Measurement of Damping In Structures by the Power Input Method

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INTRODUCTION

Damping refers to the extraction of mechanical energy from a vibrating system usually by conversion of this energy into heat. Damping serves to control the steady-state resonant response and to attenuate traveling waves in the structure. There are two types of damping: material damping and system damping. Material damping is the damping inherent in the material while system or structural damping includes the damping at the supports, boundaries, joints, interfaces, etc. in addition to material damping.

Since utilizing damping materials is the most common way to reduce resonance responses, accurate measurements of damping are crucial to the proper design, optimization, and modeling of systems from a vibration reduction standpoint. Damping or loss facor measurements are rarely straightforward due to the complexity of the dynamic interaction of system joints, trim, and geometry. Furthermore, a variety of nomenclature exists to denote damping. These include: damping ratio (ζ), log decrement (δ), loss factor (η), loss angle (ϕ), tanδ, specific damping capacity (ψ), quality factor (Q), etc. Equal to the number of different descriptors for damping levels, there are different test methods.

The various damping test methods can broadly be classified into three groups: a) frequency-domain modal analysis curve-fitting methods, b) time domain decay-rate methods, and c) other methods based on energy and wave propagation [1-3]. Each method has its own set of advantages and drawbacks. One method, the power input method (PIM) from group (c) above, is a powerful method for obtaining frequency-averaged loss factors of structures under steady state vibration.

The PIM is based on a comparison of the dissipated energy of a system to its maximum strain energy under steady state vibration. Some errors may be introduced through the measurement technique, but the PIM is fundamentally unbiased at the natural frequencies of well-defined modes or when the loss factors are frequency band averaged over many modes. These "frequency-averaged" damping values are widely used in the automotive industry in vehicle computer models based on Finite Element Method (FEM) and Statistical Energy Analysis (SEA).

This paper presents a practical application of the PIM to measure a vehicle side glass damping quickly and accurately. The loss factors between a tempered glass and a laminated glass are compared. The laminated glass is made of a layer of polyvinyl butyral (PVB) bonded between two sheets of glass under heat and pressure. The polyvinyl butyral provides damping to reduce vibrations in the glass, resulting in a significant reduction in both road and wind noise in a vehicle. The Errors introduced through mass loading are reduced by utilizing a laser vibrometer to measure the vibration response instead of traditional piezoelectric acelerometers.

POWER INPUT METHOD (PIM)

The loss factors of a structural system can be defined as

$$\eta(\omega) = \frac{\Delta E}{E_{SE}}, \qquad (1)$$

where E_{SE} is the strain energy, ΔE is the energy dissipated from damping, and η is the damping loss factor in the frequency band ω considered. Assuming a stationary input energy at a fixed location, ΔE can be replaced with E_{in} because the input energy must be equal to the dissipated energy under steady state conditions.

Unfortunately, neither E_{SE} nor E_{in} can be measured directly. The input energy can be calculated with a simultaneous measurement of the force and velocity at the point of energy input. The numerator of Eq. (1) can then be computed as

$$E_{in} = \frac{1}{2\omega}\mathrm{Re}[h_{ff}(\omega)]G_{ff}(\omega), \qquad (2)$$

where, h_{ff} is the driving point mobility function, and G_{ff} the power spectral density of the input force. Obtaining an estimate for E_{SE} requires making a few assumptions. First, since the strain energy cannot be calculated directly from force and velocity measurements, it must be replaced with twice the kinetic energy – which holds true at the natural frequencies of the system. Second, the system being measured must be approximated by a summation as opposed to a volume integral when taking measurements. This allows the kinetic energy to evaluated by

$$E_{KE} = \frac{1}{2}\sum_{i=1}^{N} m_i G_{ii}(\omega), \qquad (3)$$

where, E_{KE} is the system kinetic energy, N is the number of measurement locations, m_i is the mass of the discrete portion of the system, and G_{ii} is the power spectral density of the velocity response at each measurement location. Finally, assuming the system is linear, allows

$$\left| h_{if}(\omega) \right|^2 = \frac{G_{ii}(\omega)}{G_{ff}(\omega)}, \quad (4)$$

where, h_{if} is the transfer mobility function. With the given assumptions, all measurement points uniformly spaced throughout the system, and equal mass portions, equation (1) can be approximated by combining equations (2), (3), and (4), as:

$$\eta(\omega) = \frac{\mathrm{Re}[h_{ff}(\omega)]}{\omega m \sum_{i=1}^{N} | h_{if}(\omega) |^2}. \quad (5)$$

To obtain accurate loss factors estimations, it is essential to have highly accurate measurements of the driving point FRF (h_{ff}); otherwise, large errors could be introduced.

The PIM allows finding internal loss factors with fewer steps than with experimental SEA techniques. The loss factors can then be utilized in SEA models for noise prediction and parametric studies.

LASER VIBROMETER

Laser vibrometers use optical interferometry to measure surface velocities ranging from 0 to 125 mm/s/V and from 0 to 30 MHz depending on equipment arrangements. For the current testing, a Polytec neon-helium laser (OFV-302) and scanning head (OFV-040) were used. The vibrating surface orthogonal to the laser causes the frequency of a laser beam to be shifted due to the Doppler effect [4]. The shift in signal beam frequency is related to the velocity of the vibrating surface and the wavelength (λ) of the laser ($\lambda \cong$ 6.32E-7 m for helium-neon lasers) through the equation

$$f_S = \frac{2v}{\lambda}, \quad (6)$$

where f_s is the frequency shift of the beam and v is the vibrating surface velocity. The signal beam and a known frequency reference beam are then combined to create an interference signal. This signal contains the velocity of the measured system .

The primary advantage of using a laser vibrometer is the non-contact nature of the transducer, which eliminates mass loading of the structure due to response measurement transducers. For methods employing piezoelectric transducers, additional steps and mathematical equations have been developed to alleviate mass loading measurement discrepancies [5-6]. Unfortunately, these techniques take time to implement and are prone to noise induced errors. Also, given the proper system, laser measurements can be taken more quickly than with accelerometers because of the reduction in setup time and time spent moving accelerometers between different measurement locations.

EXPERIMENTAL SETUP AND PROCEDURE

For damping measurements of a commercial vehicle door glass assembly, a fully assembled door with the window installed and completely sealed in the weather stripping was suspended with bungee cords as shown in Fiure 1. Reflective materials must be applied on the glass to facilitate proper reflection of the laser beam. Reflective tape (3-M 510-10X) sections of 3mm square were used to create a 4 cm grid across the approximately 50x70 cm visible portion of the vehicle side glass. This grid spacing was chosen to create enough points on the surface to reduce the errors introduced through approximating the surface integral with a summation. As the number of portions increase, the estimates of loss factor measurements asymptotically approach a constant value. The reflective grid used in window panel is also seen in Figure 1

A shaker was attached to the window on the opposite side as the reflective tape grid through a stinger-force transducer combination as illustrated in Figure 2. The driving point was chosen away from the symmetric axis and the edges to excite as many modes as possible. For the estimation of the loss factors, the glass was excited using burst random excitation from 100 Hz to 8 kHz. Figure 3 shows an example of a typical measured FRF at the driving point.

RESULTS

Figure 4 shows the loss factor results. These were extracted using the PIM on data collected from two window specimens. It can be seen that the laminated PVB glass has much higher damping than the regular tempered glass over the entire frequency range. At some frequencies, the damping of PVB glass is more than twice that of the regular glass. The high loss factor values at the lower frequencies (approximately 250 Hz octave band are consistent with the driving point FRF figure. Dominant bending modes (also centered around 250 Hz) are more influenced by the internal system damping than less prominent modes. Also, the loss factors found through the PIM maintain the same trend found in most loss factor estimates, that is, decrease in damping with increasing frequency.

CONCLUSIONS

An experimental technique for determining the internal loss factors (damping) of structures was investigated. The power input method (PIM) coupled with the use of a laser vibrometer allows for quick, and accurate measurements of loss factors of flat panel type structures. The effectiveness of this method has been shown through a case study involving laminated and tempered vehicle side glass windows.

REFERENCES

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Figure Captions

Figure 1: Hanging door assembly with reflective grid in place.

Figure 2: Schematic of the experimental setup.

Figure 3: Driving Point FRF measured with the tempered glass.

Figure 4: Frequency-averaged loss factor comparison results.

Figure (1):

Figure 2:

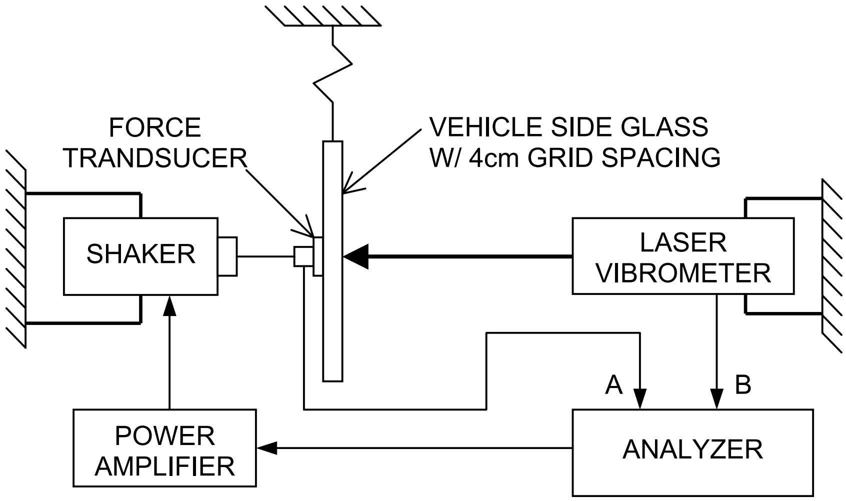
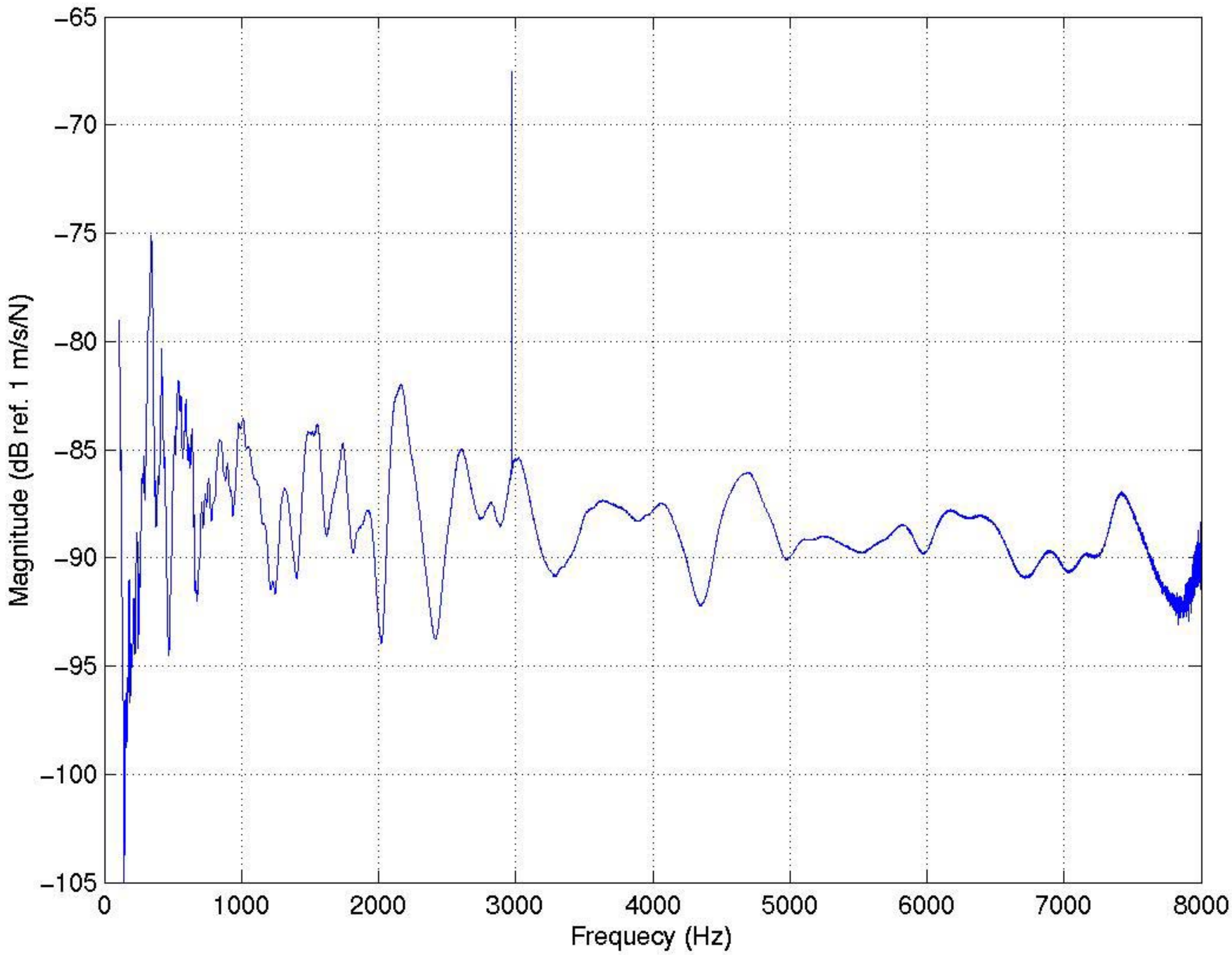
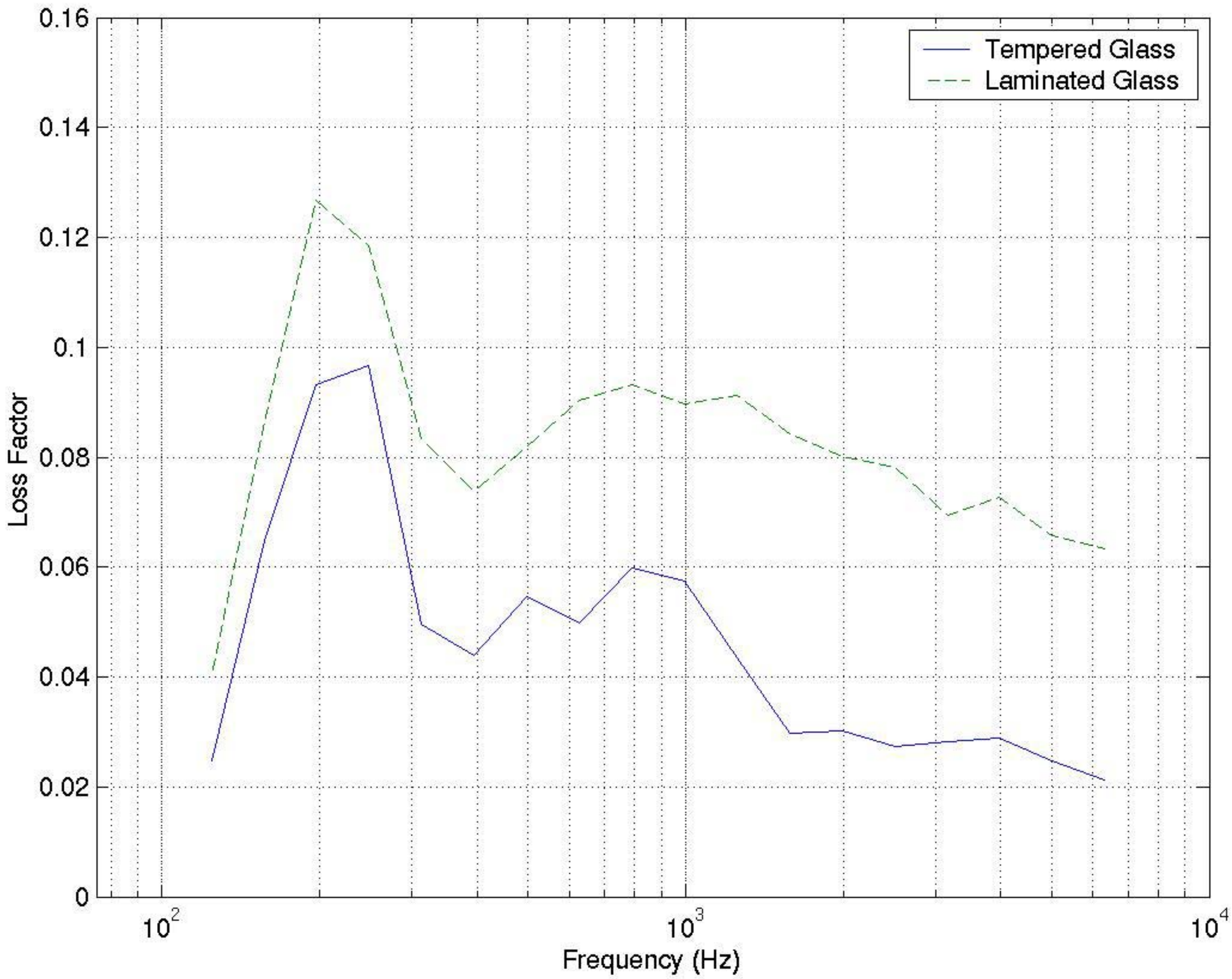

FORCE
TRANDSUCER
VEHICLE SIDE GLASS
W/ 4cm GRID SPACING
SHAKER
LASER
VIBROMETER
A
B
POWER
AMPLIFIER
ANALYZER

Figure 3:



-65
-70
-75
-80
-85
-90
-95
-100
-105
Magnitude (dB ref. 1 m/s/N)
0
1000
2000
3000
4000
5000
6000
7000
8000
Frequecy (Hz)

Figure 4:


Tempered Glass
Laminated Glass
0.16
0.14
0.12
0.1
0.08
0.06
0.04
0.02
0
Loss Factor
10^2
10^3
10^4
Frequency (Hz)