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ELECTROMECHANICAL HYDRAULIC MOUNT

Nader Vahdati

Petroleum Institute, Faculty of Mechanical Engineering, Abu Dhabi, UAE
email: nvahdati@pi.ac.ae

A hybrid electromechanical hydraulic engine mount design is proposed. The traditional single pumper hydraulic engine mount design consists of two chambers and an inertia track connecting the two chambers, but in the hydraulic mount design of this paper, only the top chamber is filled with a fluid thus fluid never enters the bottom chamber. A piston with graphite filled PTFE seals separates the top chamber from the bottom chamber. The piston, placed in the inertia track, is supported by a metal spring and a long stroke linear voice coil. The voice coil is connected to an RLC circuit with a variable capacitance. The electronics are used to vary the location of the notch frequency, when desired. Equations of the motion for this hybrid electromechanical hydraulic engine mount design are derived using bond graph modelling technique. Simulation results clearly indicate that the proposed hydraulic mount design concept is feasible when mount input displacement are small. In this new hydraulic mount design, the notch frequency can be varied using a variable capacitance.
Keywords: Semi-active, Electromechanical, Hydraulic engine mounts

1. Introduction

Engine or motor mounts support the engine and transmission weight and isolate the chassis and the cabin from engine and transmission noise and vibrations. The motor mounts can be as simple as a rubber to metal bonded engine mount [1], or a hydraulic or a fluid-filled mount (sometimes also called hydroelastic or hydromount) [2 to 5], or as complex as an active motor mount, consisting of an actuator and an elastic element with or without a fluid.

Hydraulic engine mounts, used since 1945, consist of two hollow chambers filled with a fluid (such as glycol). As the mount is given a sinusoidal input displacement, the fluid moves back and forth between the two fluid chambers. The fluid will eventually goes to resonance and at that frequency, called notch frequency, the dynamic stiffness of the mount drops, thus cabin noise and vibration reduction is achieved at that frequency.

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 tolerances and material variabilities, no two identical fluid mount designs act the same. 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 proposed in this paper, the notch frequency can be modified without the need for any redesigns using electronics.

In this paper, a hybrid electromechanical hydraulic engine mount design is proposed. A variable capacitor is used to vary the location of the notch frequency, when desired. Equations of the motion for this hybrid electromechanical hydraulic engine mount are derived using bond graph modelling technique. Simulation results clearly indicate that the proposed concept is feasible and the notch frequency can indeed be varied using a variable capacitance.

2. Hybrid electromechanical hydraulic engine mount design

In this paper, a hybrid electromechanical hydraulic engine mount design is proposed, see Fig 1. Only the top fluid chamber is filled with a fluid (shown in blue color); thus, fluid never enters the bottom fluid chamber. A piston with a graphite filled PTFE seal separates the top fluid chamber from the bottom fluid chamber. Graphite filled PTFE seals provide very low friction seal. The piston is supported by a metal spring and a long stroke linear voice coil. The piston is placed in a long inertia track. The voice coil is connected to an RLC circuit with a variable capacitance. The capacitance, connected to the voice coil, is used to vary the location of the notch frequency, when desired. This new hydraulic engine mount design concept helps reduce the Engineering Change Orders (ECO) since electronics can be used to tune the location of the notch frequency during the manufacturing or in the field, as and when needed.

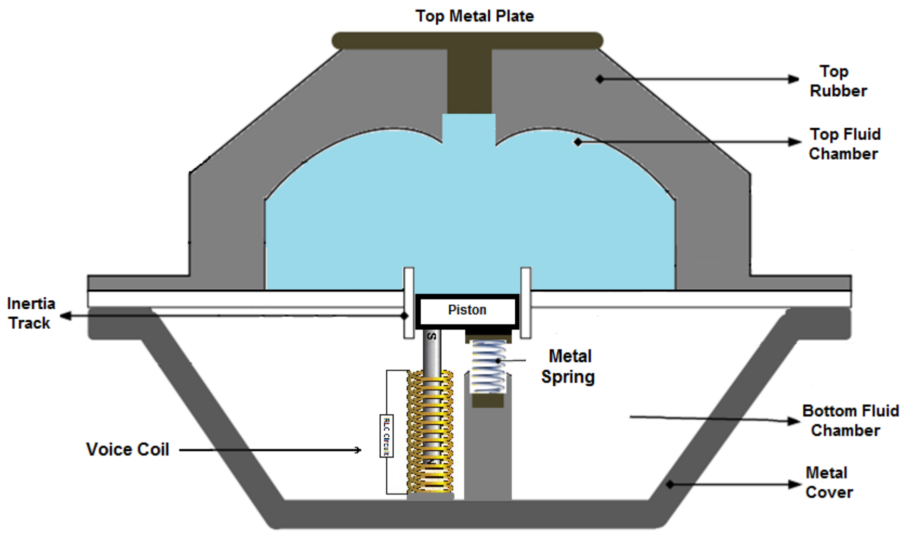

Figure 1 Hybrid electromechanical hydraulic engine mount design

The bond graph modelling technique [6] is used to develop the mathematical model of Fig. 1. The bond graph model of Fig. 1 is shown in Fig. 2:

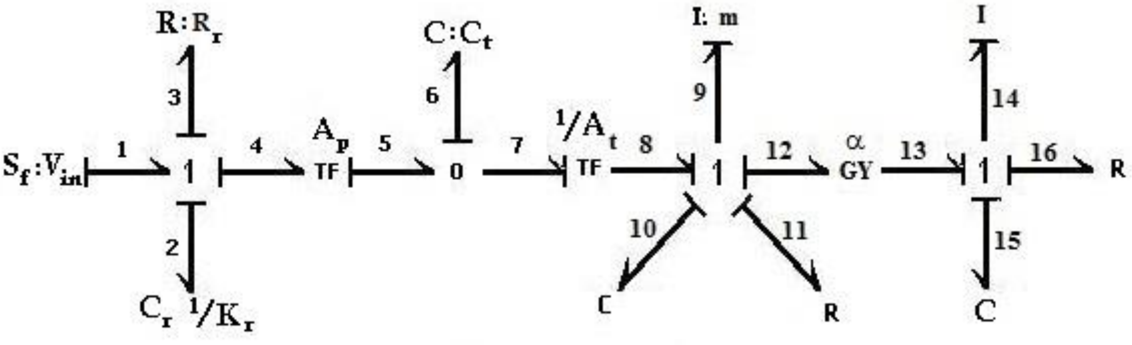

Figure 2 Bond Graph Model of Figure 1

2.1 Mathematical Model

Equations of the motion for this hybrid electromechanical hydraulic engine mount design of Fig. 1 are derived using the bond graph model of Fig. 2. State Space equations, derived from the bond graph model of Fig. 2, are shown below:

$$\dot{q}_2 = V_{in} \tag{1}$$

$$\dot{q}_6 = A_p V_{in} - \frac{A_t}{I_9} p_9 \tag{2}$$

$$\dot{p}_9 = \frac{A_t}{C_6} q_6 - \frac{R_{11}}{I_9} p_9 - \frac{1}{C_{10}} q_{10} - \frac{\alpha}{I_{14}} p_{14} \tag{3}$$

$$\dot{q}_{10} = \frac{p_9}{I_9} \tag{4}$$

$$\dot{p}_{14} = \frac{\alpha}{I_9} p_9 - \frac{R_{16}}{I_{14}} p_{14} - \frac{1}{C_{15}} q_{15} \tag{5}$$

$$\dot{q}_{15} = \frac{p_{14}}{I_{14}} \tag{6}$$

The Output equations are the input force at bond 1, velocity of the piston at bond 9, and top fluid chamber pressure. Therefore we get:

$$F_{in} = \frac{q_2}{C_2} + R_r V_{in} + \frac{A_p}{C_6} q_6 \tag{7}$$

$$V_{piston} = \frac{p_9}{I_9} \tag{8}$$

$$e_6 = \frac{q_6}{C_6} \tag{9}$$

In the above state space equations, $\dot{q}$ and $\dot{p}$ are representatives of flow and effort, respectively. Flow, in the mechanical energy domain represents the velocity, while in the electrical energy domain, it represents the current. Effort in the mechanical energy domain stands for force, but in the electrical energy domain it represents the voltage. In the above state space equations, the states are defined as:

q_2	Motion across the top elastomer
q_6	Volume change in the top fluid chamber
p_9	Momentum of the piston
q_{10}	Motion across the metal spring
p_{14}	Flux linkage variable across the voice coil
q_{15}	Electrical charge in the capacitor

To simulate the above state space and output equations, the following baseline parameters (see Table 1) were used for the MATLAB simulations.

Table 1 Baseline engine mount parameters for MATLAB simulations

Symbol	Discretion	Value	Dimension
K_r	Rubber vertical stiffness	8.756E5	N/m
R_r	Hysteresis damping of the rubber	Tanδ=0.04	$N.s/m$
A_p	Effective Piston Area	0.0052	m^2
K_{vt}	Top chamber Volumetric Stiffness	6.31E10	N/m^5

C_t	Top Chamber compliance (C_t=1/K_{vt})	1.585E-11	m^5/N
A_t	Inertia Track Area	3.66E-4	m^2
A_p/A_t	Mechanical Advantage	14.1	unitless
m	Mass of the Piston including coil	0.71	kg
K	Metal Spring Stiffness	20,000	N/m
R_{11}	Damping experienced by the piston	0.67	N-s/m
α	Force sensitivity of the voice coil	22.2	N/Amp
I_{14}	Inductance of the voice coil	2.1	mH
R_{16}	Resistance of the voice coil	3.0	Ω
C_{15}	Capacitance added to the voice coil	Variable	μF
Stroke	Voice coil stroke	+/- 31.75	mm
X_{in}	Input displacement	+/- 0.5	mm

2.2 Simulation Results

The voice coil, used in this paper, is a commercially available voice coil. Fig. 1 shows the dynamic stiffness of the new proposed engine mount design as capacitance is varied. As capacitance changes from 500 μF to 1000 μF, the notch frequency moves by 3.15 Hz. This change in the location of the notch frequency allows us to use a variable capacitance to fine tune the notch frequency location of the hydraulic mount without the need for a mount redesign. Fig. 1 indicates that as capacitance is increased, the notch depth decreases. This figure indicates that high capacitance should be avoided since the notch depth can be affected when capacitance is high.

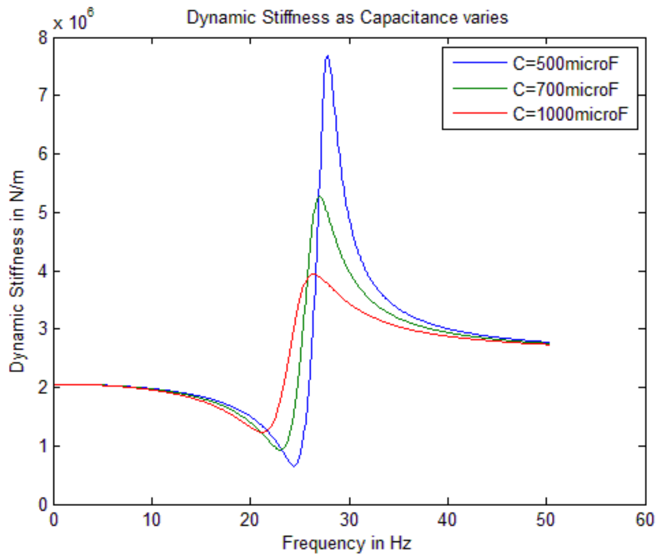

Figure 1 Dynamic Stiffness of the Mount versus Frequency as Capacitance Varies

Fig. 2 shows the motion of the piston versus frequency as capacitance is varied. This figure shows that the displacement amplitude of the piston (or the voice coil) is within the stroke limit of the voice coil.

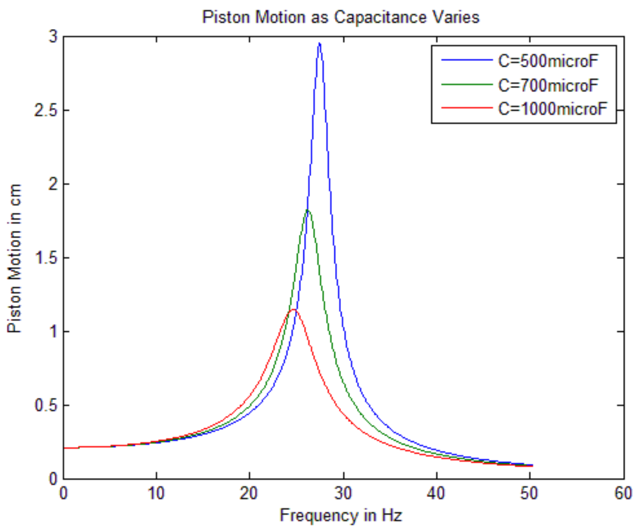

Figure 2 Piston Motion versus Frequency as Capacitance Varies

Figure 3 shows the amplitude of the dynamic pressure of the top fluid chamber versus frequency as capacitance is increased. As capacitance is increased, the dynamic pressure of the top fluid chamber reduces, which is good. At 500 μF, the maximum dynamic pressure is about 7 bars (about 100 psi). A static pressure of 100 psi will be necessary to avoid fluid cavitation. This pressure is a bit high for fluid mount applications. More design optimization will be necessary to reduce the dynamic pressures.

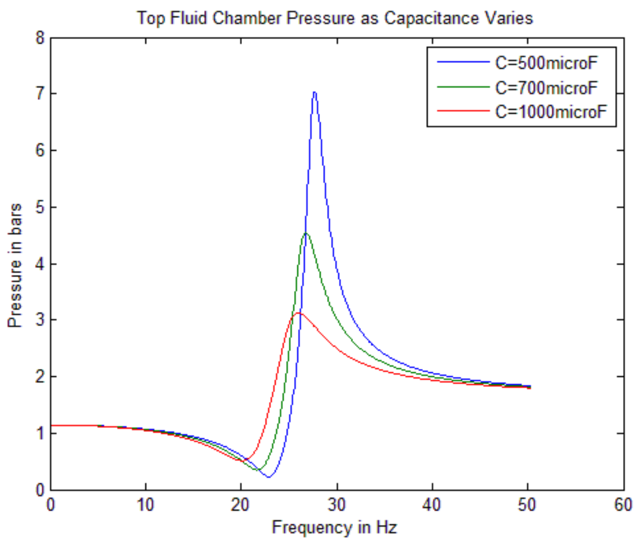

Figure 3 Top Fluid Chamber Pressure versus Frequency as Capacitance Varies

3.0 Conclusions

The simulation results suggest that the hybrid electromechanical hydraulic engine mount design proposed in this paper can be used to tune the location of the notch frequency using a variable capacitance. The fluid mount design, proposed in this paper, is not suitable for applications where the input displacements are large. When the input mount displacement are high, the piston displacement will be large and so does the top fluid chamber pressures. The piston should never leave the track. So end stops need to be added to the design of Fig. 1 in order to prevent the piston from exiting the track. This of implies that at some large motions, the piston will bottom out, meaning, the proposed mount design can behave nonlinearly when the motions become too high.

Even for small input displacements, the dynamic pressures can be a bit high and more design optimization will be necessary to reduce the top fluid chamber pressures.

For the fluid mount design of this paper, the motion of the piston was as high +/- 3 centimetres. Since the piston seats in the inertia track, the inertia track needs to be at least 6 cm or longer in order to accommodate the piston required motion. In the bond graph model of Fig. 2, the fluid inertia in the track was ignored. A more detailed bond graph model may be necessary to see the impact of the fluid inertia in the track on the dynamic stiffness.

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