Modification of a Standard Aeroacoustic Valve Noise Model to Account for Friction and Two-Phase Flow

G. M. Singh, E. Rodarte, N. R. Miller, and P. S. Hrnjak

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For additional information:

Air Conditioning and Refrigeration Center
University of Illinois
Mechanical & Industrial Engineering Dept.
1206 West Green Street
Urbana, IL 61801

(217) 333-3115

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P. S. Hrnjak and N. R. Miller, Principal Investigators

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George Singh, Enrique Rodarte, Norman R. Miller, Predrag S. Hrnjak

ABSTRACT

This paper presents several modifications to the 1995 IEC standard model for predicting the aerodynamic noise generated by control valves in ideal gas. The modifications account for frictional pressure drop from the valve inlet to the point of maximum velocity within the valve as well as for the effect of two-phase flow on the aerodynamic noise generated by the valve and the subsequent attenuation downstream of the valve. The development of the standard model as well as the key assumptions and approximations made in the fundamental aeroacoustic equations of Lighthill and Curle (upon which the standard model is based) are examined in justification of the presented modifications. Experimental results for expansion devices in refrigerant are given which support the presented modifications and experimental results for two-phase attenuation in tubes are also presented.

1. INTRODUCTION

For a number of years, it has been known that flow control and throttling valves can be a significant source of noise in industrial facilities [1,2,3,4,5,6]. As such, much work has been done on noise from control valves in air and water systems [3,4]. In 1995, the IEC released a standard method [7] for predicting the aerodynamic noise generated by control valves using ideal gases. This standard method was based on the free jet noise studies first published by Lighthill [8,9] and the confined jet studies of Curle [10]. This standard model, which has proven to be very successful at predicting the aerodynamic noise downstream of valves throttling air [11], is, however, limited to single phase gases and isentropic valves (valves with no frictional losses or heat transfer between the inlet and the point of maximum velocity). The standard model then, as written, can not be used for non-isentropic throttling devices, such as the capillary tubes often used in refrigeration, nor can it be used to predict the aerodynamic noise generated by two-phase flows, such as the two-phase flow of refrigerant. This paper first describes the standard model and then examines the basic principles and assumptions upon which the model is based. In this light, appropriate modifications to the existing model are proposed to account for both frictional pressure drop and two-phase flows. Finally, the modified model predictions are compared to experimental data for expansion devices in refrigerant.

2. IEC Standard Valve Noise Model

The IEC standard valve noise model for ideal gases is as presented in IEC 534-8-3:1995. Specifically, at the point of maximum velocity (minimum pressure), the mechanical stream power, W_m, is [7]:

$$W_m = \frac{mv^2}{2} \qquad (1)$$

where v is the velocity and m is the mass flow rate though the valve. If the valve is unchoked (i.e. the maximum velocity in the valve is less than the speed of sound in the fluid) and the flow is assumed isentropic from the inlet to this point of maximum velocity, then for an ideal gas [12]:

$$v = \sqrt{2\left(\frac{\gamma}{\gamma-1}\right)\left[1-\left(\frac{P}{P1}\right)^{(\gamma-1)/\gamma}\right]\frac{P1}{\rho 1}} \qquad (2)$$

where γ is the ratio of specific heats, P is the pressure at that point, P1 is the inlet pressure, and $\rho 1$ the inlet density.

If the valve is choked, v becomes c, the speed of sound in the valve fluid. For an ideal gas,

$$c = \sqrt{\gamma RT} \qquad (3)$$

where R is the specific ideal gas constant, and T is the absolute temperature.

Some fraction of the energy at this point will be converted into sound. In terms of the mechanical stream power (Eqn. 1):

$$W_s = \eta W_m \qquad (4)$$

where W_s is the sound power at this point and η, called the acoustical efficiency factor, represents the fraction of mechanical power converted into sound power at the point of maximum velocity.

The total internal sound pressure level (in dB) downstream of the valve exit is then given by:

$$TSPL = 10\log_{10}\left[\frac{(x)(3.2x10^9)W_s\rho_d c_d}{D_i^2}\right] \qquad (5)$$

where the subscript "d" signifies downstream (normally assumed to be 1m from the valve exit), Di is the internal pipe diameter of the downstream pipe, and x is a factor to account for jet exit angle. For an orifice or capillary tube, where the jet exits along the tube axis, 100% of the exit noise is transmitted downstream, thus x=1. For a globe valve, the jet exits at an angle and as much as 75% of the noise is dissipated near the valve exit, leaving as little as 25% to travel downstream [13]. Thus for a globe valve, x=0.25. The factor of $3.2x10^9$ comes from converting sound power into sound pressure [1] with a reference pressure of $20x10^{-6}$ Pa.

Finally, note that Eqn. (5) does not account for any attenuation of the sound waves from the valve exit to the location in the downstream tubing where the internal sound pressure is measured. This is normally justified for ideal gas flow (as will be shown below), but would need to be accounted for in a two-phase flow.

3. Pressure Recovery

At pressure ratios below the critical, some control valves or expansion devices can exhibit pressure recovery. That is, the valve/device design allows an increase, or recovery, in pressure from the minimum pressure to the existing downstream pressure (see Figure 1).

This process represents the conversion of potential energy (inlet pressure) to kinetic energy (velocity head), and back to potential energy (exit pressure), where the pressure that is not recovered represents energy lost to waste heat and sound [14]. Assuming only a tiny amount of energy is lost to sound (as will be shown later), the amount of pressure recovery depends on the amount of waste heat (or entropy) generated in the expansion process. A device which allows for a smooth and gradual expansion, like a smooth converging-diverging nozzle (Figure 2A), will exhibit high pressure recovery, while a simple orifice (Figure 1A), which contracts and expands suddenly, will generally exhibit lower pressure recovery.

The pressure distribution along a converging-diverging nozzle and a control valve are illustrated qualitatively for gas flows in Figure 2. The pressure distribution along an orifice or capillary tube is illustrated in Figure 3, as indicated by experimental evidence for orifice and capillary tubes in two-phase refrigerant flow [15,16,17,18] and as predicted by compressible flow theory for gas flows in tubes [19]. As such, orifice (short) tubes and capillary tubes with two-phase or pure vapor refrigerant flow can be considered to have negligible pressure recovery.

A factor to describe the amount of pressure recovery in a valve was first introduced by Baumann [20] and can be written as:

$$F_L^2 = \frac{P1 - P2}{P1 - P} \qquad (6)$$

where P2 is the downstream pressure and P is the minimum pressure, or the pressure at the point of maximum velocity in the valve. For valves without extended interiors or with little frictional

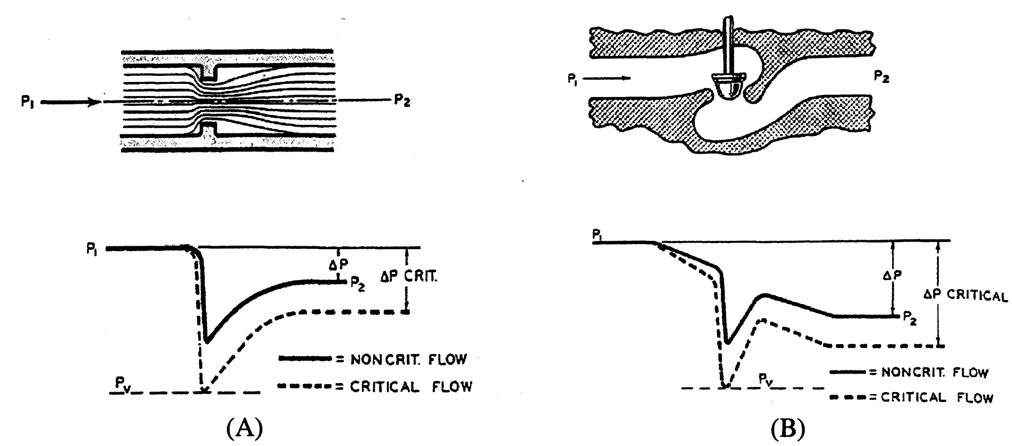


Figure 1. Schematic of pressure gradients in (A) simple orifice and (B) control valve
Figures from [20]. Modified for clarity only

pressure drop, this point occurs at the vena contracta. For orifice and capillary tubes in two-phase or vapor flow, this point occurs at the tube exit. The point of maximum velocity is the choking point in any device when the pressure ratio is above the critical.

For devices with little or no pressure recovery then, such as orifices, orifice tubes, or capillary tubes in two-phase or pure vapor flow, $F_L=1$. The pressure recovery factor for a valve can be obtained through the valve manufacturer or determined experimentally by standard procedures [21,22].

For pressure ratios greater than the critical, shock waves form and pressure recovery occurs via shock-wave recompression (See Figure 2). As this recompression is non-isentropic, more of the energy at the choking point may convert into sound energy. Thus, choked valves are more efficient radiators of sound (as will be shown below).

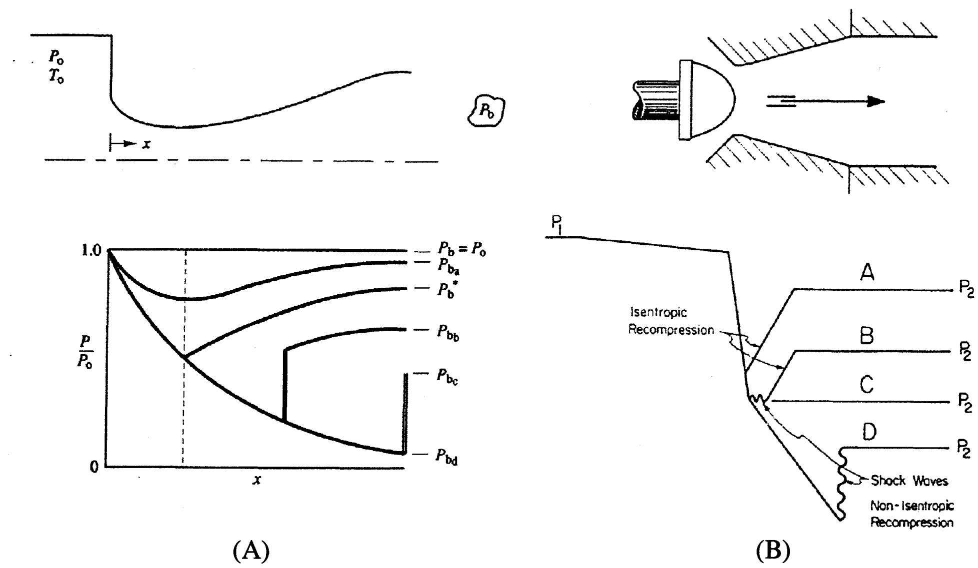


Figure 2. Pressure gradients for gas in (A) converging-diverging nozzle and (B) control valve for various back (downstream) pressures.
The heavy vertical lines in (A) are normal shock waves.
Dashed line in (A) signifies the critical pressure ratio for choking
(A) from [19] and (B) from [14]. Modified for clarity only.

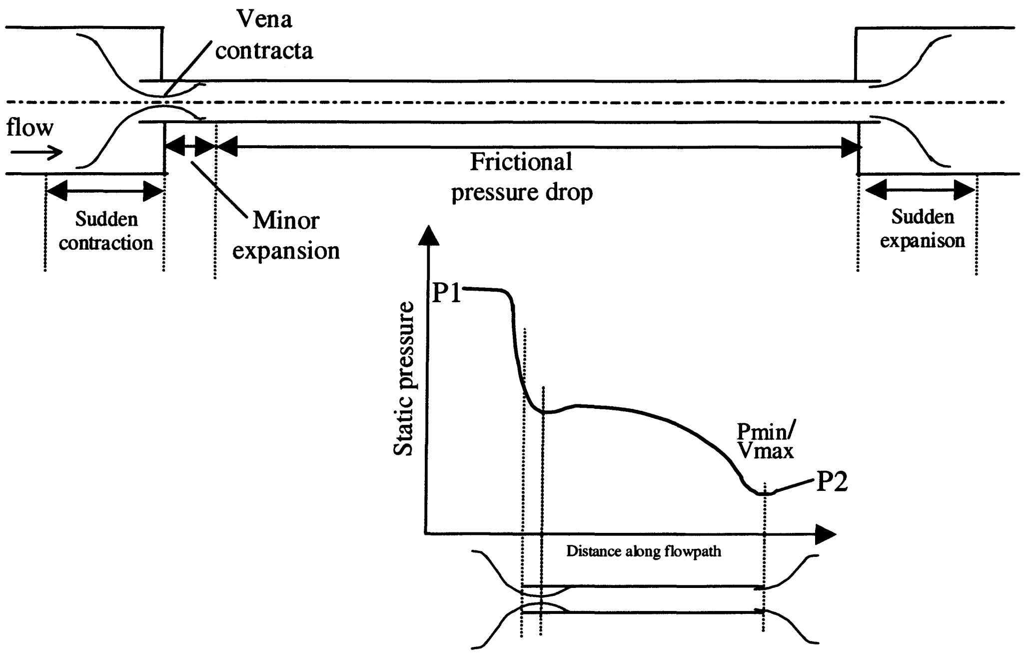


Figure 3. Qualitative pressure distributions along orifice and capillary tubes in refrigerant

4. Acoustical Efficiency Factor

The acoustical efficiency factor (a function of F_L) contains the only empiricism in the standard model. Lighthill shows that for a subsonic jet, the turbulent flow noise acts as a continuous distribution of quadrupole sources, where each fluid element in the jet wake acts as a quadrupole source proportional in strength to the intensity of the turbulence [8]. It can be shown that η is approximately proportional to M^5, where M is the jet Mach number. As Lighthill shows, however, this approximation is derived directly from conservation of mass and momentum and is valid for any *continuous, inviscid fluid without body forces.* As any low-temperature free jet is, in fact, *continuous, inviscid, and generally fast enough to neglect body forces*, this result in principle should be as applicable to two-phase jets as to single phase ones.

Experiments on single-phase free jets showed that $\eta \propto M^5$, and that $\eta = 0.001M^5$ as M approached 1 [1,14]. Curle later showed that for continuous, inviscid confined jets in the absence of body forces, the fluid elements near the surface of the tube walls act not as quadrupole but as dipole noise sources and radiate sound such that $\eta \propto M^3$, especially at low mach numbers [10,14]. A curve fit for $0.35<M<1$, with $\eta = 0.001$ at $M=1$ gives $\eta = 0.001M^{3.6}$ for confined jets in the subsonic regime [14]. Since subsonic pressure recovery represents energy isentropically recovered beyond the point of maximum velocity (and hence not dissipated through turbulence into waste heat or sound), F_L^2 is the fraction of energy at that point which is recovered and hence not available for sound. Hence, η for confined subsonic valve flows may be written as

$$\eta = 0.0001M^{3.6}F_L^2. \qquad (7)$$

Curve fits in conjunction with experiment and supersonic free jet noise theory show that

$$\eta = 0.0001\, M^{6.6\, F_L^2} \tag{8}$$

for jet mach numbers above M=1 [6]. If pressure ratios across the valve continue to increase, a "mach disk" may form in the tube downstream of the valve exit and the acoustical efficiency levels off while a factor of $\sqrt{2}$ is introduced into the expression for η:

$$\eta = 0.0001\left(\frac{M^2}{2}\right)\left(\sqrt{2}\right)^{6.6F_L^2} \tag{9}$$

for confined supersonic valve jets at high pressure ratios.

5. Modifications to the Standard Model

The careful explanation of the standard model given above serves to accurately account for all of the assumptions and approximations present in the model to help justify the following proposed modifications:

In the valve noise model of IEC 534-8-3:1995, the speed of sound at the point of maximum velocity (choking point for choked flow) is given by the ideal gas speed of sound, Eqn. (3). For any real gas flow, however, the speed of sound would more accurately be calculated from the definition of sound speed:

$$c = \sqrt{\left.\frac{\partial P}{\partial \rho}\right)_s} \tag{10}$$

where the subscript "s" indicates the derivative is taken at constant entropy. Note this modification in no way affects any other principle or equation in the model. Thus, one would expect to simply replace the ideal gas speed of sound with Eqn (10) for any gas whose speed of sound differed significantly from the ideal gas speed of sound.

The speed of sound for any homogeneuous *two-phase* fluid can also be calculated numerically from Eqn. (10). Thus, if the fluid at the point of maximum velocity through the valve or expansion device is assumed to be well-mixed, one would expect to likewise replace the ideal gas speed of sound (Eqn. 3) with the homogenous two-phase speed of sound (calculated numerically from Eqn. 10):

$$c \approx \sqrt{\left.\frac{\Delta P}{\Delta \rho}\right)_s} \tag{11}$$

where P is the saturation pressure and the density is the linear two-phase density:

$$\rho = x\rho_{sv} + (1-x)\rho_{sl} \tag{12}$$

where x is the quality (0-1) and ρ_{sv} and ρ_{sl} are the saturated vapor and liquid densities, respectively, at that saturation pressure.

Similarly, one would expect to replace the density and speed of sound terms in Eqn. (5) with their homogeneous two-phase values.

As the standard model assumes isentropic flow from the entrance of the valve to the point of maximum velocity, non-isentropic effects that may exist in this region in some valves or expansion devices must also be considered when applying the model. Friction is seemingly the most common (and perhaps only) non-isentropic effect one would expect to encounter in a real valve or expansion device. As such, by replacing the measured inlet pressure (P1) with the actual (frictionally reduced) pressure near the point of maximum velocity, one would expect the standard model to behave as accurately as for any near-isentropic device.

6. Experimental Results

The modifications to the standard model outlined above can be justified by comparison of model predictions, with and without modification, to experimental results. The experimental test facility used was designed specifically for measuring the incident internal sound pressure downstream of various valves and expansion devices in refrigerant and a full description of the facility can be found in [23]. All experimental results are for pure Refrigerant 134a and were measured over the range 0-20kHz with a resolution of 25Hz and a Hanning window.

Figure 4 shows results for block-type thermostatic expansion valves (TXV's) of different sizes and F_L factors. The valves are throttling pure R-134a vapor (assumed to be nearly an ideal gas). Since TXV's do not contain extended interiors, they may be considered to be of the valve type addressed in the standard (i.e. isentropic from the inlet to the vena contracta). Since these data are for an ideal gas, the standard model was applied *as written* (i.e. without modification). Note the excellent agreement between model predictions and experimental results over a wide range of pressure ratios.

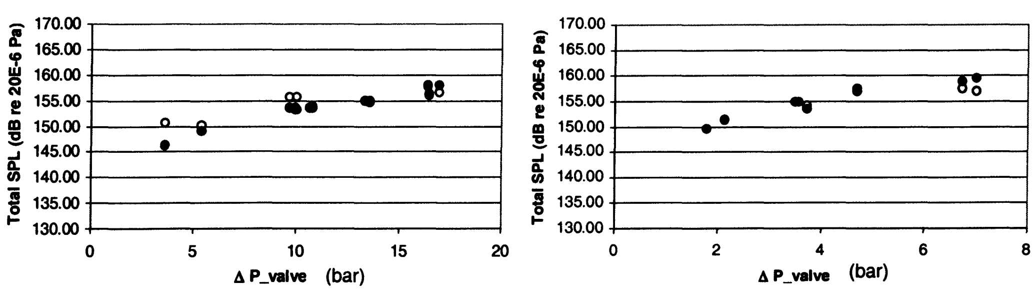


Figure 4. Model predictions vs. experimental results for two different sized thermostatic expansion valves
Right: F_L=0.45 Left: F_L=0.65
Open circles are model predictions. Closed circles are experimental data

Figure 5 shows results for orifice (short tube) and capillary (long tube) expansion devices. Since orifice tubes are short, they may generally be considered isentropic from the inlet to the exit plane. However, capillary tubes have greatly extended interiors and usually generate significant frictional (non-isentropic) pressure drop from the inlet to the point of maximum velocity (exit plane). As such, when the standard model is applied as written, the short orifice tubes show excellent agreement between model predictions and experimental results, but the long capillary tubes show significant overprediction by the model (indicating the model does not account for energy lost via friction along the tube length). However, when the inlet pressure specified in the model is replaced by the actual (frictionally reduced) pressure near the exit plane (computed using standard Fanno flow relations, 1/85 of the tube length from the exit plane [19]), the model predictions are in very good agreement with the experimental results, as shown in Figure 5b. Figure 6 shows results for an orifice tube with two-phase flow of R-134a. Note the model as written again overpredicts the experimental results. However, when the two-phase speed of sound and density are substituted in the model in place of their ideal gas values and the two-phase attenuation is accounted for in Eqn. (5) as described below, the model predictions agree very closely with the experimental results, as shown in Figure 6.

7. Two-phase Attenuation

Although the problem of sound wave attenuation for gas flows in tubes has been well studied and is fairly well understood, there appears to be few published results for the attenuation of sound in a confined two-phase flow. The attenuation in two-phase flow was measured in our test section

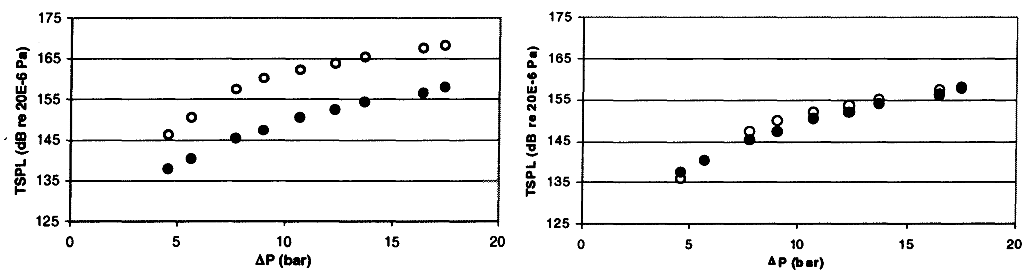
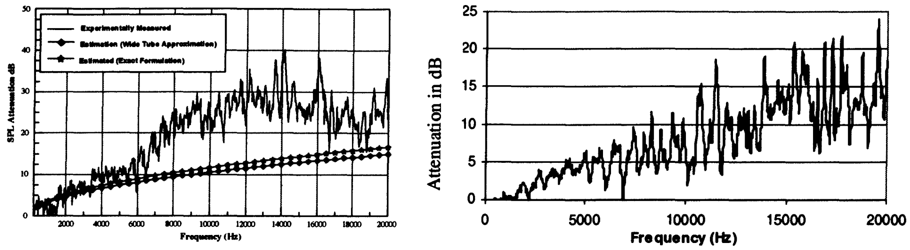


Figure 6. Model predictions vs. experimental data for two-phase flow of R134a through an orifice tube

Left: Model as written Right: Modified for two-phase speed of sound, density, and attenuation.

for R134a between 55% and 95% quality (see Figure 7B). As suspected, the attenuation was substantial and significantly greater than the attenuation present in a single-phase flow (Figure 7A). Further, the results appeared to be a minor function of the quality of the flow but basically independent of the specific flow conditions (mass flow and pressure). Using the results of Figure 7B, a correction was made to Eqn. (5) for each data point to account for two-phase attenuation (which is not accounted for in the model).



(A) (B)

Figure 7. Attenuation of sound waves in Refrigerant-134a
(A) vapor over 7.24 m
(B) two-phase flow (85% quality) over 0.635 m. Figure (A) from [24]
0.5 in O.D. copper tubing. Orifice tube (1.7 mm I.D., 38.4 mm length) used as noise source.

$$Sp(f)_{corrected} = Sp(f)_{measured} + (d/0.635\ m)([14+(9.5-10x)]dB/20{,}000Hz)f(Hz)$$

"Sp" is sound pressure. (d/0.635 m) corrects for the difference between where the attenuation was measured and where the sound pressure was measured relative to the noise source. "d" is the distance (in m) from the exit of the expansion device to the point of internal noise measurement.

8. Summary and Conclusions

The isentropic valve noise model of IEC 534-8-3:1995, originally developed to predict the aerodynamic noise generated by isentropic control valves throttling ideal gases, can be modified to account for both (1) frictional pressure drop along the length of the valve and (2) two-phase flows. The frictional pressure drop can be accounted for by simply replacing the inlet pressure with the pressure near the end of the frictional length. This can be done using a Fanno-Flow analysis for vapor flow or published correlations (such as [17] and [25] for two-phase flow. Two-phase flow effects can be accounted for in the model by replacing the ideal gas speed of sound and gas density in the original model with the homogenous two-phase density and the homogeneous two-phase speed of sound (computed numerically), assuming a well-mixed flow. Further, the sound attenuation downstream must be accounted for when two-phase flow exists. As little published data appears to exist for sound wave attenuation in two-phase flows, the attenuation must be determined experimentally for the fluid and conditions of interest.

Comparison of both the original and the modified model to experimental data for expansion devices with refrigerant flow show very good agreement between the two. These results seem to support both the validity and accuracy of the model, as well as the proposed modifications. Also, it appears that the one empiricism in the model, the acoustical efficiency factor η, is valid for both single and two-phase flows (as shown by the good agreement of the modified model to the experimental results). This is significant because although η was shown qualitatively to be valid for any type of flow, the only quantitative experimental results to date were for single-phase flows of gas.

Finally, it should be noted that the IEC model predicts not only the internal sound pressure level downstream of the valve (as was presented here) but also predicts the "peak frequency" in the internal sound pressure spectrum as well as the coincident frequencies and transmission loss through downstream pipe walls. Our experimental data with R134a vapor flows shows very good agreement with all of these model predictions, as shown in [24]. Although these factors should be unaffected by frictional losses in the valve, they may very well be affected by two-phase flow. The effect of two-phase flow, then, on the peak frequency and pipe wall transmission needs to be studied further.

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