

Effect of Local Stiffness Coupling on the Modes of a Subframe-Bushing System

Scott Noll, Jason Dreyer and Rajendra Singh
The Ohio State University

ABSTRACT

The elastomeric joints (bushings or mounts) in vehicle structural frames are usually described as uncoupled springs (only with diagonal terms) in large scale system models. The off-diagonal terms of an elastomeric joint have been previously ignored as they are often unknown since their properties cannot be measured in a uniaxial elastomer test system. This paper overcomes this deficiency via a scientific study of a laboratory frame that is designed to maintain a high fidelity with real-world vehicle body subframes in terms of natural modes under free boundaries. The steel beam construction of the laboratory frame, with four elastomeric mounts at the corners, permits the development of a highly accurate, yet simple, beam finite element model. This allows for a correlation study between the experiment and model that helps shed light upon the underlying physical phenomenon. In particular, the effect of local stiffness coupling of elastomeric bushings or mounts is demonstrated through computational modeling and experimental validation. It is seen that the joint stiffness matrices strongly influence the modal properties of a laboratory subframe-mount system. For instance, a strong correlation between the rigid body modes ($r = 1$ to 6) and the first three elastic modes ($r = 7$ to 9) is only possible when the coupling (non-diagonal) terms are included in the bushing model.

CITATION: Noll, S., Dreyer, J. and Singh, R., "Effect of Local Stiffness Coupling on the Modes of a Subframe-Bushing System," *SAE Int. J. Passeng. Cars - Mech. Syst.* 6(2):2013, doi:10.4271/2013-01-1904.

INTRODUCTION

Elastomeric components are used as engine mounts, suspension bushings, torque struts, shock and strut mounts, jounce bumpers, cradle mounts, body frame mounts, and exhaust hangers. Their geometries are shaped to provide favorable properties in certain directions based upon the diagonal terms [1]. Nonetheless, non-diagonal (coupling) terms are often unknown though they are intrinsic to the design of complex automotive assemblies [2, 3]. Elastomeric joints present many modeling challenges such as amplitude and frequency dependent properties. Their small amplitude dynamic stiffness and damping characteristics influence vehicle noise, vibration, and harshness performance; their large-amplitude static force-deflection characteristics are crucial for vehicle ride and handling performance; on the other hand, when a vehicle is driven on a rough road, elastomeric components experience large-amplitude dynamic loads [4].

To complicate the matter further, the dynamic properties of an elastomeric joint embedded within an assembly may differ from those of components due to preload and boundary conditions [2]. Real world automotive frames, engine cradles, and suspension subframes are complicated structures;

consequently, there is a need for a laboratory frame design permitting combinations of bushings and frames for experimental and analytical characterization. In view of the above-mentioned challenges, this article attempts to fill this void by designing and constructing a simple, yet high fidelity laboratory frame and investigate the influence of non-diagonal elastomeric joint properties on the modal properties through computational modeling and experimental validation.

PROBLEM FORMULATION

The scope of this investigation is limited to the frequency range of the first nine modes which includes six rigid body and three elastic deformation modes (say up to 300 Hz). Only natural frequencies and mode shapes will be examined. Since only lower frequencies are examined, the elastomeric mounts will be modeled as massless with lumped linear stiffness. Thick beam (Timoshenko) theory will be used for beam flexure and the joints are selected with low damping loss factors (say less than 5% in the small amplitude regime), thus the assembled structure is assumed to have normal modes. The specific objectives of this article are as follows: 1. design and construct a laboratory ladder beam frame with corner mounted (four) elastomeric mounts in the context of

automotive subframes (engine cradle or rear suspension), 2. conduct modal measurements in freely suspended and constrained boundary conditions, 3. develop a computational model of the frame and frame-bushing system experiment, 4. examine the influence of off-diagonal stiffness terms on the eigensolution.

DESIGN OF A LABORATORY FRAME

A ladder frame design was selected due to its simplicity in construction, modeling, and robustness in capturing the modal properties of real-world frames. The laboratory ladder frame (Figure 1) is comprised of two symmetrical rail beams that connect together via two symmetrical cross-member beams. The dimensions of the laboratory frame were selected using modal similitude with that of real-world frames for the first three elastic deformation modes that are briefly described in Table 1. A simple design equation is used to estimate the natural frequency of the first rail beam bending mode which can be approximated with the free-free condition beam flexure formula:

$$f_{\text{rail}} = \frac{(\beta_r L)^2}{2\pi} \sqrt{\frac{E_1 I_1}{\rho_1 A_1 L^4}} \quad (1)$$

where $\beta_r L \approx (2r+1)\pi/2$, E is young's modulus, I is the area moment of inertia, A is the cross-sectional area and L is the length of the rail beam. For beam models with the the same geometry, the natural frequency ratio can be scaled for different materials (from material 1 to 2) as:

$$\bar{f}_{1/2} = \sqrt{\frac{E_1 \rho_2}{\rho_1 E_2}}. \quad (2)$$

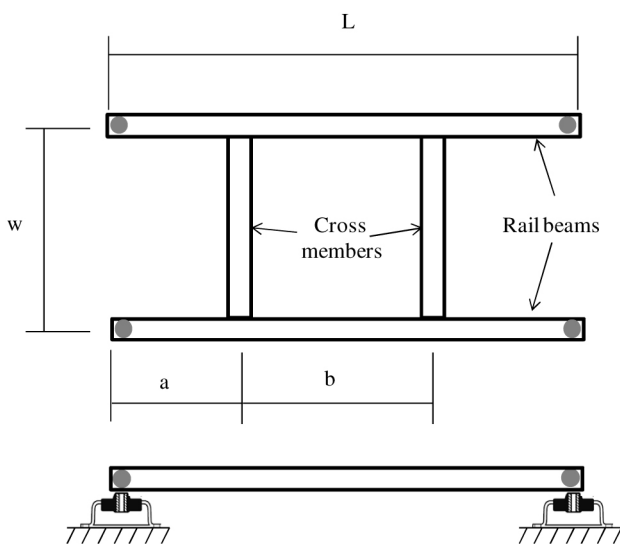


Figure 1. Schematic of the laboratory frame. Bushing supports are at the four corners.

Table 1. Comparison of measured normalized natural frequencies ($\bar{\omega}_r$) of elastic deformation modes between vehicle and laboratory frames under freely suspended conditions. Natural frequencies normalized by $r = 7$ of the vehicle frame.

Mode, r	Vehicle Frame, $\bar{\omega}_r$	Laboratory Frame, $\bar{\omega}_r$	Mode Description
7	1.0	0.9	Torsion
8	1.5	1.4	Rail bending
9	1.8	1.8	Cross-member bending (vehicle) Out-of-phase rail bending (lab)

COMPUTATIONAL MODEL OF FRAME

The computational model is comprised of 20 beam finite elements shown in Figure 2. A two-node Timoshenko beam element is superposed with a longitudinal rod element to generate a two-node six degree of freedom element. This formulation assumes that elastic longitudinal and beam bending are uncoupled. The work of Friedman and Kosmatka [5] contains a complete derivation of the finite element stiffness and mass matrices for a Timoshenko beam. The finite element model of the laboratory frame is implemented into a self-written script. The bushings are modeled with lumped dynamic stiffness matrices at nodes 1, 7, 10, and 16. The axi-symmetric bushings are assumed to have a stiffness matrix of the form:

$$\mathbf{k}_{\text{bushing}} = \begin{bmatrix} k_{11} & 0 & 0 & 0 & 0 & k_{61} \\ 0 & k_{22} & 0 & 0 & 0 & 0 \\ 0 & 0 & k_{11} & -k_{61} & 0 & 0 \\ 0 & 0 & -k_{61} & k_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & k_{55} & 0 \\ k_{61} & 0 & 0 & 0 & 0 & k_{44} \end{bmatrix}, \quad \delta = \begin{bmatrix} \delta x \\ \delta y \\ \delta z \\ \delta \theta_x \\ \delta \theta_y \\ \delta \theta_z \end{bmatrix} \quad (3)$$

when one end is attached to ground. Note that the vertical and torsional stiffnesses, k_{22} and k_{55} , are uncoupled (lacking terms in the same row or column) and that the x and z translational and rotational stiffness terms are assumed identical as are the coupling terms. The negative signs for k_{61} that couple the δz and $\delta \theta_x$ is due to the sign convention of the kinematic coupling.