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Double-notch single-pumper fluid mounts

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Abstract

Passive fluid mounts are widely used in the automotive and aerospace applications to isolate the cabin from the engine noise and vibration. In the case of aerospace fixed wing applications, when fluid mounts are used, they are placed in between the engine and the fuselage, and the notch frequency of each fluid mount is tuned to either N1 frequency (engine low speed shaft imbalance excitation frequency) or to N2 frequency (engine high speed shaft imbalance excitation frequency) at the cruise condition. Since current passive fluid mount designs have only one notch, isolation is only possible at N1 or at N2, but not both. Here, in this paper, a double-notch passive fluid mount design will be presented, which has two notch frequencies, and therefore can provide vibration and noise isolation at two frequencies. In this paper, the new fluid mount 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 have the lowest dynamic stiffness at 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 are obtained. The design location of the notch frequency (see Fig. 2) depends on the application, but with most

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applications, the ‘‘notch frequency’’ is designed to coincide with the longest period of constant speed. For example, in the case of fixed wing applications, the notch frequency may be designed to coincide with the cruise speed rather than the take-off and the landing speeds. Since most of the airplane's flight time is spent at the cruise speed, it makes most sense to reduce the cabin noise and vibration at the cruise speed rather than at the take-off or landing speeds.

But, at the cruise speed, there are many imbalance (disturbance) excitation frequencies and ideally one wants to isolate the cabin from all engine imbalance excitation frequencies, but with the current fluid mount technology, the fluid mount notch frequency can only be tuned to one and only one frequency. For example, for most turbofan engines the largest imbalance excitation amplitudes normally occur at N1 (engine low-speed shaft imbalance excitation frequency) and at N2 frequencies (engine high-speed shaft imbalance excitation frequency), but since the current fluid mount design technology only offers isolation at one frequency, a fluid mount designer has no choice but to choose isolation at N1 or at N2.

Literature and patent review was conducted to see if any fluid mount design has been documented and published that can provide vibration and noise isolation at two distinct frequencies, but none were found. Here, in this paper, a new single-pumper fluid mount design will be described, which can offer cabin vibration and noise isolation at two frequencies. The new design concept and its mathematical model, and simulation results will be presented.

2. 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 (or double acting) fluid mounts [10] and single-pumper fluid mounts [3]. The focus of this paper will be on single-pumper fluid mounts.

A single-pumper 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.

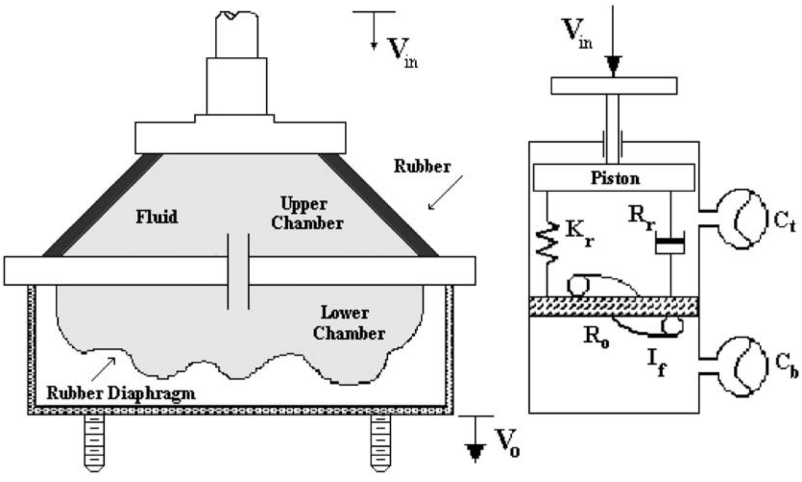

Fig. 1. A typical single-pumper fluid mount and its physical model (courtesy of Ref. [10]).

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

Fig. 3 shows the bond graph [7] model of Fig. 1. From the bond graph model, the following state space equations can be derived:

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

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

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

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

In the above state space equations, q_3, q_6, and q_9 are the generalized displacement variables, and P_{10} the momentum variable. 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, the following input force will be obtained:

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

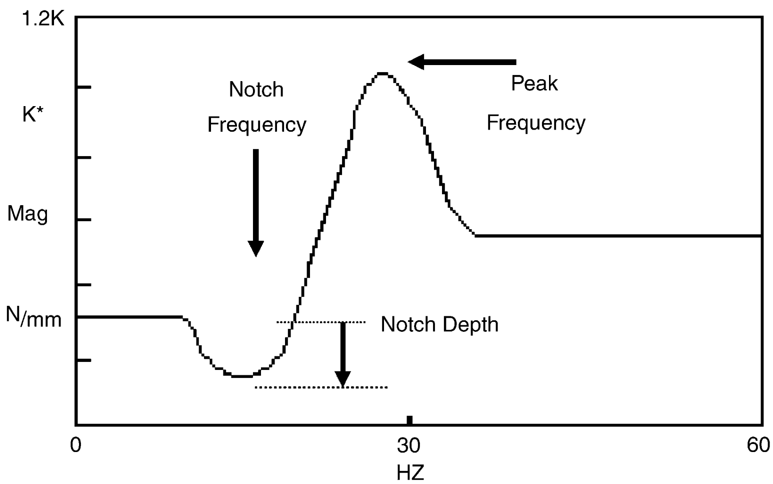

Fig. 2. Dynamic stiffness of a typical fluid mount versus frequency (courtesy of Ref. [10]).

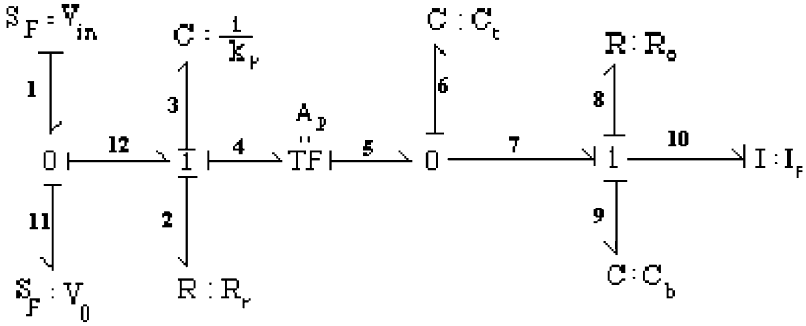

Fig. 3. Bond graph model of the single-pumper fluid mount of Fig. 1.

The dynamic stiffness is defined as $K^* = F_{in}/X_{in}$ (where $V_{in} = \dot{X}_{in}$). Laplace transformations will be used on Eqs. (1)–(5), to get the following dynamic stiffness equation:

$$K^* = K_r' + \frac{K_r''}{\omega}\,S + A_p^2 K_{vt}'\,\frac{S^2 + \frac{R_0}{I_f}\,S + \frac{K_{vb}'}{I_f}}{S^2 + \frac{R_0}{I_f}\,S + \frac{K_{vt}'+K_{vb}'}{I_f}}. \tag{6}$$

In the above state space and dynamic stiffness equations, the following symbols are defined:

V_0 and V_{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 ($C_6 = 1/K_{vt}$), N/m^5
K_{vb}	bottom chamber volumetric or bulge stiffness ($C_9 = 1/K_{vb}$), 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
ω	circular frequency, rad/s
S	equal to jω

The fluid inertia is given by

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

where ρ is the fluid density (kg/m^3) and L the 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 given by

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

Eq. (8) shows that the notch frequency depends on the piston area (A_p), the top and bottom chamber volumetric stiffnesses (K_{vt} and K_{vb}), axial stiffness (K_r), and fluid inertia (I_f). The parameters that were just defined are the ones a fluid mount designer uses to place and design the location of the notch frequency.

Fig. 2 shows that the dynamic stiffness is the lowest at only one frequency which is at the notch frequency, and it is at this frequency where reduced cabin noise and vibration are achieved. If the fluid mount dynamic stiffness could be lowered at two distinct frequencies, cabin noise and vibration isolation can be provided at two frequencies.

3. Double-notch passive fluid mount design

In the previous section, a single-notch passive fluid mount design was described, and earlier it was mentioned that with a single-notch fluid mount design, cabin noise and vibration reduction can only be possible at one frequency (at the notch frequency). Here, in this section, a double-notch passive fluid mount design concept will be described. In this new design, two imbalance disturbance inputs can be filtered out from the cabin.

Fig. 4 shows the new fluid mount design concept. In this new design, instead of conventional two fluid chambers, there are three fluid chambers and two fluid inertia tracks. At the top fluid

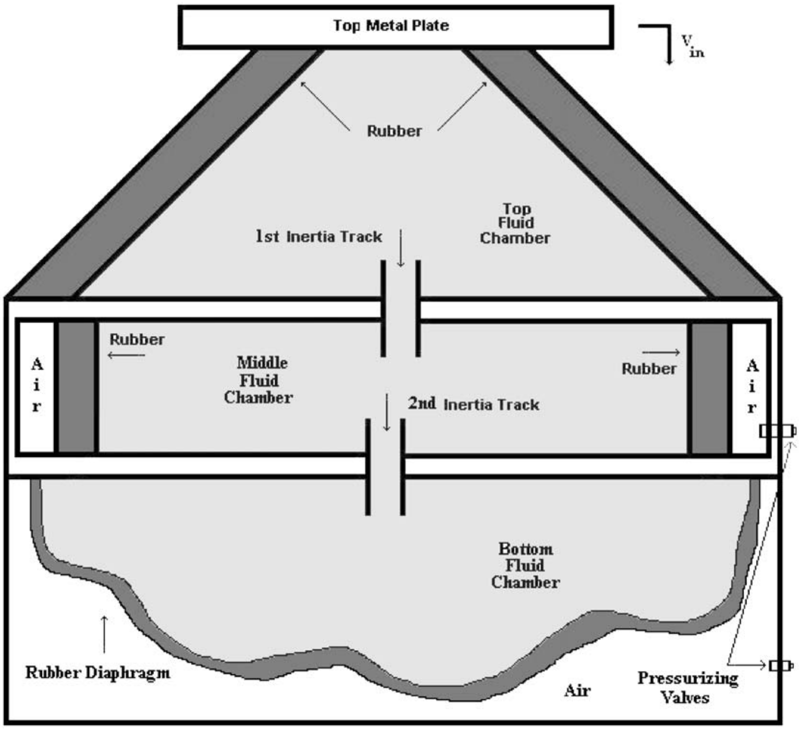

Fig. 4. Variable spring rate fluid mount design.

chamber, a costumed designed rubber component (shown in Fig. 4 as a cone-shaped rubber component) acts like a spring in the axial direction, acts like a piston pumping fluid, and acts like a volumetric spring in the volumetric or bulge direction containing the fluid. The top fluid chamber is connected to the middle fluid chamber with an inertia track. In the middle fluid chamber, a costumed designed rubber piece (shown in Fig. 4 as a cylindrical-shaped rubber piece) provides stiffness in the bugle direction. This bulge or volumetric stiffness can be easily varied by pressuring the air behind the cylindrical rubber piece. The middle fluid chamber is connected to the bottom fluid chamber with another inertia track. In the bottom fluid chamber a soft rubber diaphragm provides the volumetric stiffness and contains the fluid. This volumetric stiffness can also be varied by pressuring the air behind it, as shown in Fig. 4.

The physical and bond graph models of Fig. 4 are shown in Figs. 5 and 6, respectively. In the bond graph model of Fig. 5, the rubber damping in the bulge direction has been assumed negligible. In general, to get the deepest notch it is desired to keep the rubber damping and inertia track flow losses to a minimum. To achieve this goal, low damped elastomers are generally used in fluid mounts; so, the rubber damping can be generally ignored. So it is reasonable to neglect damping in the bulge direction. In the axial direction, the damping of the rubber (K_r'') was included in the bond graph model, but in the simulation, it was set to zero, based upon the same reasoning. The inertia track flow losses are not negligible since it is generally very difficult to bring the inertia track fluid flow losses to zero.

Let us assume that one end of the mount is held fixed ($V_0 = 0$, see Fig. 5) and the other end is subject to an input velocity V_{in}. The state space equations, from the bond graph model of Fig. 6,

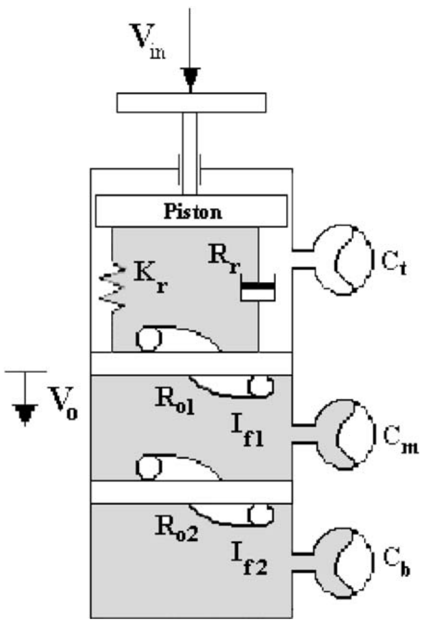

Fig. 5. Physical model of Fig. 4.

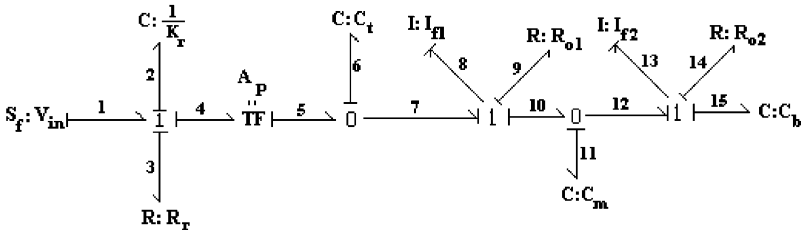

Fig. 6. Bond graph model of Fig. 5.

can be derived as

$$\dot{q}_2 = V_{in}, \tag{9}$$

$$\dot{q}_6 = A_p V_{in} - \frac{P_8}{I_8}, \tag{10}$$

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

$$\dot{q}_{11} = \frac{P_8}{I_8} - A_m \frac{P_{13}}{I_{13}}, \tag{12}$$

$$\dot{P}_{13} = \frac{q_{11}}{C_{11}} - R_{14} \frac{P_{13}}{I_{13}} - \frac{q_{15}}{C_{15}}, \tag{13}$$

$$\dot{q}_{15} = \frac{P_{13}}{I_{13}}. \tag{14}$$

The input force (effort on bond 1) is given by

$$F_{in} = \frac{q_2}{C_2} + R_3 V_{in} + A_p \frac{q_6}{C_6}. \tag{15}$$

In the above state space equations, q_2, q_6, q_{11}, and q_{15} are the generalized displacement variables, and P_8 and P_{13} are the momentum variables. If Laplace transformation is used on all the state space and the output equations, the following dynamic stiffness equation ($K^* = F_{in}(s)/X_{in}(s)$) can be obtained:

$$K^* = (K'_r + A_p^2 K_{vt}) + \frac{K''_r}{\omega} S - A_p^2 K_{vt} \frac{1}{I_{f1} S^2 + R_{01} S + (K_{vt} + K_{vb}\beta)}, \tag{16}$$

where

$$\beta = \frac{I_{f2} S^2 + R_{02} S + K_{vb}}{I_{f2} S^2 + R_{02} S + (K_{vm} + K_{vb})}. \tag{17}$$

The lowest dynamic stiffness (occurring at the notch frequencies) occurs when the numerator of Eq. (16) is set to zero, and the maximum dynamic stiffness (occurring at the peak frequencies) occurs when the denominator is set to zero. It was shown in Ref. [10] that the notch and peak frequency locations are insensitive to damping; so, to find the notch and peak frequencies, all the damping factors (R_{01}, R_{02}, and K_r'') are set to zero. So, if the denominator of Eq. (16) is set to zero, the following peak frequencies will be obtained:

$$f_{\text{peak1}} = \frac{1}{2\pi}\sqrt{\frac{(K_{vt}+K_{vm})}{2I_{f1}} - \sqrt{\left(\frac{K_{vt}+K_{vm}}{2I_{f1}}\right)^2 - \frac{(K_{vt}K_{vb}+K_{vb}K_{vm}+K_{vm}K_{vt})}{I_{f1}I_{f2}}}}, \tag{18}$$

$$f_{\text{peak2}} = \frac{1}{2\pi}\sqrt{\frac{(K_{vt}+K_{vm})}{2I_{f1}} + \sqrt{\left(\frac{K_{vt}+K_{vm}}{2I_{f1}}\right)^2 - \frac{(K_{vt}K_{vb}+K_{vb}K_{vm}+K_{vm}K_{vt})}{I_{f1}I_{f2}}}}. \tag{19}$$

If the numerator of Eq. (16) is set to zero, the following notch frequencies will be obtained:

$$f_{\text{notch1}} = \frac{1}{2\pi}\sqrt{\frac{\alpha_1}{2} - \sqrt{\left(\frac{\alpha_1}{2}\right)^2 - \alpha_2}}, \tag{20}$$

$$f_{\text{notch2}} = \frac{1}{2\pi}\sqrt{\frac{\alpha_1}{2} + \sqrt{\left(\frac{\alpha_1}{2}\right)^2 - \alpha_2}}, \tag{21}$$

where the parameters α_1 and α_2 are

$$\alpha_1 = \frac{K_{vb}+K_{vm}}{I_{f2}} + \frac{K_{vm}}{I_{f1}} + \frac{K_rK_{vt}}{I_{f1}(K_r' + A_p^2K_{vt})}, \tag{22}$$

$$\alpha_2 = \frac{K_{vb}K_{vm}}{I_{f1}I_{f2}} + \frac{K_rK_{vt}(K_{vb}+K_{vm})}{I_{f1}I_{f2}(K_r' + A_p^2K_{vt})}. \tag{23}$$

To simulate the design of Fig. 4, MATLAB Program and the above state space equations with the following baseline parameters were used.

K_r'	real component of the vertical stiffness, 2.1E6 N/m ($C_2 = 1/K_r'$)
K_r''	imaginary component of the vertical stiffness, 0 N/m ($R_3 = K_r''/\omega$)
A_p	effective piston area, 0.009 m^2
d_{t1}	first inertia track diameter, 0.0254 m
d_{t2}	second inertia track diameter, 0.0127 m
L_{t1}	first inertia track length, 0.1778 m^2
L_{t2}	second inertia track length, 0.1778 m^2
ρ	fluid density, 1770 kg/m^3
R_{01}	first inertia track flow resistance, equal to R_9, 6.4E6 N s/m^5
R_{02}	second inertia track flow resistance, equal to R_{14}, 6.4E6 N s/m^5
K_{vt}	top chamber volumetric or bulge stiffness, 1.1E11 N/m^5 ($C_6 = 1/K_{vt}$)
K_{vm}	middle chamber volumetric or bulge stiffness, 4.21E10 N/m^5 ($C_{11} = 1/K_{vm}$)
K_{vb}	bottom chamber volumetric or bulge stiffness, 2.1E9 N/m^5 ($C_{15} = 1/K_{vb}$)

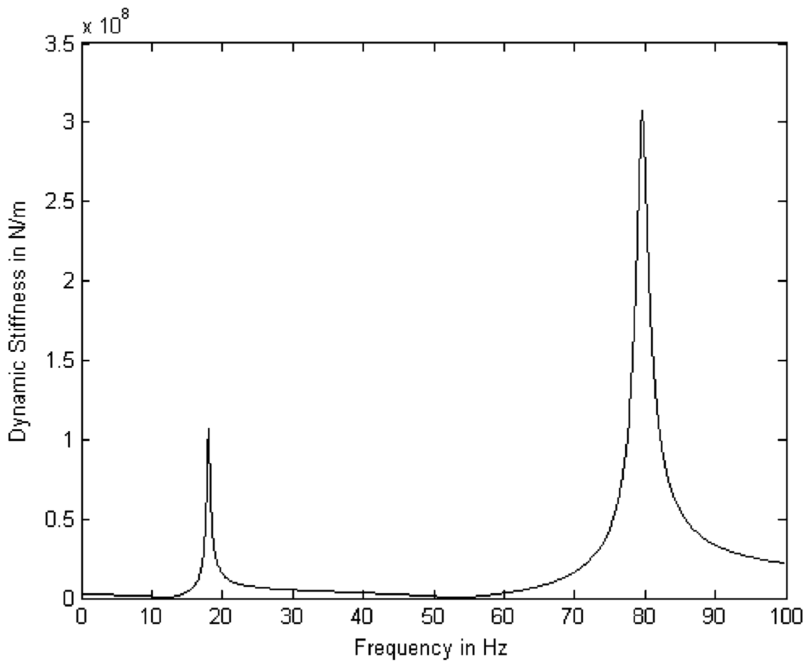

Fig. 7. Dynamic stiffness of the double-notch fluid mount (MATLAB simulation).

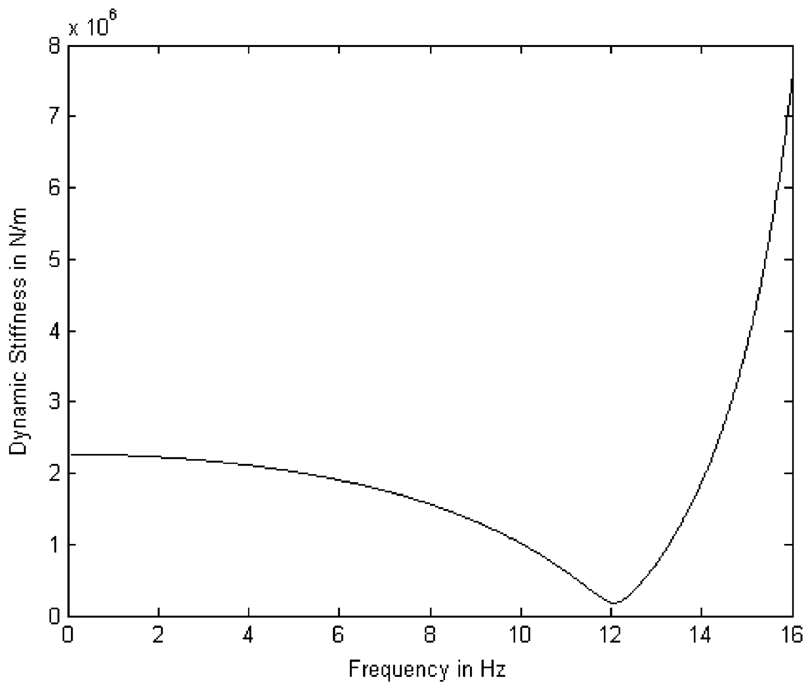

Fig. 8. First notch frequency of the double-notch single-pumper fluid mount.

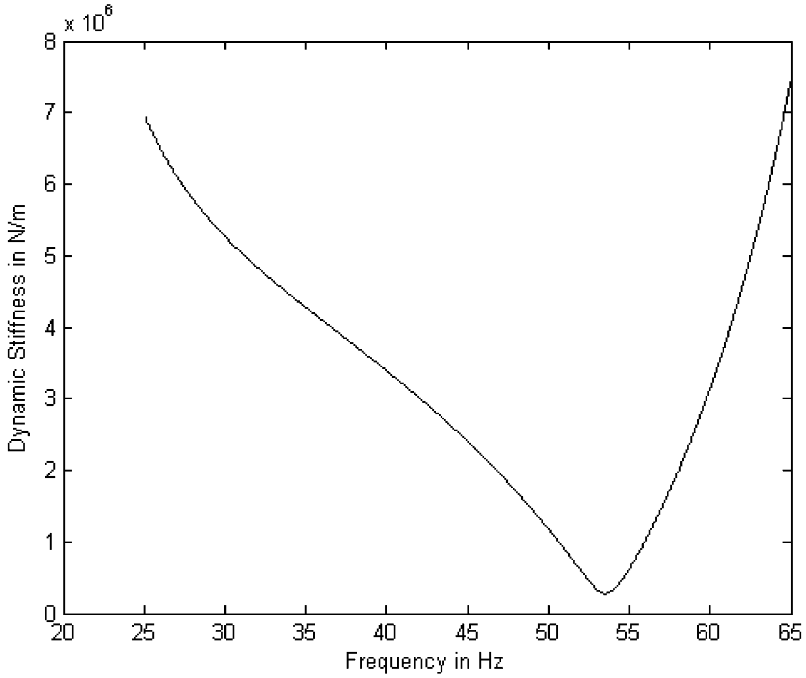

Fig. 9. Second notch frequency of the double-notch single-pumper fluid mount.

MATLAB program, with the above baseline parameters, was used to simulate the state space of Eqs. (9)–(15). Figs. 7–9 show the new fluid mount dynamic stiffness versus frequency and show that indeed that are two notches and two peaks. The same figures show that the first and the second notch frequencies occur at 12.1 and 53.5 Hz, respectively, and the peak frequencies at 18.75 and 76.5 Hz, respectively. Of course, one can place the notch and peak frequencies to any desired location by altering fluid mount parameters. For example, Figs. 10 and 11 show that if one varies the second inertia track diameter, one can alter the location of the first notch frequency and the first peak frequency.

Often, it is necessary to change or alter the location of the notch frequencies after the fluid mount is manufactured. Also, if notch frequency or frequencies need to be retuned, ideally it can be done without a need for any fluid mount redesigns, particularly redesign of the rubber components. In this new fluid mount design, the volumetric stiffnesses, K_{vm} and K_{vb}, can be easily varied if the air (or gas) pressure behind the rubber can be increased or decreased. This tunability can be very useful both to the OEM and to the customer. The two notch frequencies can be fine tuned with the help of K_{vm} and K_{vb}, without the need for any fluid mount redesign. If it is needed to fine tune the fluid mount notches in the field, one can do so by changing gas pressure. Fine tuning the notch frequencies in the field can provide better cabin noise and vibration isolation than tuning the fluid mount notches at the OEM's manufacturing site. Fig. 12 shows that as the volume stiffness K_{vm} is varied, the second notch frequency can be relocated.

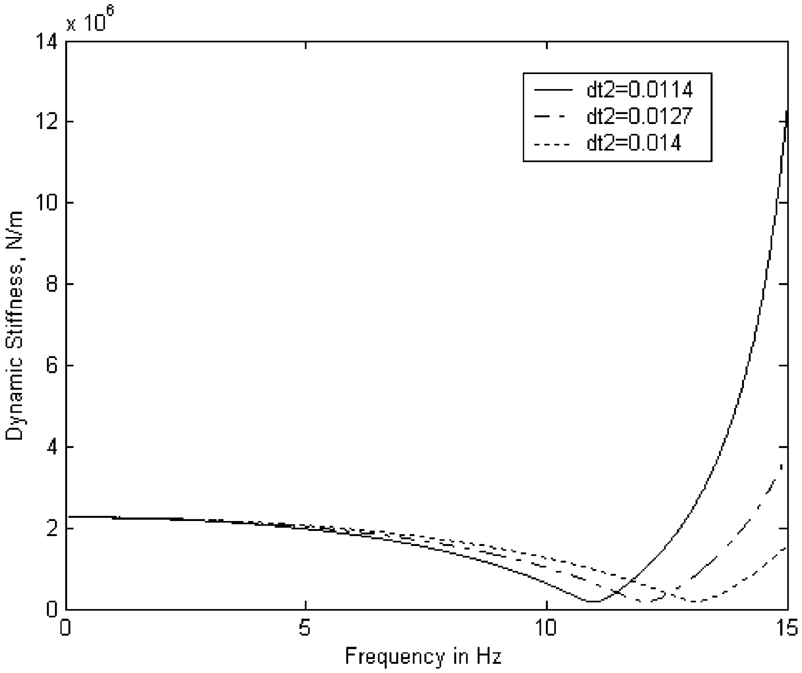

Fig. 10. Dynamic stiffness as the second inertia track diameter is varied from 0.0114 to 0.014 m.

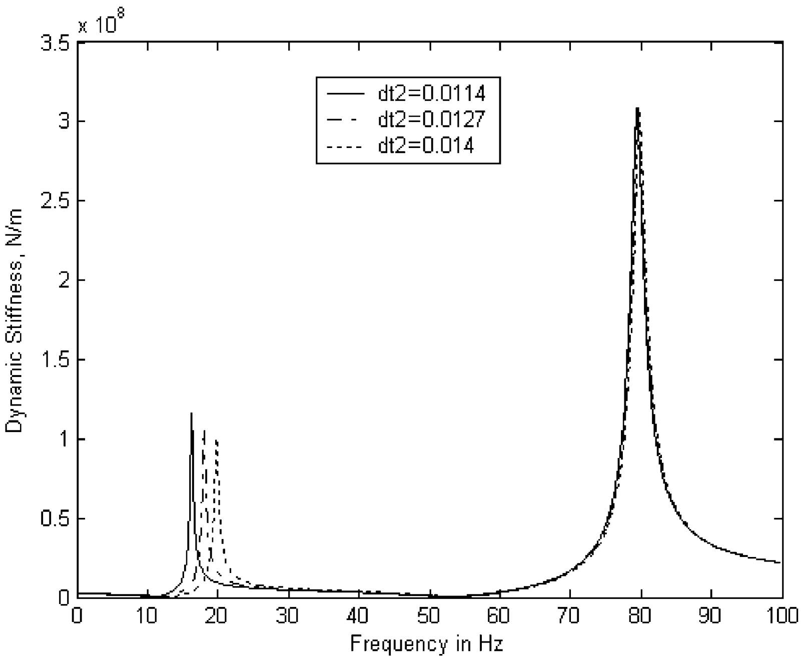

Fig. 11. Dynamic stiffness as the second inertia track diameter is varied from 0.0114 to 0.014 m.

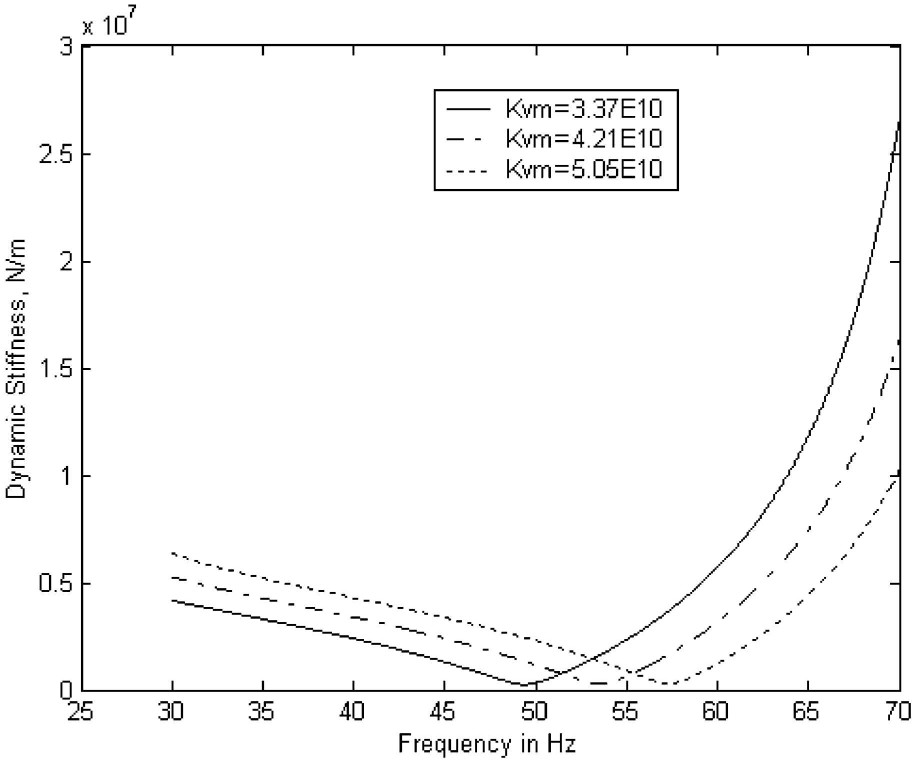

Fig. 12. Dynamic stiffness as the middle volume stiffness, K_{vm}, is varied.

4. Sensitivity analyses

The design location of the notch frequency or frequencies (see Figs. 2 and 7) 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 may be 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 the cruise speed rather than during the take-off or landing speeds. 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. Hence, to achieve the 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. Hence, the fluid mount parameters such as inertia track length or diameter, or fluid density, or other properties are varied after the first manufacturing pass till the mount is tuned to the correct notch frequency. Since retuning of the notch frequencies may be necessary, to see which

Table 1
Sensitivity of the notch and peak frequencies to 20% change in fluid mount parameters

Baseline parameters	% Change in parameters	% Change in fnotch1 = 12.1 Hz	% Change in fpeak1 = 18.75 Hz	% Change in fnotch2 = 53.5 Hz	% Change in fpeak2 = 76.5 Hz
$K_r = 2.1E6$	± 20	± 5	± 0	± 2	± 0
$D_{t1} = 0.0254$	± 20	± 2.5	± 1.5	± 17.5	± 21
$D_{t2} = 0.0127$	± 20	± 17.7	± 21.4	± 2.2	± 1.3
$L_{t1} = 0.1778$	± 20	± 1.1	± 0.65	± 9	± 11
$L_{t2} = 0.1778$	± 20	± 9	± 11	± 1.2	± 0.7
$\rho = 1770$	± 20	± 10.3	± 10.3	± 10.3	± 10.3
$K_{vt} = 1.1E11$	± 20	± 1.25	± 2.3	± 0.5	± 7.5
$K_{vm} = 4.21E10$	± 20	± 2.3	± 7	± 7.4	± 2.5
$K_{vb} = 2.1E9$	± 20	± 1.3	± 0.7	± 0.15	± 0
$A_p = 0.009$	± 20	± 10.5	± 0	± 4.4	± 0

fluid mount parameters have the greatest impact on the notch and peak frequencies, a sensitivity analysis was conducted for this new fluid mount design.

In this analysis, each fluid mount parameter of the new design was varied by 20% and the percent change in the notch and the peak frequencies were recorded. Table 1 indicates the results. With the baseline parameters that were chosen here in this paper, Table 1 indicates that the first notch frequency is most sensitive to the second inertia track dimensions, and fluid density (basically sensitive to the second inertia track fluid inertia, $I_f = \rho L/A_t$), and the piston area. Table 1 also indicates that the second notch frequency is most sensitive to the first inertia track dimensions, and fluid density (basically sensitive to the first inertia track fluid inertia), and the piston area. One can simply state that the two inertia tracks and the effective piston area greatly impact the two notch frequencies.

Table 1 indicates that the peak frequencies are completely independent of the piston area, and the axial stiffness K_r, but again most sensitive to inertia track dimensions and fluid density, and are a weak function of volumetric stiffnesses.

The data suggest that, to independently vary notch frequencies from the peak frequencies, one can use inertia track dimensions, fluid density, and piston area to set the notch frequencies, and use volume stiffnesses to set the peak frequencies.

5. Conclusions

For fixed wing applications, the current passive single-pumper fluid mount designs have only one notch frequency; therefore, cabin noise and vibration isolation is only possible at N1 or at N2, but not both. Here, in this paper, a new single-pumper fluid mount design has been presented, which has two notch frequencies. The new design was described and its mathematical model was presented. It was shown that indeed this new design can provide vibration and noise isolation at two frequencies if the inertia track flow losses and rubber damping are kept to a minimum. Keeping fluid flow and rubber damping to a minimum is not a hindrance since most fluid mount manufacturers are currently doing it.

The new fluid mount design provides additional benefits, and that is the ability to tune the notch frequencies in the field by changing gas pressure. The ability to tune the notch frequency in the field is very beneficial in the sense that one can vary the notch frequency locations till lowest cabin noise and vibration are achieved.

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