Motion and Vibration Control Using Passive and Active Hydraulic Mounts: Improved Nonlinear and Quasi-Linear Models, Latest Design Trends and Some Insights

R. Singh

The Donald D. Glower Chair in Engineering and
Professor, Dept. of Mechanical Engineering
Director, NSF I/UCRC Smart Vehicle Concepts Center
The Ohio State University, Columbus, OH 43210 USA
e-mail: singh.3@osu.edu

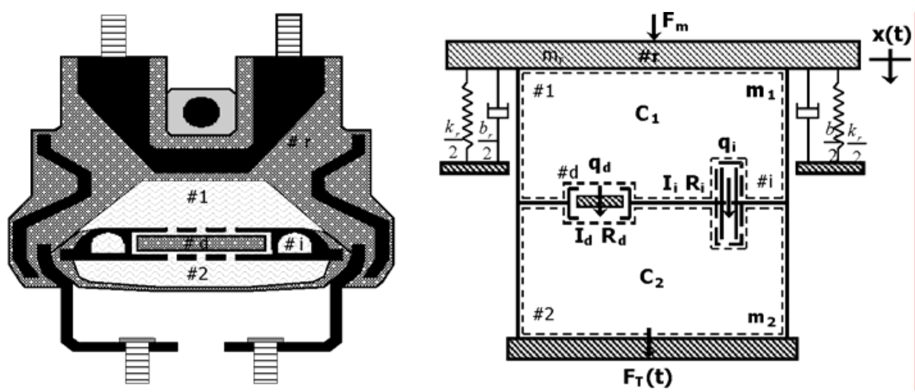
ABSTRACT

Hydraulic engine mounts are utilized in many vehicles to control motion and vibration. These are inherently nonlinear devices and their steady-state characteristics are extensively reported and used for design, usually in the form of spectrally-varying and amplitude-sensitive stiffness spectra. Although several linear and nonlinear mathematical models have been proposed, some key questions still remain. For instance, how should we comparatively evaluate the competing linear, quasi-linear and nonlinear (fluid or analogous mechanical system) formulations? Next, we examine the dynamic system design issues as a combination of passive and active hydraulic (and/or elastomeric) mounts is utilized to isolate engines and other machines. Requirements include not only the minimization of forces transmitted into the rigid or compliant foundations but also a careful examination of multi-dimensional motion control and sub-system

decoupling issues. Thus, passive and active mount parameters (including stiffness and damping rates, mount locations and orientation angles) must be properly analyzed especially at the lower frequencies. This article will briefly discuss the dynamics of such multi-degree-of-freedom passive or active isolation system by focusing on their eigensolutions, coupling dynamics and motion control issues. Latest work on smart mounts and dynamic load sensing issues will be discussed, and some directions for research will be suggested.

1. INTRODUCTION

Hydraulic engine mounts are often designed to provide improved stiffness and damping characteristics which vary with frequency and excitation amplitude (Singh et al., 1992; Kim and Singh, 1995; Geisberger et al., 2002; Adiguna et al., 2003; Tiwari et al., 2003). Figure 1(a) illustrates a typical schematic of the mount; refer to Singh et al. (1992) for a detailed description of the internal parts, their functions and basic parameters. Hydraulic engine mounts are utilized in many vehicles to control motion and vibration. These are inherently nonlinear devices and their steady-state characteristics are extensively reported and used for design, usually in the form of spectrally-varying and amplitude-sensitive stiffness spectra based on non-resonant type sinusoidal testing methods. Although several linear and nonlinear mathematical models have been proposed, some key questions still remain. Further, how should we comparatively evaluate the competing linear, quasi-linear and nonlinear (fluid or analogous mechanical system) formulations?



(a) (b)

Fig. 1: Example Case. (a) Hydraulic Engine Mount with Inertia Track and Decoupler; (b) Lumped Fluid Model of a Generic Hydraulic Engine Mount

Such mounts are usually modeled by lumping the fluid system into several control volumes as shown in Figure 1(b) (Singh et al., 1992; Adiguna et al., 2003). System parameters include the fluid compliances C_1 and C_2 of the top (#1) and bottom (#2) chambers, stiffness k_r and viscous damping b_r of the elastomeric rubber element (#*r*), fluid resistance R_i and inertance I_i of the inertia track (#*i*), inertance I_d and resistance R_d of the decoupler (#*d*). Further, the dynamic displacement excitation $x(t)$ is applied (non-resonant sinusoidal testing) under a mean load F_m, and the force $F_T(t)$ transmitted to the rigid base is often viewed as a measure of mount performance (Singh et al., 1992). We utilize both fluid system parameters (such as compliances, C_1 and C_2, in pressure and volume units) and mechanical system parameters (such as stiffness k_r in force and displacement units). Refer to Singh *et al.* (1992) for details. Continuity equations for the lower and upper chambers of Figure 1(b) the pressure and flow relations where $q_i(t)$ and $q_d(t)$ are the flow rates through the inertia track and decoupler respectively, A_r is the effective piston area, and $p_1(t)$ and $p_2(t)$ are dynamic pressures in the top and bottom chambers respectively.

2. MOUNT NONLINEARITIES AND IDENTIFICATION OF PARAMETERS

The following nonlinearities are usually found in most applications: (a) chamber compliances $C_1(p_1)$ and $C_2(p_2)$, (b) flow resistances $R_i(q_i)$ and $R_d(q_d)$ through the inertia track and decoupler respectively, (c) decoupler switching action and associated flow $q_d(t)$, and (d) vacuum phenomenon in the upper chamber (Kim and Singh, 1995; Adiguna et al., 2003). Kim and Singh (1995) and then Tiwari *et al.* (2003; also Aduniga et al., 2003) have described the mathematical or empirical relationships based on deliberate bench experiments. Tiwari *et al.* (2003) described a series of laboratory experiments that must be undertaken to characterize various nonlinearities of hydraulic engine mounts. Essentially they refined the experimental methods that were first proposed by Kim and Singh (1995) and successfully extended the nonlinear formulation with empirically obtained functions or curve-fits to the prediction of responses to ideal transient excitations. Further, Geisberger *et al.* (2002) used a hydraulic cylinder with a two-chamber vessel to test various components including upper or lower fluid chambers, inertia track and decoupler. Such experimental approaches are necessary for research studies but they pose significant difficulties for mount manufacturers and users (vehicle designers) as they may have tens or even hundreds of engine mount designs at their disposal but do not have the luxury of time, or even the facility, to fully characterize the parameters using the suggested research procedures (Kim and Singh, 1995; Geisberger et al., 2002; Tiwari et al., 2003). What is ideally desirable would be an approach that employs limited (and off the shelf) information such as measured data in terms of dynamic stiffness spectra $\tilde{K}(f, X)$ over the frequency (*f*, Hz) range of interest at certain displacement excitation amplitudes (*X*, mm). We have developed new estimation methods that would quickly develop linear or quasi-linear models with reasonable accuracy over both lower (typically up to 50 Hz) and higher frequency (say from 50 to 300 Hz) regimes, as well as in time domain.

In our proposed approach, we first assess the following constraints from the perspective of system user or manufacturer: (a) the mount is viewed as a black-box component with very limited information provided by the mount vendors to protect their proprietary designs; (b) only steady state $\tilde{K}(f, X)$ data are available; (c) experimental facilities to conduct bench tests as suggested by researchers (Kim and Singh, 1995; Geisberger et al., 2002; Adiguna et al., 2003; Tiwari et al., 2003) are not available; and (d) time is of essence since the product design cycles are now very stringent. Accordingly we develop new or refined estimation procedures, and focus on the quasi-linear model in this article. Note that our method would be able to estimate parameters over two frequency regimes: (i) the lower frequency regime (typically up to 50 Hz) that is usually controlled by the inertia track resonance, and (ii) the decoupler resonance that dominates over the higher frequency regime (say from 50 to 300 Hz). Details are given by He and Singh (2005).

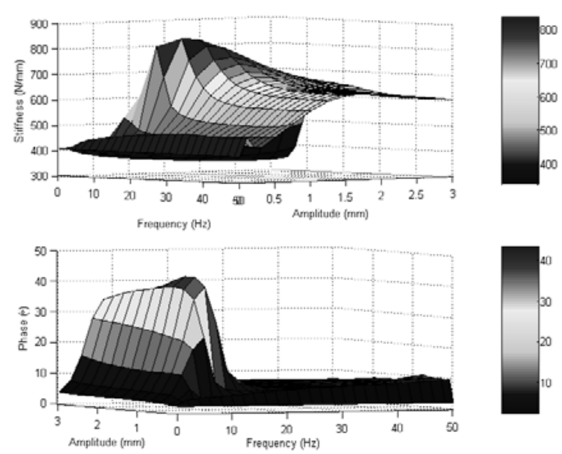


Fig. 2: Typical Measured $\tilde{K}(f, X)$ of Mount D with Discrete Excitation Amplitude *X* and Frequency *f*

3. TIME DOMAIN MODELS OF HYDRAULIC MOUNTS AND COMPARISONS WITH MEASUREMENTS

The steady-state characteristics have been extensively reported, usually in the form of K(f, X), based on experimental and analytical studies (Kim and Singh, 1993, 1995; Colgate et al., 1995; Royston and Singh, 1997; Golnaraghi and Jazar, 2001; Jeong and Singh, 2001; Yu et al., 2001; Geisberger et al., 2002; Jazar and Golnaraghi, 2002; Foumani et al., 2003). However, under transient and realistic loading conditions, new nonlinear phenomena emerge. These include asymmetric step responses (Adinuga et al., 2003; Tiwari et al., 2003) and the stiffening of chambers under increased mean load. Such responses cannot be explained by the existing models (Kim and Singh, 1993, 1995; Colgate et al., 1995; Royston and Singh, 1997; Golnaraghi and Jazar, 2001; Jeong and Singh, 2001; Yu et al., 2001; Geisberger et al., 2002; Jazar and Golnaraghi, 2002; Foumani et al., 2003; Adinuga et al., 2003; Tiwari et al., 2003; He and Singh, 2005; Lee and Kim, 2005). Therefore, an in-depth study is needed to investigate the discontinuous nature of the top and bottom chamber compliances.

Although several nonlinear models have been proposed and validated to some extent, some key questions still remain: Are the nonlinear models based on statically measured parameters (Kim and Singh, 1993; Tiwari et al., 2003) applicable to real-life excitations or operational conditions? Which nonlinearities would be excited under transient conditions, especially when a rapid change in the loading condition takes place, or when the preload Fm itself may also vary with time? Further, how should we comparatively evaluate the competing linear, quasi-linear and nonlinear formulations? Accordingly, we formulate the following objectives: (1) Propose an improved multi-staged $C_1(p_1)$ formulation to capture the asymmetric responses including dynamic stiffening and softening (vacuum) effects, where p1 is the dynamic top chamber pressure. (2) Develop a new semi-analytical model with time-varying $C_1(t)$ to predict the asymmetric step responses. (3) Propose a refined $C_2(x_m)$ formulation to explain the stiffening effect observed under increased mean loads, where x_m is the mean displacement. (4) Validate proposed nonlinear and semi-analytical linear time-varying formulations by comparing predictions with measurements (and with quasi-linear models) under step and realistic excitations.

In order to quantify the above-mentioned multi-staged nonlinearities, experiments were conducted using a take-apart mount (designated as D). Parameters of this mount are as follows: k_r = 320e3 N/m, b_r = 0.5e3N s/m, A_r = 3.31e-3 m^2, R_i = 3.45e7 kg/s m^4, inertia track length l_i = 0.236 m, I_i = 2.8le6 kg/m^4, decoupler damping b_d = 100N s/m, decoupler gap length l_g = 1.1e-3 m, and m_d = 6e-3 kg. The linearized nominal chamber compliances are C_{10} = 2.5e-11 m^5/N and C_{20} = 2.4e-9 m^5/N. As a first estimate, effective top chamber compliances C_{1e} are approximated by using the quasi-linear model (He and Singh, 2005) to best curve-fit either the step overshoot or decaying transient given various step excitations. Comparisons of predictions and measurements in Figs. 3–4 show that the quasi-linear model (with a constant C_{1e}) fails to concurrently predict both the overshoot and the decaying transient of step responses due to significant changes in C1e during the step transition.

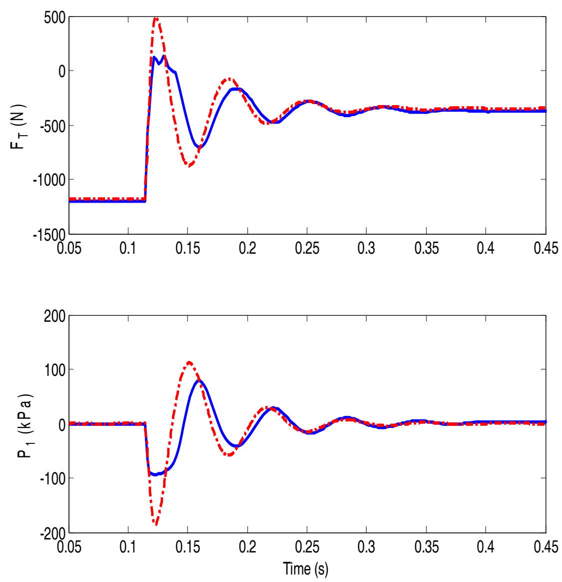


Fig. 3: Transient Responses to Non-Ideal Step-up (x_m from -3.7 to -1.32 mm) Excitation. Key: ——, Measurement; – · –, Predictions by Quasi-Linear Model with Effective C_{1e} (estimated using the decaying transients)

Compare the estimated C_{1e} values in Table 1(a) with the linearized C_1 values in Table 1(b) that were previously measured (Tiwari et al., 2003) at several F_m levels. Note that C_{1e} of the step-up overshoot (from -3.7 to 0 mm) is consistent with the measured C_1 under no preload. Conversely, C_{1e} estimated from the step-down overshoot (from 0 to -3.7 mm) coincides with C_1 measured under F_m = -1200N (or x_m = -3.7 mm). Meanwhile, C_{1e} values estimated from the decay transients are consistent for all step responses. Also, these C_{1e} values (ranging from 2.17e-11 to 2.99e-11 m^5/N) match well with the nominal C_{10} value (2.5e-11 m^5/N), which is a linearized (and averaged) value based on several operational conditions (Tiwari et al., 2003).

The fact that C_{10} lies between the effective C_{1e} values estimated from step-up and step-down overshoots implies that: First, during the unloading (or step-up) process, a dynamic softening effect occurs, which could be explained by the vacuum phenomenon due to a release of dissolved gas under reduced pressure, as suggested by Kim and Singh (1993) and Adiguna et al. (2003). Second, a dynamic stiffening effect takes place during the loading (or step-down) process. Third, a linear region exists between the softening and hardening regions, during which a linearized (quasi-linear) model should suffice (He and Singh, 2005).

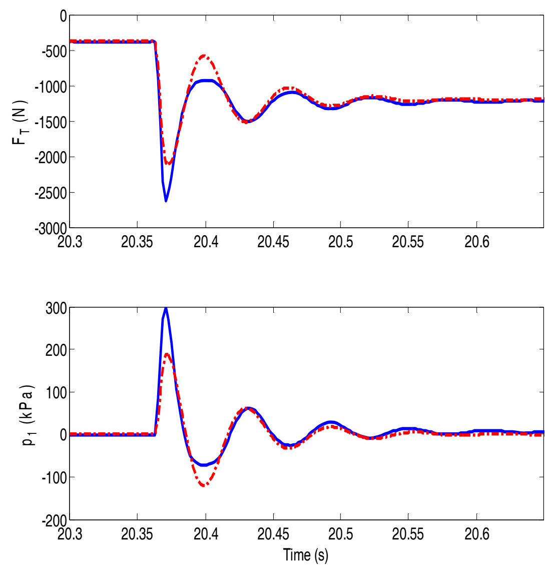


Fig. 4: Transient Responses to Non-Ideal Step-Down (x_m from -1.32 to -3.7 mm) Excitation. Key: ——, Measurement; – · –, Predictions by Quasi-Linear Model with Effective C_{1e} (Estimated using the decaying transients)

The measured realistic profile of Fig. 5 is implemented in the elastomer test machine as displacement excitations to both fixed and free decoupler hydraulic engine mounts with responses measured in time domain. A comparative study is then conducted using perditions from three competing formulations: (i) quasi-linear model (He and Singh, 2005), (ii) nonlinear fluid model with a constant C_{20}, and (iii) ‘‘true’’ nonlinear $C_2(x_m)$ model. For the fixed decoupler mount, Fig. 5 confirms that the mean pressure built-up effect can only be captured by the $C_2(x_m)$ model though the quasi-linear model is capable of predicting most transient responses.

Table 1: Estimated or Measured Top Chamber Compliance C_1 Values

(a) Effective C_{1e} values estimated from responses to step-up and step-down excitations

Non-ideal displacement excitation	*C1e (m5/N) from first overshoot value*	*C1e (m5/N) from decaying transient curve*
		*C*1e (m5/N)
Step-up from -3.7 to 0 mm	7.63e-11	2.99e-11
Step-up from -3.7 to -1.32 mm	5.28e-11	2.45e-11
Step-down from 0 to -3.7 mm	1.09e-11	2.39e-11
Step-down from -1.32 to -3.7 mm	1.26e-11	2.17e-11

(b) Measured C_1 values from a static test

Condition	*Static load Fm (N)*	*Measured C1(m5/N)*
Above *pa*	0 −800 −1200 −1200	7.29e-11 1.05e-11 1.09e-11 2.5e-11 (C_{10})
Below *pa*	C_1 = -7e-45 p17 + 2.5e-11 (here p1 is in Pa)	

Chief contribution of this study has been the development of new estimation procedures that employ measured sinusoidal dynamic stiffness data of fixed or free decoupler mount over low and high frequency regimes to characterize amplitude and frequency dependent parameters. Compared with the previously reported laboratory experiments (Kim and Singh, 1995; Geisberger et al., 2002; Adiguna et al., 2003; Tiwari et al., 2003), our estimation method requires minimal experimental effort. It can be efficiently implemented by mount manufacturers or vehicle designers to quickly develop quasi-linear models and then predict responses to mount excitations with reasonable accuracy. The main limitation of the proposed method is that the predicted response is based on the linearized approximations around the operating points and all nonlinearities are characterized by operational-dependent parameters. Consequently, the estimated quasi-linear model should not be viewed as the true non-linear formulation and as such it might not capture all of the significant events in time domain. An example has been shown where the pressure build-up effect, under a realistic excitation, was not predicted. Better prediction would obviously require an improved nonlinear $C_1(p_1)$ model that should incorporate the multi-staged stiffness characteristics including the dynamic stiffening effect corresponding to a time-varying mean load.

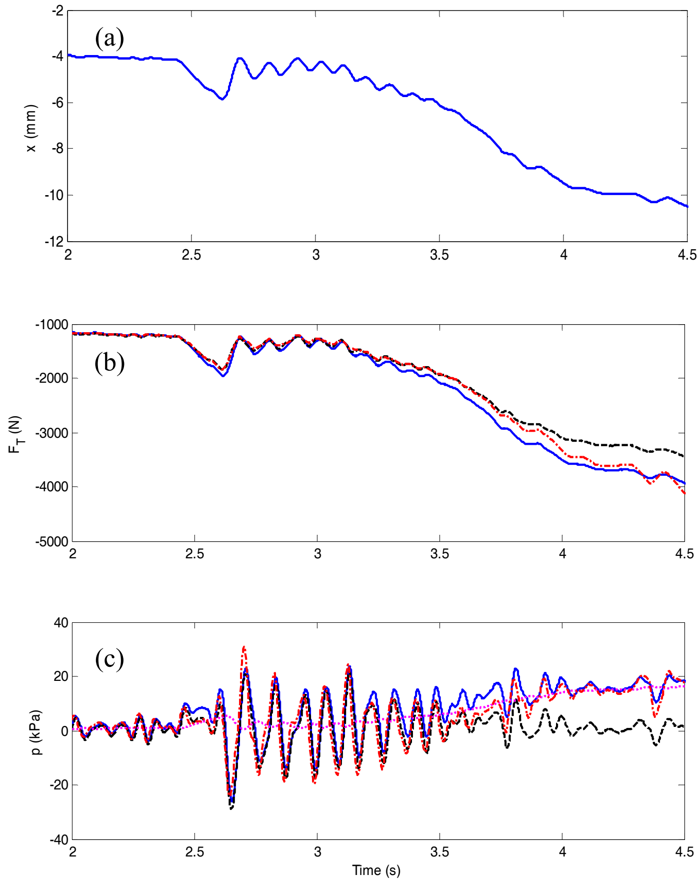


Fig. 5: Transient Responses to a Realistic Profile for a Fixed Decoupler Mount: (a) Displacement Excitation $x(t)$; (b) Transmitted Force $F_T(t)$; and (c) Chamber Pressures $p_1(t)$ and $p_2(t)$. Key: – · –, Measurements of $F_T(t)$ or $p_1(t)$; – – -, Prediction of $F_T(t)$ or $p(t)$ by a Quasi-Linear Model; ——, Prediction of $F_T(t)$ or $p(t)$ by a Nonlinear Model; ········, Prediction of p2(t) by Nonlinear Model with Effective $C_{2e} = c_{20}/1.8$

4. POWERTRAIN SYSTEM MODEL WITH PASSIVE AND ACTIVE MOUNTS

Next, we examine the dynamic system design issues as a combination of passive and active hydraulic (and/or elastomeric) mounts is utilized to isolate engines and other machines. Requirements include not only the minimization of forces transmitted into the rigid or compliant foundations but also a careful examination of multi-dimensional motion control and sub-system decoupling issues. Thus, passive and active mount parameters (including stiffness and damping rates, mount locations and orientation angles) must be properly analyzed especially at the lower frequencies. This article will briefly discuss the dynamics of such multi-degree-of-freedom passive or active isolation system by focusing on their eigensolutions, coupling dynamics and motion control issues.

Several active or smart engine mounting devices (Gennesseaux, 1995; Shibayama et al., 1995; Aoki et al., 1999; Lee and Lee, 2002; Matsuoka et al., 2004) have been designed to reduce noise and vibration, especially under high dynamic torque conditions. Typical design criteria (Spiekermann et al., 1985; Ashrafiuon and Nataraj, 1992; Ashrafiuon, 1993; Harris, 1995; Jeong and Singh, 2000; Tao et al., 2000; Park and Singh, 2008) include decoupling of powertrain motions and motion control. Motion control is achieved through reduction in resonant peaks, natural frequency placement, reduced vibration transmissibility and increased acoustic comfort. Even though the powertrain (rigid body) is a 6 degree-of-freedom (DOF) isolation system, most of analytical work has been conducted in the context of single-degree-of-freedom systems (Gennesseaux, 1995; Shibayama et al., 1995; Aoki et al., 1999; Lee and Lee, 2002; Matsuoka et al., 2004). In several cases, high damping solutions have been implemented to control resonance(s) at lower frequencies, and then more compliant mounts have been sought at higher frequencies to minimize the force transmitted in the base; this implies an introduction of frequency-dependent isolators through passive, adaptive or active means. The above-mentioned design may not satisfy the powertrain motion decoupling considerations, and thus the selection of an active mount in the context of a 6-DOF system remains an empirical science. We seek to overcome this deficiency by proposing a new analytical model that will examine a combination of active and passive mounts.

The literature on multi-degree-of-freedom active isolation systems is sparse. For example, Gardonio et al. (1997), Kim and Lee (2003), and Royston and Singh (1996) have limited their analyses to 3-DOF systems (e.g. transverse-axial-pitch motions or pitch-roll-bounce motions). Also, prior researchers have attempted to minimize the forces transmitted into the rigid or compliant foundations without considering the multi-dimensional motion control and coupling issues. In particular, Gardonio et al. (1997) and Kim and Lee (2003) have suggested that passive and active mount parameters should be properly selected prior to the isolation control problem, especially at the lower frequencies (say up to 50 Hz). We extend the prior multi-degree-of-freedom active isolators work (Royston and Singh, 1996; Gardonio et al., 1997; Kim and Lee, 2003) by focusing on the eigensolutions, coupling dynamics and motion control issues.

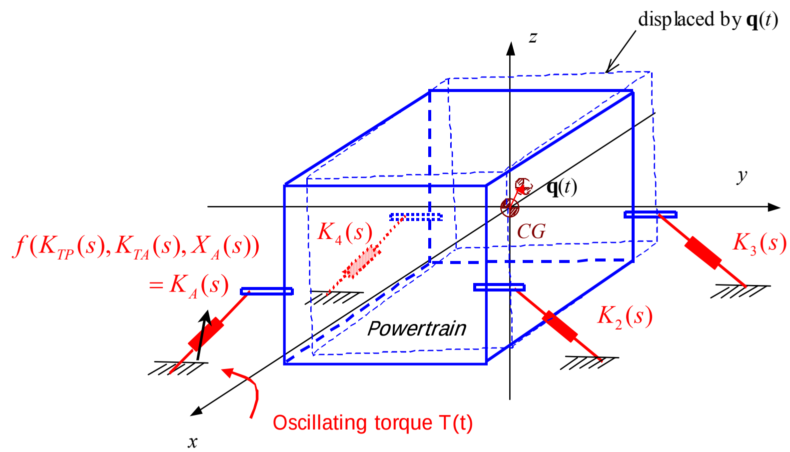


Fig. 6: Multi-Degree-of-Freedom Powertrain Isolation System with One Active Mount and Three Passive Mounts. Each Mount is Described by Tri-Axial Elements with Frequency Dependent Stiffness k(ω) and Damping c(ω) Properties. Here $K_i(s)$, (i = 2, 3, and 4 (here n is the number of mounts)) and $K_A(s)$ are the Dynamic Stiffness Terms of Passive and Active Elements (in a specific direction) Respectively.

Figure 6 illustrates a typical 6-DOF rigid body powertrain mounting system with linear time invariant active and passive mount models; active mount models are depicted in Figs. 7 to 9. Effective control of powertrain motions is essential over the lower frequency range up to 50 Hz since the rigid body modes significantly dominate and could couple with other vehicle system modes (Gardonio et al., 1997; Kim and Lee, 2003). Actuator displacement in active mounts is adjusted sinusoidally (Hillis et al., 2005; Nakaji et al., 1999) by a displacement actuator, $x_A(t) = X_A e^{j(\omega t + \phi_A)}$ where ω is the angular frequency of active displacement input, j is the imaginary unit, X_A is the active displacement amplitude, and ϕ_A is the phase angle of active displacement with respect to the external torque excitation. Examination of constant X_A and ϕ_A (at a time) would permit us to develop analytically tractable models in the context of a multi-degree-of-freedom isolation system. Our model analytically examines the parameters of active and passive mounts (such as stiffness, damping and active input), their locations and orientation angles. Analysis is limited to only the engine torque excitation.

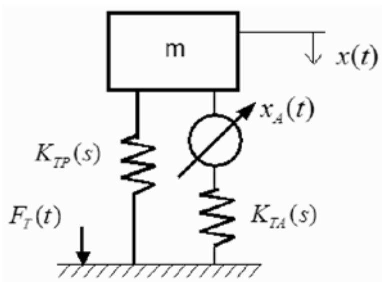


Fig. 7: Single-degree-of-Freedom System with an Active Mount

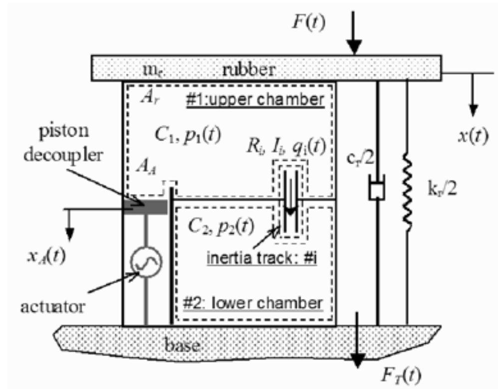


Fig. 8: Schematic of an Active Engine Mount Based on Hydraulic Mount. Here, C_1 and C_2 are the Compliances of Upper and Lower Chambers, $p_1(t)$ and $p_2(t)$ are the Chamber Pressures, R_i, I_i, and $q_i(t)$ are the Fluid Resistance, Inertance, and Flow Rate in the Inertia Track, A_T is the Equivalent Piston Area of Rubber, A_A is the Actuator Piston Area, and k_r and c_r are the Rubber Stiffness and Viscous Damping Terms (Voigt model assumed).

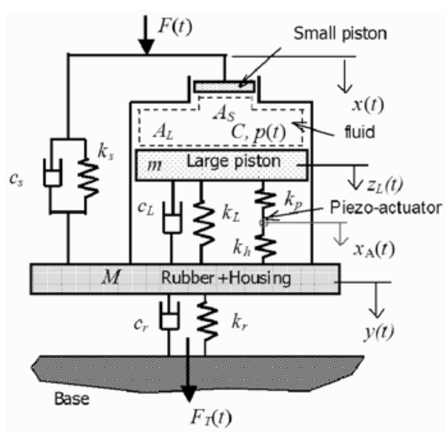


Fig. 9: Active Engine Mount Based on Piezoelectric Actuator. Here k_p and k_h are the Linearized Stiffness Terms for Piezoactuator and Holder, A_s and A_L are the Areas of Small and Large Pistons, C and p(t) are the Compliance of and Pressure in the Cylinder Control Volume, k_s and c_s are the Stiffness and Viscous Damping Coefficient of Small Piston Rubber, k_L and c_L are the Stiffness and Viscous Damping Coefficient of Large Piston Rubber, and k_r and c_r are Stiffness and Viscous Damping Coefficient of Main (Cushion) Rubber.

Consider a 6-DOF isolation system consisting of a rigid body (with powertrain mass *m*, and inertia I_{ij}, *i* and $j = x$, *y*, *z*,) under an oscillating torque (*T(t)*) and 4 tri-axial mounts which are assumed to be attached to a rigid base. Out of these, one is an active mount (given by subscript *A*) with dynamic stiffness $K_A(s) = f[K_{TP}(s), K_{TA}(s), X_A(s), X(s)]$ in a specific direction where *s* is the Laplace variable, *X(s)* is the powertrain displacement and X_A(s) is the actuator displacement. The other three are passive devices with dynamic stiffness K_i(s), $i = 2, 3, 4$ in a specific direction. Each mount element (in any direction) is assumed to have frequency-dependent, stiffness *k(ω)* and viscous damping *c(ω)* properties.

The model is then used to investigate the complex eigensolutions and frequency responses of the powertrain system of Fig. 6, when excited by an oscillating torque, with one active mount and 3 passive mounts. The active mount model is first formulated for fluid piston type active device such as a hydraulic engine mount (Genneseaux, 1995; Aoki et al., 1999; Lee and Lee, 2002; Matsuoka et al., 2004) or piezoelectric mount (Shibayama et al., 1995). Our method is validated by comparing analytical predictions in frequency domain with results from the direct inversion (numerical) method where we could simply use different *k* and *c* values at each frequency. This model can be used to examine multi-directional motion control and vibration isolation issues in the context of a multi-degree-of-freedom mounting system.

Gennesseaux (1993) examined four active isolation schemes and concluded that an actuator structure in parallel with a rubber element is the most preferred design, as the active actuator should be designed to generate only the dynamic force, and the static force should be provided by the rubber element. Based on this concept, a new analytical model for active mounts with actuator displacement input is proposed in Figs. 7 to 9. Since this work is limited to the lower frequency regime, the cross point transfer function (K_T(s)) concept is also applied to represent the dynamic property of an active mount. In this model, the force transmitted into the rigid base (F_T(s)), consists of a passive force (F_{TP}(s)) and an active force (F_{TA}(s)). See Park and Singh (2008, 09 and 10) for detailed studies and results.

5. CONCLUSION

In this article, we identify and quantify discontinuous compliance nonlinearities of hydraulic engine mounts. First, asymmetric nonlinearities are identified in transient step-up and step-down responses by using a quasi-linear mount model with parameters that are estimated from measured dynamic stiffness data. Second, an improved multistaged top chamber compliance model is developed which confirms the existence of highly nonlinear region(s) during the step transitions as well as during the decaying transients. Third, new semi-analytical solutions for both step-up and step-down responses have been successfully constructed by using a linear time-varying system formulation that incorporates time-varying compliance. Fourth, a mean displacement-dependent model is proposed for the bottom chamber compliance. It clearly explains the stiffening effect in measurements under higher mean loads. Finally, competing quasi-linear, linear time-varying and nonlinear formulations are comparatively evaluated for step and realistic excitations. Transient measurements confirm the validity of models, as well as their applicability and limitations. Based on the theory presented in this article, interactive simulation software could be developed that is already being used to examine or characterize some practical mounts. Finally, we intend to develop computational procedures that could incorporate quasi-linear or nonlinear mount formulations into large scale finite element or multi-body dynamic models of vehicles.

Further, a new 6-DOF rigid body model with a combination of active and passive mounts is proposed. To facilitate this development, a refined transfer function model for fluid-piston displacement type active mounts is developed and then is incorporated into mounting system, resulting in a spectrally-varying linear time-invariant system formulation. Our model is partially verified by comparison with numerically obtained frequency response functions; also, complex eigensolutions match with measured natural frequencies for one powertrain example. Future work includes extension of this work to other active isolation systems. Further, properties of an active mount could be specified from the system perspective (say decoupled motions, resonance control and reduced transmissibility) and then passive and active paths could be optimized to yield the desired performance over the frequency range of interest.

Space limitations prevent us from further discussion. Therefore, the latest work on smart mounts and dynamic load sensing issues will be discussed in the oral presentation, and some directions for future research will be suggested. See Yoon and Singh (2010) and Barszcz (2010) for details.

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