Prediction of Noise Generated by Expansion Devices Throttling Refrigerant

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ABSTRACT

A method is presented for predicting the noise generated by expansion devices throttling refrigerant, based on an existing standard model for predicting the noise downstream of large, industrial control valves throttling air. The presented method can be used to predict the noise downstream of any expansion device of any size for any pure two-phase or pure vapor refrigerant flow. Comparison of the model predictions to experimental data for orifice tubes, capillary tubes, and refrigerant valves show excellent agreement over a wide range of operating conditions. Experimental results for the attenuation of sound waves in a two-phase refrigerant flow are also presented, as well as direct comparisons of the noise generated by different expansion devices over similar operating conditions.

INTRODUCTION

Aerodynamic noise (or "flow noise") on the low pressure side of refrigeration and air-conditioning systems can be a significant problem in some units. The primary source of this noise is often the expansion device. The jet of refrigerant exiting the expansion device can generate significant aerodynamic noise, which can then propagate and interact with downstream components, such as piping and heat exchangers, inducing vibrations or even resonance, as illustrated in Figure 1.

To date, much work has been done on noise from control valves in air and water systems [1,2], but little appears to have been done on noise from expansion devices (valves, orifice tubes, and

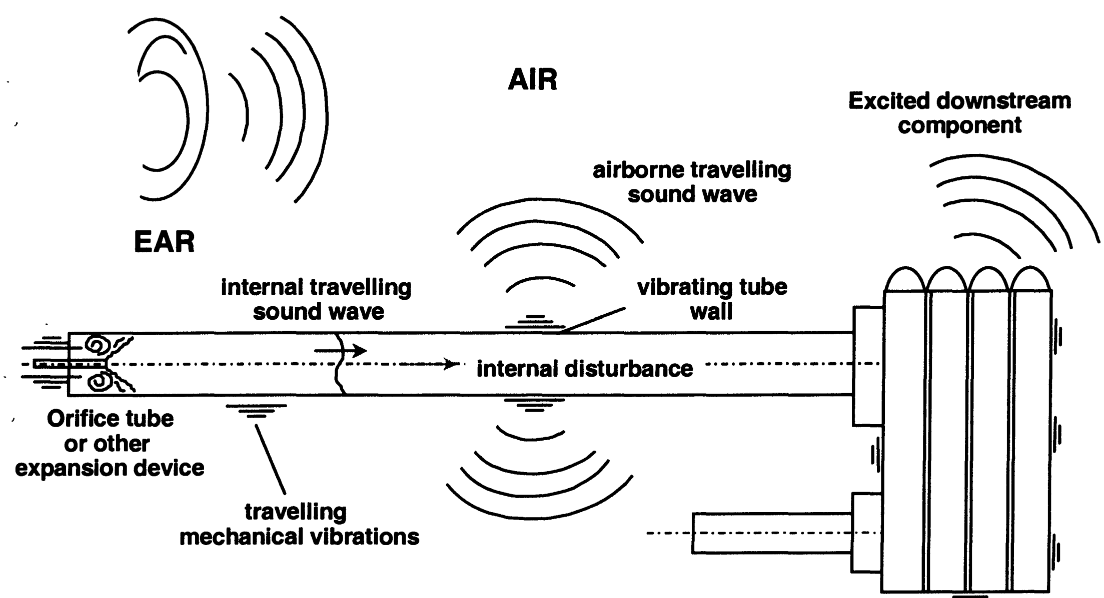


Figure 1. Overview of expansion noise

capillary tubes) in refrigeration systems. In 1995, the IEC released a standard method [3] for predicting the aerodynamic noise generated by control valves. The model showed that valve noise is primarily a function of pressure ratio across the valve, as different physical mechanisms of noise generation occurred (see Figure 2). However, the standard model is limited to ideal gases and isentropic valves (valves with no frictional losses or heat transfer between the inlet and the choking point). The standard model then, as written, can not be used for capillary tubes, nor can it be used during two-phase flow. This paper presents a method for predicting the noise from all types of refrigerant expansion devices during pure refrigerant single-phase vapor or two-phase flow. This method is based on the standard model, but includes modifications for friction and two-phase flow. A full justification of these modifications is presented in [4]. Data is presented describing some of the basics of expansion noise in refrigerant, including comparisons of the presented model predictions to experimental data, comparisons of the noise generated by different expansion devices with similar operating conditions, and experimental results for two-phase attenuation in tubes.

PREDICTION OF EXPANSION NOISE FOR CHOKED FLOW

At the choking point in any expansion device, the mechanical stream power is:

$$W_m = \frac{mc^2}{2} \tag{1}$$

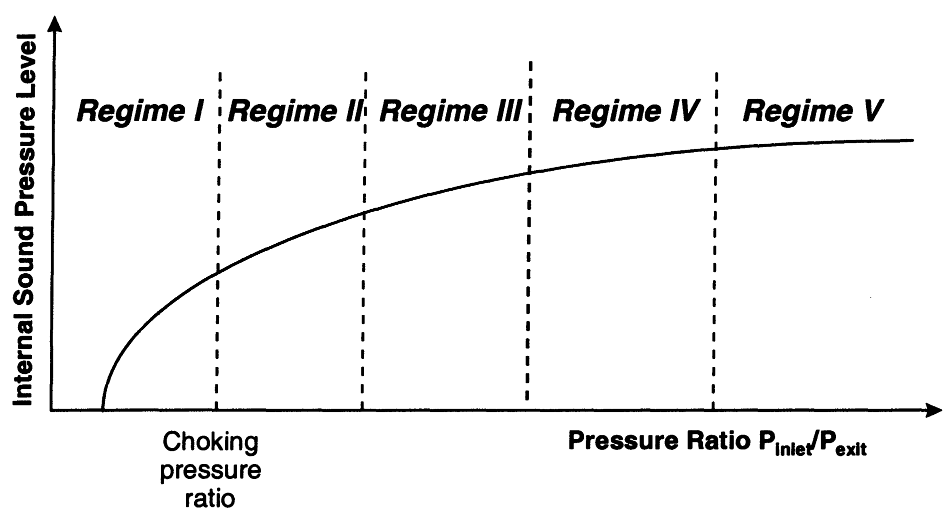


Figure 2. The five regimes of downstream expansion noise

Regime I: (Unchoked flow) Isentropic recompression. *Dipole noise due to turbulent mixing.*
Regime II: (Choked flow) Isentropic recompression. *Interaction between shock cells and turbulent choked flow mixing.*
Regime III: (Choked flow) No isentropic recompression. *Turbulent flow-shear noise mechanisms.*
Regime IV: (Choked flow) Mach disc forms. *Noise mechanism is shock cell-turbulent flow interaction.*
Regime V: (Choked flow) Constant acoustical efficiency. *Decrease in P_{exit} will no longer increase noise*

Qualitative reproduction from [3]

The speed of sound can be calculated numerically from its definition for both single-phase and homogeneous two-phase flows:

$$c = \sqrt{\left(\frac{\partial P}{\partial \rho}\right)_s} \approx \sqrt{\left(\frac{\Delta P}{\Delta \rho}\right)_s} \tag{2}$$

$$\rho = x\rho_{sv} + (1-x)\rho_{sl} \qquad \text{(Two - phase flow)} \tag{2a}$$

$$c = \sqrt{\gamma RT} \qquad \text{(Ideal gas flow)} \tag{2b}$$

For vapor flow through thermostatic expansion valves or orifice tubes, P and T at the choking point can be calculated from standard ideal gas isentropic flow relations [3,5]:

$$P = P1\left(\frac{2}{\gamma+1}\right)^{\gamma/\gamma-1} \tag{2c}$$

$$T = T1\left(\frac{P}{P1}\right)^{\gamma-1/\gamma} \tag{2d}$$

For vapor flow through capillary tubes, a Fanno-flow analysis may be used, as described in [5]. For two-phase flows though valves or orifice tubes, the choking-point pressure and quality can be calculated by assuming isentropic and isenthalpic flow. Finally, for two-phase flows through capillary tubes, pressure, quality, and velocity can be calculated by correlation, as described in [6].

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

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

where η, the acoustical efficiency factor, is given in Table 1.

Table 1. Table of Acoustical Efficiency Factors (assembled from [3])

Regime	Pressure Limit*	Acoustical Efficiency Factor*	Jet Mach Number, Mj
$P1 > P2 \geq PII$	$P_{II} = P_1 - F_L^2(P_1 - P_{III})$	$\eta = 0.0001 M_j^{3.6} F_L^2$	Mj=v/c
$PII > P2 \geq PIII$	$P_{III} = P_1\left(2/\gamma+1\right)^{\gamma/\gamma-1}$	$\eta = 0.0001 M_j^{6.6F_L^2}\left(\frac{P_1 - P_2}{P_1 - P}\right)$	$M_j = \sqrt{\left(\frac{2}{\gamma-1}\right)\left[\left(\frac{P1}{\alpha P2}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} - 1\right]}$
$PIII > P2 \geq PIV$	$P_{IV} = \left(P_1/\alpha\right)\left(1/\gamma\right)^{\gamma/\gamma-1}$	$\eta = 0.0001 M_j^{6.6F_L^2}$	Same as regime II
$PIV > P2 \geq PV$	$P_v = (P_1)/(22\alpha)$	$\eta = 0.0001\frac{M_j^2}{2}\sqrt{2}^{6.6F_L^2}$	Same as regime II
$PV > P2$		$\eta = 0.0001\frac{M_j^2}{2}\sqrt{2}^{6.6F_L^2}$	$M_j = \sqrt{\left(\frac{2}{\gamma-1}\right)\left[(22)\left(\frac{\gamma-1}{\gamma}\right) - 1\right]}$

*Although these pressure limits and the forms of η were principally determined for valves in air, they have been shown experimentally to be valid for single or two-phase refrigerant flows [4].

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

$$TSPL = 10\log_{10}\left[\frac{(x)(3.2x10^9)W_s\rho_d c_d}{D_i^2}\right] \tag{4}$$

Here 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 and x=1. The standard factor for an arbitrary valve whose jet exits at an angle is 0.25.

Model for Unchoked flow

For unchoked flow, replace "c" in Eqn (1) with "v". For valves or orifice tubes with ideal gases, v may be calculated from the standard isentropic equation:

$$v = \sqrt{2\left(\frac{\gamma}{\gamma-1}\right)\left[1-\left(\frac{P}{P1}\right)^{(\gamma-1)/\gamma}\right]\frac{P1}{\rho 1}} \tag{5}$$

For two-phase flows and/or capillary tubes, v must be determined by correlation, as in [6].

CHARACTER OF EXPANSION NOISE

Expansion noise is generally white noise over the audible frequency range (0-20kHz) for both single-phase and two-phase flows. For choked flow, a soft "haystack"-like peak may appear in the spectrum at a frequency equal to the convection velocity divided by the shock spacing [7]. The equations for estimating the peak frequency are given in Table 2.

Table 2. Peak Frequency

Regime	PEAK FREQUENCY
I	$f_p = \frac{0.2 * V_{vc}}{D_j}$
II	$f_p = \frac{0.2 * M_j * c}{D_j}$
III-V	$f_p = \frac{0.35 * c}{1.25 D_j \sqrt{M_j^2 - 1}}$

PREDICTION OF EXTERNAL PIPE-WALL ACCELERATION

If the internal sound pressure spectrum downstream of a valve is known, the external pipe-wall acceleration at a point can be calculated as well. Based on coincidence between the acoustic modes of the internal sound waves and structural modes of the tubing, only certain frequencies of the internal spectrum are transmitted though the tube. A full description of this method for refrigeration systems is given in [8].

EXPERIMENTAL PROCEDURE

In each experiment, the internal sound pressure and external pipe-wall acceleration was measured 1m. downstream of the exit of various expansion devices over a wide range of operating conditions. The internal noise measurements were made with tiny microphones (PCB piezoelectric 105B02, 2.54 mm measuring diaphragm) mounted flush to the inside wall of the downstream tubing, as shown in Figure 3. Pure R-134a was used in all experiments. Measurements were made over the audible frequency range (20-20kHz) using resolution of 25Hz and a Hanning window. All measurements were corrected for reflections (sound waves reflected off of downstream components) and confirmed by experiment to be free from internal flow disturbances, flow oscillations, or vibrations generated by other system components. The total internal sound pressure was then calculated for each measurement as follows:

$$\text{TSPL (dB)} = 10\text{Log}_{10}\left(\frac{\sum_{25\text{Hz}}^{20\text{kHz}}(\text{Sound\_pressure})^2}{(\text{reference\_pressure})^2}\right) \tag{6}$$

Please note that *for two-phase flow, the sound pressure spectrum used in Eqn.(6) must be corrected for attenuation*, as described below.

The experimental test facility used is described in full detail in [9] and [10].

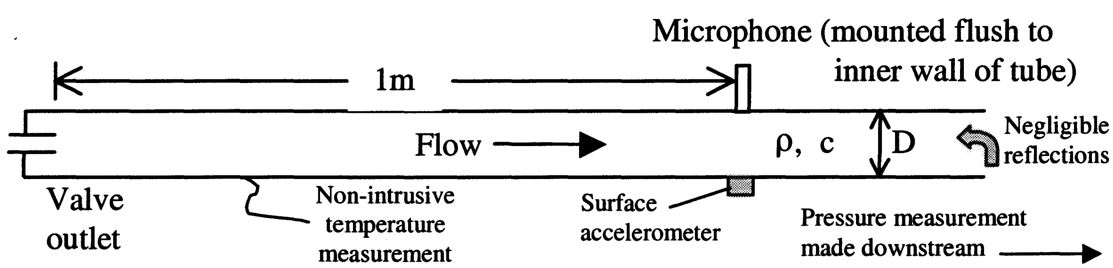
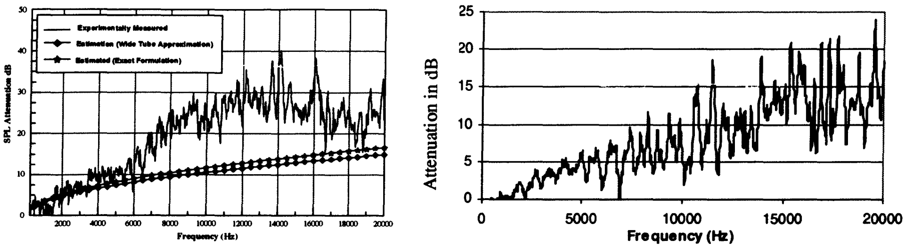


Figure 3. Basic experimental set-up

EXPERIMENTAL RESULTS

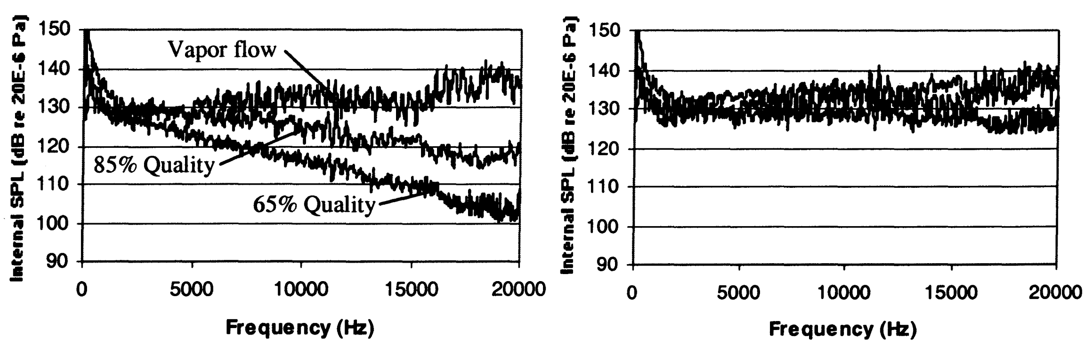
Figure 4 shows the results of both single-phase vapor and two-phase attenuation as measured in our system. Note that the single-phase attenuation is negligible over reasonable distances, but that two-phase attenuation appears much more significant. Further, the two-phase attenuation appeared to be a minor function of the quality of the flow but independent of the specific flow conditions (mass flow and pressure). *As noted, the measured spectra were corrected for two-phase attenuation before using Eqn (6) to compute the total sound pressure level.* The corrected results are shown in Figure 5. Although the problem of sound wave attenuation for gas flows in tubes has been well studied and is fairly well understood [11], there appears to be few published results for the attenuation of sound in two-phase flow in tubes.



(A) (B)

Figure 4. Attenuation of sound waves in pure Refrigerant-134a
(A) vapor over 7.24 m and
(B) two-phase flow (85% quality) over 0.635 m. Figure (A) from [11]
0.5 in O.D. copper tubing. Orifice tube (1.7 mm I.D., 38.4 mm length) used as noise source.

Figure 6 shows results for two block-type thermostatic expansion valves of different sizes and F_L factors. The valves throttled pure R-134a vapor. Note the excellent agreement between predicted noise (using the method of this paper) and experimental results over a wide range of pressure ratios. Also note the increase in noise with pressure difference and the white-noise character of the measured spectra.



(A) **(B)**

Figure 5. Internal sound pressure spectra
(A) As measured 1 m downstream of expansion device
(B) Corrected for two-phase attenuation as measured in Figure 4b*
*Simple linear correction over frequency and distance and adjusted for the effect of flow quality (0.55 <x < 0.95):
$Sp(f)_{corrected} = Sp(f)_{measured} + (d/0.635\ m)([14+(9.5-10x)]dB/20{,}000Hz)f(Hz)$

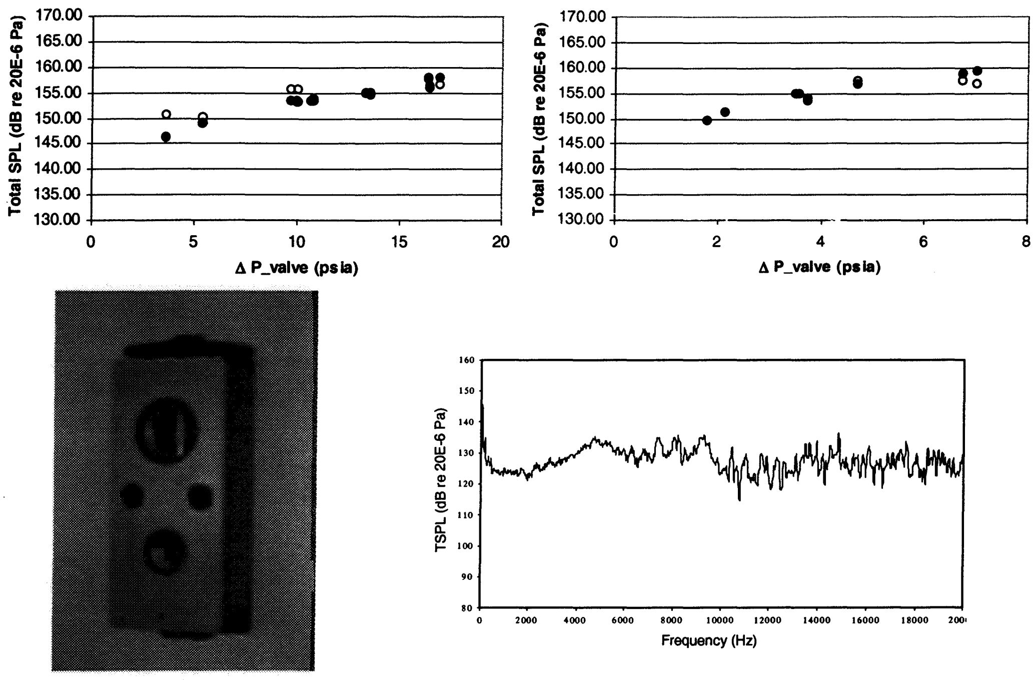


Figure 6. Experimental results for two different block-type TXVs
Top: Predicted noise and experimental results (Right: F_L=0.45, Left: F_L=0.65)
Open circles are model predictions. Closed circles are experimental data
Bottom: Photo of valve (left) and typical internal sound pressure spectra (right)

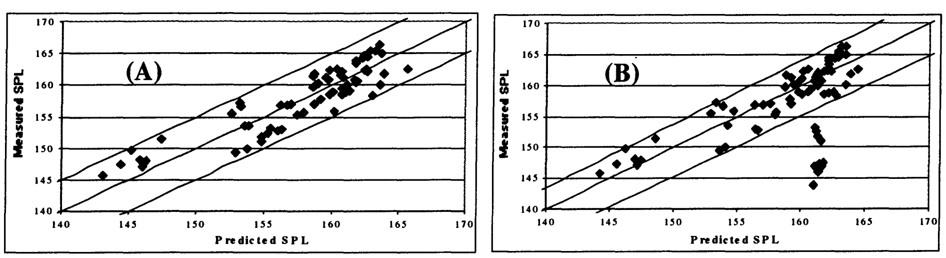


Figure 7. Predicted noise vs. experimental results for orifice and capillary tubes throttling pure R134a vapor
(A) Noise prediction method used as presented
(B) Uncorrected for frictional pressure drop in Table 1
Outlying points in (B) are capillary tube data. Lines are ± 5db (± 3.5%).

Figure 7 shows results for orifice and capillary tubes in vapor flow. There is again good agreement between the predicted and measured noise, as seen in Figure 7A. The importance of accounting for frictional pressure drop in capillary tubes is illustrated in Figure 7B. Since orifice tubes are short, they were assumed isentropic from the inlet to the exit plane. However, capillary tubes generate significant frictional pressure drop from the inlet to the choking point (exit plane – see Figure 8). When the measured inlet pressure is used in computing η, the capillary tubes show significant overprediction by the model, indicating the model does not account for energy or acoustic efficiency lost via friction along the tube length. This can be seen over a wider range of operating condition in Figure 9, where the capillary and orifice tubes were sized to yield nearly identical outlet conditions (pressure, quality/superheat) for any given set of inlet conditions (pressure, quality/superheat, mass flow) over the range of pressure differences shown.

Figure 10 shows results for an orifice tube with two-phase flow of R-134a. Again, note the good agreement between predicted noise and experimental results, as seen in Figure 10a. Figure 10b highlights the importance of correcting for two-phase speed of sound, density, and attenuation.

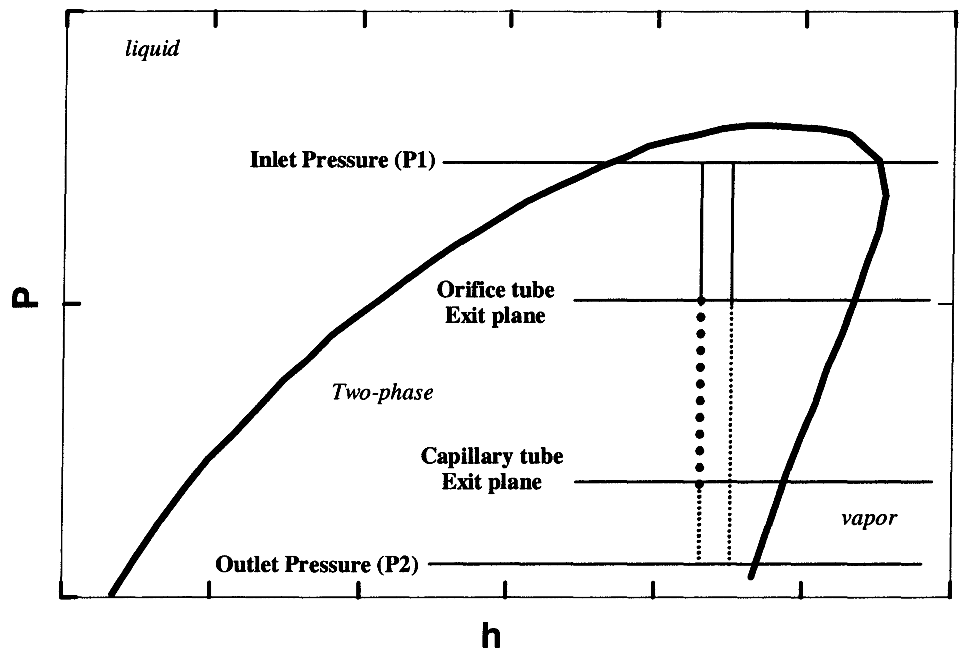


Figure 8. Qualitative view of expansion in choked orifice and capillary tubes
Solid lines: Inlet contraction
Dotted lines: Frictional pressure drop
Dashed lines: Sudden expansion
Although shown for two-phase flow, process is qualitatively similar for vapor flow

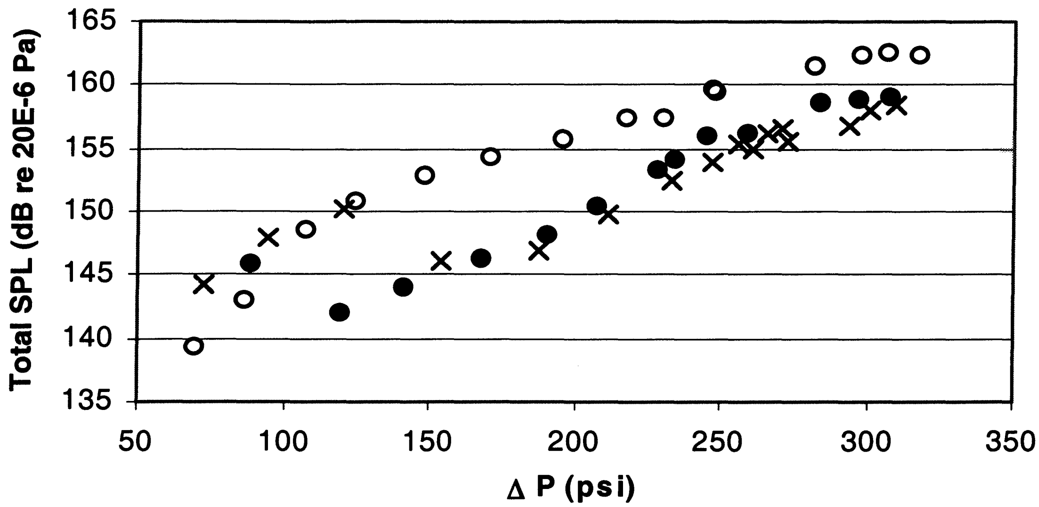
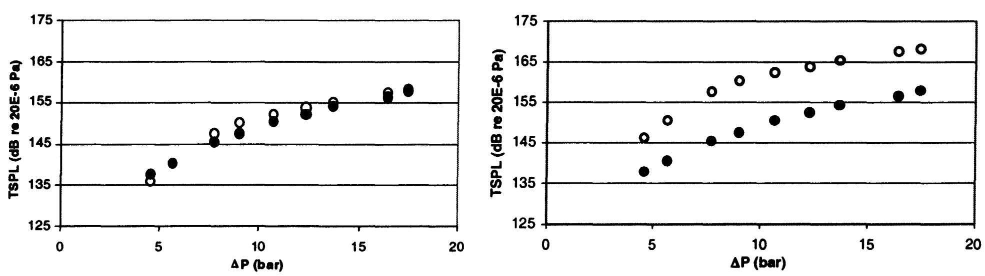


Figure 9. Noise from orifice and capillary tubes with matching inlet and outlet conditions

Open circles: Orifice tube, 1.22 mm I.D., 38.4 mm length
Closed circles: Cap tube, 1.626 mm I.D., 813 mm length
Crosses: Cap tube, 1.905 mm I.D., 1.73 m length
Mass flow rate: 75 lbm/hr (9.45 g/s), Outlet pressure: 4 bar



(A) (B)

Figure 10 – Predicted noise vs. experimental results for orifice tube in two-phase flow

(A) Noise prediction method used as presented
(B) Uncorrected for two-phase density, speed of sound, or attenuation
Orifice tube: 1.45 mm I.D., 38.4 mm length
Mass flow rate: 75 lbm/hr (9.45 g/s), Outlet pressure: 4 bar

SUMMARY AND CONCLUSIONS

A method is presented which can be used to predict the noise from any single-orifice valve, orifice tube, or capillary tube in refrigerant. The model is based on the valve noise model of IEC 534-8-3:1995, but modified to account for both frictional pressure drop along the length of the device and two-phase flows. Comparison of predicted noise to experimental show very good agreement between the two for several expansion devices over a wide range of operating conditions.

Experimental results show expansion noise to be white noise over the audible frequency range and to increase with an increase in exit velocity, mass flow, or pressure ratio. Further, for expansion devices over the same operating conditions, increased frictional pressure drop or an angle to the exiting jet can significantly decrease the expansion noise. Finally, the attenuation of sound waves in two-phase flow has been measured in refrigerant and shown to be significant, especially as compared to the attenuation present in single-phase flows.

It should be noted that again all experimental results reported here were done with an anechoic termination for high frequencies and corrected for reflections at low frequencies (see [9, 10]). In a typical system downstream components can cause significant reflections, vibrations, and even resonance (see Figure 1) and may add significantly to the measured noise. Further, as noted, all experiments were done using pure R-134a. As such, the effects of oil concentration in the system were not studied per se, but one might speculate that the noise results would be qualitatively the same, changed in the model by changing the values of attenuation, density, and speed of sound accordingly. This topic might well be worth further study. Finally, mesh screens attached to the exit of orifice tubes were found to significantly reduce the noise (by roughly 20 dB TSPL) over a wide range of operating conditions, as can be seen in [9] and [10].

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APPENDIX

Example: Orifice Tube in Vapor Flow
Orifice tube: D=1.22 mm, L=38.4 mm, P1=24.8 bar, P2=4.3 bar, T1=84.1°C, T2=41.4°C, m=75.1 lbm/hr, pure R134a vapor flow, <u>measured TSPL = 162.3 dB.</u>

Look up the following properties:
$\rho1=129.4$ kg/m^3, $\rho2=18.2$ kg/m3, $\gamma=1.05$, R=0.0823 kg/kmol-K

Assume negligible entrance effects*. Since orifice tube is short, neglect frictional pressure drop. Thus, assume isentropic flow from the inlet to the choking point (exit plane). Calculate the temperature and pressure at the choking point using isentropic ideal gas relations:

P=15.0 bar (Eqn 2c)
T=83.2°C (Eqn 2d)

Calculate the speed of sound at the choking point**:

c=171.7 m/s (Eqn 2b)

Calculate the mechanical stream power of the flow through the choking point:

Wm=139.3 W (Eqn 1)

Calculate the pressure regime cut-offs (Table 1):

PII=15.0bar
PIII=15.0bar
PIV=9.1bar
PV=1.1bar

Calculate η:

Mj=1.87 (Table 1)
$\eta = 0.0017221$ (Table 1)

The sound power exiting the valve is:

Ws=0.2398851 W (Eqn 3)

And the total internal sound pressure level downstream of the valve is:

c2 = 161.3 m/s (Eqn 2b**)
<u>TSPL = 162.8 dB</u> (Eqn 4)

* As can be calculated using the methods in [12]
** For two-phase flow, use Eqns. 2 and 2a.

NOMENCLATURE

Wm	Mechanical stream power
m	Mass flow rate
c	speed of sound
v	Velocity
ρ	density
ρsv	denity of saturated vapor
ρsl	density of saturated liquid
x	quality (0-1)
g	ideal gas specific heat ratio
P	Pressure at the choking point in an expansion device
T	Temperature at the choking point in an expansion device
P1	Pressure at the inlet of an expansion device
T1	Temperature at the inlet of an expansion device
ρ1	Density at the inlet of an expansion device
Ws	Sound power
η	Acoustical efficiency factor
F_L	Pressure recovery factor
α	Ratio of inlet pressure to outlet pressure at critical flow conditions = PIII/PII
PII	Pressure at which expansion noise regime II begins (See Figure 2)
PIII	Pressure at which expansion noise regime III begins (See Figure 2)
PIV	Pressure at which expansion noise regime IV begins (See Figure 2)
PV	Pressure at which expansion noise regime V begins (See Figure 2)
ρd	Density downstream of expansion device
cd	Speed of sound downstream of expansion device
Di	Internal diameter of downstream tubing
d	Distance (in m) from the exit of the expansion device to the point of the internal noise measurement.

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