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Evaluating Damping Treatments using Radiated Sound Power normalized by Input Mechanical Power

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1. INTRODUCTION

In the commercial vehicle market today, damping material is widely used in conjunction with acoustic absorbers as a means to reduce interior noise levels. Typical experimental methods for designing damping treatment for a vehicle subsystem include impact hammer tests, power injection method, and laser vibrometry. The ability to optimize the damping treatment design for a given vehicle subsystem helps to minimize the amount of full-vehicle testing necessary to achieve interior noise level targets. The test method described in this paper proposes a simple way to evaluate damping treatment performance quickly, with little instrumentation.

A vehicle subsystem, such as an engine tunnel, is mounted and sealed in a transmission loss window between two reverberation rooms. On the appropriate side of the subsystem, a broadband steady state mechanical vibration is introduced using a shaker and the input power is measured with an impedance head. The resulting radiated sound power is measured with a rotating microphone in the diffuse sound field on the opposite side of the subsystem. Both the impedance head and diffuse field microphone signals are processed with a narrowband FFT analyzer. Noise control treatments are evaluated and compared by normalizing the radiated sound power FFT by the corresponding input mechanical power FFT. Damping treatments can be installed and characterized quickly and easily, allowing for rapid optimization of radiated noise control solutions.

This paper will discuss the test method that was developed, the equipment used, and a general overview of test results.

2. METHODOLOGY

All testing was performed at Aearo Technologies LLC, a 3M company, in Indianapolis, IN, inside the Acoustic Technology Center.

A. Subsystem Installation

A steel engine tunnel was mounted and sealed in an ASTM E90 transmission loss window between two reverberation rooms. The engine side of the tunnel was oriented towards the sound source room, leaving the cab side facing the receiving room. Due to the irregular shape of the engine tunnel, a mounting fixture was built to replicate a truck frontwall and floor so that there were no leak paths between the reverberation rooms. The fixture was constructed out of a lumber frame, filled with sound insulation material, and covered with two layers of cement board in order to achieve a high level of transmission loss. The tunnel was rigidly connected to the fixture and the fixture rigidly connected to the transmission loss window. Figure 1 shows the tunnel mounted in the transmission loss window from the source room and receiving room.

(a) (b)

Figure 1: (a) Mounted engine tunnel from source room. (b) Mounted engine tunnel from receiving room.

B. Test Equipment

Testing was performed with the engine tunnel mounted inside the test window of an ASTM E90 test suite. This consists of two reverberation rooms connected by a 48” x 48” window. The test suite is fully equipped to perform ASTM E90 per the standard and meets all the qualifications associated with it. On the engine side of the engine tunnel, a medium-sized broadband shaker was installed and attached to the tunnel via a rigid stinger and impedance head (Figure 2).

Figure 2: Shaker mounted to tunnel via rigid stinger and impedance head

The shaker is mounted on a cinder block and connected to the windowsill through vibration isolation pads to prevent additional energy from entering the engine tunnel. The impedance head will measure the input force and acceleration the shaker injects into the engine tunnel. These measurements will allow the calculation of mechanical power induced by the shaker. Eight reference accelerometers are placed at locations on the engine tunnel where large deflections are expected in order to aid in the characterization of the vibration response. Although, not used directly used in the calculations for the final results, these accelerometers played an important role in the design of damping treatments that were tested. In the middle of the receiving room on the cab side of the engine tunnel, there is a rotating boom diffuse field microphone (Figure 3).

Figure 3: Rotating boom diffuse field microphone in receiving reverberation room

This microphone is used to measure the average diffuse sound pressure level radiated by the cab side of the engine tunnel into the receiving room. The radiated sound power can then be calculated using this measurement and known parameters about the receiving reverberation room.

All measurement signals were connected to a data acquisition system and processed to create FFTs. The FFTs were then used to calculate the radiated sound power normalized by the input mechanical power.

C. Test Procedure

The test procedure begins with the calibration of the diffuse field microphone. This is done within the data acquisition software and is only performed at the beginning of each test day, unless the environmental conditions inside the test suite have changed dramatically. After the microphone has been calibrated the rotating boom is turned on at a rotational speed of 1 revolution per 32 seconds. The test suite is vacated and doors and ventilation shafts are closed. A broadband signal from 16 to 12800 Hz is injected into the engine tunnel via the shaker and allowed to reach steady state for a few seconds. Using the data acquisition software, a 64 second measurement of the impedance head, diffuse field microphone, and 8 accelerometers is taken. The measurement length of 64 seconds allows the microphone to make two full revolutions in order to get a time and spatial average of the diffuse sound pressure level in the receiving room. After the measurement is completed, an ASTM E90 test is performed to evaluate transmission loss and obtain the receiving room equivalent absorption, *A*. The equivalent absorption will be used in calculating the radiated sound power from the tunnel into the receiving room.

After both tests have been completed, the results can be obtained with simple post processing (outlined in section D. Analysis). The next damping treatment to be evaluated can then be installed on the engine tunnel and the measurements repeated.

D. Analysis

The measurement signals of the impedance head, diffuse field microphone, and 8 accelerometers were processed using 16 Hz to 12800 Hz FFTs with 4 Hz resolution. 766 averages were taken over the 64 second duration of the measurements. The radiated sound power of the engine tunnel was calculated based on the diffuse field sound pressure level, p, using Equation 1:

$$W_{rad} = \frac{p^2}{\rho c} \cdot \frac{A}{4}, \tag{1}$$

where $A = \bar{\alpha} \cdot S$, the equivalent absorption of the receiving reverberation room. The mechanical input power injected into the engine tunnel is calculated using the time averaged force and velocity data collected from the impedance head as in Equation 2:

$$W_{in} = F \cdot v. \tag{2}$$

Finally, the radiated sound power is normalized by the mechanical input power with Equation 3:

$$W_{norm} = {}^{W_{out}}/_{W_{in}} \tag{3}$$

It is the normalized radiated sound power that is used to compare the different damping treatments applied to the engine tunnel.

3. RESULTS

The results produced from this test method are used to compare different damping treatments applied to the same vehicle subsystem. Multiple tests of a given damping treatment resulted in less than 1% variation. Removal and reinstallation of identical treatments had an effect of less than 2% on the results, giving reassurance of the repeatability of the test method. Comparison to similar test methods was outside of the scope of this paper.

This test method is very useful when the damping material and geometry have not yet been defined. The ability to quickly evaluate and modify geometries helps to downselect treatments that will ultimately be tested on a full vehicle, where test time and iterations may be limited. The results shown in Figure 4 are the normalized radiated sound power from the initial incumbent damping design and the optimized lower weight, lower cost, and higher performing design.

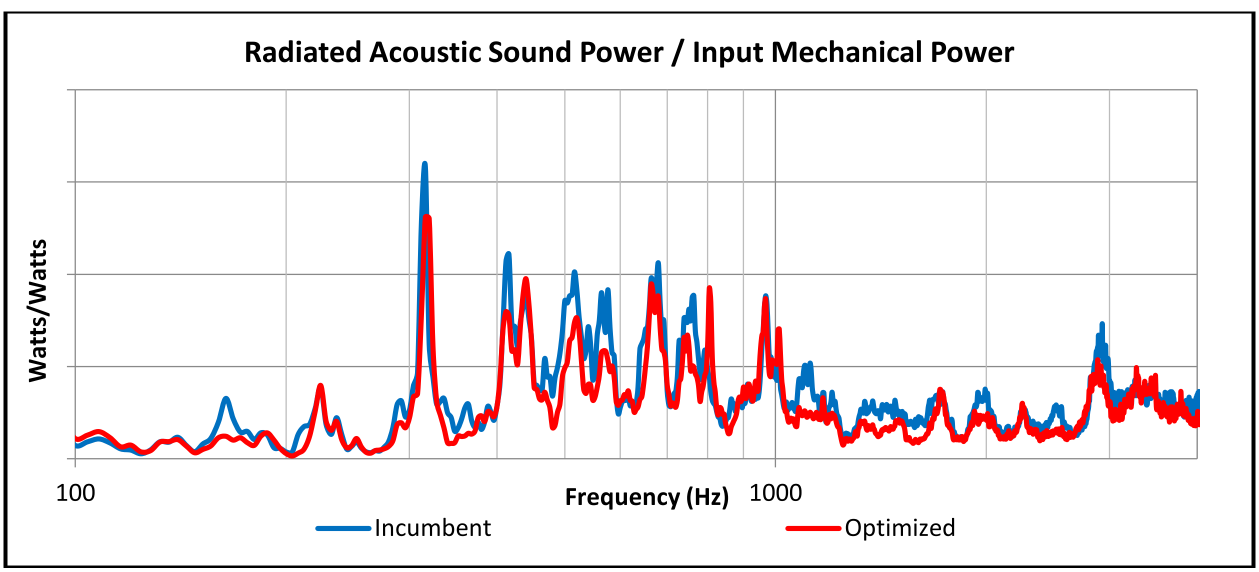

Figure 4: Normalized Radiated Acoustic Sound Power of Damping Optimization

The results shown in Figure 5 are the normalized radiated sound power from 3 different damping treatments and the Bare (untreated) case (Narrow Band and 1/3 Octave Level).

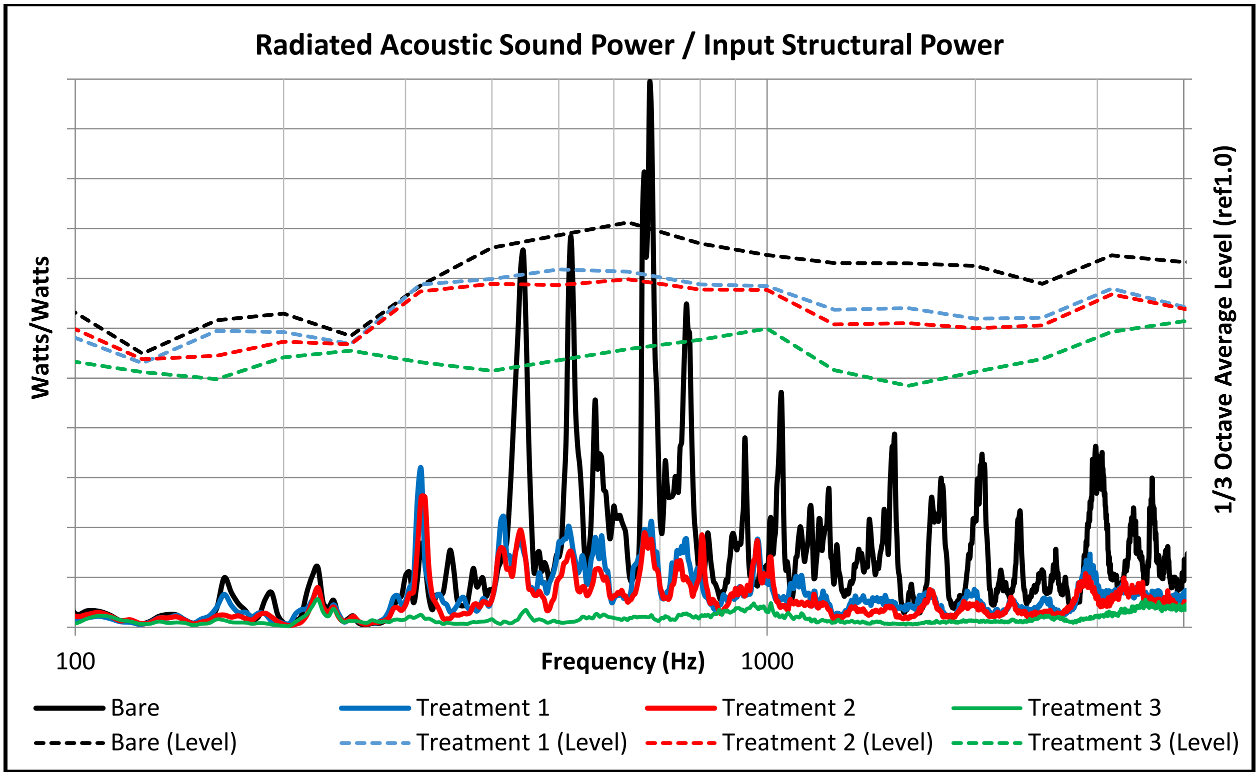

Figure 5: Normalized Radiated Acoustic Sound Power of Damping Treatments

Treatment 1 represents the target performance. Treatment 2 is the result of many optimization iterations. Treatment 3 represents the damping performance limit, while Bare shows the response of the tunnel if left untreated. The transmission loss performance is also measured in order to characterize the equivalent absorption of the receiving room. This provides a secondary data source by which the damping treatments can be compared. Figure 6 shows the ASTM E90 transmission loss performance of the same damping treatments featured in Figure 5.

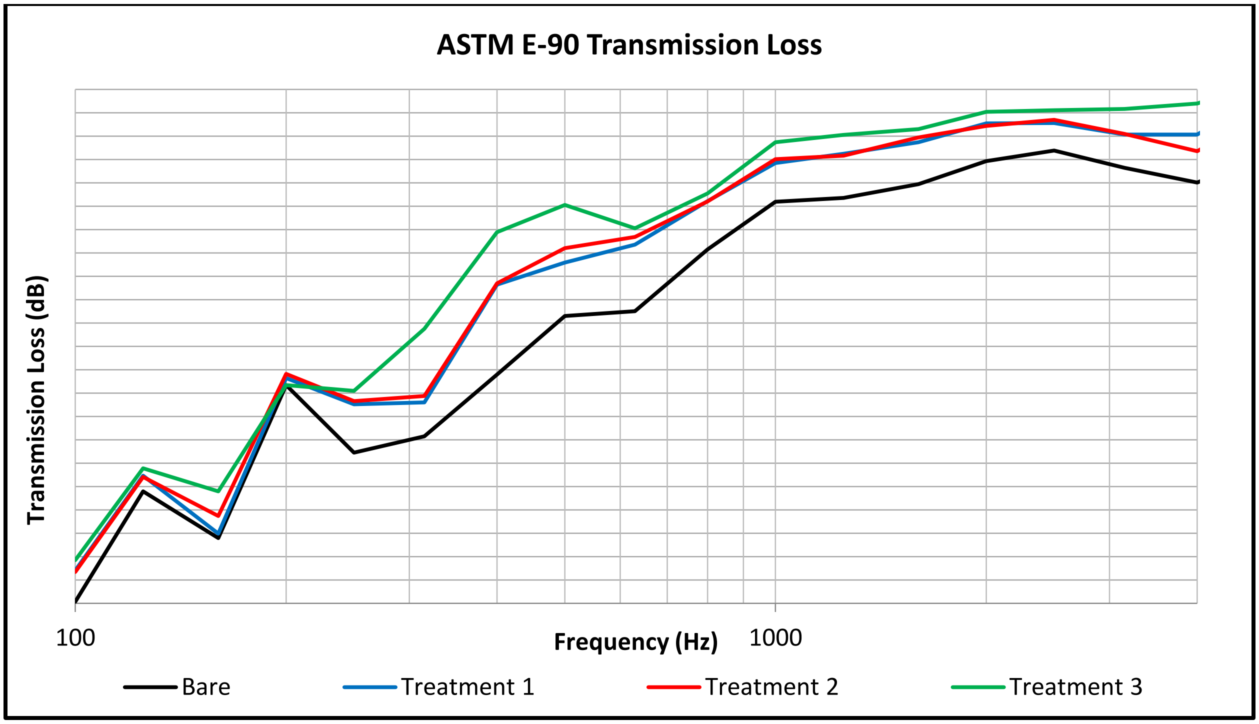

Figure 6: ASTM E90 Transmission Loss of Damping Treatments

The results of the transmission loss test show the same performance ranking as the radiated sound power test. This gives confidence that the radiated sound power test is an effective method for evaluating damping treatments on vehicle subsystems. Although treatment 2 has less mass than treatment 1, the transmission loss is higher due to the improvement in vibration damping.

4. CONCLUSION

The test method outlined in this paper is an effective way to quickly evaluate and compare damping treatments on vehicle subsystems. Due to the easily accessible subsystem and the minimal amount of test equipment, damping treatments can be exchanged easily without requiring the disassembly of the test setup. When testing a full vehicle, access to certain subsystems can be extremely limited and may require a fair amount of time to disassemble surrounding components to change a noise treatment. By doing a large amount of optimization on the subsystem outside of the full vehicle, final validation test time can be greatly reduced. Another advantage of testing the subsystem outside of the full vehicle is the increase in repeatability and reduction in test variation. This is because, the inputs of the system are totally controlled and the meaningful outputs can easily be measured. Measuring a given damping treatment, multiple times resulted in less than 1% variation. This speaks well to the high level of precision of the test. Removal and reinstallation of identical treatments had an effect of less than 2% on the results, giving reassurance of the repeatability of the test method.

Future work involving the development of this test method will include comparisons to similar test methods, such as impact hammer tests, power injection method, and laser vibrometry. This will add a level of confidence that the results from this test are indeed valid and should be used in the optimization of damping treatments. The ability to quantify the airborne noise that is radiated from a subsystem due to structural vibration is an interesting way of determining how effective damping treatments will be.