

Volumetric and Dynamic Performance Considerations of Elastomeric Components

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ABSTRACT

Elastomeric joints such as mounts and suspension bushings undergo broadband excitation and are often characterized through a cross-point dynamic stiffness measurement; yet, at frequencies above 100 Hz for many elastomeric components, the cross- and driving-point dynamic stiffness results significantly deviate. An illustrative example is developed where two different sized mounts, constructed of the same material and are shaped to achieve the same static stiffness behavior, exhibit drastically different dynamic behavior. Physical insight is provided through the development of a reduced order single-degree-of-freedom model where an internal resonance is explained. Next, a method to extract the parameters for the reduced order model from a detailed finite element bushing model is provided. Further, a new benchmark experiment is used to validate the simulated behavior and provide insight into certain frequency dependent behavior where internal elastic modes of elastomeric component are present. Finally, the effect of the internal resonance is examined within the context of a hybrid vibro-acoustic vehicle system model. It is demonstrated that the internal mount resonance exhibits a significant influence on the sound pressure sensitivity that is dependent upon the size of the component.

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INTRODUCTION

Elastomeric components are widely used in vehicle isolation systems such as engine mounts, suspension bushings, shock and strut mounts, cradle and body frame mounts, and exhaust hangers [1]. These components present many challenges in modeling such as amplitude and frequency dependent properties and the dynamic properties of assemblies may differ from those of components due to preload and boundary condition effects [2, 3, 4].

In large scale vehicle simulations, elastomeric components are commonly represented as lumped stiffness and damping elements [5] as the inclusion of the continuous three-dimensional elements and relevant material data is computationally prohibitive and this information may not be available early in design stages. Nevertheless, the internal mass contribution to the dynamic stiffness at higher frequencies cannot be ignored [3, 6-7]. For example, Hamada [3] demonstrates an exhaust hanger design through finite element simulations that component geometry can be used to tune the internal dynamics of the isolator. The internal dynamics of the isolator is also referred to as internal resonances or wave effects [7].

Elastomeric components are often characterized using a cross-point uniaxial, non-resonant sinusoidal measurement [8]. The connection models in commercial simulation software [9-10] seem to incorporate data from such cross-point non-resonant characterization methods. Further, connector models do not include options for separate

driving- and cross-point dynamic stiffness. To model such behavior, an analyst is compelled to include a separate substructure to represent the elastomeric component.

In view of these above mentioned complexities, this article attempts to clarify frequency dependent behavior of elastomeric components under the small strain assumption where internal resonances strongly influence the component's dynamic stiffness. This will be achieved by developing detailed finite element models of components and reducing the observed frequency responses to analytical expressions based on simple lumped models. Further, benchmark experiments are conducted to illustrate the proposed concepts and to validate of the models. Finally, a hybrid FRF based substructure vehicle system model [10] is used to evaluate the effect on the sound pressure level sensitivity.

PROBLEM FORMULATION

The scope of this article is limited to elastomeric mounts or bushings where the material is considered to be linear elastic with frequency dependent structural damping. Only the axial direction of a cylindrical component (as displayed in Fig. 1) is examined. The specific objectives of this article are as follows: 1. Design two illustrative cylindrical mounts that are scaled to have identical axial static stiffness of the same material and compare the dynamic behavior, 2. Develop a lumped parameter model that captures the

underlying physical phenomenon, 3. Conduct benchmark validation experiments; and 4. Examine the effect of the internal resonance on a vibro-acoustic vehicle system model.

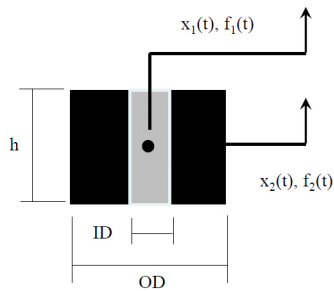


Figure 1. Schematic of the mount / bushing computational model. The inner and outer surfaces are kinematically coupled to reference points, x_1 and x_2 , respectively.

A schematic of the computational mount models is illustrated in Fig. 1 and are comprised of cylindrical geometries. The dimension and material properties of the two mounts (cases A and B) are given in Table 1. The dimensions are scaled such that the axial static stiffness is identical between the mounts; thus, the larger mount (A) has a 50% larger diameter and a volume four times greater than that the smaller mount (B). These mounts are subsequently analyzed via a detailed finite element simulation.

Table 1. Dimensions and material properties of the scaled 100 and 67 mm diameter mounts (A and B).

Parameter	Case A	Case B	Ratio (A/B)
OD, mm	100	67	1.5
ID, mm	32	32	1
h, mm	78	50	1.56
mass, kg	0.550	0.136	4.0
E, MPa	10	10	1
γ , loss factor	0.10	0.10	1
ρ , kg-m ⁻³	1000	1000	1
k, axial stiffness, N-mm ⁻¹	1305	1312	1.0

DYNAMIC STIFFNESS MATRICES VIA COMPUTATIONAL MODEL

The elastomeric mount models contain only a representative elastomer and the inner and outer surfaces are constrained through use of kinematic couplings each to a single reference point, x_1 and x_2 . The models are used to extract a two-terminal dynamic stiffness matrix at a given frequency (ω , rad/s) as

$$\begin{bmatrix} f_1(\omega) \\ f_2(\omega) \end{bmatrix} = \begin{bmatrix} D_{11}(\omega) & D_{12}(\omega) \\ D_{21}(\omega) & D_{22}(\omega) \end{bmatrix} \begin{bmatrix} x_1(\omega) \\ x_2(\omega) \end{bmatrix}, \quad (1)$$

where each element of the dynamic stiffness matrix is computed by imposing the following ideally fixed boundary conditions

$$D_{11}(\omega) = \left. \frac{f_1}{x_1} \right|_{x_2=0}, \quad D_{12}(\omega) = \left. \frac{f_1}{x_2} \right|_{x_1=0},$$

$$D_{21}(\omega) = \left. \frac{f_2}{x_1} \right|_{x_2=0}, \text{ and } D_{22}(\omega) = \left. \frac{f_2}{x_2} \right|_{x_1=0}. \quad (2a-d)$$

Additionally, all other degrees of freedom, except for the actuation direction, at the reference points are set to zero. The reaction forces at the reference points are then output and divided by the input displacement to compute the dynamic stiffness elements. These boundary conditions are equivalent to a displacement controlled stepped sinusoidal test commonly employed to characterize a physical component while the other end is fixed to the base. The cross-point terms, D_{12} or D_{21} , are typically reported from a commercial elastomer test machine [8].

Figures 2 and 3 contain the elements of dynamic stiffness matrix and the results are shown normalized with respect to the static stiffness value. Both mounts exhibit similar trends. First, the driving point dynamic stiffness magnitude terms, D_{11} and D_{22} , initially start at the static stiffness value and decrease to a minimum value of above the loss factor, 0.10, before reversing and increasing to a value of 10 times greater than the static value. For both mounts, the driving point stiffness terms are not equal; thus, the mounts behave differently depending upon which surface, the inner or outer, is being actuated by a displacement input or constrained. The cross-point terms, D_{12} and D_{21} , (shown in Figs. 2b and 3b) are identical (as expected) and do not exhibit the same reduction in dynamic stiffness as is observed for the driving point terms; however, the amplification is observed in the frequency range shown for the larger mount with a 100 mm diameter.

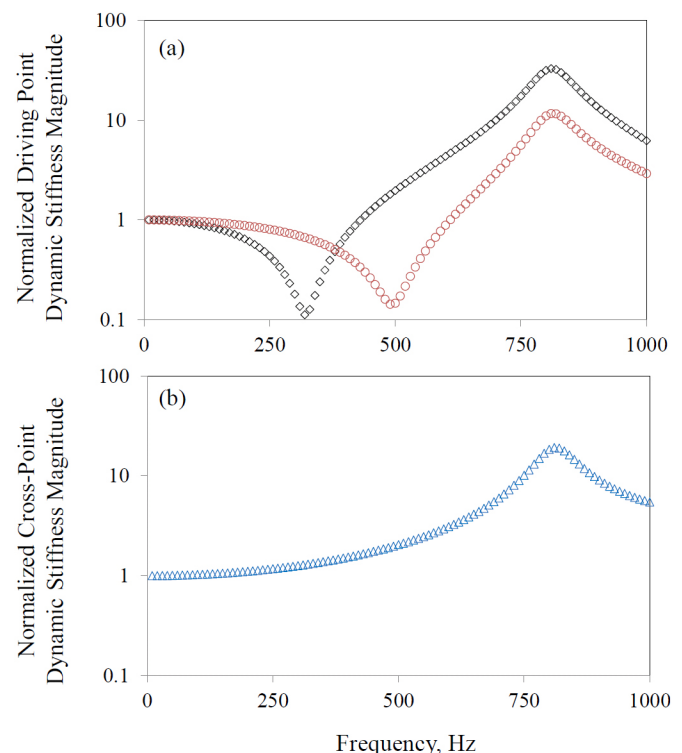


Figure 2. Dynamic stiffness terms for mount A, 100 mm diameter mount. (a) Driving point terms; (b) Cross-point terms. Key: $\square$ D_{11} , $\circ$ D_{22} and $\triangle$ D_{12} . Note that $D_{12} = D_{21}$.

When comparing dynamic properties of mounts A and B, the driving point dynamic stiffness term, D_{11} , achieves its minimum value at 320 and 680 Hz, respectively, a factor 2.1. This result is explained by the 4.0 ratio of A/B mount mass as shown in Table 1 and is discussed in detail later in this article.

EXPERIMENTAL VALIDATION

Controlled benchmark non-resonant test experiments are performed using solid cylindrical elastomeric components and are arranged in a horizontal configuration for characterization in the shear directions which are shown in Fig. 4. The shear configuration is achieved with a pair of cylinders to ensure appropriate boundary conditions. The diameter and height of each cylinder is 25.4 mm. Metallic caps are bonded to both sides of each cylinder and are threaded for ease in assembly. The non-resonant test employs a uniaxial hydraulically actuated, close loop servo-controlled elastomer test machine [8]. A sinusoidal displacement, $x(t)$, is applied to the top side of the specimen with a peak-to-peak displacement of 0.1 mm, and the transmitted force, $f(t)$, at the bottom of the specimen is measured at the same frequency. The dynamic stiffness is defined as:

$$\tilde{k}_d(\omega) = \frac{f}{x} e^{i\delta} = |\tilde{k}_d| \angle \delta = k + ih \quad (3)$$

where δ is the loss angle, k is the storage stiffness and h is the dissipative stiffness. The dynamic stiffness is rewritten by defining the structural loss factor, $\gamma = \tan \delta$, as

$$\tilde{k}_d(\omega) = k(1 + i\gamma). \quad (4)$$

Fig. 5 shows the measured cross-point dynamic stiffness for a frequency range up to 600 Hz. At 600 Hz, the test machine is no longer able to hold the specified tolerance for controlled displacement and thus the test is ended for this experiment. The experimental results are normalized by the dynamic stiffness observed at 100 Hz.

A linear finite element model is constructed of the benchmark experiment. The material nonlinearities that dominate the measured frequency dependent behavior of the experiment below 100 Hz are intentionally ignored in the model. The material is modeled as linear elastic with structural (hysteretic) damping that is proportional to stiffness and set equal to the measured loss factor for stiffness. The dynamic elastic modulus ($\tilde{E}$) is scaled to match the component dynamic stiffness measurement at 100 Hz. The measured and simulated cross-point dynamic stiffness values are compared in Fig. 5, and a good agreement is achieved above 100 Hz. The dynamic stiffness is amplified by a factor of two from 100 to 600 Hz in both the simulation and experiment.

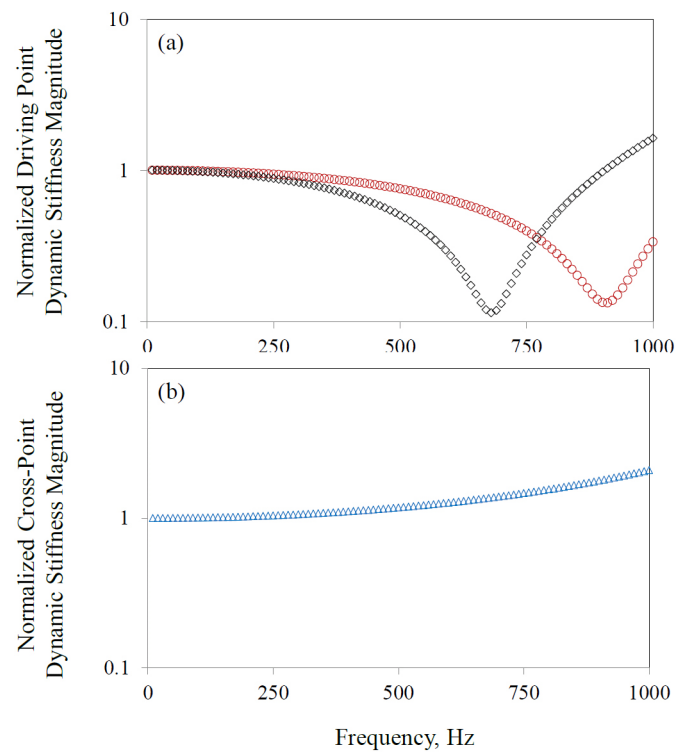


Figure 3. Dynamic stiffness terms for Case B, 67 mm diameter mount; (a) Driving point terms; (b) Cross-point terms; key: $\diamond D_{11}$, $\circ D_{22}$ and $\triangle D_{12}$. Note that $D_{12} = D_{21}$.

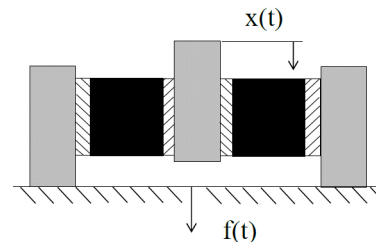


Figure 4. Schematic of the non-resonant test with cylindrical elastomeric elements configured in shear loading.

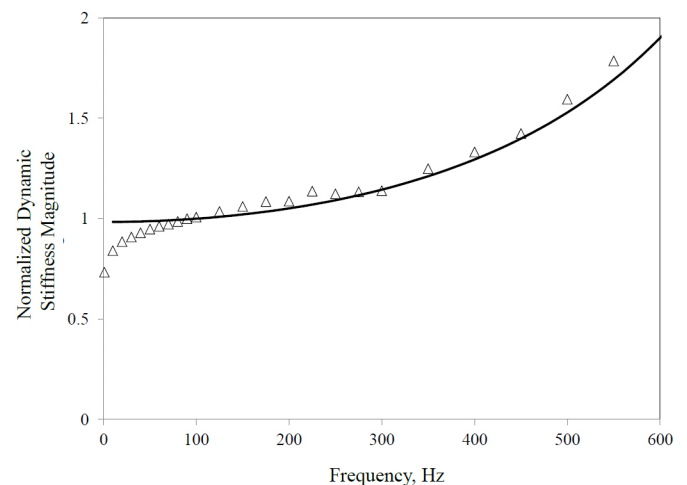


Figure 5. Comparison of cross-point dynamic stiffness term for benchmark experiment shown in Fig. 4; key: $\triangle$ Measurement, — Finite element simulation.