

Dynamic Analysis of a Hydraulic Body Mount with Amplitude and Preload Dependence

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ABSTRACT

The application of hydraulic body mounts between a pickup truck frame and cab to reduce freeway hop and smooth road shake has been documented in literature and realized in production vehicles. Previous studies have demonstrated the potential benefits of these devices, often through iterative prototype evaluation. Component dynamic characterization has also shown that these devices exhibit significant dependence to preload and dynamic amplitude; however, analysis of these devices has not addressed these dependences. This paper aims to understand the amplitude and preload dependence on the spectrally-varying properties of a production hydraulic body mount. This double-pumping, three-spring mount construction has a shared compliant element between the two fluid-filled chambers. A physics-based reduced-order model of the mount assembly is developed using parameters derived from inspection of the component geometry and bench experiments on the different elastomeric components and the fluid system. The dynamic properties of the mount are characterized using step sine testing, and the model is validated in the frequency domain. The model analysis provides insight into which features within the mount assembly drive the dynamic amplitude and preload dependence.

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INTRODUCTION

Within the last decade, the use of hydraulic body mounts between a pickup truck frame and cab to vehicle improve ride quality and reduce after-shake has become increasingly popular [1, 2]. Hydraulic, or fluid-filled, mounts are commonly used in automotive applications ranging from vehicle suspensions to powertrain and are known for the ability to provide high narrow band damping compared to simple elastomeric mounts. Static and dynamic properties for hydraulic mounts are dependent on geometry, preload, material properties, and manufacturing and assembly conditions, often inducing observed nonlinear phenomena such as hysteresis and amplitude dependence. Recent studies have focused heavily on characterization and lumped parameter modeling of single-pumping hydraulic engine or body mounts [3, 4] and double-pumping hydraulic suspension bushings [5, 6, 7], providing insight into the mean and dynamic displacement dependence of each device's spectrally-varying properties. In order to obtain unique and physically meaningful parameters for these models, pressure within the mount chambers must be measured. A few experimental and analytical studies have evaluated hydraulic mounts using embedded pressure sensors within prototype devices [6, 7], and fewer studies have attempted this on a production mount.

Dynamic performance of production hydraulic body mounts is often tuned by iteratively testing different physical prototypes. Proper modeling of these mounts could improve the representation of these mounts in system-level vehicle models and reduce the number of physical prototypes and test iterations required in the development phase. New hydraulic body mount designs, such as the ones

described by Hassan et al. [1], have been evaluated using iterative component- and system-level testing, but no analytical studies have been reported. This paper will focus on addressing this gap with a systematic approach to evaluate one of these designs. Accordingly, the objectives of this paper are to (i) design a laboratory experiment to measure the dynamic stiffness and chamber pressures of a production hydraulic body mount; (ii) develop a physics-based reduced-order model of the mount assembly; (iii) identify parameters for the reduced-order model using bench experiments and inspection of the component geometry; (iv) validate the model using the dynamic stiffness and pressures measured in the laboratory experiment under different loading conditions; and (v) exercise the model to gain a better understanding of the effects of preload and amplitude dependence and provide insight into how to modify the mount design to achieve a specific dynamic performance.

A cross-section of the production hydraulic body mount investigated is shown in [Figure 1](#). It consists of three elastomeric springs which connect the steel inner collar and plate to the steel outer shell. The inner collar is bolted to the cab with a through bolt, and the shell is bolted to the frame. The elastomeric springs also seal the fluid within the mount. Two fluid-filled chambers are connected by two identical parallel inertia tracks running around the circumference of the outer shell. When the inner collar is moved upward with respect to the outer shell, the inner plate and middle elastomeric spring pump fluid from the top chamber, through the inertia tracks, to the bottom

chamber, shown in Figure 2. Likewise, when the inner collar is moved downward, fluid is pumped from the bottom chamber to the top chamber.

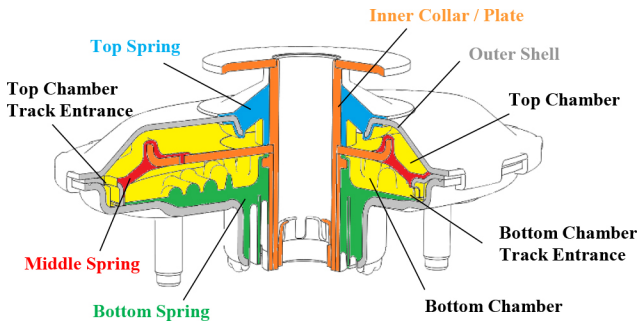


Figure 1. Cross-section of the production hydraulic body mount.

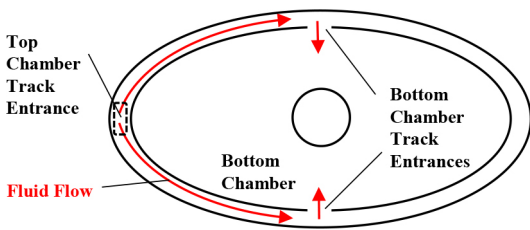


Figure 2. Parallel inertia track fluid flow from the top chamber to the bottom chamber (viewed from the bottom of the mount).

MODEL FORMULATION

A schematic of the proposed reduced-order lumped parameter model of the hydraulic body mount is shown in Figure 3. A dynamic input displacement x is applied to the inner collar, and a transmitted force f_T is measured at the grounded outer shell. The complex stiffness elements of the top, bottom, and middle elastomeric springs are given by $\tilde{k}_1$, $\tilde{k}_2$, and $\tilde{k}_3$, respectively, where $\tilde{k}$ takes the form of $k(1+j\eta)$ with η as the associated material loss factor for the given elastomeric spring, k as the effective real stiffness, and $j = \sqrt{-1}$. The complex stiffness, as written, can be applied in the frequency domain, but must be represented as a dry friction element in the time domain (omitted in the subsequent equations for brevity). The top chamber compliance C_1 is assumed to be dominated by the associated compliance from the top spring; likewise, the bottom chamber compliance C_2 is assumed to be dominated by the bottom spring. The middle spring acts as a common compliance element between the two chambers. The associated compliance C_3 due to this middle spring is given by $A_3^2/2k_3$, where A_3 is the effective pumping area of the middle spring. The middle spring is assumed to have a small effective lumped mass m_3 and its associated motion x_3 . If A_3 is set to zero, the formulation reduces to that of a hydraulic suspension bushing presented in previous literature [5, 6, 7]. The effective pumping areas of the top and bottom chambers are given by A_1 and A_2 , respectively. The dynamic pressures in the top and bottom chambers, respectively, are denoted by p_1 and p_2 . The volumetric flowrate for the effective inertia fluid path is given by q_{12} , with its respective inertance I_{12} and resistance R_{12} .

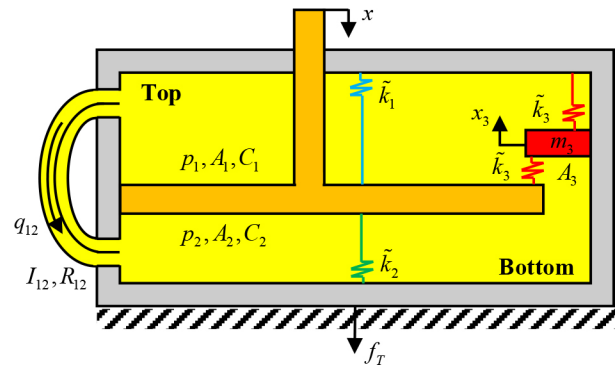


Figure 3. Schematic of the reduced-order lumped parameter model of the production hydraulic body mount.

The mean force transmitted through the mount subjected to a mean displacement input $\bar{x}$ and mean fluid pressure $\bar{p}$ is given by

$$\bar{f}_T = (A_2 - A_1)\bar{p} + \left(k_1 + k_2 + \frac{k_3}{2}\right)\bar{x}. \quad (1)$$

Applying fluid continuity to the top and bottom chambers, respectively, the following two equations are obtained.

$$C_1\dot{p}_1 = -q_{12} - A_1\dot{x} + A_3\dot{x}_3 \quad (2)$$

$$C_2\dot{p}_2 = q_{12} + A_2\dot{x} - A_3\dot{x}_3 \quad (3)$$

Applying the conservation of momentum through the inertia tracks connecting the two chambers give the following equation.

$$p_1 - p_2 = I_{12}\dot{q}_{12} + R_{12}q_{12} \quad (4)$$

The equation of motion for the lumped mass of the inner spring is given by

$$m_3\ddot{x}_3 = A_3(p_2 - p_1) - 2\tilde{k}_3x_3 - \tilde{k}_3x. \quad (5)$$

Finally, the transmitted force through the mount given a dynamic input displacement is calculated as

$$f_T = (A_2 + A_3)p_2 - (A_1 + A_3)p_1 + (\tilde{k}_1 + \tilde{k}_2)x - \tilde{k}_3x_3. \quad (6)$$

The dynamic stiffness magnitude $|K_d|$ and loss angle $\angle K_d$ can be calculated at each frequency ($\omega/2\pi$ in Hz) using a sine regression algorithm in the time domain or an iterative frequency domain method that updates nonlinear resistance terms (if included in the formulation) by intermediate calculations of the volumetric flow rate. The peak-to-peak pressures in the top and bottom chambers, $|P_1|$ and $|P_2|$, respectively, can also be calculated at each frequency.

EXPERIMENTAL STUDIES

In order to validate the model, a production mount is assembled with a pressure sensor (PCB 102A05) mounted on the outer shell attached to each chamber, shown in Figure 4. The instrumented mount is then assembled in a test fixture, where the outer shell is attached to the rigid base with a force transducer and the inner collar is attached to the hydraulic actuator. Static and dynamic properties are characterized using an elastomer test machine (MTS 831.50, [8]) using a step sine procedure. A mean displacement of 2 or 3 mm (in compression) is applied to the specimen while the transmitted force is measured due to a sinusoidal displacement input at peak-to-peak amplitudes of 0.5, 1, and 2 mm from 1 to 50 Hz in 1 Hz increments. These values are selected based on design preload and dynamic performance ranges for these mounts [1]. The $|K_d|$ and $\angle K_d$ are calculated at each frequency using a sine regression algorithm, and $|P_1|$ and $|P_2|$ are reported at each frequency. Harmonic distortion of the measured signals due to physical nonlinearities of the mount system are not considered in this study.

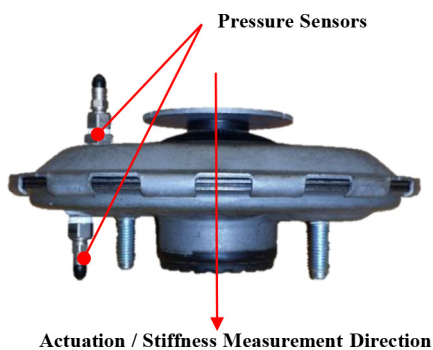


Figure 4. Production hydraulic body mount assembled with pressure sensor instrumentation.

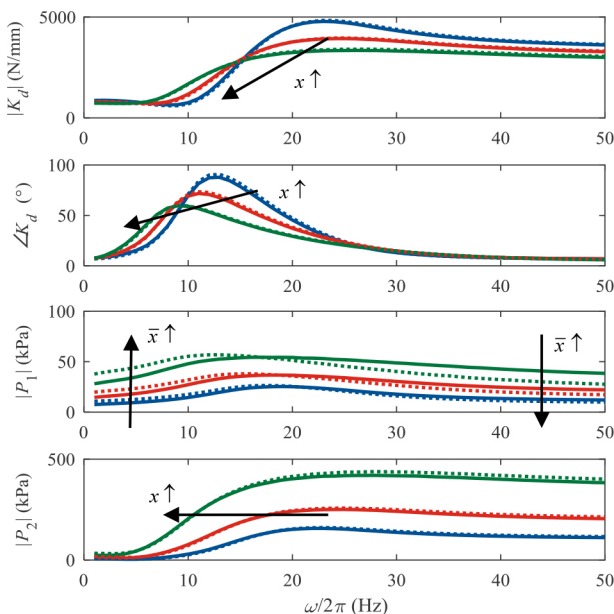


Figure 5. Dynamic stiffness, loss angle, and peak-to-peak chamber pressures from the laboratory experiment as a function of frequency. Key: solid, 2 mm mean; dotted, 3 mm mean; blue, 0.5 mm peak-to-peak; red, 1 mm peak-to-peak; green, 2 mm peak-to-peak.

Experimental results are shown in Figure 5. As is typical with many of these devices, there is a significant excitation amplitude dependence, with $|K_d|$ increasing with decreased excitation amplitude. The results also show a decrease in peak $\angle K_d$ and frequency at peak $\angle K_d$, as excitation amplitude increases. The $|K_d|$ and $\angle K_d$ have minimal dependence on mean displacement. The results show that the pressure in the top chamber is about seven times lower than in the bottom chamber. The pressure is also sensitive to the excitation amplitude, with a shift in the maximum $|P_1|$ and $|P_2|$ increasing with increased excitation amplitude. In addition, $|P_1|$ exhibits a noticeable dependence to mean displacement, with an increase at lower frequencies and a decrease at higher frequencies.

PARAMETER IDENTIFICATION

The effective inertance for this parallel track configuration is calculated as $I_{12} = \rho L_i / 2A_i$, where ρ is the density of the fluid, L_i is the length of one of the inertia tracks, and A_i is the cross-sectional area of each inertia track [3, 9]. The values of A_i and L_i are calculated from inspection of the solid model, shown in Figure 1, and ρ is an assumed value for a 55%/45% propylene-glycol/water mixture by volume at 25°C [10].

The lumped mass m_3 associated with the middle spring is estimated from the associated volume calculated from the solid model and an assumed density of the elastomeric material. Within the 1 to 50 Hz frequency range, this interfacial mass has negligible effect on dynamic stiffness and chamber pressures.

Due to the complex sheet metal entrance and exit geometries within the mount, the fluid resistance is calculated using a flow bench experiment, shown in Figure 6. The experiment consists of a flow control valve and pressure gage immediate upstream of the specimen. For convenience, city water is used as the test fluid. The flow control valve is adjusted to obtain desired inlet pressures, and the volumetric flow rate through the specimen is calculated from the measured time to fill a graduated cylinder under ambient pressure at the outlet of the specimen to a specified volume. This procedure is repeated for different inlet pressures, obtaining a pressure drop Δp to volumetric flow rate q curve, illustrated by the blue curve in Figure 7.

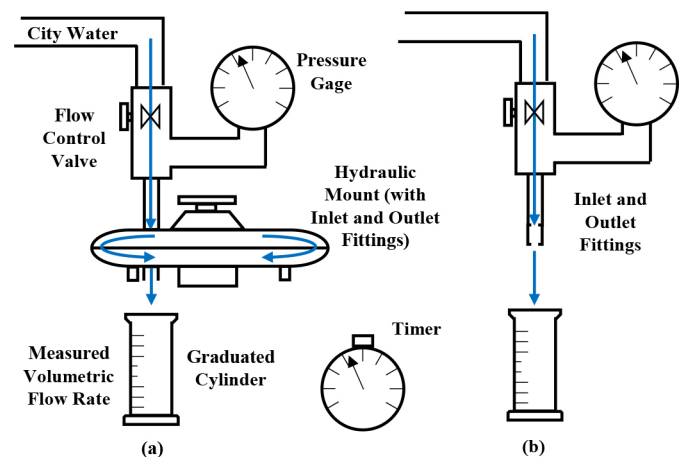


Figure 6. Flow bench experiment for (a) the hydraulic mount with inlet and outlet fittings and (b) the inlet and outlet fittings alone.