

Improving STL Performance of Automotive Carpets with Multi-layering and Effective Decoupling

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ABSTRACT

Automotive floor carpet serves the purpose of insulating airborne noises like road-tire noise, transmission noise, fuel pump noise etc. Most commonly used automotive floor carpet structure is- molded sound barrier (PE, vinyl etc.) decoupled from the floor pan with an absorber such as felt. With increasing customer expectations and fuel efficiency requirements, the NVH requirements are increasing as well. The only possible way of increasing acoustic performance (Specifically, Sound Transmission Loss, STL) in the mentioned carpet structure is to increase the barrier material. This solution, however, comes at a great weight penalty.

Theoretically, increasing the number of decoupled barrier layers greatly enhances the STL performance of an acoustic packaging for same weight. In practice, however, this solution presents problems like- ineffectiveness at lower frequencies, sudden dip in performance at modal frequencies. Also, practical constraints in achieving ideal decoupling limit the advantages of the technique.

This paper aims at exploring the effects of multi-layering of barrier material on STL performance of automotive carpets. Another focus area of this paper is the optimization of several multi-layered structures to make it effective for automotive floor carpet application in frequency band of 500Hz to 8000 Hz. Also, methods of improving decoupling in automotive carpet to improve STL performance have been studied.

An approximate mathematical model has been developed in the paper to explain the STL behavior of automotive carpets. The model has been validated by comparing results with actual test results at ARAI. Based on the model, effect of changes in parameters of backing material, absorber/decoupler, type of bonding between backing and absorber has been discussed. In addition, guidelines for a part designer have been developed to decide the kind of carpet structure to be used for given NVH requirements.

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INTRODUCTION

The floor pan of a modern monocoque car is directly connected to the dash panel of the vehicle and mechanisms like transmission suspensions and fuel pumps are present underneath it. Further, much of the airborne noise like road-tire and underbody wind noise leak into cabin through floor pan. SEA (Statistical Energy Analysis) models suggest that more than half the acoustic energy is transferred inside the cabin through floor pan [1]. Though radical solutions - like acoustically decoupled underbody for a motor vehicle patented by VW [2] - are being explored, there is much scope of improvement and optimization in existing automotive carpet solutions.

Although, Sound Absorption Coefficient (SAC) and Sound Transmission Loss (STL) both characterize the NVH performance of a carpet, this article focuses on STL performance only. Techniques like multi-layering of barrier layer and introduction of air-gap are known to increase the overall STL performance of a carpet. However, increase is observed at higher frequencies only while in lower frequency band STL slightly deteriorates.

Each vehicle has different inside-cabin NVH requirements and different noise pattern. Hence, the ideal acoustic solution for each vehicle is unique and the 'best' solution for a given vehicle cannot be generalized. Also, determining the type of carpet to be used in a

vehicle at the design stage, so that minimum iterations are required in final vehicle, is of more interest to a NVH part designer than improving NVH level of already existing vehicle.

Advanced computational tools which use FEM and SEA, are available for predicting the STL behavior of a given composition of carpet. For a given STL requirements, however, optimization of the carpet composition using these tools is a highly iterative process. Since there are a number of variables involved in fixing the composition of a carpet, an understanding of effect of each variable on STL performance, separately and in combination, is required. A mathematical model incorporating all the major variables, have been presented in the following section of the paper.

APPROXIMATE MATHEMATICAL MODEL OF AUTOMOTIVE CARPET

The typical automotive floor carpet has a multiple layer construction. The top layer facing the cabin side is mainly for aesthetic appeal and is a non-woven/woven fabric of synthetic material (PET/PP/PA). Underneath the top layer a backing layer of one or more materials (PE/EPDM/EVA/PP) is present. The top two layers are separated from steel floor pan panel by a layer of PU foam or felt. All the layers are usually thermoformed, molded and cut together to fit the floor pan. Many variations of this typical structure are possible. The foam/felt may not be bonded to the backing layer and maybe provided in selected areas only. Similarly, GSM of backing may be increased in selected areas by providing an additional backing layer of EVA or EPDM.

A simple carpet with following construction was used as a base for developing the mathematical model (Figure 1) -

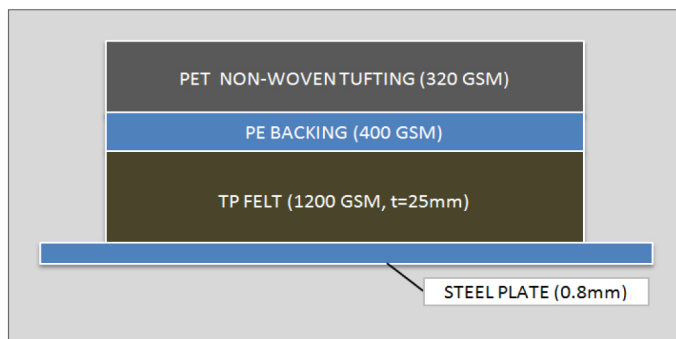


Figure 1. Construction of carpet sample-1

Top layer is a non-woven PET fabric of 320 GSM with PE of 400 GSM thermoformed with it. TP Felt of 1200 GSM (t=25mm) is also bonded to the PE backing during thermoforming.

STL performance of the given structure as per ISO 140-3 (test conducted at ARAI) in frequency range of 200 Hz to 6300 Hz is shown in Figure 2 -

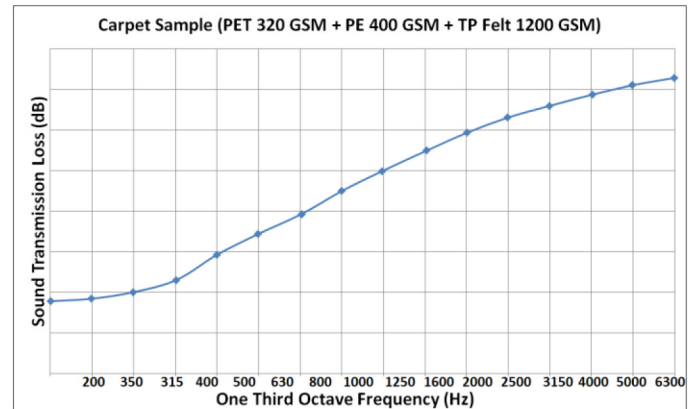


Figure 2. Actual STL Performance of carpet sample-1

(Note: Measurements of STL performance of all the samples mentioned in paper have been done at ARAI, Pune. Also, for all the samples mentioned in the paper, STL has been measured with a backing of 0.8mm steel plate)

Model 1

In this model the carpet was analyzed as a 'simply added mass' to the 0.8mm steel panel.

Mass Law

Sound transmission loss of a single thin panel below its critical frequency may be expressed as a function of its mass by equation commonly referred to as the 'Mass Law' (Sharp, 1973) [3][4]

$$STL, R = 20 \log \left(\frac{\pi \cdot m \cdot f}{\rho \cdot c} \right) - 5 \quad (1)$$

Where, m is the mass of the panel per unit area, f is the frequency in Hz, ρ is the density of air in given conditions and c is the speed of sound in given conditions.

On substituting values at room temperature, the equation assumes a simple form as below -

$$STL, R = 20 \log(m \cdot f) - 47 \quad (2)$$

For 0.8mm steel panel, mass per unit area, m_s

$$= 7800 \times 0.8 \times 10^{-3} = 6.24 \text{ Kg m}^{-2}$$

For the given carpet structure, mass per unit area, m_c

$$= 1.92 \text{ Kg m}^{-2}$$

For Model 1, Total mass per unit area, $m_t = m_s + m_c$

$$= 8.16 \text{ Kg m}^{-2}$$

Theoretically expected STL performance of carpet sample-1 according to this model w.r.t. the actual STL performance is shown below in Figure 3 -

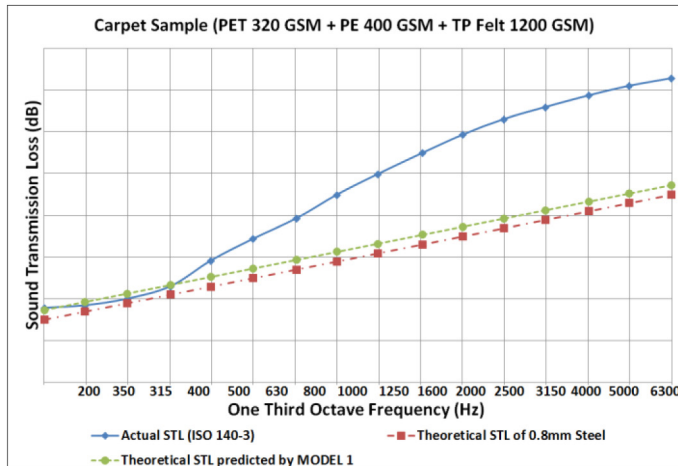


Figure 3. Actual vs. theoretical (Model 1) STL performance of carpet sample-1

Evidently, Model 1 is inadequate to explain the actual observed trend of STL vs. frequency. Therefore, automotive carpet should not be treated as a simply added mass to body panel for NVH study.

Model 2

In this model, carpet has been analyzed as a double leaf barrier - body panel and barrier layer (backing with fabric) being two different barriers separated by an air cavity (Figure 4).

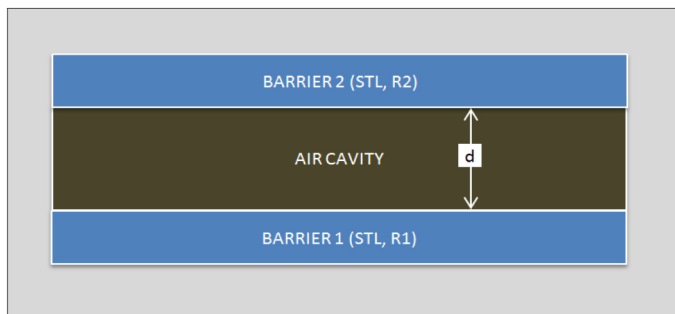


Figure 4. Double leaf barrier with air-cavity

Expressions have been developed by Sharp [3] [4] to predict the behavior of a double leaf barrier system in different frequency ranges.

Below the critical frequency or the system resonance frequency, f_c , transmission loss is given by -

$$STL, R = 20 \log[f(m_1 + m_2)] - 47 \quad [\text{For } f < f_c] \quad (3)$$

The equation is essentially same as the 'Mass Law'.

Between f_c and lowest cavity resonance, f_a , transmission loss is given

by -

$$STL, R = R_1 + R_2 + 20 \log(f \cdot d) - 29 \quad [\text{For } f_c < f < f_a] \quad (4)$$

Where R_1 and R_2 are the transmission losses of the two leaves predicted by mass-law separately and d is the depth of the cavity.

The lowest resonance frequency, f_a of the cavity (assumed to be air) between the panels is given by

$$f_a = \frac{c}{2\pi \cdot d} \quad (5)$$

Substituting speed of sound at room temperature as 345 m/s,

$$f_a = \frac{55}{d} \quad (6)$$

Above f_a , transmission loss is no longer affected by the spring-mass resonances and is given by:

$$STL, R = R_1 + R_2 + 6 \quad [\text{For } f_c < f < f_a] \quad (7)$$

Shortcoming of the proposed model is that an air cavity has been assumed in the model instead of an absorber. However, since the model is intended to serve as a comparative tool such assumptions are necessary to deliver simple expressions without a need of complex computing.

Calculation of Critical Frequency, f_c

To be able to use this model, it is imperative to determine the resonance frequency of the system. The carpet system was acoustically analyzed by considering it as a spring-mass system in a Simple Harmonic motion (SHM) and the de-coupler resonance frequency has been derived using standard SHM mathematical solution [1].

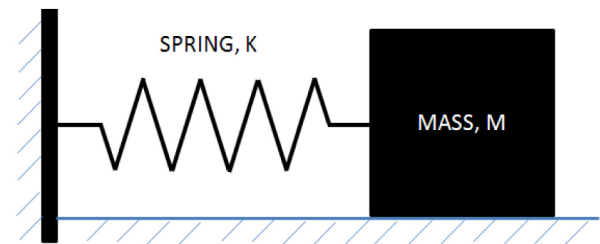


Figure 5. Simple harmonic oscillator system

For the given system (Figure 5) of spring constant, K and mass, M , resonant frequency occurs at,

$$f_c = 0.5 \left(\frac{1}{\pi} \right)^2 \sqrt{\frac{K}{M}} \quad (8)$$

Where K = stiffness of the spring and M is mass of the block.