Minneapolis, Minnesota
NOISE-CON 2005
2005 October 17-19

Piping noise transmission loss calculations using finite element analysis.

Richard Eberhart, Fred W. Catron, Allen C. Fagerlund
Fisher Controls International,
301 South 1st Street,
Marshalltown, IA, 50158
Fred.Catron@EmersonProcess.com

Denis G. Karczub
SVT-Engineering Consultants, Australia
Denis.Karczub@svt.com.au

J. Adin Mann III
Iowa State University
2032 Black Engineering
Ames, IA 50011, USA
jamann@iastate.edu

ABSTRACT

The prediction of noise radiated by piping downstream of a control valve is subject to various uncertainties. One of the significant sources of uncertainty is the pipe-wall transmission loss. Due to the difficulties in experimentally measuring pipe-wall transmission loss accurately, and practical difficulties of taking into account pipe length and boundary conditions, an analytical/numerical approach for the calculation of transmission loss is required. The feasibility of coupled structural-acoustic finite element based calculations of transmission loss is being investigated for this purpose. This should also assist in the refinement of analytical/statistical calculations of transmission loss and noise radiation.

BACKGROUND

Predicting the noise radiated from piping system internal flows is a complex problem. In general there are three steps in this process. First, the noise inside the pipe such as the noise generated by flow through a valve or tee junction is predicted. Second, the propagation of noise inside the piping system is modeled so that the sound distribution inside each pipe section is represented. Third, a transmission loss model is applied to the internal sound field to predict the external sound field generated from each piping section. The transmission loss model captures how the internal sound field excites the pipe walls resulting in external sound radiation.

The accuracy of transmission loss modeling has a significant impact on the accuracy of predicted external noise levels, and is a likely factor in sometimes large errors of several decibels in predictions obtained using IEC control valve standard 534-8-4 [1]. The historical approach for modeling transmission loss has been to assume infinite length piping and develop calculation schemes focused on capturing modal effects using wave-number co-incidence concepts. Furthermore, a statistical model that averages the interaction of the internal sound and pipe-wall vibration over several modes was used. This approach was developed and successfully implemented by Fagerlund and Chou [2,3], although the model is limited to applications and frequency ranges with a high modal density.

In finite length piping, discrete frequency phenomena may tend to dominate the radiated sound field as illustrated by Karczub, Fagerlund & Catron [4]. Discrete frequency phenomena are sensitive to the length of the pipe, pipe diameter, wall thickness and internal acoustic boundary conditions, in addition to the internal excitation source. A general transmission loss model that results from a statistical model cannot be fully sensitive to these discrete frequency phenomena. The approach proposed in this paper is to use the statistical model over the frequency ranges that it is accurate, and then augment the transmission loss model with a modal model for discrete frequency phenomena. This approach takes advantage of the precision of the modal models where needed and the simplicity of the statistical model where applicable.

One means of developing a frequency-domain modal transmission loss model is to generate a modal model of the internal sound field and a separate modal model of the pipe wall vibration using the assumption of uncoupled modes. Finite element analysis (FEA) is the computational technique used for the modal calculations. The goal of this paper is to introduce the concept of using finite element analysis as a practical means of implementing modal transmission loss calculations to address discrete frequency phenomena. FEA may be used to develop estimates of variations from the statistical model for transmission loss, to calculate transmission loss in specific critical applications, and to provide a research tool to better understand transmission loss in the low- to mid-frequency regime in order to enhance statistical methods.

The system studied here is taken from the test work reported by Karczub et. al. [4]. That paper presented studies of a 3 m long test pipe that was closed at one end and open to atmosphere at the other. A loud speaker mounted in the closed end excited the acoustic modes. Sound pressure levels were measured on the inside and outside of the pipe wall over a range of frequencies. In that paper, some initial pipe-wall vibration finite element results were presented and compared to the measured data. In this paper, additional work is presented including acoustic finite element calculations of the higher-order acoustic modes and coupling of the acoustic field inside the pipe to the pipe-wall structural response.

TEST DATA

The pipe reported on in [4] is shown in Figure 1. The pipe is closed at the inlet end, with a small off-center opening for mounting of the loud speaker used to excite the pipe internal acoustic field. The other end of the pipe is open giving classical closed-open acoustic boundary conditions. The pipe is 200NB Schedule 40 with 101.4 mm internal radius and 8.2 mm wall thickness. The carbon steel pipe is 3000 mm long. The inlet end of the pipe is closed using a 10 mm thick steel disc welded over the end of the pipe, and the outlet end of the pipe is un-flanged giving classical fixed-free structural boundary conditions. The pipe is supported by thin cables which were assumed to have a negligible effect on the structural response of the pipe-wall. There were several axial and circumferential tapping points where measurements could be performed using a pressure transducer, Fig. 1.

Figure 2 shows the transmission loss and the external sound level over the frequency range of 1600 Hz to 2000 Hz. There are clear discrete frequencies that are evident in the acoustic response of the pipe wall.

Figure 1. *Experimental Arrangement (Pipe is closed at left-hand end with loud speaker; pipe is open at the right-hand end (not shown); microphone tappings at several axial and circumferential locations)*

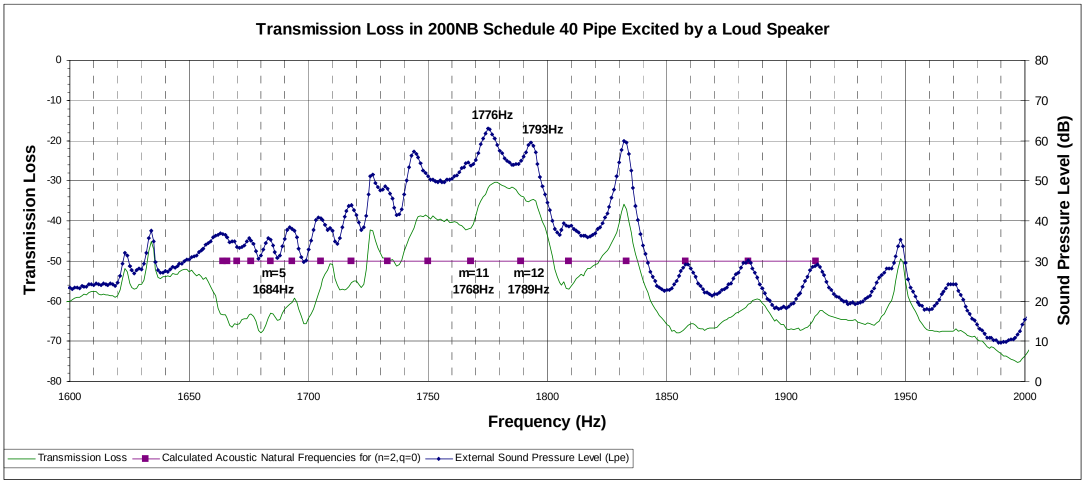

Figure 2. *Measured pipe-wall transmission loss spectrum at frequencies covering the axial modes m=1 to 18 of circumferential mode n=2. (Karczub, Fagerlund & Catron[4])*

FINITE ELEMENT ANALYSIS (FEA) PROCEDURE

Due to the complex interaction between acoustic and structural response, finite element analysis methods were used to simulate the test over a selected frequency range. The finite element models were prepared and solved using Version 8.1 of ANSYS

Structural and acoustic modal analyses were initially performed to establish the boundary conditions that would best correlate with how the test pipe was supported. Next, a coupled acoustic-structural harmonic analysis was run to simulate the response of the pipe to the acoustic modes present in the frequency range of interest.

STRUCTURAL MODAL ANALYSIS RESULTS – SCHEDULE 40 PIPE

Structural modes were determined from 1600 to 1900Hz. The results varied significantly with the boundary conditions used. The best agreement with experimental results was obtained using no constraints (free-free). As constraints were required for the harmonic analysis, several ways of constraining the model were investigated. The best correlation was obtained by fixing the center region of the disk used to close one end of the pipe. The modes of interest from this constraint were in agreement with those from the unconstrained model. In general, the calculated modes were approximately 102% of the values observed experimentally. Analysis results are compared with experimental values in Table 1. Mode plots for the constrained model are shown in Figures 3 and 4 for two of the structural modes.

Table 1. Structural Modal Analysis Results Summary

Mode (n,m)	Test (Hz)	Analysis (Hz)
(1,6)	1405	1604
(3,8)	-	1608
(3,9)	1637	1682
(3,10)	1730	1772
(2,11)	1776	1806
(1,7)	1799	-
(3,11)	1835	1878

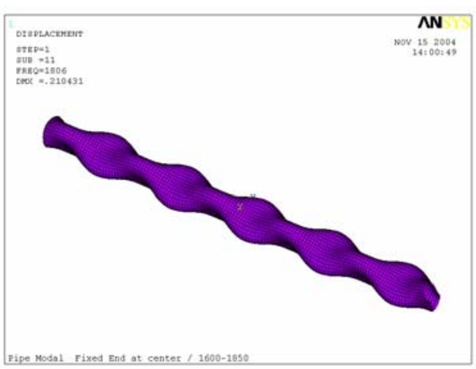

Figure 3. Structural Mode (2,11) at 1806 Hz.

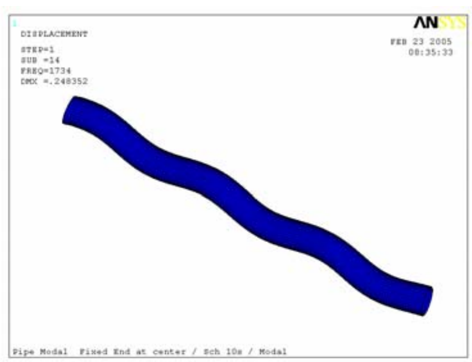

Figure 4. Structural Mode (1,6) at 1604 Hz.

A harmonic analysis was also performed from 1796 Hz to 1900 Hz for structural response of the pipe only. An arbitrary pressure of 344000 N/m^2 (50 psi) was applied to one element face located at an antinode for the mode (2,11) at 1806Hz. This location was also an antinode for the mode (3,11) at 1878Hz. The structural natural frequencies at 1806 Hz and 1878 Hz were confirmed. No distinct anti-resonance was noted.

ACOUSTIC MODAL ANALYSIS RESULTS – SCHEDULE 40 PIPE

Acoustic modes were determined from 1800 Hz to 1900 Hz. A rigid boundary condition was imposed on the air internal to the pipe by increasing the elastic modulus of the pipe material by a factor of 100. In general, the calculated acoustic mode natural frequencies were approximately 102% of the theoretical values. Analysis results are compared with theoretical values in Table 2. The theoretical values are taken from a wave number analysis performed by Karczub et. al. [4]

Acoustic mode plots are presented in Figures 5 and 6 where acoustic pressure is shown contoured. Analysis of all the modes revealed a 30 Hz offset between the FEA calculated modes and those predicted by theory/experiment. The cause of the shift is being investigated. Where required, a 30Hz shift is applied to the results for comparison with experimental data.

Table 2. Acoustic Modal Analysis Results Summary

Mode (n,q,m)	Theory (Hz)	Analysis (Hz)
(1,0,25)		1802
(2,0,11)	1768	1808
(2,0,12)	1789	1830
(1,0,26)		1852
(2,0,14)	1833	1880

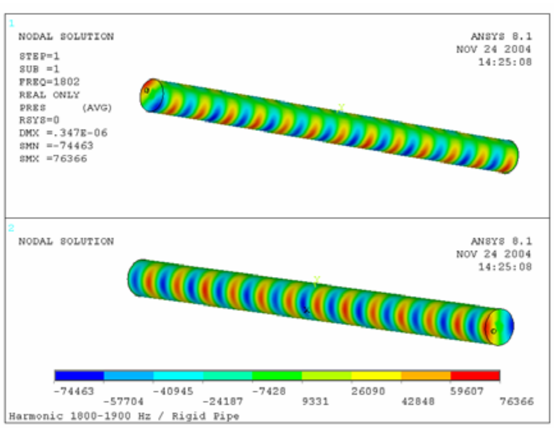

Figure 5. Acoustic Mode (1,0,25) at 1802 Hz.

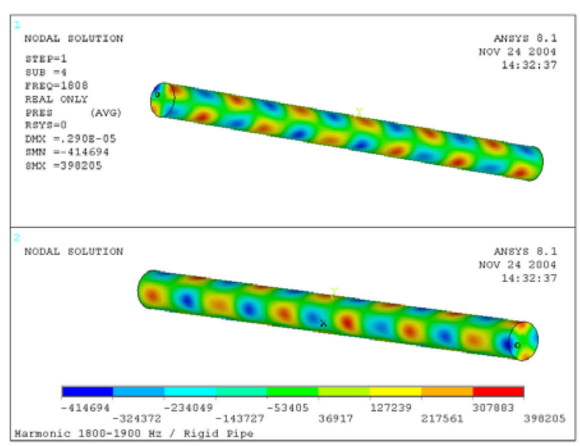

Figure 6. Acoustic Mode (2,0,11) at 1808 Hz

COUPLED HARMONIC ANALYSIS RESULTS

A harmonic analysis was performed with coupled degrees of freedom between the acoustic elements representing the air internal to the duct and the structural elements simulating the duct. Air external to the pipe was not represented. Damping effects were not included in the model. An arbitrary 30000 N/m^2 (5 psi) harmonically varying pressure excited the internal air. The pressure at the open end of the duct was set to 0. In acoustic elements, the pressure is deviation from mean pressure rather than absolute pressure. The frequencies from 1800 to 1900 Hz were investigated in 2Hz increments. The intent was to simulate the coupling of the acoustic (2,0,11) and structural (2,11) modes at 1808 Hz and 1806 Hz respectively. The mode at 1806 Hz corresponds to the mode observed at 1776 Hz in the experimental test (n=2,m=11). The spike observed in the test data at around 1830 Hz was also of interest.

Figure 7 plots sound pressure level over the frequency range of interest at five locations on the inside of the pipe wall. Sound pressure level peaks at 1818 Hz. Secondary peaks can also be noted at 1802 Hz, 1830 Hz, 1854 Hz and 1880 Hz.

Figure 8 provides contours of pressure at the inside of the pipe wall at 1818 Hz. The deformed shape of the pipe is also represented. The increased response at 1818 Hz reflects the partial coincidence between the structural resonance at 1806 Hz (2,11) and the acoustic resonance around 1808 Hz (2,0,11). It is of interest that coincidence effects occur well above the natural frequencies of the nearly coincident structural and acoustic modes.

Figure 9 shows the acoustic and structural response at 1802 Hz. Similar plots were viewed for 1830 Hz, 1852 Hz, 1854 Hz and 1880 Hz respectively, corresponding to the secondary peaks observed in the internal sound pressure as shown in Figure 7. The maximum structural deformation response (denoted DMX on plots) at these frequencies is 3 to 4 orders of magnitude less than noted at 1818 Hz. The peaked response at these frequencies is primarily from acoustic resonance rather than from acoustic-structural coincidence. As expected, acoustic-structural coincidence results in a much larger response due to resonance effects.

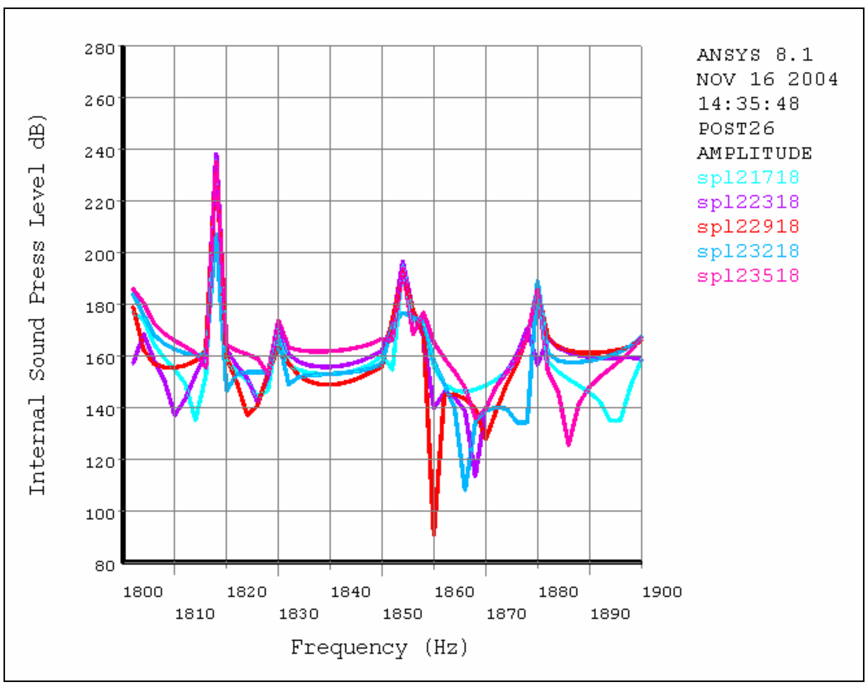

Figure 7. Sound Pressure Level versus Frequency for Selected Locations at Inside of Pipe Wall.

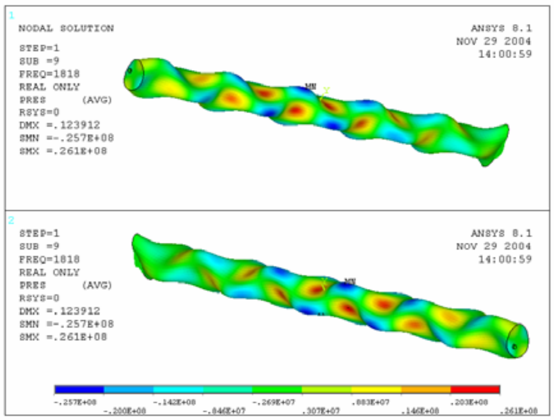

Figure 8. Coupled Response at 1818 Hz. (Maximum response frequency)

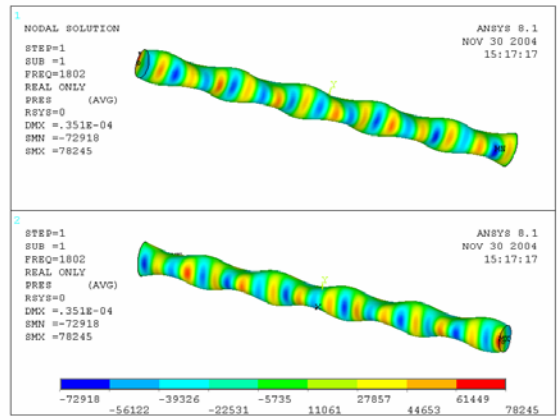

Figure 9. Coupled Response at 1802 Hz. (Acoustic resonance frequency)

PREDICTING TRANSMISSION LOSS

The pipe wall displacement as a function of frequency was extracted from the ANSYS results for an anti-node remote from the pipe ends. Similarly the internal sound pressure was obtained. The results are shown in Figure 10. Sound levels are higher than expected, but are due to the arbitrary scaling factor used in the excitation model. Since a transmission loss prediction is being developed, only the relative difference between the internal and external sound levels is important. If needed as an absolute prediction, the excitation of the internal field can be scaled such that the internal sound level predictions are more reasonable.

The pipe wall vibration displacement is then used to calculate the external sound radiation using a radiation efficiency model. Subtraction of the calculated internal sound pressure level from the external sound pressure level results in the transmission loss. Figure 11 shows a comparison of the calculated and measured transmission loss. The predictions capture the broad harmonic content of the measured transmission loss.

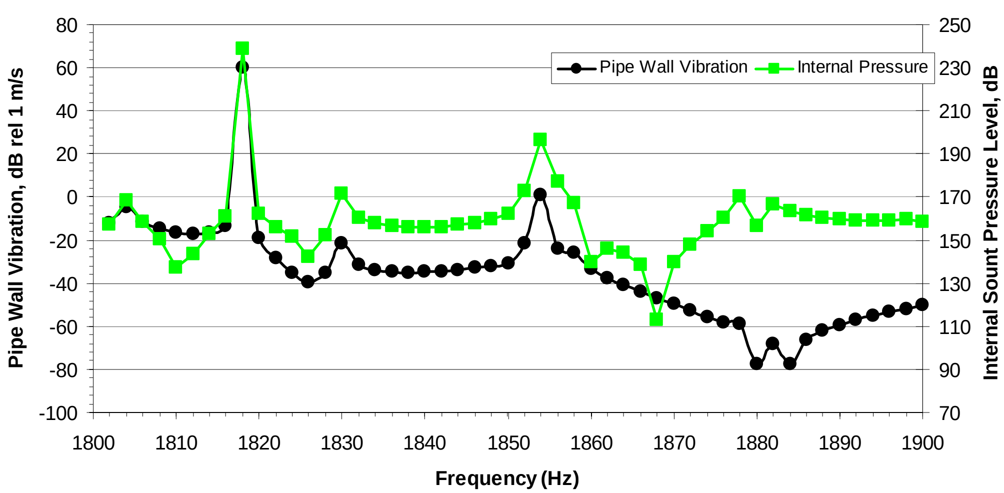

Figure 10. FEA calculated sound pressure level and vibration response.

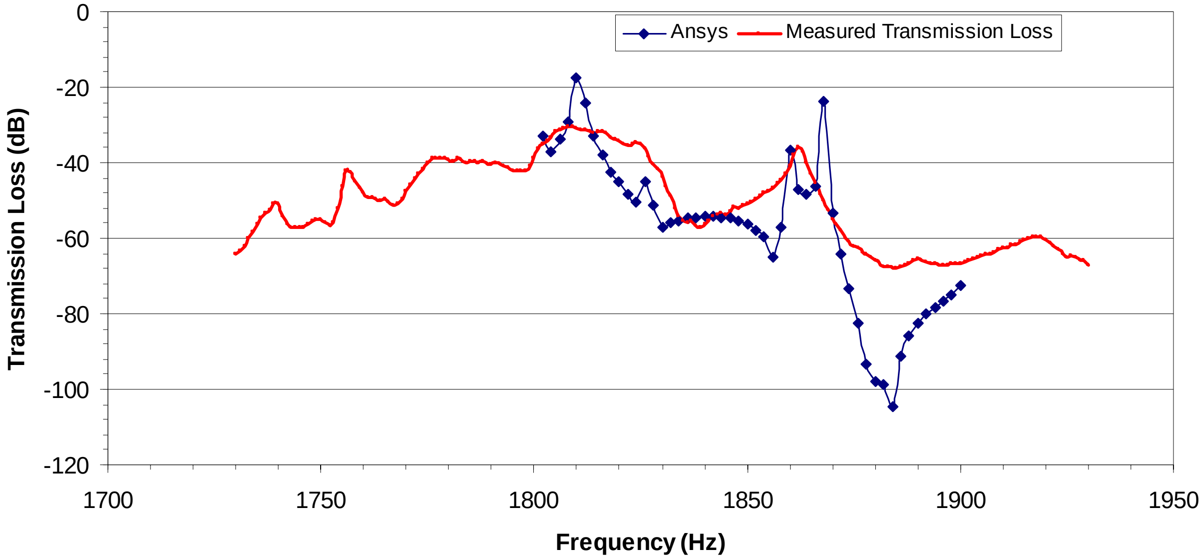

Figure 11. Comparison of the measured and calculated transmission loss.

CONCLUSIONS

An outline for an approach to modeling discrete-frequency features contained in the transmission loss of finite pipes was presented. The approach uses a finite element analysis method to model the coupled internal acoustic and pipe wall vibration fields. The results capture resonance peaks identified in the measured data and provide a good model of peaks in the measured transmission loss.

Future work will focus on performing measurements and analysis on additional pipes, and a more complete analysis of the case presented here given that these results are only preliminary. Pipes of different wall thickness and diameter will be studied. For each pipe, the length and acoustic boundary conditions will be varied. A large library of data should help refine the proposed analysis process.

ACKNOWLEDGEMENTS

The continued support of Fisher Controls International for testing and computational modeling at the R.A. Engel Technical Center is gratefully acknowledged.

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] 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.
[3] 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.
[4] D.G. Karczub, A.C. Fagerlund and F. C. Catron , "Discrete Frequency Characteristics of Pipe-Wall Transmission Loss," *Proc. Inter-Noise 2004*