than FL^2 = 25.3 dB;
$\mathbf{C_2}$ = 60 log (0.7/0.28) = 24 dB(A);
$\mathbf{C_3}$ = −120 log (0.7/0.64) = −4.7 dB(A);
$\mathbf{C = C_1 + C_2 + C_3 = 44.5}$ **dB(A)**;
LpAe1m = A + B + C−rw = 17.4 + 35 + 44.5 − 3 = **93.5 dB(A)** at 1 m from the pipe and based on Schedule 40 steel pipe.

Test data showed 93 dB(A); see Fig. 1. For more results see the legend of Table 1 and Fig. 3, most predictions fall within ±+0.5 dB of the measurement curve.

4 Description of the Various Parts of the ABC Equation

In *PART A*, 10 log(D) tells the effect of pipe size on sound level. 12 log (C_v/D^2) tells the combined slopes of the given flow capacity (C_v) per given pipe size, plus the slope of 8 log(t_p/D), the statistically derived ratio between pipe diameter and pipe wall thickness, (t_p). C_v for gallon per minutes of water constitutes the mass flow required in any sound power equation.

The number 4 is a constant accounting for conversion factors.

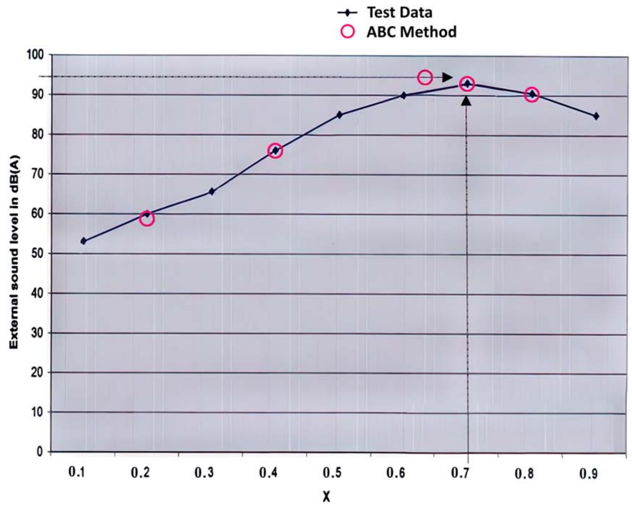

Fig. 1 Test data from typical sound pressure slopes of a globe valve measured 1 m from a Schedule 40 steel pipe. Source: Samson AG. Other ABC prediction results at different *X* ratios shown include: *x* = 0.2, SPL = 58.6; *x* = 0.4, SPL = 76.8; *x* = 0.64, SPL = 95.1; *x* = 0.8, SPL = 90.1 (see circled results—all SPL in dB(A)).

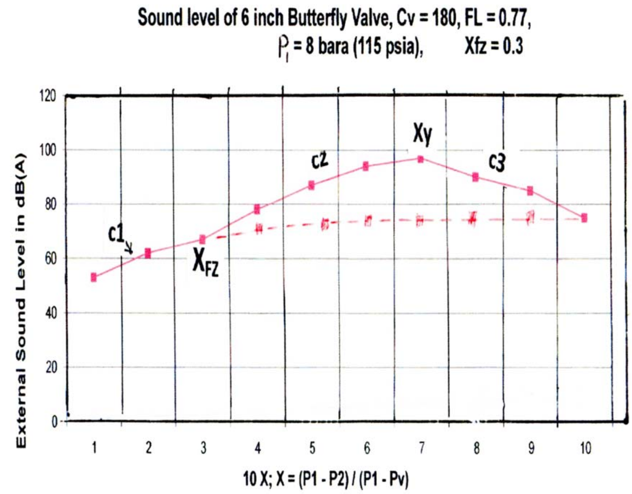

Fig. 2 The various logarithmic regimes of turbulence and cavitation. *C1* = turbulent sound (function of Uvc^3), *C2* = increasing cavitation (function of Uvc^6), and *C3* = decreasing cavitation (function of Uvc^{12}).

PART B indicates how many times the inlet pressure $P1$ is a factor in a typical noise equation. The answer is 3.5 times, hence 35 log($P1$).

PART C determines what kind of flow behavior is encountered. Either turbulent (C_1), turbulent plus beginning of cavitation ($C_1 + C_2$), or C_3, turbulence and cavitation sound from Xy less decreasing cavitation sound. Please refer to Fig. 2.

5 Proof of Accuracy of Prediction

Any sound level estimating method requires verification in order to establish trust among the users of such a method. My approach to providing such verification is to utilize laboratory test data or test data published by reputable publications. In addition to the data from Fig. 1, here is a selection of such data. It shows the stated measured sound levels, the calculated results using the above equations, and the resultant offset. In reviewing these data, one must consider that those offsets include not only ABC prediction accuracies (which, theoretically, should be 100%, since the method is based on absolute scientific laws) but also reflecting errors in the test measurement (typically assumed to be ±−3% of combined measurement instruments).

Additional graphical representations as proof of prediction are shown in Figs. 3–5.

Code for identifying sources of test data:

U = State of Utah, Water Research Laboratory, Projects: 3846, 4105, 4214
Ba1 = H. Baumann article 11-2014
Ba2 = ISA Hand Books
Y = Yeary Assoc.
Mo = J. Monsen article March 2022
Ma = Masoneilan Co., legacy test data from period 1958 to 1972
K = J Kiesbauer article March 2004
S = Siemens A.G. (R&D)

Table 1 ABC method prediction versus actual test data

Size (in.)	Valve type	*P*1 (psia)	*X*	C_v	*Xfz*I FL	*A*	*B*	*C*	Estim dB(A)	Test dB(A)	Offset dB S
4	Rotary Plug	145	0.7	18	0.3 I 0.9	9.4	40.6	41.5	91.5	92	−0.5 Mo
4	Rotary Plug	145	0.2	18	0.3 I 0.9	9.4	40.6	9.6	59.6	61.5	−1.9 Mo
3	Rotary Sleeve	145	0.5	114	0.3 I 0.55	21	40.6	27.6	89.2	89	+0.2 U
3	Rotary Sleeve	145	0.2	114	0.3 I 0.55	21	40.6	9	70.7	61.5	+9.2 U
4	Rotary Plug	145	0.1	100	0.1 I 0.7	18.4	40.6	1.3	60.3	62	−1.7 Ma
4	Rotary Plug	145	0.55	100	0.1 I 0.7	18.4	40.6	65	123.9	129	−5.1 Ma
2	Globe valve	145	0.5	2	0.3 I 0.85	2.8	40.6	35.1	78.5	80	−1.5 U
2	Globe valve	145	0.3	2	0.3 I 0.85	2.8	40.6	15.7	59.1	60	−0.9 U
3	Rotary Plug	130	0.25	117	028 I 0.87	21.2	39	11.8	72	77.5	−5.5 Ma
3	Rotary Plug	130	0.5	117	028 I 0.87	21.2	39	36.1	96.3	103	−6.7 Ma
6	Butterfly	115	0.25	180	0.3 I 0.8	18.6	371	12.1	67.8	66.5	+1.3 Y
6	Butterfly	115	0.65	180	0.3 I 0.8	18.6	18.6	44.1	99.8	100	−0.2 Y
6	Butterfly	145	0.9	180	0.3 I 0.6	18.6	40.6	24.2	83.4	74	+9.4 Ba1
6	Butterfly	145	0.45	826	0.4 I 0.66	26.6	40.6	18.1	85.3	85	+0.3 Ba1
6	Butterfly	145	0.65	826	0.4 I 0.66	26.6	40.6	31.8	99	95	+4 Ba2
2	V-Port Globe	145	0.6	3.5	0.53 I 0.95	5.7	40.6	24.6	72.9	70.5	+2.4 K
2	V-Port Globe	145	0.8	3.5	0.53 I 0.95	5.7	40.6	33.7	80	83	−3 K
3	Rotary sleeve	145	0.5	63.5	0.3 I 0.5	18	40.6	35.6	83.7	86	−2.3 S

Notes: S denotes the source of the test data (see table below).
Any shown offsets exceeding ±3 dB are invariably attributed to wrong input data. One typical example is when general catalog data are used as input instead of data taken directly from the tested valve.

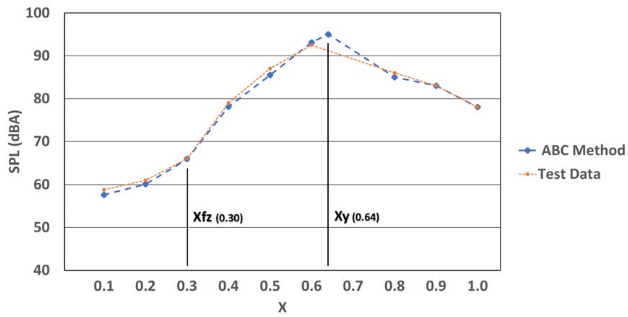

Fig. 3 Comparison of a 4 in. rotary plug valve. Test data: $C_v = 22$, $P1 = 145$ psia, FL = 0.9, and $Xfz = 0.3$.

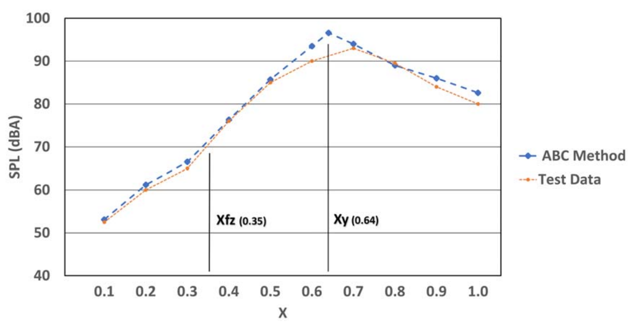

Fig. 4 Comparison of a 4 in. V-Port valve. Test data: $C_v = 93.5$, $P1 = 145$ psia, FL = 0.9, and $Xfz = 0.35$.

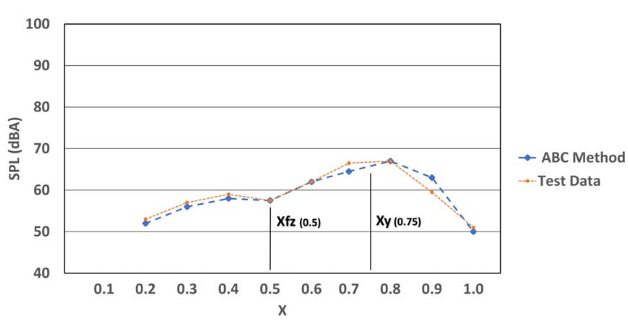

Fig. 5 Comparison of a 2 in. globe valve with anti-cavitation trim. Test data: $C_v = 4.88$, $P1 = 250$ psia, FL = 0.98, and $Xfz = 0.5$

6 Other Methods

A current method devised by a group of control valve engineers, led by Kiesbauer [1], this author included, and published as IEC Standard 60534-8-4 with 22 standard equations, is not much in use probably due to the complexity of required calculations. It is primarily based on test-derived, curve-fitting, empirical data, and lacks independent verification. The published prediction accuracy goals claimed ±5 dB with "standard" control valves, if tested *Xfz* is used. No verifiable data to support the above claim have ever been published by the IEC Standards Committee. The value of this *Xfz* factor can only be obtained from selected valve manufacturers. This claim is based only on selected measurements with water in the standard authors' laboratories. That certainly is insufficient to meet industrial needs. It excludes industrial users from utilizing this method having neither a proper *Xfz* factor for their valves nor any other required factors part of the equation for large or high-pressure valves, nor data available for fluids other than water. Examples listed in the appendix of the IEC Standard only serve to familiarize a reader with the use of the many equations. The results of these sample calculations should not imply that they represent actual test data. Although, a comparison calculation for the three examples with ABC method predictions using the same input shows similar results with differences averaging 4 dB(A).

In contrast, the herein proposed, very simple ABC method has few equations and is based entirely on fluid-mechanical and acoustic laws. The method has no restrictions as to the size or types of valves as shown. Based on available test data, accuracy ranges from 75% of cases within ±3 dB, and an average prediction error of −0.6 dB(A). Finally, the ABC method is an "open" method and includes all types of valves, not only "STANDARD" types. Based on the above analyses, it is apparent that the IEC hydro standard gives even worse results than the previously discussed IEC aero standard [5].

Table 2 Summation of comparison data from Table 1

Accuracy range between	IEC % of cases	ABC % of cases
+/−3dB	12%	32%
+/−5 dB	15%	75%
+/−10 dB	47%	0%

Note: What this table means is that a customer has a 75% change in getting a +/−5 dB correct answer compared while only 15% get a same level of prediction accuracy with an IEC standard.

7 Summation

For decades, the industry has been looking for a reliable method to correctly and reliably predict valve noise associated with liquid turbulence and cavitation. The publication of an IEC Standard 60534-8-4 seemed to be the answer. However, general use of this standard is limited since the promised prediction accuracy of between ±5 dB was seldom obtained.

This was the reason that I decided to propose a viable alternative. This author realizes the general need for overcoming the resistance by the IEC supporting establishment and for gradual acceptance by industry of my method may be difficult to obtain. Realizing those difficulties, it is important to show that a viable, better, and easier to use alternative to the IEC standard exists.

The alternative, this paper proposed is a scientific-based method to predict turbulent and cavitation derived sound pressure levels with sufficient accuracy to meet industrial requirements. The method is verified using sufficient numbers of published laboratory or field-based test data. It is hoped that the ABC method, someday, will replace the current IEC standard. From the above analyses, it is apparent that, based on the evidence provided here, the IEC hydro standard gives substantially inferior results compared to the ABC method.

All calculations were performed by Jon Monsen, a member of ISA Committee 75.07 [16] using identical test data. Here is an accuracy comparison between the IEC Hydro Standard and the ABC method. The percentages given in Table 2 denote the percentage of test cases meeting the specified accuracy requirements. Table 2 gives a summary of the accuracy comparison between IEC and ABC methods as found from test data comparisons.

Here are other statistics based on the ABC/IEC comparison:

Minimum error: IEC = 0.5 dB; ABC = 0.2 dB

Maximum error: IEC = 24.6 dB; ABC = 9.2 dB

Prior to the development of the ABC method, I disseminated my own fundaments based aerodynamic sound level prediction method [11] through a number of acoustic textbooks [13–15], without known dissent. This fundamentals based method then served as the basis for the current ABC method.

Conflict of Interest

There are no conflicts of interest. This article does not include research in which human participants were involved. Informed consent not applicable. This article does not include any research in which animal participants were involved.

Data Availability Statement

The datasets generated and supporting the findings of this article are obtainable from the corresponding author upon reasonable request.

Nomenclature

- D = nominal external pipe diameter, in.
- X = pressure drop ratio, $(P_1 - P_2)/(P_1 - Pv)$
- C_v = valve sizing coefficient, 1 $C_v = 1$ gallon of water at 1 psi pressure
- P_v = vapor pressure of water at 20 C = 0.4 psia = 2.8 kPa
- ci = speed of sound of liquid, water = 1498 m/s
- sp = specific gravity, water = 1
- t_p = pipe wall thickness, in.
- FL = pressure recovery factor
- LpLAe = sound pressure level on the A scale, measured at 1 m from the outlet pipe wall, dB
- $P1$ = absolute inlet pressure, psia = 0.145 kPa
- $P2$ = absolute outlet pressure, psia = 0.145 kPa
- Uvc = jet velocity at the orifice, m/s
- Xfz = X at which cavitation commences (specified by the manufacturer—approximately FL^2)
- Xy = pressure ratio at which sound pressure reaches maximum
- Xy = $(Xfz + 1)/2$, as defined by this author
- ζ = acoustical efficiency factor for liquids, dimensionless. This factor was created by the author; it is (Uvc/ci) × 10^{-4}

Abbreviations

- IEC = International Electrical Commission, Geneva, Switzerland
- ISA = International Society of Automation, Research Triangle Park, NC, USA

All symbols are generally in accordance with ISA Standard S75.05, control valve terminology.

Note: The Xfz factor, the most important part of any liquid sound prediction method, typically is determined by laboratory vibration measurements or by audio means. The Xfz factor is determined when, after slowly increasing the pressure drop through the valve till a sharp increase of vibration is measured or sensed. The pressure ratio, when this happens is Xfz. This rather costly, and, error-prone method should be refined. I advocate a mathematical determination to calculate the value of Xfz. I proposed such a method in my following paper: Baumann, H. D., 2017, "De-mystifying the Coefficient of Incipient Cavitation," Valve World, **22**(3), pp. 72–75.

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