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A Case Study Comparing 1-D and 3-D Analytical Modeling Methods for Vehicle Intake System Design

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ABSTRACT

There is intense competition among automakers to create ever-quieter vehicles and powertrains. Exterior and interior noise of many vehicles is significantly influenced by noise coming from the engine intake system. In order to address this source, significant effort needs to be expended on the noise design of an engine's intake system. Cost and time constraints tend to make intake designers ignore many complexities in their modeling efforts, specifically in assuming rigid walls, and using 1-D model methods. This paper presents a case study comparing acoustic transmission loss (TL) results for a particular intake system, comparing a 1-D method, the 3-D boundary element analysis (BEA) method, both coupled and uncoupled to the structure, and the new MSC.ACTRAN code by FFT/MSC.Software for analyzing coupled structure/fluid systems. The accuracy of the result is addressed by comparing to measurements of a physical prototype. Time required to set up and execute models with each method are compared, as well as some of the modeling issues encountered.

1. INTRODUCTION

The noise emanating from the inlet of the engine air intake system is one of the more important issues in the noise design of vehicles. Intake system designers typically face challenging constraints of space, weight, cost and timing. This tends to favor the use of simpler analysis methods, particularly in manufacturing environments where the noise design may be considered by decision makers to be a secondary priority. This is the case with low cost vehicles and utility / commercial vehicles. Otherwise, the most complete, 3-D coupled acoustic prediction methods would always be the designer's method of choice. However, the practical constraints of the automotive marketplace dictate that there is still a place for simple, cheap and quick noise estimation tools such as the transfer matrix method[1].

Several excellent papers have recently been written on the topic of analytical intake system NVH design[2-5]. Shaw et al conclude that 1-D methods can be acceptable up to the cutoff frequency of the various ducts in the intake system, or from low frequencies up to the point at which a half wavelength in air equals the largest of the tube width dimensions[2]. However, they note that certain geometries of ducts and chambers can cause response that does not match the 1-D predictions, such as tube eccentricity, and chamber aspect ratio. They further explain how a study of the transmission loss (TL) of an induction system can be used with other information to

predict induction system radiated noise, and to treat problem frequencies in the induction system. This paper focuses only on a study of TL measurement and prediction methods.

Marion and Ye show a useful formulation to include wall compliance effects into their transfer matrix (1-D) formulation[3]. They also discuss the topic of how intake system wall compliance affects the quality of a prediction made with a rigid wall assumption. Modern intake systems are frequently constructed from nylon and other polymer materials that are quite compliant statically and that exhibit wall resonances in the frequency range of interest. The conclusion the present authors draw from this body of work is that the first intake system wall bending resonances must be well above the significant acoustic frequencies of interest for either uncoupled 1-D or 3-D analysis methods to be adequate as intake system design tools.

The authors frequently perform intake design predictions using the transfer matrix (4 pole) method, and also using BEA without taking wall compliance into account. This paper is intended to provide a case study testing the conclusions of this prior work regarding limitations of 1-D and 3-D uncoupled methods.

2. MEASUREMENTS OF FORD GT AIRBOX

Recent work on the Ford GT[*] by the authors allowed both analytical models and prototype parts to be available for this case study. The Ford GT airbox was viewed as an attractive test piece because it has internal chambers with sizable cross section dimensions, is constructed of nylon, and exhibits relatively low wall stiffness. It was expected, when selected, that it would highlight the difficulties of relying on the simpler modeling methods. Figure 1 shows a CAD representation of the Ford GT airbox. The oval tube shaded red is the single outlet that feeds to

Figure 1: Ford GT airbox

the engine supercharger. The two openings to the right and left shaded blue are the dual inlets that draw fresh air from right and left body cavities on the Ford GT. It was decided not to model the entire intake system, but rather just the airbox as a demonstration of the methods.
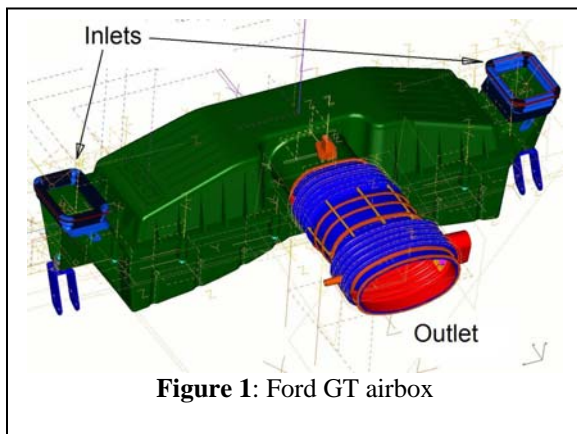
The Ford GT airbox right and left side inlets feed air into a lower right and left chamber. The air in the lower chambers feeds through the symmetric air filters into symmetric upper right and left chambers. These connect to the central outlet tube. The airbox upper chambers can be removed to reveal the paper air filters as can be seen in Figure 2. For the measurement of the airbox, acoustic energy was applied to an extension of the outlet tube, with length L1 = approximately a half meter (See Figure 3).
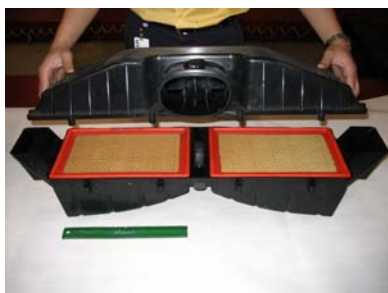

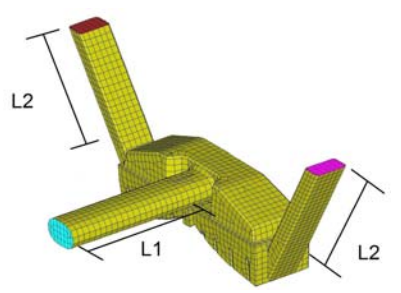**Figure 2**: Air filters in the Ford GT airbox

Figure 3: Airbox inlet and outlet terminations

Transmission loss (TL) across the airbox was calculated from the resulting microphone data as calculated by a ratio of sound energy at the outlet tube to the sound energy at the inlet tube using the two microphone method on each extender tube[6]. Anechoic terminations were constructed for the two inlet tube extensions of length L2 = a half meter, as shown in Figure 3.

Measured data from the Ford GT airbox are now presented. Several sets of data were taken to test various modeling assumptions. The Ford GT airbox is symmetric about the center of the outlet tube. Analytical models might frequently be simplified by assuming symmetry, allowing the analyst to model only half of the airbox and to infer the results for the other half. This assumption was tested, and appears to be acceptable, as can be seen in Figure 4. Next, the effect of the air filters is seen in Figure 5. The flow restriction of the filters appears to smooth out the peaks above 900 Hz. Modeling of the acoustic properties of air filters is clearly an advanced topic that is not addressed in this work. Next, the effect of wall compliance was addressed. Experimentally, this was approximated by burying the airbox in sand to greatly increase the stiffness and effective mass of the walls. This was meant to much more closely approximate the acoustical condition of a rigid wall assumption. These results are seen in Figure 6. The main result seems to be a downward shift in the frequencies of the higher peaks. Finally, it was noted that the dip in the TL curve at about 275Hz is not predicted by the analysis methods being
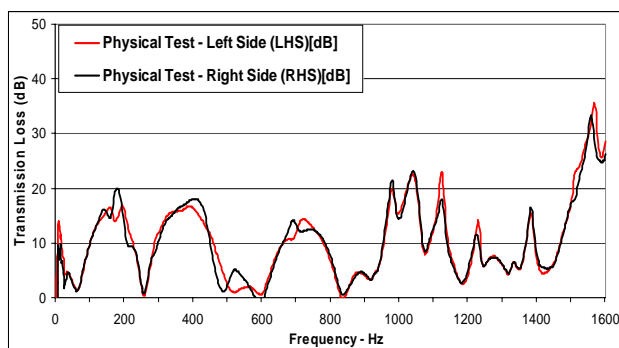

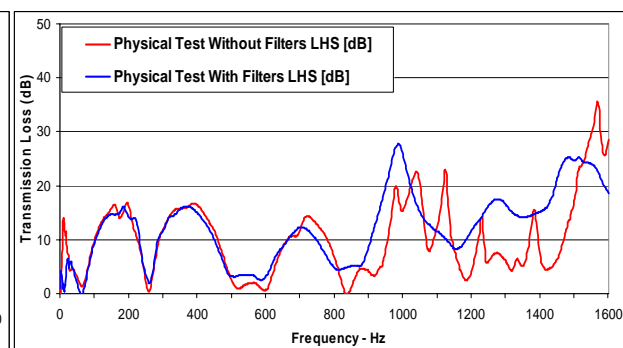**Figure 4:** Measurement testing right – left symmetry

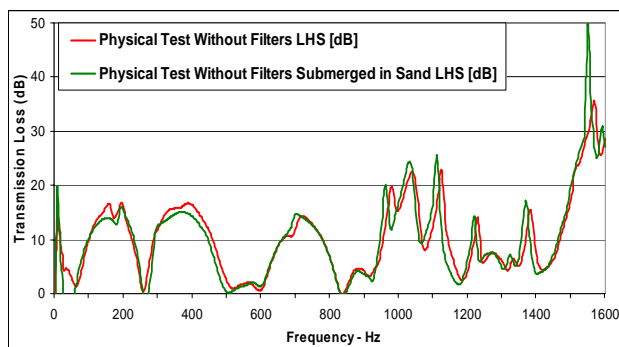**Figure 5:** Measured effect of filters

Figure 6: Measured effect of stiffened walls
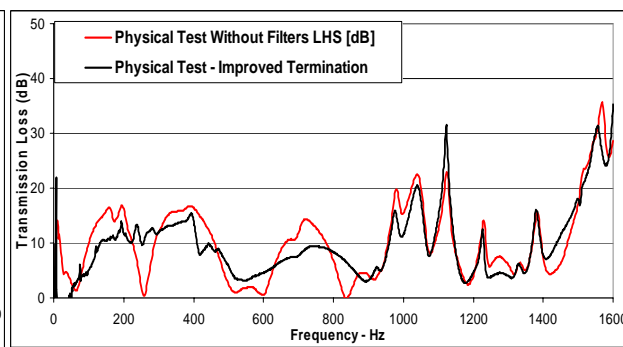

Figure 7: Measured TL after improved anechoic termination

studied. In discussions with Prof. Bolton,[6] he indicated that the two microphone measurement method can have numerical issues in certain circumstances. A significant effort was made to improve the anechoic terminations. They were lengthened from a half meter to approximately two meters. A very effective sound absorption material was installed, and damping treatment was applied to the extender tube walls. Figure 7 displays the resulting measurement. The 275Hz dip in the earlier measurement has been associated with reflections

from the previous "anechoic" termination, and the rise between 600 – 800Hz was associated with a termination tube resonance.

3. NUMERICAL MODELING THE FORD GT AIRBOX TL

Now we move on to a review of analytical results that can be compared to the measured data.

A. 1-D Transfer Matrix Model

With no filter installed, the 1-D model of the left half of the Ford GT airbox can be modeled as an inlet tube with an expansion chamber and the outlet tube. TL was computed according to Munjal's equations[1]. The resulting TL prediction is shown in Figure 8. It is clear from the measured data that acoustical cross modes appear in the Ford GT airbox above 900Hz. Other analytical results (discussed later) show airbox panel modes appearing at 300Hz and above. The
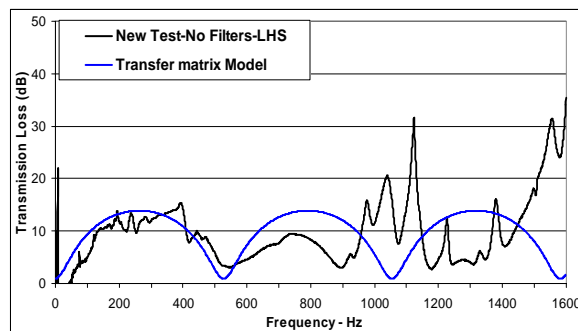
Figure 8: Transfer matrix TL prediction

measured data shows what appears to be modal activity as low as 100Hz. Yet the basic acoustic response as predicted by the transfer matrix approach seems quite good to 700Hz. Once transfer matrix equations are ready to use[†], this model takes under one hour to set up and run, with later iterations taking even less time. This method can be enhanced with different element types into a very quick and functional tool.

B. 3-D BEA Model

SYSNOISE[‡] was used to calculate the Ford GT airbox TL with a variety of assumptions. The simplest version was to run the acoustic model only of the airbox assuming rigid walls and no damping. This is shown in the dark blue curve of Figure 9. This model method takes a moderate (approx 2 man-days) amount of time and effort when starting from CAD geometric data. This result seems quite good even though the cross modes above 900Hz clearly have unrealistic sharpness. A slightly better model, with not much additional effort, was to use a complex speed of sound to include damping in the air, shown in the light blue curve of Figure 9. Finally, the effects of side wall compliance in the airbox was studied using the BEA method. An ABAQUS FEA model was generated from CAD data of the
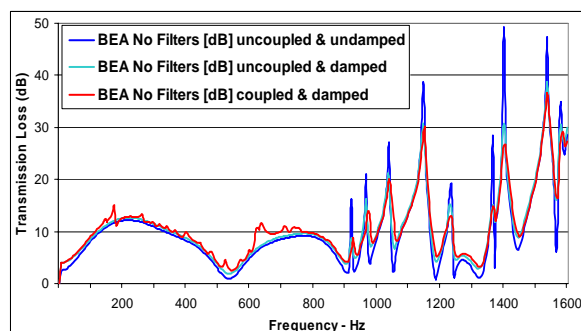
Figure 9: BEA model results

airbox. The structural modes of the airbox were computed and results output to the BEA software. SYSNOISE is able to make use of this information (though no structural damping is included) as a frequency dependent compliance at the boundary. The resulting BEA prediction is the red curve shown in Figure 9. This effort approximately doubled the time required to run a prediction[§]. The side wall resonances have a visible effect on the TL prediction from 150Hz up. Aside from the bump at about 630Hz, no

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[†] The authors have developed an NI/LabView based software tool to perform transfer matrix analyses.

[‡] SYSNOISE is a popular BEA software. It has recently been replaced by Virtual.Lab Acoustics. For more information see www.lmsintl.com.

[§] Since further model geometric data has to be read into an FEA solver and a separate FEA analysis has to be run on the structure, this procedure is likely to more than double the analysis time.
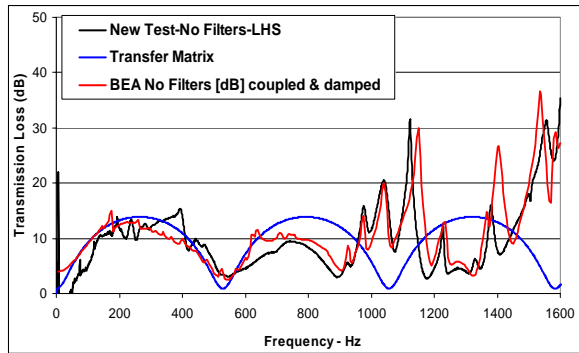
Figure 10: Transfer matrix and BEA results compared to measured data.

significantly new information seems to be gained from this increase in complexity and cost. The transfer matrix and BEA methods are compared to the measured result in Figure 10.

C. 3-D Impedance Model

Another 3-D method was investigated. A new formulation for general 3-D wave transmission modeling has recently come to the attention of the authors. MSC.Software Corp. (MSC) and Free Field Technologies (FFT) have available an acoustic analysis package called MSC.ACTRAN[**].

This is an entirely different formulation from traditional FEA/BEA, even though it is a "finite element" method which functions similarly at the user interface. The internal formulation, however, is derived from a calculation of impedance matrices through a solution of the general wave equations in 3-D. The solution is formulated to use complex math, meaning that losses (damping), reflections at impedance discontinuities, acoustic absorption and structural coupling are included everywhere automatically. Different from the multi-step process described in section B, above, where to include structural coupling the undamped structural modal data is exported to the BEA model, MSC.ACTRAN solves a mixed structure/fluid discretized model in a single step, including damping everywhere. Fluids are simply treated with solid elements and a fluid material type.
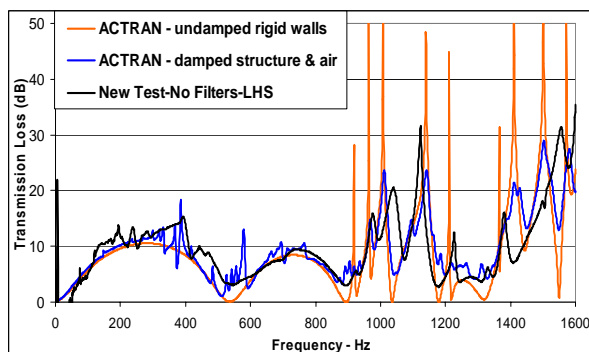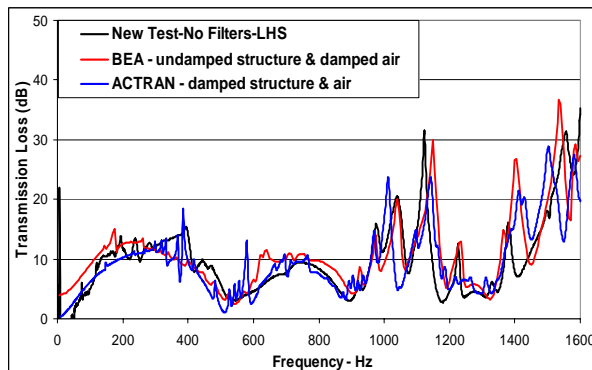

Figure 11: MSC.ACTRAN results

This method was exercised first using the rigid boundary assumption. The TL of only the air spaces in the Ford GT airbox with anechoic terminations were evaluated. The results are shown in the red curve of Figure 11. To set up and run this model takes effectively the same effort and time as an uncoupled BEA analysis, starting from CAD data of the airbox.

The results of the fully coupled model, including damped structural and acoustic response, are shown in the blue line of Figure 11. This model predicts modal activity beginning below 100Hz. The cross modes above 900Hz are predicted. Finally, the BEA coupled results, and the full MSC.ACTRAN results are compared to measurements in Figure 12. Both coupled BEA and MSC.ACTRAN results are good over the entire frequency range, with peaks only slightly more pronounced than the measurement. A further result of the MSC.ACTRAN analysis was production of visualizations of the coupled airbox wall deflections at any desired frequency.

Figure 12: Comparison of 3-D methods to measured result

[**] For information about this commercially available software, see www.mscsoftware.com.

The following three examples are computed wall responses due to the acoustic loading from a speaker at the outlet tube of the Ford GT airbox. See Figure 13.
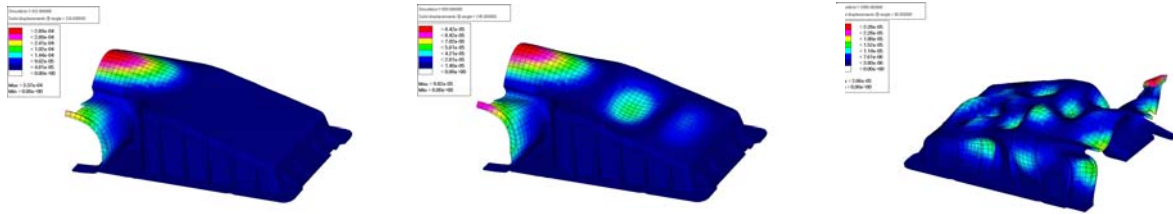

Figure 13: Some Ford GT top cover MSC.ACTRAN calculated wall dynamic deflection shapes at 433Hz, 509Hz and 2000Hz

5. CONCLUSIONS

Various methods of analyzing the TL of an automotive intake system were investigated. The limitations of 1-D and simpler 3-D methods were investigated. Prior work[2-5] indicated that transfer matrix methods would give good TL predictions in the frequency range below where acoustic cross modes occur. This case study confirmed that conclusion[††]. Prior studies[2-3] indicated that 1-D and 3-D uncoupled methods that assume rigid walls, would give good results in the frequency range below the first structural modes of the wall panels. This work confirmed that conclusion as well. However, for the frequency region above the start of sub-panel modes, up to the start of acoustic cross-modes, both 1-D and 3-D uncoupled analysis methods gave acceptable results for design purposes. Thus, in a time and cost constrained design environment, the simpler methods are seen to give acceptable results in the case that was studied.

ACKNOWLEDGEMENTS

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[††] Subject to comments made about some irregular geometries degrading the 1-D calculations.