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Variable volumetric stiffness fluid mount design

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Abstract. Passive fluid mounts are commonly used in the automotive and aerospace applications to isolate the cabin from the engine noise and vibration. Due to manufacturing and material variabilities, no two identical fluid mount designs act the same. So, fluid mounts are tuned one by one before it is shipped out to customers. In some cases, for a batch of fluid mounts manufactured at the same time, one is tuned and the rest is set to the same settings. In some cases they are shipped as is with its notch frequency not being in its most optimum location. Since none of the passive fluid mount parameters are controllable, the only way to tune the mount is to redesign the mount by changing fluid, changing inertia track length or diameter, or changing rubber stiffness. This trial and error manufacturing process is very costly. To reduce the fluid mount notch frequency tuning cycle time, a new fluid mount design is proposed. In this new fluid mount design, the notch frequency can be easily modified without the need for any redesigns. In this paper, the new design concept, and its mathematical model and simulation results will be presented.

1. Introduction

Passive fluid mounts are widely used in the automotive and aerospace applications to reduce or control cabin noise and vibration. The fluid mount is placed in between the engine and the fuselage or the car frame and tuned to a particular frequency, called "notch frequency". It is at this frequency where the dynamic stiffness of the fluid mount is the lowest; therefore, greatest cabin noise and vibration reduction is obtained. The design location of the notch frequency depends on the application, but with most applications, the "notch frequency" is designed to coincide with the longest period of constant speed. For example, in the case of aerospace applications, the notch frequency maybe designed to coincide with the cruise speed rather than the take off and the landing speeds. Since most of the airplane flight time is spent at the cruise speed, it makes most sense to reduce the cabin noise and vibration at this speed rather than during the take off or landing.

To obtain greatest cabin noise and vibration reduction at the cruise speed (or frequency), the notch frequency needs to be as close to the cruise frequency as possible or else cabin noise and vibration reduction may not be optimum. So to achieve greatest cabin noise and vibration reduction, fluid mount manufacturers make great efforts to tune the fluid mount notch frequency to the desired frequency. Unfortunately, due to tolerances on all the fluid mount dimensions, material property variations, and variation in elastomer molding processes, no two identical fluid mount designs have the same notch frequency on the first manufacturing pass. So the fluid mount parameters such as inertia track length or diameter, or fluid density, or other properties are varied till the mount is tuned to the correct

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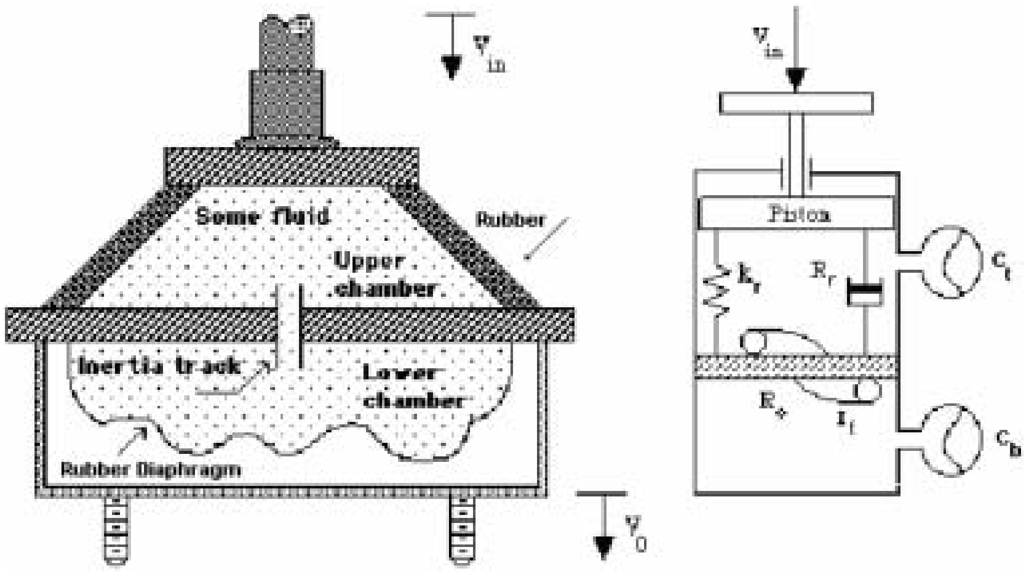

Fig. 1. A Typical Single Pumper Fluid Mount and its Physical Model.

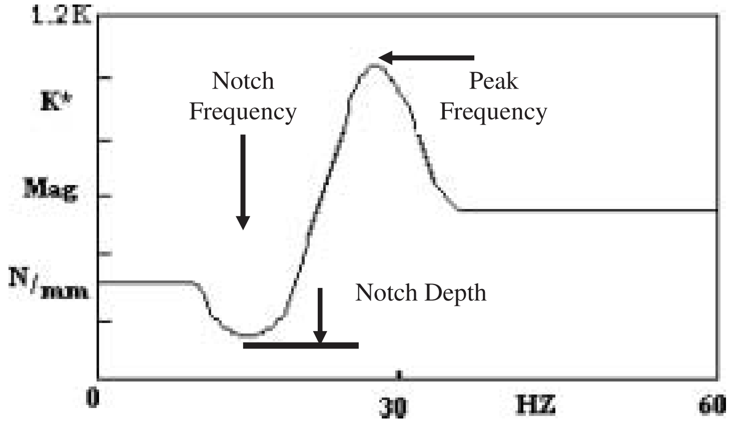

Fig. 2. Dynamic Stiffness of a Typical Fluid Mount versus Frequency.

notch frequency. This manufacturing process is very costly.

Here in this paper, a new fluid mount design is described such that the notch frequency can be easily tuned without the need to disassemble or redesign the mount. The focus of this paper will be on single pumper fluid mounts.

1.1. Single pumper fluid mounts

Fluid-filled (or fluid) mounts have been referred to in many publications by many different names, such as hydraulic mounts [1,4,6,8], hydroelastic mounts [9], or Fluidlastic mounts [5]. There are two types of fluid mounts, double pumper fluid mounts [10] and single pumper fluid mounts [3,10]. The focus of this paper will be on single pumper fluid mounts since it is widely used in the automotive and the helicopter applications.

A passive fluid mount, as shown in Fig. 1, consists of a fluid contained in two elastomeric cavities (or fluid chambers) that are connected together through an inertia track. When a sinusoidal motion is applied to the fluid mount, the fluid will oscillate between the two fluid chambers. The oscillating fluid having mass, bounces between the two chamber volume stiffnesses and the vertical (or axial) stiffness, and eventually goes to resonance at a frequency called "notch frequency". At this frequency, the fluid mount dynamic stiffness decreases considerably and thus the transmitted force;

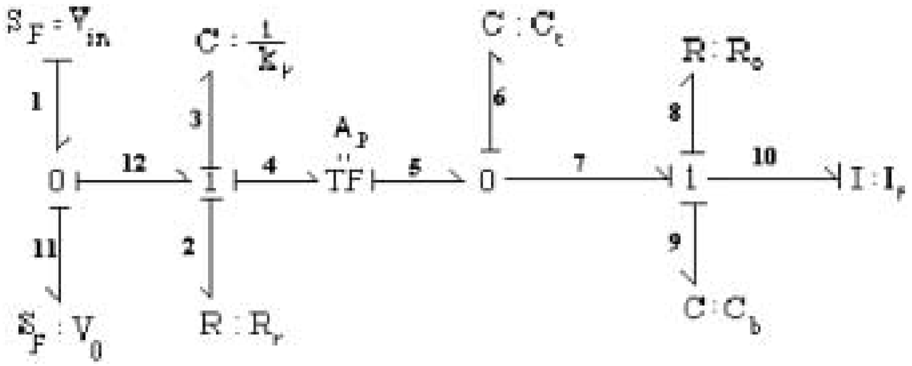

Fig. 3. Bond Graph Model of the Single Pumper Fluid Mount of Fig. 1.

therefore, providing cabin noise and vibration reduction. To place the "notch frequency" to a desired location, the fluid mount designer needs to use appropriate inertia track length, diameter, fluid density and viscosity, and rubber stiffnesses. Figure 2 shows a typical dynamic stiffness of a passive fluid mount versus frequency.

Figure 3 shows the bond graph [7] model of Fig. 1. It is important to mention that the volumetric damping of the rubber in the bulge direction has not been considered in this bond graph model. From the bond graph model, the following state space equations can be derived:

$$\dot{q}_3 = V_{\rm in} - V_0 \tag{1}$$

$$\dot{q}_6 = A_p(V_{\rm in} - V_0) - \frac{P_{10}}{I_{10}} \tag{2}$$

$$\dot{q}_9 = \frac{P_{10}}{I_{10}} \tag{3}$$

$$\dot{P}_{10} = \frac{q_6}{C_6} - \frac{q_9}{C_9} - R_8\frac{P_{10}}{I_{10}} \tag{4}$$

In the above state space equations, q_3, q_6, and q_9 are the generalized displacement variables, and P_{10} the momentum variable. Physically, q_3 represents the relative displacement across the mount $(x_{\rm in} - x_0)$, q_6, and q_9 variables represent volume change in the upper and lower fluid chambers, and the P_{10} represents the pressure momentum (integral of pressure) in the inertia track. The derivative of P_{10} represents the fluid pressure drop $(\Delta P = I_f dQ/dt)$ in the inertia track due to the fluid mass.

In the bond graph model of Fig. 3, it is assumed that the pressure drop (due to flow losses) in the inertia track is a linear function of flow rate $(\Delta P = RQ)$, the effective piston area is a constant and not a function of frequency or deflection, the volumetric stiffnesses are constants and not functions of frequency and strain, and volumetric damping assumed negligible.

If one end of the fluid mount is held fixed $(V_0 = 0)$ and sinusoidal displacement is applied to the other end and force is measured, we get the following input force (effort on bond 1, see Fig. 3):

$$F_{\rm in} = \frac{q_3}{C_3} + R_2V_{\rm in} + A_p\frac{q_6}{C_6} \tag{5}$$

The dynamic stiffness (defined as $K^* = F_{\rm in}/X_{\rm in}$) is equal to:

$$K^* = K'_r + \frac{K''_r}{\omega}S + A_p^2K'_{vt} \frac{S^2 + \frac{R_0}{I_f}S + \frac{K'_{vb}}{I_f}}{S^2 + \frac{R_0}{I_f}S + \frac{K'_{vt}+K'_{vb}}{I_f}} \tag{6}$$

In the above equations, the following symbols are defined:

V_0 and $V_{\rm in}$	Velocities across the mount, m/s
A_p	Effective piston area, m^2
A_t	Inertia track area, m^2
I_f	Inertia track fluid inertia, same as I_{10}, N-S^2/m^5
R_0	Inertia track flow resistance, same as R_8, N-S/m^5
K_{vt}	Top chamber volumetric or bulge Stiffness, $(C_6 = 1/Kvt)$, N/m^5
K_{vb}	Bottom chamber volumetric or bulge stiffness, $(C_9 = 1/Kvb)$, N/m^5
$K'r$	Real component of the vertical stiffness, $(C_3 = 1/K'r)$, N/m
$K''r$	Imaginary component of the vertical stiffness, $(R_2 = K''r/\omega)$, N/m
*	Circular frequency, rad/s

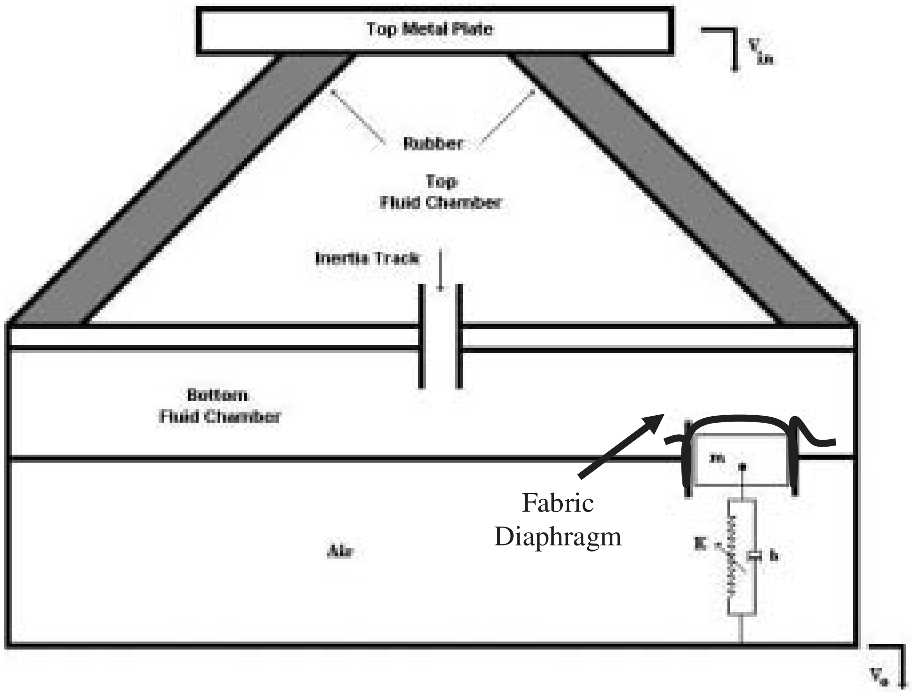

Fig. 4. Variable Spring Rate Fluid Mount Design.

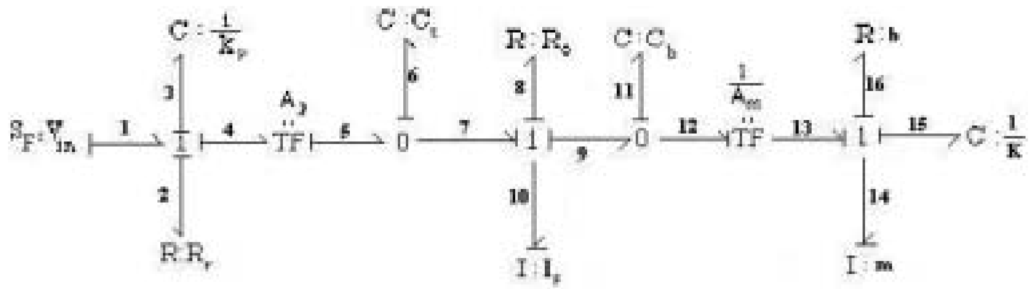

Fig. 5. Bond Graph Model of Fig. 4.

S Equal to jω

The fluid inertia is equal to:

$$I_f = \frac{\rho L}{A_t} \tag{7}$$

where ρ and L are:

ρ Fluid Density, kg/m^3
L Inertia Track Length, m

The above parameters are the fluid mount parameters that a designer uses to design a fluid mount. The fluid mount notch frequency, for a single pumper fluid mount assuming zero rubber damping and no inertia track flow losses, is approximately equal to:

$$f_{\text{notch}} = \frac{1}{2\pi}\sqrt{\frac{A_p^2 K_{vt} K_{vb} + K_r(K_{vt} + K_{vb})}{I_f(K_r + A_p^2 K_{vt})}} \tag{8}$$

Equation (8) shows that the notch frequency depends on the piston area, A_p, on the top and bottom chamber volumetric stiffnesses, K_{vt} and K_{vb}, axial stiffness K_r, and fluid inertia, I_f. These parameters that we just defined above can be used for notch frequency placement, but some parameters are easier to vary than others. To understand which parameters are easier to vary and which ones are not, we will need to define them in more details.

It is important to mention that when we talk about varying a parameter, we have two choices, one through a redesign or one through a parameter modulation. The

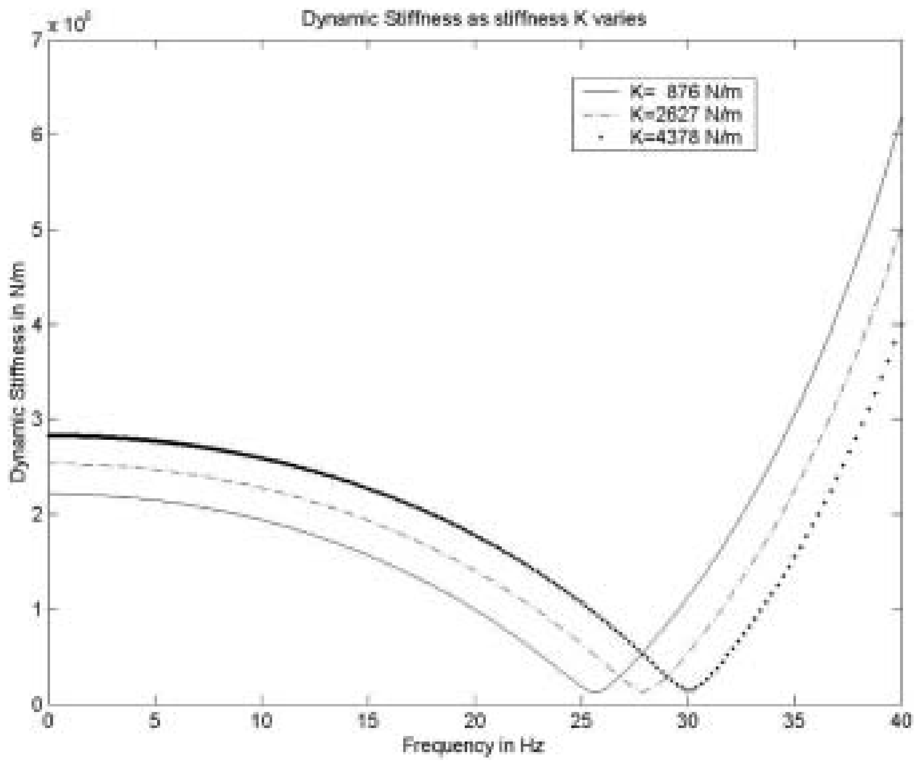

Fig. 6. Dynamic Stiffness as Spring K varies (MATLAB simulation).

emphasis of this paper is on the 2nd approach, but in the fluid mount parameters discussion section, we will discuss both approaches. It is important to also mention that parameter modulation can be automatic or manual. Automatic parameter modulation in real-time is wonderful to have, but it makes the design more expensive since it requires sensors and a controller.

In this paper, the new fluid mount design that is described will be such that the modulated parameter can be varied both in real-time or non real-time but manually; thus, requiring no sensors and a controller.

1.2. Fluid mount parameters

1.2.1. Piston area

If we consider the fluid mount design of Fig. 1, the conical shaped rubber component not only acts like a spring (K'_r, K''_r) but also acts like a piston pumping fluid between the upper and lower fluid chambers. In a sense we can think of the conical shaped rubber component as a piston with an effective area, A_p, pumping fluid in between the two fluid chambers.

To vary piston area through a redesign, involves modification to the design of the rubber component and therefore the rubber mold, which is very costly. It is also important to mention that when one changes the conical shaped rubber design for the purpose of varying piston area, axial and top chamber volume stiffnesses change too.

It is the belief of the authors that it is impractical to vary piston area in real-time (when rubber is providing the pumping action) since piston area, axial stiffness, and volume stiffness are coupled parameters. A change in one parameter affects other parameters.

1.3. Vertical (or axial) stiffness and top chamber volume stiffness

For the same reason we just described for the piston area, one cannot easily vary vertical (or axial) stiffness without impacting other parameters. A change in any of the stiffnesses such as vertical stiffness (K'_r) or top chamber volume stiffness, results in a change in piston area (A_p) too.

All commercially available fluid mount designs, use conventional passive elastomers as the load carrying element. To create a semi-active fluid mount design, Magneto-rheological (MR) elastomers maybe considered as a replacement of conventional passive elastomers. With MR-elastomers, the stiffness of the MR

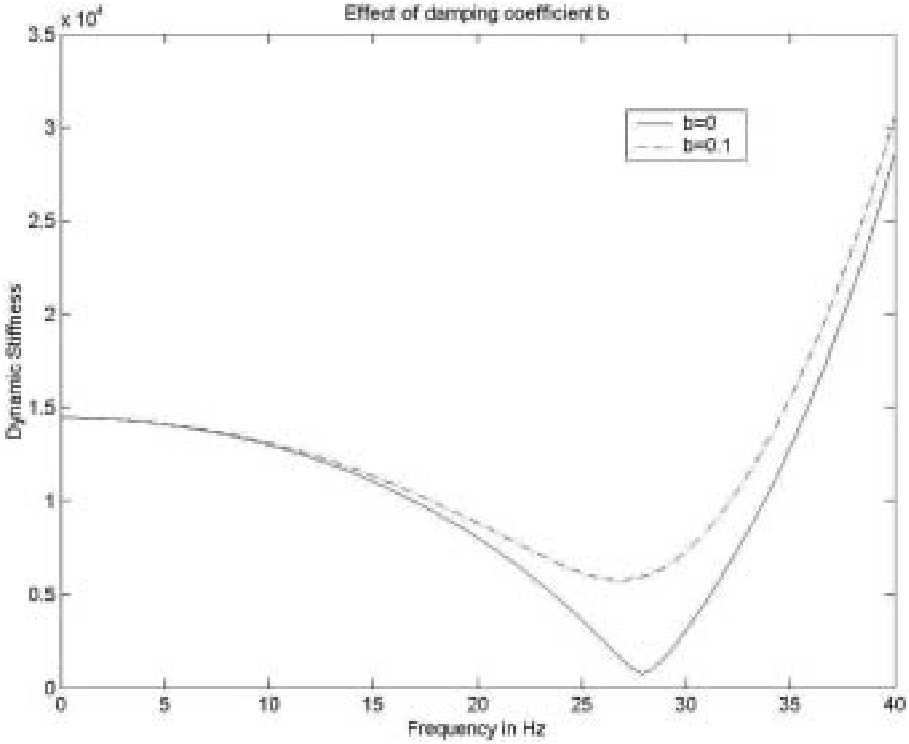

Fig. 7. Effect of Damping Coefficient, b, on Dynamic Stiffness.

rubber can be varied by varying magnetic field, thus the notch frequency, but one needs to bear in mind that coupling of the piston area, vertical stiffness, and volumetric stiffness parameters still remains using MR-elastomers.

Another issue to also bear in mind is that with MR-elastomers, damping increases with increase in the magnetic field. Increase in damping results in the loss of the notch frequency. So, in general, MR-elastomers maybe inappropriate to use with semi-active fluid mount isolators, but very appropriate to use with semi-active fluid dampers. The focus of this paper is on fluid mount isolators.

If one elects to change piston area, or volume stiffness, or axial stiffness through a rubber redesign for the purpose of notch frequency placement, one needs to understand that each rubber redesign implies a redesign of the rubber mold, which is very costly.

1.3.1. Inertia track length, diameter, and fluid density

It may appear that the easiest parameters to vary, for notch frequency placement, are the inertia track length, the track diameter, or the fluid density since it does not involve rubber redesign. But this is not entirely true. Redesign of the inertia track involves re-machining of the inertia track, which could be costly, and possibly can involve increase in flow losses. Currently it is a common practice by most fluid mount manufacturers to tune fluid mounts by changing inertia track length, diameter, or fluid, but varying these parameters can cause problems with the notch depth (see Fig. 2). To provide the best vibration and noise isolation, we would ideally like to have zero dynamic stiffness at the notch frequency. The notch depth greatly depends upon fluid damping and to a lesser degree to rubber damping. By making the inertia track length longer or diameter smaller, one can directly impact flow losses and therefore the notch depth. Changing fluid is also not always a practical thing to do since changing fluid changes fluid viscosity and that can impact notch depth.

1.4. Bottom chamber volume stiffness

From all of the fluid mount parameters; it is the belief of the authors that by far the best and easiest parameter to vary is the bottom chamber volumetric stiffness and that is because the bottom chamber volume stiffness is not coupled to any other parameter and it can be readily varied with the design that the authors will describe.

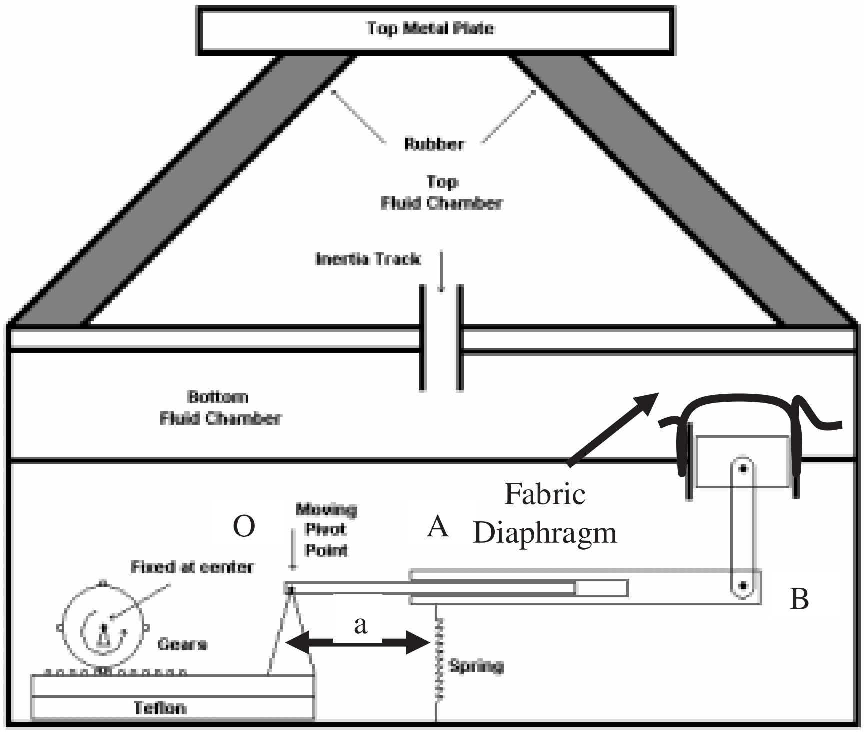

Fig. 8. Variable Bottom Chamber Volume Stiffness Fluid Mount Design.

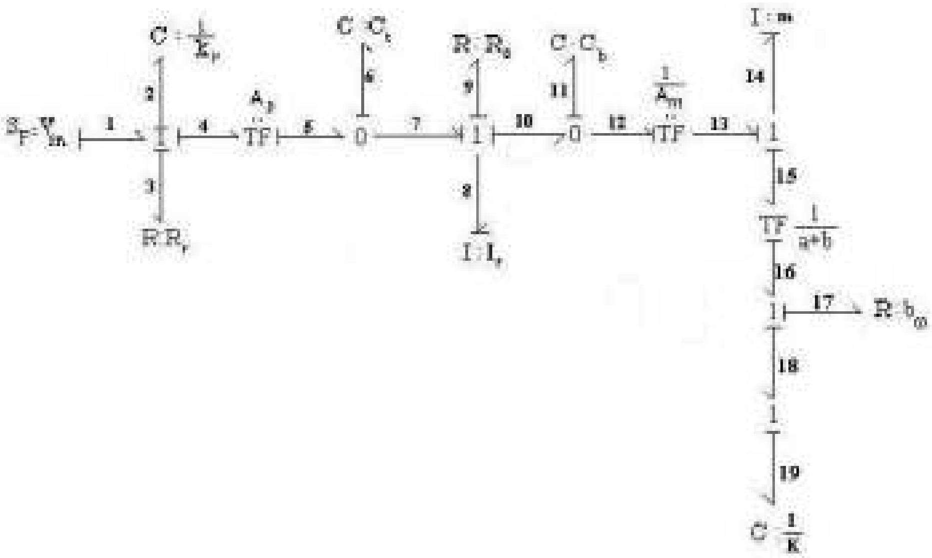

Fig. 9. Bond Graph Model of Fig. 8.

1.5. *Definition of volumetric stiffness*

When the fluid pressure in a cavity increases, the rubber is forced to balloon out, or bulge out. So the change in cavity pressure divided by the change in cavity volume is defined as the volumetric stiffness. The volumetric stiffness equation can be described as,

$$K_v = \frac{\Delta P}{\Delta V} \tag{9}$$

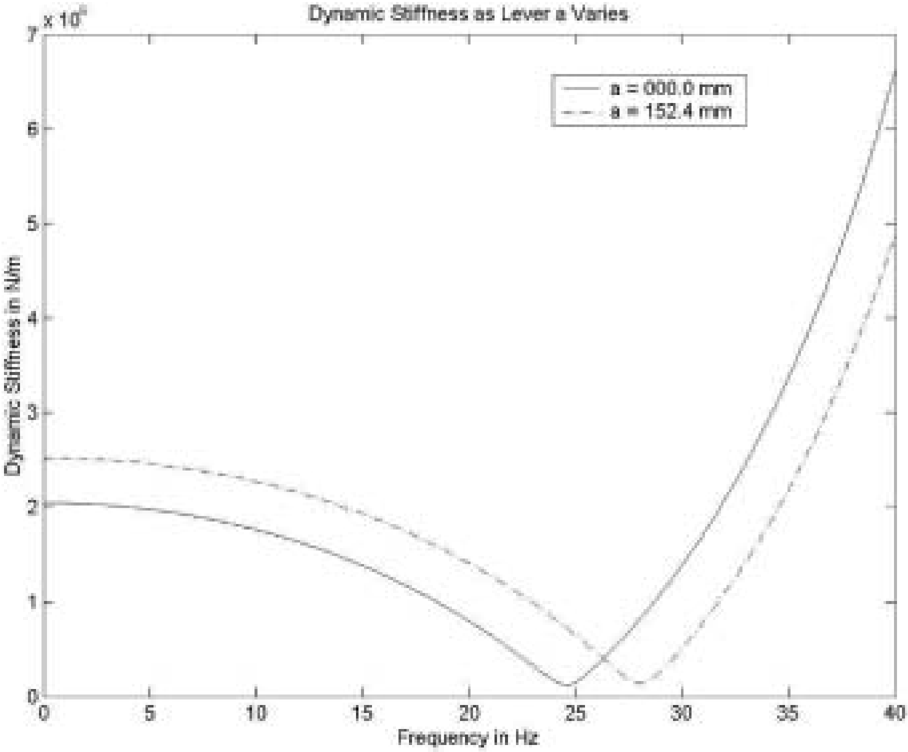

Fig. 10. Dynamic Stiffness as Parameter a is Varied from 0 to 152.4 mm (0 to 6 inches).

1.6. Variable spring fluid mount design

In the previous sections, the authors concluded that the easiest parameter to vary in real-time is the bottom chamber volumetric stiffness. To demonstrate this design concept, the model of Fig. 4 will be used. Figure 4 shows a piston (labeled as m) that is supported by a spring K and a damper b in the bottom fluid chamber. A fabric diaphragm (with high volumetric stiffness) is placed on the top of the piston, sealing the bottom fluid chamber. When the fluid enters the bottom fluid chamber, the increase in the fluid pressure forces the diaphragm to inflate or to displace, and this displacement causes the piston, supported by a spring and a damper, to move. Basically, the fabric diaphragm transfers power from the bottom fluid chamber to the mass, spring, and damper system without impacting the volumetric stiffness of the bottom fluid chamber.

In this design we assume that we are able to vary the stiffness of the spring K. By vary this spring rate we can vary the volumetric stiffness of the bottom chamber; therefore, the notch frequency location. Let's assume that one end of the mount is fixed $(V_o = 0)$ and the other end subject to an input velocity V_{in}. The bond graph model of Fig. 4 is shown in Fig. 5.

The state space equations, obtained from the bond graph model of Fig. 5, will be as follows:

$$\dot{q}_3 = V_{\text{in}} \tag{10}$$

$$\dot{q}_6 = A_p V_{\text{in}} - \frac{P_{10}}{I_{10}} \tag{11}$$

$$\dot{P}_{10} = \frac{q_6}{C_6} - \frac{q_{11}}{C_{11}} - R_8 \frac{P_{10}}{I_{10}} \tag{12}$$

$$\dot{q}_{11} = \frac{P_{10}}{I_{10}} - A_m \frac{P_{14}}{I_{14}} \tag{13}$$

$$\dot{P}_{14} = A_m \frac{q_{11}}{C_{11}} - \frac{q_{15}}{C_{15}} - R_{16} \frac{P_{14}}{I_{14}} \tag{14}$$

$$\dot{p}_{15} = \frac{P_{14}}{I_{14}} \tag{15}$$

The input force is equal to:

$$F_{\text{in}} = \frac{q_3}{C_3} + R_2 V_{\text{in}} + A_p \frac{q_6}{C_6} \tag{16}$$

In the above state space equations, q_3, q_6, q_{11}, and q_{15} are the generalized displacement variables, and P_{10} and P_{14} are the momentum variables.

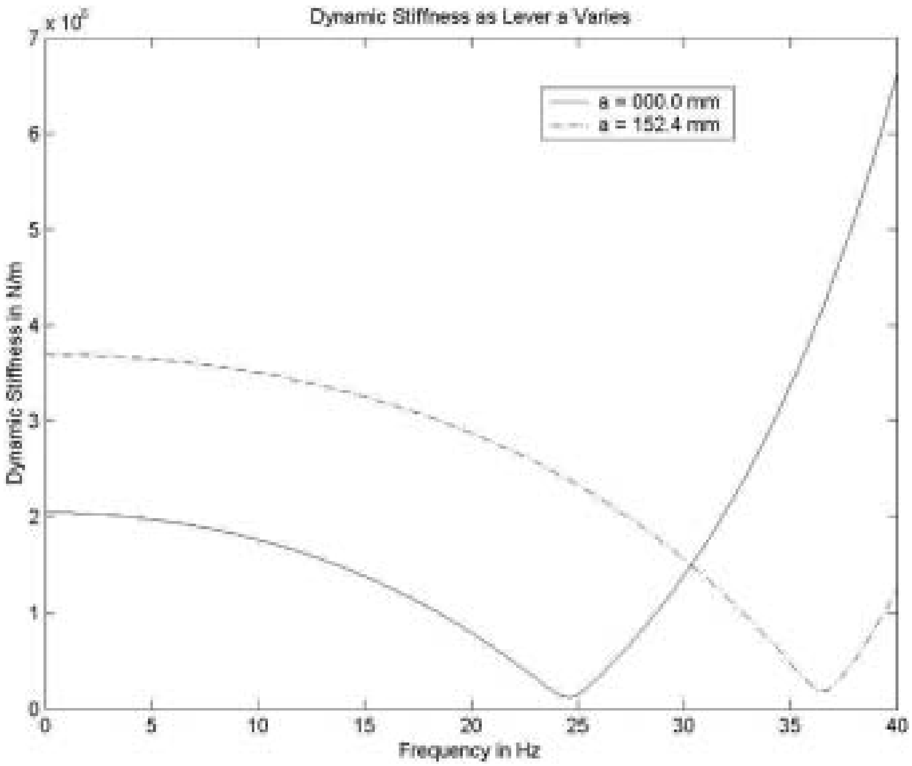

Fig. 11. Dynamic Stiffness with $A_m = 322.6$ mm^2 (0.5 in^2) and $0 < a < 152.4$ mm.

To simulate the design of Fig. 4, MATLAB Program and the above State Space equations were used. The following parameters, typical of an automotive fluid mount parameters, were used for the MATLAB simulation.

- V_{in} Input velocity, m/s
- A_p Effective piston area, 0.009 m^2
- A_t Inertia track area, 0.000366 m^2
- A_m Area of the mass m, 6.45E-4 m^2
- I_f Inertia track fluid inertia, equal to I_{10}, 8.564E5 N-S^2/m^5
- R_o Inertia track flow resistance, equal to R_8, 6.4E6 N-S/m^5
- K_{vt} Top chamber volumetric or bulge Stiffness, $(C_6 = 1/K_{vt})$, 1.1E11 N/m^5
- K_{vb} Compressibility of the fluid, $(C_{11} = 1/Kvb)$, 2.1E14 N/m^5
- K Stiffness of spring K, $(C_{15} = 1/K)$, 876 N/m, 2627 N/m, and 4378 N/m
- K'_r Real component of the vertical stiffness, $(C_3 = 1/K'_r)$, 2.05e6 N/m
- K''_r Imaginary component of the vertical stiffness, $(R_2 = K''_r/\omega)$, 0 N/m
- m Mass m, equal to I14, 0.00453 kg
- R_{16} Damping coefficient b, equal to 0 N-s/m

Figure 6 shows that as the stiffness of the spring rate K is varied from 876 N/m to 4378 N/m (therefore varying bottom chamber volume stiffness), the notch frequency varies from 25 Hz to 30 Hz. Figure 6 also shows that the static stiffness is increased with increase in spring rate K. The relationship between spring rate K and the bottom chamber volume stiffness is given by:

$$K_{vb} = \frac{K}{A_m^2} \tag{17}$$

Figure 7 indicates that the damping coefficient b has to be kept to a minimum if we need to maintain a descent notch depth. In general, to maintain a descent notch depth, all dampings need to be kept to a minimum.

So to create a variable stiffness fluid mount design, we need to develop a design where we can easily vary stiffness K, keep damping coefficient b to a minimum, and keep mass m reasonably small.

One of the ways of to do this, is to have an airbag (an air spring) in place of the spring K and vary the airbag pressure to vary stiffness. But one of disadvantages of an airbag is that one needs to maintain pressure for long

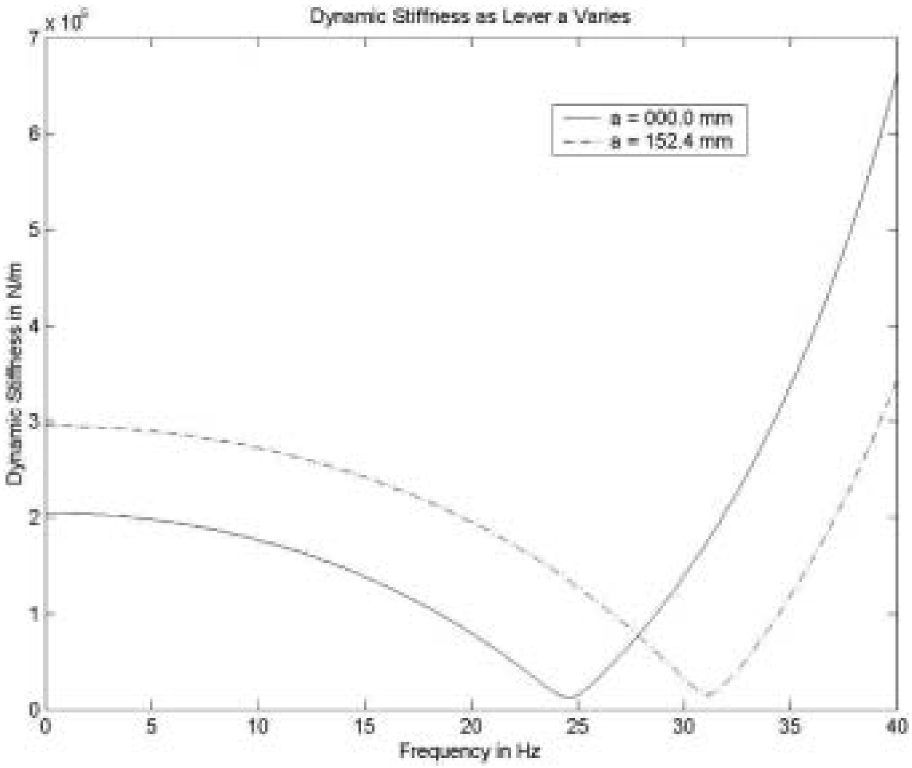

Fig. 12. Dynamic Stiffness versus Frequency with $K = 7004$ N/m (40 lbf/in).

periods of time and that could be a problem. The other problem with airbags is that their stiffness is non-linear and a function of deflection; so, notch frequency can vary as deflection to the mount is increased.

1.7. Variable bottom chamber volume stiffness fluid mount design

Figure 8 shows a possible fluid mount design where the volume stiffness of the bottom chamber can be varied by moving the fulcrum (or the pivot point) in real-time. Figure 8 shows a pinion and rack gear system. The rack's bottom surface is coated with Teflon for the purpose of ease of sliding. The pinion rotates about its fixed center. When one rotates the pinion, the rack moves. This movement causes the fulcrum (or the pivot point) of the lever to change; therefore, the parameter a. When the distance a is varied, the apparent stiffness at B is changed; so does the volume stiffness of the bottom chamber. One can manually rotate the pinion, or have a small DC motor rotate the pinion as necessary.

The bond graph model of Fig. 8 is shown in Fig. 9. From the bond graph model of Fig. 9, the following state space equations can be obtained:

$$\dot{q}_2 = V_{\text{in}} \tag{18}$$

$$\dot{q}_6 = A_p V_{\text{in}} - \frac{P_8}{I_8} \tag{19}$$

$$\dot{q}_8 = \frac{q_6}{C_6} - \frac{q_{11}}{C_{11}} - R_9 \frac{P_8}{I_8} \tag{20}$$

$$\dot{q}_{11} = \frac{P_8}{I_8} - A_m \frac{P_{14}}{I_{14}} \tag{21}$$

$$\dot{P}_{14} = A_m \frac{q_{11}}{C_{11}} - \frac{R_{17}}{(a+b)^2} \frac{P_{14}}{I_{14}} - \frac{a}{a+b} \frac{q_{19}}{C_{19}} \tag{22}$$

$$\dot{q}_{19} = \frac{a}{a+b} \frac{P_{14}}{I_{14}} \tag{23}$$

The output equation is given as follows:

$$F_{\text{in}} = \frac{q_2}{C_2} + R_3 V_{\text{in}} + A_p \frac{q_6}{C_6} \tag{24}$$

To simulate the design of Fig. 8, MATLAB Program and the above State Space equations were used. The following parameters were used for the MATLAB simulation.

A_p Effective Piston area, 0.009 m^2

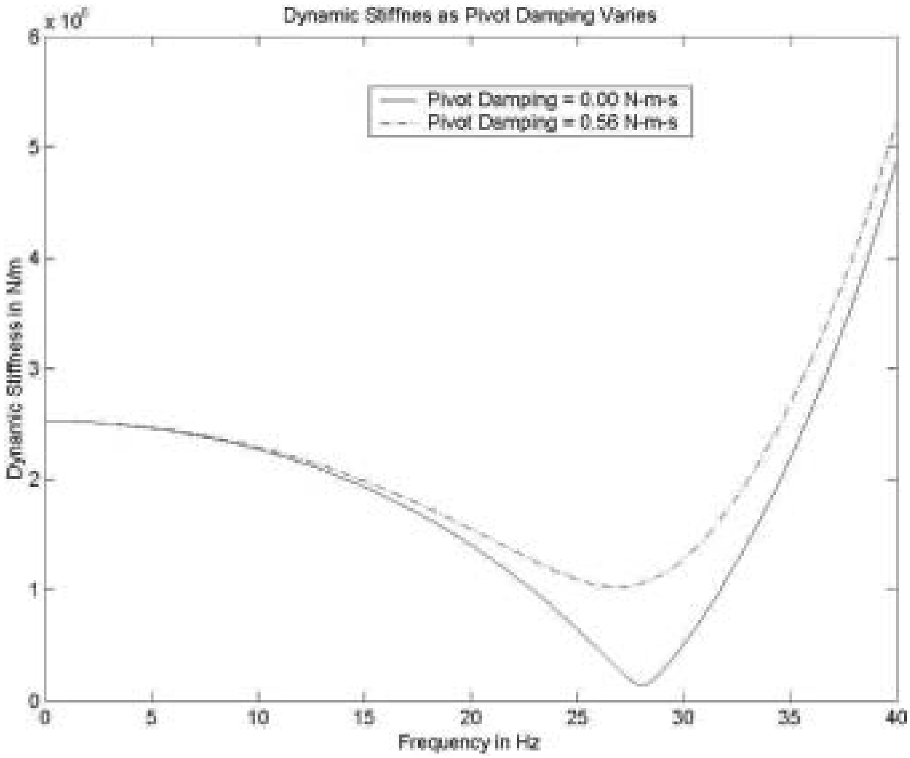

Fig. 13. Dynamic Stiffness versus Frequency as Pivot Damping Varies.

- A_t Inertia track area, 0.000366 m^2
- A_m Cross sectional area of the mass m, 6.45E-4 m^2
- I_f Inertia track fluid inertia, equal to I_8, 8.564E5 N-S^2/m^5
- R_o Inertia track flow resistance, equal to R_9, 6.4E6 N-S/m^5
- K_{vt} Top Chamber Volumetric or bulge Stiffness, $(C_6 = 1/K_{vt})$, 1.1E11 N/m^5
- K_{vb} Compressibility of the fluid, $(C_{11} = 1/K_{vb})$, 2.1E14 N/m^5
- K Stiffness of spring K, $(C_{19} = 1/K)$, 3502 N/m
- K'_r Real component of the vertical stiffness, $(C_2 = 1/K'r)$, 2.05e6 N/m
- K''_r Imaginary component of the vertical stiffness, 0 N/m
- m Mass m, equal to I_{14}, 0.00453 kg
- a Lever length from O to A, varied from 0 to 152.4 mm
- b Lever arm from A to B, set to 25.4 mm
- R_{17} Damping at the pivot point O, equal to b_ω

In the simulation, the parameter b is set to 25.4 mm (1 inch) and the lever arm a is varied from 0 to 152.4 mm (0 to 6 inches). Figure 10 shows the dynamic stiffness of the new fluid mount design versus frequency as a is varied. The simulation shows that the notch frequency moves about 5 Hz as a is varied from 0 to 152.4 mm (0 to 6 inches). The effective volume stiffness of the bottom chamber in this new design is given by:

$$K_{vb} = \frac{a^2 K}{(a+b)^2 A_m^2} \tag{25}$$

In some situations, it may not be permissible to have a lever as long as 177.8 mm ($a+b = 7$ inches). To make the bottom chamber volume stiffness more sensitive to changes in the parameter a, we can set the spring rate K to a larger value or make the area A_m smaller. Figure 11 shows that when A_m is reduced to 322.6 mm^2 (0.5 in^2), the notch frequency location can be varied from 25 to 37 Hz for the same variation in the parameter a. Figure 12 shows that when K is increased to 7004 N/m (40 lbf/in), the notch location can be varied from 25 Hz to 32 Hz for the same variation in the parameter a. Basically, Figs 11 and 12 indicate that if one needs to have a smaller lever arm due to space constraints, but needs the same notch frequency modulation capability, one can increase stiffness K or reduce area A_m.

In Figs 10 through 12, the pivot damping was set to zero, but in Fig. 13, the damping is increased from 0 to 0.56 N-m-s/rad (5 in-lb-sec/rad). Figure 13 shows that when pivot damping is increased, the notch depth is reduced. This result indicates that the pivot damping (or friction at the pivot) needs to be kept small, ideally zero if possible.

2. Conclusions

Due to manufacturing and material variabilities, no two identical fluid mount designs act the same. So, fluid mounts are tuned one by one before they are shipped to the customers. Since none of the passive fluid mount parameters are controllable, the only way to tune or retune the mount is to redesign the mount. This trial and error manufacturing process is very costly. To reduce the fluid mount notch frequency tuning cycle time, a new fluid mount design was proposed.

The variable bottom chamber volumetric stiffness fluid mount design, described in this paper, provides the ability to tune the notch frequency location on the fly. This design reduces the tuning cycle time, and eliminates redesigns and re-manufacturing; therefore, saving time and reducing cost. This design provides additional benefits, and that is the ability to tune the notch frequency of the fluid mount in the field. The ability to tune the notch frequency in the field can be very beneficial in the sense that one can vary the notch frequency location till lowest cabin noise and vibration is achieved. With the current passive fluid mount technology, this capability does not exist.

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