

Dynamic Stiffness of Hydraulic Bushing with Multiple Internal Configurations

Tan Chai, Rajendra Singh and Jason Dreyer
The Ohio State University

ABSTRACT

Fluid filled bushings are commonly used in vehicle suspension and sub-frame systems due to their spectrally-varying and amplitude-dependent properties. Since the literature on this topic is sparse, a controlled laboratory prototype bushing is first designed, constructed, and instrumented. This device provides different internal combination of long and short flow passages and flow restriction elements. Experiments with sinusoidal displacement excitations are conducted on the prototype, and dynamic stiffness spectra along with fluid chamber pressure responses are measured. The frequency-dependent properties of several commonly seen hydraulic bushing designs are experimentally studied and compared under two excitation amplitudes. Further, new linear time-invariant models with one long and one short flow passages (in parallel or series) are proposed along with the limiting cases. The analytical models are validated in frequency and time domains using transmitted force and chamber pressure measurements. The proposed formulation provides insights into various performance features.

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INTRODUCTION

Elastomeric and hydraulic bushings are widely used in vehicle suspension, body and engine sub-systems for improved handling and ride performance characteristics and to reduce vibration and noise [1,2,3,4,5,6]. Such devices are often expected to provide a high viscous damping coefficient (c) and stiffness (k) for large amplitude excitations at lower frequencies. Further, lower k and c values are needed for controlling structure-borne noise at moderate to higher frequencies [1,2,7]. Since elastomeric bushings cannot satisfy such conflicting requirements, many fluid-filled bushing designs have been developed and utilized in vehicles [1-2], as evident from some articles [7,8,9,10] and patents [11,12,13,14,15,16,17,18,19,20]. Even though many patents [11,12,13,14,15,16,17,18,19,20] claim certain performance features, they provide no analytical justifications or specific stiffness properties. Further, very few scientific articles have addressed the hydraulic bushing characterization and modeling issues [7,8,9,10]. Therefore, this article aims to fill this void and experimentally and analytically study the dynamic characteristics of a family of hydraulic bushing design.

A typical fluid-filled bushing consists of two almost identical elastomeric chambers between inner and outer metal sleeves. These chambers are usually filled with the anti-

freeze mixture, and they communicate fluid via a long passage (inertia track), and/or a short passage (or a controlled leakage orifice [1]). The relative deflection between the inner and outer metal sleeves causes chamber pressures to vary, and thus fluid flows back and forth through inertia track type and orifice-like passages, which provides effective damping over the desired frequency range, depending on the fluid system configuration and flow conditions. An accurate estimation of the dynamic stiffness of bushings is important for the prediction of dynamic loads in a vehicle sub-frame and suspension system. Although fluid-filled bushings exhibit frequency-dependent and amplitude-sensitive properties, a linear time-invariant (LTI) model must be developed first to estimate dynamic properties and to diagnose and tune such devices over the lower frequency regime (say from 1 to 50 Hz). Several different configurations of the hydraulic bushing should be experimentally and analytically examined to gain a better understanding of the underlying physics. Only radial damping is considered because the relative displacement between inner and outer metal parts is assumed to be only in the radial direction. Accordingly, specific objectives for this article are as follows: (a) Design a laboratory device with controlled internal configurations and conduct experiments on a production bushing and the laboratory prototype under sinusoidal excitation; (b) Analyze measured dynamic

stiffness results and compare the dynamic properties of different configurations and examine their spectrally-varying properties amplitude-dependence; (c) Develop linear time-invariant (LTI) models of a fluid-filled bushing with alternate configuration using the lumped fluid parameter approach; (d) Validate the proposed models by comparing predictions with measurements under sinusoidal excitation only.

EXPERIMENTAL STUDIES

The static and dynamic properties of a practical fluid-filled bushing (designated Y1) is evaluated first using the non-resonant elastomer test machine (MTS 831.50, [21]) under a mean load f_m . A sinusoidal displacement $x(t) = A \sin \omega t$ is applied to the inner metal part in the radial direction, from 1 to 50 Hz in 1 Hz increments under amplitude X , where ω is the excitation frequency (rad/s), and $X = 2A$ is the peak to peak (pp) amplitude. The force transmitted to the rigid base $f_T(t)$ is typically measured. The dynamic stiffness is defined as:

$$K_d(\omega) = |K_d(\omega)| \cos \phi_K(\omega) + i |K_d(\omega)| \sin \phi_K(\omega), \quad (1)$$

where $i = \sqrt{-1}$, and $|K_d(\omega)|$ and $\phi_K(\omega)$ are the magnitude and loss angle of $K_d(\omega)$, respectively. In this paper, results are presented in the dimensionless form, and thus $|K_d|$ is normalized by its static stiffness. The frequency ratio is defined as the excitation frequency ($f = \omega/2\pi$) divided by the frequency f_ϕ^* , at which the loss angle of a laboratory bushing prototype is maximum. One typical set of the measurements under three excitation amplitudes ($X = 0.1, 0.5$ and 1.0 mm (pp)) is shown in Figure 1. Significant amplitude and frequency dependence is observed from the $K_d(\omega)$ spectra. Both $|K_d|$ and ϕ_K decrease as X is increased. The loss angle achieves its peak at f_ϕ^* and then gradually decreases with ω , while $|K_d|$ continues to increase until $f_{|K|}^*$, which is about two times f_ϕ^* , and then settles down.

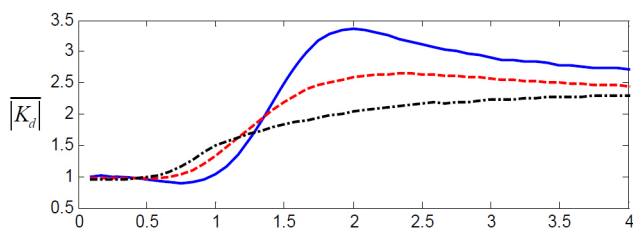


Figure 1.

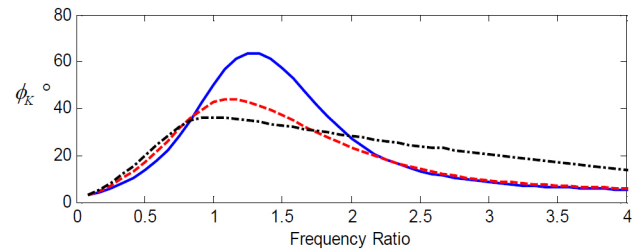


Figure 1 (cont). Measured dynamic stiffness spectra of a typical production bushing (Y1) for three excitation amplitudes (X). Key: —, $X = 0.1$ mm; - -, $X = 0.5$ mm; ···, $X = 1.0$ mm.

Fluid passages in practical bushings usually have irregular geometry and may be constructed with elastomer or metal. Moreover, experimentation with many bushing samples from different manufacturers would pose a difficult task. Thus, a laboratory device which can provide insights into various aspects is designed and fabricated for dynamic characterization and scientific examination. This device is able to provide different combinations of long fluid passage (inertia track) and short passage with adjustable flow restriction. Initial experiments are conducted on five configurations (designated B1 to B5, as listed in Table 1 and Figure 2) of this prototype using the MTS elastomer test machine. Additionally, two limiting cases (B0 with all the fluid passages closed and BR as the drained bushing) are also evaluated in the experiment. Similar to the dynamic stiffness measurements of the production bushing, a sinusoidal displacement excitation $x(t) = A \sin \omega t$ is applied to the prototype device under a mean load. The dynamic stiffness $K_d(\omega)$ spectra, along with the steady state time histories of the transmitted force $f_T(t)$ are measured from 1 to 60 Hz with 1 Hz increment at two amplitudes, $X = 0.1$ and 1 mm. In addition, the internal pressures of two hydraulic chambers, $p_1(t)$ and $p_2(t)$, are measured by the dynamic pressure transducers installed within the fluid chambers of the prototype device.

Table 1. Configurations of the prototype hydraulic bushing

Configuration	Description (Also see Fig. 2)
B1	One long flow passage of diameter d_i
B2	One short fluid passage with a restriction of diameter d_o
B3	One short fluid passage with a restriction of diameter $d_o/3$
B4	One long passage and one short passage with a restriction of diameter d_o in parallel
B5	One long passage and one short passage with a restriction of diameter $d_o/3$ in parallel
B0	All fluid passages closed
BR	Drained bushing

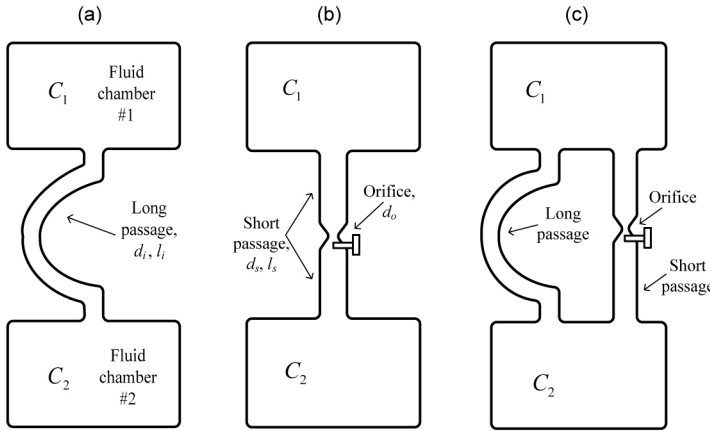


Figure 2. Multiple configurations of the hydraulic bushing constructed for controlled laboratory studies. (a) Configuration B1 with long fluid passage of diameter d_i ; (b) Configuration B2 with short flow passage with restriction of diameter d_o and configuration B3 of diameter $d_o/3$; (c) Configuration B4 with parallel long and short flow passages with d_o and B5 with $d_o/3$.

ANALYTICAL MODELS

Fluid Model I with Two Parallel Flow Passages

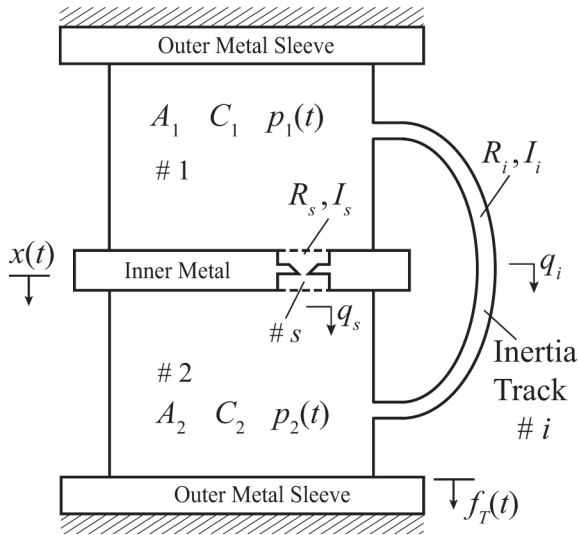


Figure 3. Fluid model I of the hydraulic path with a long flow passage (inertia track, #i) and a short flow passage with restriction (#s).

A fluid system model of the hydraulic path of a bushing with two passages in parallel is proposed in Figure 3, with many simplifying assumptions, based on lumped parameter method. These include an internal long passage such as the inertia track (#i) though shown as external tubing for the sake of clarity in Figure 3, and/or a short passage (#s) with a flow

restriction or controlled leakage element (such as the orifice). Assuming the linear system, the long and short flow passages are represented by the fluid inertances I_i and I_s and fluid resistance R_i and R_s , respectively. The compliances of two fluid chambers (#1 and #2) are C_1 and C_2 , respectively, with effective pumping areas A_1 and A_2 . The rubber element (#r, parallel with the hydraulic path, not shown in Figure 3) is modeled by rubber stiffness k_r and viscous damping coefficient c_r .

Static and dynamic displacement excitations are applied to the inner metal part while the outer metal sleeve is relatively fixed. The mean force $\bar{f}$ transmitted to the outer sleeve under a static displacement $\bar{x}$ is as follows, where $\bar{p}$ is the fluid chamber pressure under static equilibrium:

$$\bar{f} = k_r \bar{x} + (A_2 - A_1) \bar{p}. \quad (2)$$

The dynamic displacement excitation $x(t)$, from the static equilibrium, is applied to the bushing under a mean load $\bar{f}$. By applying the continuity equation to two fluid control volume elements (1 and 2), the following equations are obtained, where $q_i(t)$ and $q_s(t)$ are the volumetric flow rates through the long track and short passage, and $p_1(t)$ and $p_2(t)$ are the dynamic pressures inside the two fluid chambers,

$$-A_1 \dot{x}(t) - q_i(t) - q_s(t) = C_1 \dot{p}_1(t), \quad (3a)$$

$$A_2 \dot{x}(t) + q_i(t) - q_s(t) = C_2 \dot{p}_2(t). \quad (3b)$$

The application of the momentum equation to two flow passages (i and s) yields the following:

$$p_1(t) - p_2(t) = I_i \dot{q}_i(t) + R_i q_i(t), \quad (4a)$$

$$p_1(t) - p_2(t) = I_s \dot{q}_s(t) + R_s q_s(t). \quad (4b)$$

The dynamic force is transmitted to the outer sleeve by rubber and hydraulic paths. Thus, the total transmitted force $f_T(t)$ is defined below where $f_{Tr}(t)$ is the rubber path and $f_{Th}(t)$ the hydraulic path force:

$$f_T(t) = f_{Tr}(t) + f_{Th}(t) = (k_r x(t) + c_r \dot{x}(t)) + (A_2 p_2(t) - A_1 p_1(t)). \quad (5)$$

By transforming Eqs. (3), (4), (5) to the Laplace (s) domain and assuming zero initial conditions, the following dynamic stiffness $K_d(s)$ is obtained, where $K_{dh}(s)$ and $K_{dr}(s)$ are the dynamic stiffness of the hydraulic and rubber path, respectively, and $Z_i = I_i s + R_i$ and $Z_s = I_s s + R_s$ are the fluid