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Comparison of Models for Piping Transmission Loss Measurements

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ABSTRACT

A frequency dependent model for the transmission loss of piping is important for accurate estimates of the external radiation from pipes and the vibration level of the pipe walls. A statistical energy analysis model is used to predict the transmission loss of piping. Key terms in the model are the modal density and the radiation efficiency of the piping wall. Several available models for each are compared in reference to measured data. In low frequency octave bands, the modal density is low. The model of the transmission loss in these octave bands is augmented with a mass law model in the low frequency regime where the number of modes is small. The different models and a comparison of the models will be presented.

1.0 INTRODUCTION

Accurately estimating the transmission loss of piping is important to predict internal and external sound levels and also levels of the pipe wall vibration. One application could be to use the known internal pipe sound level to calculate the external sound pressure level for predictions of noise levels in the environment of the pipe. An additional application is to use the known internal sound level to predict the pipe wall vibration levels, to then predict dynamic fatigue of the pipe and fittings.

For this work, the primary application is to use measured pipe wall vibration levels to predict the internal sound levels. In a typical lab facility, piping is instrumented with pressure sensors to measure the internal sound. These are often only attached at a few locations because of cost and convenience. Further, in the field, it is unlikely that any pressure sensors would be attached to the piping, since penetrations such as pressure sensors are not desirable in field applications. Therefore, having a method to accurately predict the internal sound levels from measured pipe wall vibrations is desirable.

The driving factor behind investigating this subject matter is the current work at Fisher Controls International, LLC, in Marshalltown, Iowa. The main focus of this work is a part of developing a complete understanding of the aerodynamic noise generated by control valves and propagation into the downstream piping. The goal of this work is to predict the internal sound pressure level of the flowing compressed air inside the pipe by measuring the pipe wall vibration velocity response using an accelerometer.

The general subject of flow induced noise and vibration is a large and complex one. The subject includes internal axial flows, the transmission of large volume flows of gases, liquids, or two-phase mixtures across high pressure drops through complex piping systems comprising of bends, valves, tee junctions, orifice plates, expansions, and reductions. This investigation will only look at noise and vibration from a steel pipeline of some straight length with internal compressed air flow generating flow induced noise and vibration.

Several models available to predict the transmission loss of piping will be reviewed. These models will be coupled with other models for the radiation from piping to use the measured pipe wall vibration levels to predict the internal sound levels.

2.0 MODEL BACKGROUND

There are several current methods in the literature to calculate a transmission loss value. One method that is used by control valve venders is the IEC method [1]. The IEC control valve standard number 534-8-4 is used to predict externally radiated noise one meter from a pipe wall, one meter downstream of the control valve outlet, and utilizes highly simplified procedures for calculating transmission loss that are specifically tailored to control valve noise based on transmission loss at a single frequency, and are unsuitable for more general predictions of piping system noise.

Another method developed to investigate the response of a cylinder to sound was done by Fahy [2]. Those results were addressing large radius, thin walled cylinders, and Fahy contended that the theory could also be applied to pipe like structures, but no data was given to support it. Norton [3] provides several ways to calculate the transmission loss from various sources. One method offered by Norton provides some general formulations of the vibration response of and the radiated sound power from a thin walled cylindrical shell which is subjected to a random internal wall pressure field. Another method of calculating the transmission loss is by Fagerlund [4] [5], who provides an equation that relates the external pressure to the internal pressure using statistical methods which is highly dependant on the mode count and radiation efficiency. Szechenyi [6] outlines a method for estimating sound transmission through cylinder walls using statistical methods and applies them to considerations in aerospace structures, settling chambers, and various other large cylindrical containers.

In all of these models, one of the important parameters besides radiation efficiency is the number of structural modes within a given frequency band. The mode count is usually obtained by either using a finite element analysis program or calculating a modal density function and multiplying by the appropriate frequency bandwidth. Models are available from Norton [3], Lyon & DeJong [7], and Szechenyi [6]. Szechenyi [8] also provides a method for estimating the radiation efficiencies from cylinders.

Each of the methods stated above has a piece to the overall calculation of the predicted internal sound pressure level and the subsequent prediction for transmission loss. However, none of them bring all the pieces together into one method for predicting the internal sound pressure level from measuring the pipe wall vibration velocity response. The work described in this paper, takes the following steps by established methods: 1) the externally radiated sound pressure (p_0) from a vibrating pipe (v), 2) the pipe wall transmission loss (p_0/p), 3) the mode count (n), and 4) the radiation efficiency (σ). Using the various formations for each, three composite methods are created and compared to measured data.

3.0 MODELS

Formulations for each of the four steps are briefly presented. More detailed information is available in the references.

3.1 Externally Sound Pressure Level (p_0)

From Fagerlund and Chou [5], the relationship between the velocity of the pipe wall and the acoustic pressure, p_0^2, at a distance from the pipe centerline, r, is

$$p^2 = \rho_0^2 c_0^2 v^2 \frac{D}{2r}\frac{f}{f_c} \tag{1}$$

below the coincidence frequency, f_c, and

$$p_0^{\,2} = \rho_0^2 c_0^2 v^2 \frac{D}{2r} \tag{2}$$

above the coincidence frequency, where ρ_0 is the ambient density is, c_0 is the ambient wave speed is, v is the pipe wall velocity, and D is the pipe outside diameter. Note, that in this model, the radiation efficiency is f/f_c below the coincidence frequency and equal to 1 above the coincidence frequency.

3.2 Transmission Loss (p_0/p)

Fagerlund [4] developed an equation that related the mean square acoustic pressure inside the pipe to the mean square acoustic pressure outside the pipe at a given observation point, r,

$$\frac{p_0^2}{p^2} = \frac{5\rho_0^2 c_0^2 c^2 D(\Delta k_{ZS})G(M)\sigma\sigma_0}{18\rho_S a r t \omega^2 \Delta\omega(\rho c\sigma + \rho_0 c_0 \sigma_0 + t\rho_S \omega \eta_S)} \tag{3}$$

where the new variables are: Δk_{ZS} is the change in the axial structural wave number, $G(M)$ is a correction factor for flow, t is the pipe wall thickness, a is the pipe radius, η_s is the pipe wall damping factor, ω is the circular frequency, and Δω is the circular frequency bandwidth. The variables ρ and c with no subscript are for the internal fluid, with an s subscript are for the pipe wall, and with an 0 subscript are for the external fluid. Before Eq. 3 can be used, there are several quantities that need to be defined and derived.

Fagerlund[4] presents methods to estimate the change in axial wave number, Δk_{ZS}, based on coincidence of the modes in the pipe wall and internal sound field. In this paper, the change in axial structural wavenumber, Δk_{ZS}, will be estimated as the number of modes within a frequency band [4], $N_s(\omega)$,

$$\Delta k_{ZS}(\omega) = \frac{\pi}{L} N(\omega) \tag{4}$$

where L is the pipe length.

Based on Eqs. 3 and 4, the modal density and radiation efficiency of the pipe are needed.

3.3 Mode Count (N)

Three different methods of deriving the number of modes within a frequency band are used in this investigation:

METHOD 1: Lyon and DeJong [7]

$$N(f) = \frac{L}{2\kappa}\frac{f}{f}\left\{1 + \left[\frac{\pi/2}{\sqrt{f/f_r} + 1/2\left(f/f_r\right)^{3.5}}\right]^4\right\}^{-1/4} \tag{5}$$

METHOD 2: Norton [3].

$$N(f) = \begin{cases} \dfrac{5S}{\pi c_L t}\left(\dfrac{f}{f_r}\right)^{1/2} & \dfrac{f}{f_c} \le 0.48 \\ \dfrac{7.2S}{\pi c_L t}\left(\dfrac{f}{f_r}\right) & 0.48 \le \dfrac{f}{f_c} \le 0.83 \\ \dfrac{2S}{\pi c_L t}\left[2 + \dfrac{0.596}{F - 1/F}\left\{F\cos\left(\dfrac{1.745 f_r^2}{F^2 f^2}\right) - \dfrac{1}{F}\cos\left(\dfrac{1.745 F^2 f_r^2}{f^2}\right)\right\}\right] & \dfrac{f}{f_c} \ge 0.48 \end{cases} \tag{6}$$

where S is the surface of the pipe, and for one-third octave bands, F=1.122. Norton [3] notes that Eq. 7 does not account for the grouping of circumferential modes in cylindrical shells at frequencies below the ring frequency.

METHOD 3: Szechenyi [6] (The reader is directed to the reference for details.)

3.4 Radiation Efficiency (σ)

The radiation efficiency was calculated using a composite of a shell model from Szechenyi [8] who defines the external and internal radiation efficiency to be approximately equal to each other when averaged over a number of modes. The reader is referred to Szechenyi [8] for details of the model.

One issue to be resolved is how to handle the radiation efficiency near and above coincidence. As was observed in the measured data, there is a strong coincidence dip in the transmission loss, so the radiation efficiency of a plate, Fahy [2], is used near coincidence. Then as the plate model decreases below unity above the coincidence frequency, the radiation efficiency is set to 1. A radiation efficiency of 1 above coincidence is supported by work such as Fagerlund and Chou [5] and Szechenyi [6]

4.0 TEST SETUP

Figs. 1 and 2 show the test setup. The tests were performed on 8" SCH 40 A105 seamless steel pipe, with a wall thickness of 0.322 inches, a longitudinal wave speed of 5392 m/s, and a density of 7860 kg/m^3. A control valve served as the noise source. The test data shown in this paper were obtained for the valve operating at a pressure drop ratio of 0.50 with downstream pipe flow density of 14.1 kg/m^3, mass flow rate of 20.7 kg/s, and downstream pipe velocity of 45.5 m/s (Mach 0.13).

Figure 1 Fisher Controls research flow lab. Noise line is second from left.

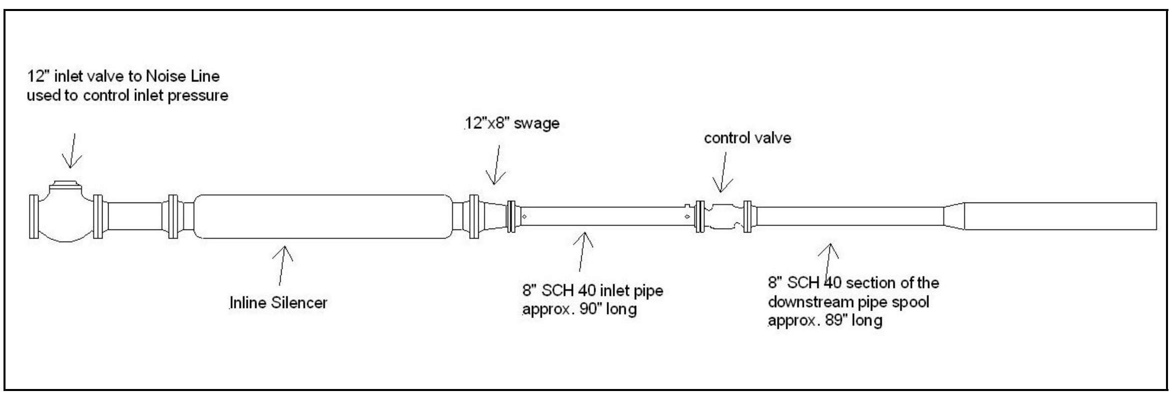

Figure 2 Schematic of the flow line used for testing. 8" SCH 40 section on the far right was the test section and the control valve was the internal noise source.

The pipe wall vibration was measured with quartz shear ICP® accelerometers, PCB model 353B16, with approximately 10 mV/g of sensitivity, and a frequency range of 1 to 10000 Hz. The accelerometers were mounted on the pipe wall using Loctite. The external sound was measured with ICP® type, PCB model 377A02, ½" diameter, prepolarized, condenser free field microphone, with sensitivity of approximately 50 mV/Pa. The microphone was mounted on a stand and the face of the microphone was one meter away, perpendicular to the pipe wall. The internal sound was measured using PCB model 106B50 with acceleration compensation, sensitivity of approximately 500 mV/psi, and resolution of 0.00007 psi or 87 dB. This pressure transducer was mounted so the face of the transducer was flush with the internal pipe wall, 37 inches downstream of the control valve. Care was taken to ensure that the pressure transducer was mounted flush with the pipe wall.

A Bruel & Kjaer multi channel spectrum analyzer was used to measure the data from the free field microphone, the pressure transducer, and the accelerometers at the same time. The data was collected using a 1/3 Octave analyzer setup, measuring 1/3-octave center band frequencies from 10 Hz up to 10 kHz.

The data below 100 Hz included noise from the piping system and large compressors in addition to the flow noise generated by the valve. Further, the external microphones were placed in the free field without an anechoic enclosure to simulate a field test environment. The data below 100 Hz is important for piping fatigue and structural design and will be studied in future work.

5.0 RESULTS

The fist step in the process was estimation of the externally radiated sound. Fig. 3 shows a comparison of the external sound radiation at one microphone location compared to the estimated sound radiation using the data from an accelerometer at the same axial position along the pipe. Above 400 Hz the results compare within 5 dB and above 800 Hz the results agree to within 3 dB. Below 200 Hz the model under predicts the measured data, indicating that the radiation efficiency is under predicted or the contributions of noise generated from other parts of the flow line and other equipment in the lab are major contributors to the measured sound.

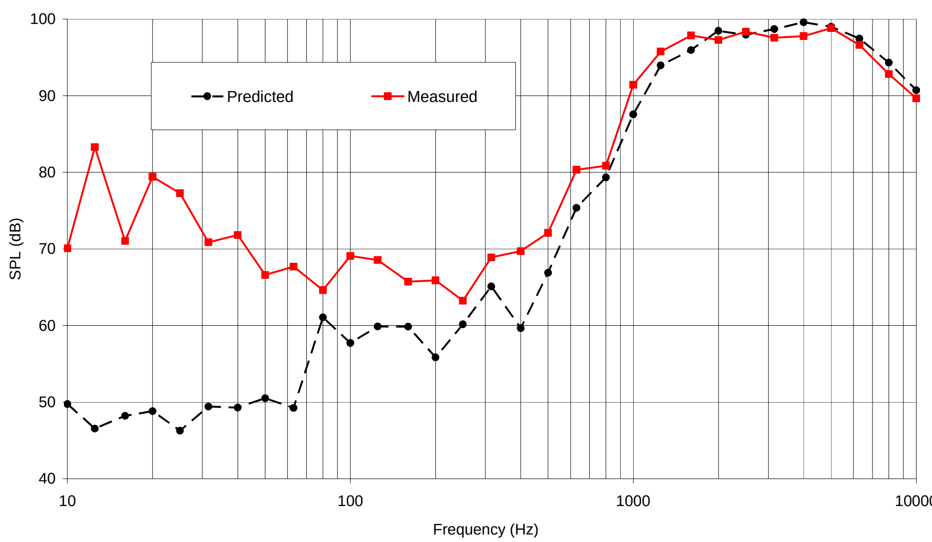

Figure 3: Comparison of the measured and predicted external sound level.

Fig. 4 shows a comparison of the mode count calculated from the three methods. The results for the Szecheni formulation, Method 3, predicts mode counts over a factor of 10 larger than either other method. In general all three methods increase with frequency, but at different rates. There are no measured values available to compare to these estimates.

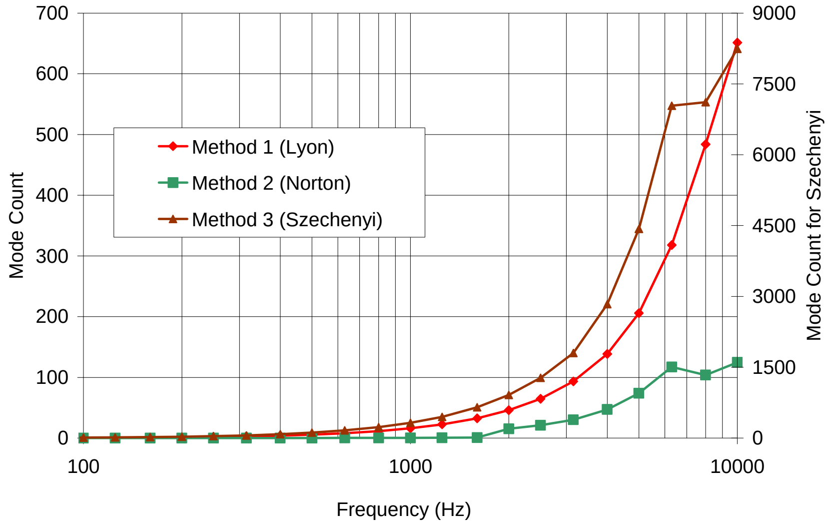

Figure 4: Comparison of the predicted modal density from the three methods.

Fig. 5 shows a comparison of the measured and predicted transmission loss using Eq. 3 and the three different mode count methods. The results for Method 1, the Lyon and DeJong mode count, provide the best agreement over the largest frequency range. Above 2000Hz, both Methods 1 and 2, provide estimates of the transmission loss within 3dB of the measured value. Note that the coincidence frequency is at 1677 Hz. The deviations below 400 Hz in the measured transmission loss and the transmission loss predicted with Method 1 appear to be related to resonance in the system since they are largest at particular frequencies. Based on the similar discrepancies in the external sound radiation, Fig 4, the influence of additional noise sources can not be disregarded below 400 Hz.

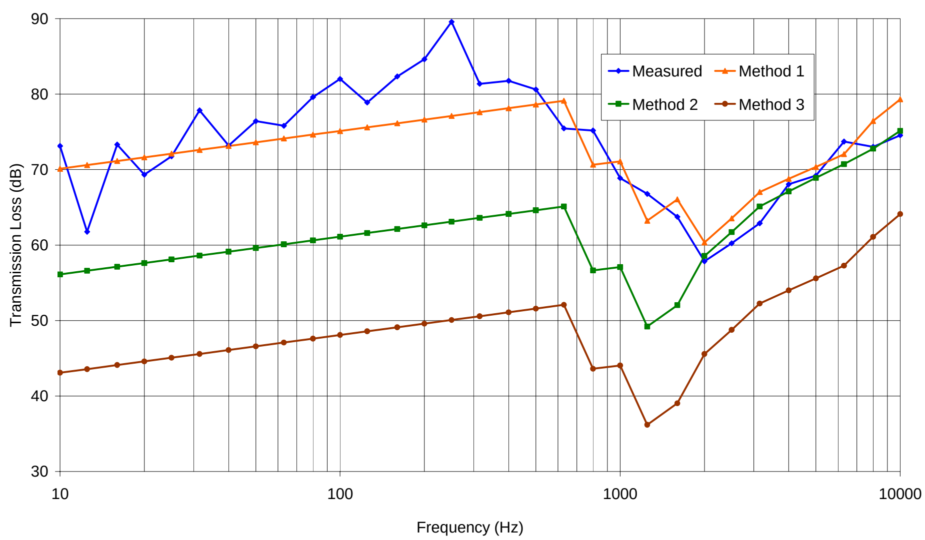

Figure 5: Comparison of the predicted transmission loss of the three methods.

Fig. 6 shows a comparison of the measures and predicted internal pressure. Method 1 provides the best match over the largest frequency range. Above, 2000 Hz, both Method 1 and 2 agree within 5 dB. There appear to be resonances that were predicted in all three methods that were not measured. These resonances are related to the measured vibration of the pipe wall and are not contained within the model of Eq. 3 and the related mode count models.

The results in Fig. 6 show a large deviation between the measured and predicted values below 400 Hz. Other than at 80Hz, the closest method, Method 1, under predicts the internal level by over 10 dB. The under prediction is likely related to the under prediction of the external sound level, as shown in Fig 3.

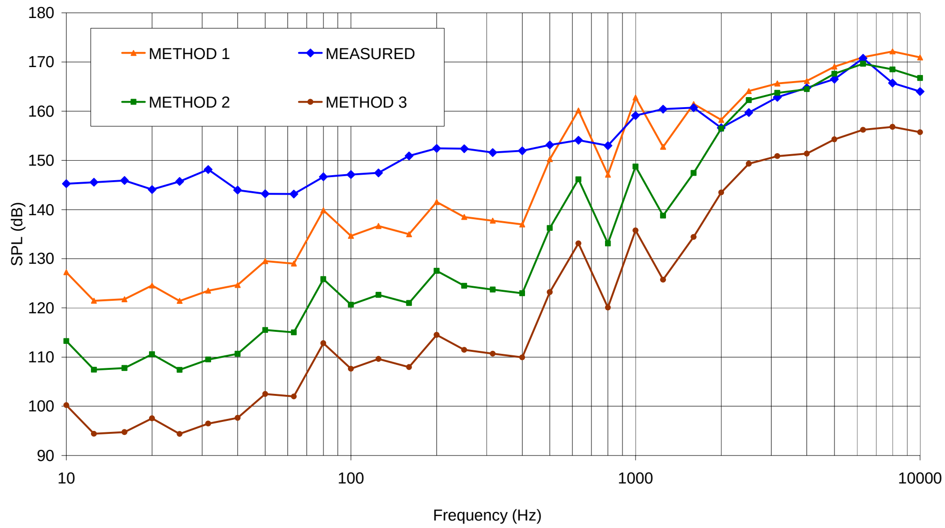

Figure 6: Comparison of the measured and predicted internal sound level.

6.0 CONCLUSIONS

The process of predicting the sound pressure inside a pipe by using the measured pipe wall vibration was explored. For all the work, existing prediction methods were used that implemented a statistical energy analysis approach. The models generated results within 3 dB above the coincidence frequency and within 5 dB between the coincidence frequency and roughly half the coincidence frequency. Below this range, all methods under predicted the internal level by 10 dB or more. The discrepancy is related to the under prediction of the external level or other noise sources in the lab entering the measurements. Future work should focus on improving the prediction below the coincidence frequency.

7.0 REFERENCES

[1] IEC (International Electrotechnical Commission) Control Valve Standard IEC 534-8-3, “Part 8: Noise Considerations – Section 3: Control Valve Aerodynamic Noise Prediction Method” (1995).

[2] Fahy, F.J., “Response of a Cylinder to Random Sound in the Contained Fluid”, Journal of Sound and Vibration (1970) **13** (2), 171-194.

[3] M.P. Norton, Fundamentals of Noise and Vibration Analysis for Engineers, First Edition, Cambridge University Press, 1989.

[4] A.C. Fagerlund, “A Theoretical and Experimental Investigation on the Effects of the Interaction Between an Acoustic Field and Cylindrical Structure on Sound Transmission Loss”, Ph.D. Thesis, University of Iowa, 1979.

[5] A.C. Fagerlund & D.C. Chou, “Sound Transmission Through a Cylindrical Pipe Wall”, ASME Paper 80-WA/NC-3 presented at the ASME Winter Annual Meeting, Nov. 1980.

[6] E. Szechenyi, “Sound Transmission Through Cylinder Walls Using Statistical Considerations”, Journal of Sound and Vibration (1971) **19** (1) 83-94.

[7] R.H. Lyon and R.G. DeJong, Theory and Application of Statistical Energy Analysis, Second Edition, Butterworth-Heinemann, 1995.

[8] E. Szechenyi, “Modal Densities and Radiation Efficiencies of Unstiffened Cylinders Using Statistical Methods”, Journal of Sound and Vibration (1971) **19** (1) 65-81.