Development of variable multiple-notch and multi-function fluid mounts

Somayeh Heidari

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DEVELOPMENT OF VARIABLE MULTIPLE-NOTCH AND MULTI-FUNCTION FLUID MOUNTS

SOMAYEH HEIDARI

School of Aerospace and Mechanical Engineering

A thesis submitted to the Nanyang Technological University
in partial fulfillment of the requirement for the degree of
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2010

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List of Symbols

MR	Magneto-Rheological
ER	Electro-Rheological
SMA	Shape Memory Alloy
MRE	Magneto-Rheological Elastomers
AVTA	Adaptive Tuned Vibration Absorbers
ETVA	Electromechanical Tuned Vibration Absorber
EMVA	Electro-Magnet Vibration Absorber
GY	Gyrator
RMS	Root-Mean-Square
DSP	Digital Signal Processor
A/D	Analog to Digital
D/A	Digital to Analog
SSR	Solid-State Relay
RLC	Resistance-Inductance-Capacitance
PZT	Lead Zirconate Titanate (piezoelectric ceramic material)
K^*	Dynamic stiffness
K_{eff}	Effective stiffness
K_v	Volumetric stiffness
K_{vt}	Top chamber volumetric or bulge Stiffness
K'_r	Real component of the vertical stiffness
K''_r	Imaginary component of the vertical stiffness
I_{eff}	Effective moment of inertia

List of Symbols

d_{31}	PZT electromechanical coupling coefficient
ϵ	PZT permittivity
ϵ_{33}^T	Dielectric permittivity, at constant stress
s_{11}^E	PZT mechanical compliance at constant electric field
s_{11}	PZT mechanical compliance
S_1	Strain
T_1	Stress
μ	Mass per length
M_v	Bending moment
$\bar{z}$	Neutral axis of a multi-layer PZT beam
YI	Flexural rigidity
C_p	Inherent capacitance of PZT
C_s	Shunted capacitance
s	Laplace variable
ω	Circular frequency
ρ	Fluid Density

Abstract

Passive fluid mounts have been in use in automotive and aerospace industry since 1940s. They provide noise and vibration isolation at a frequency coined “notch frequency” which is basically designed to coincide with the longest period of constant speed. To obtain the greatest noise and vibration reduction at any desired frequency, the notch frequency needs to be close to that desired frequency.

Conventional passive fluid mounts have two limitations. Firstly, due to tolerances on all the fluid mount dimensions, variations of elastomer material property and molding process, the notch frequency never ends up at the right location on the first manufacturing pass. Secondly, they provide the best vibration isolation at only one invariable notch frequency.

For aerospace applications, the engine has two or more disturbances due to imbalance of the fan ($N1$) and the high-pressure compressor ($N2$) and a multiple notch fluid mount design is required.

For automotive applications, the internal combustion engine has two dynamic disturbances: primary (the inertia forces of the rotating and reciprocating parts) and secondary (the firing pulses due to the explosion of the fuel in the cylinders). Engine manufacturers attempt to eliminate the primary disturbances. However, the secondary disturbances can affect the vehicle compartments and it is desirable to eliminate their effect. Since, in automotive applications engine RPM can vary, variable notch fluid mounts are needed.

The above requirements have prompted us to develop fluid mount technologies, which have multi-functional mode capability and the notch frequency, or frequencies are easily tunable and variable.

For the aerospace applications, a variable multiple-notch semi-active fluid mount is proposed by using a conventional passive fluid mount plus two piezoelectric cantilever beams, shunted with capacitive ladder circuits. The fluid mount “notch frequencies” can be easily tuned by changing shunt capacitance conditions and the mount performs well, where multiple disturbance frequencies exist.

For the automotive applications, a multi-functional (passive, semi-active and active) fluid mount design is proposed which has a variable notch frequency and can suppress the effect of secondary disturbances and manufacturing variations efficiently. The design consists of a conventional passive fluid mount and a voice coil shunted with a capacitive circuit. The voice coil can change the volumetric stiffness of the bottom chamber of the fluid mount semi-actively or actively. So, the mount is able to act as a passive, semi-active and an active mount; and it has the capability to be converted to a damper as and when needed.

For the applications, where the wiring and power supply to the mount can be important issues, an energy harvester device is added to the mounting system. The energy harvester converts the vibration energy of the engine to the useful electrical power, used for running controller and sensors; so it adds self-contained, self-powered capability to the fluid mount design.

Abstract

Finally, the proposed fluid mount is developed to an active design by applying current to the voice coil. An adaptive controller is utilized to control the input current such that the transmitted force cancels out completely in a wide frequency range.

Chapter1 Introduction

1.1 Mount Technologies

In most vibration textbooks, vibration is referred to as small amplitude oscillation, and shock is referred to as large amplitude oscillation.

In automotive applications, the car engine experiences both small amplitude vibrations due to the imbalance of the engine rotating parts and cylinder firings, and large amplitude vibrations due to sudden starts and stops of the car and the car going over bumps and potholes.

In the case of aerospace applications, engine shock waves are present but less common and majority of the time, the major vibration that the engine sees, comes from the imbalance of the engine rotating components (fan, $N1$ frequency, and high speed compressor, $N2$ frequency) and flow induced engine vibrations.

In both automotive and aerospace applications, small and large engine vibrations as well as shock inputs are present, so the engine mounting systems have to have several functions. Generally, the engine mounts need to satisfy three design requirements: (1) firmly support engine weight, (2) isolate structure (cabin or fuselage) from the engine vibrations, and (3) control and limit engine motion when large vibration inputs or shocks or engine resonances are present.

To firmly support engine weight, the engine mounts need to be stiff. To isolate structure from engine vibrations, the engine mounts need to be soft and low damped,

and to control engine motion, in the event of a large vibration or shock input or operating near engine resonances, high damping is needed. This inherent problem sets up competing objectives that require compromise. But compromise would result in mediocre quality for all the three attributes. Various engine mount technologies have been developed which satisfy all the aforementioned design requirements to some extent and they can be classified into three main groups:

1. Passive engine mounts
2. Semi-active engine mounts
3. Active engine mounts

The details of the developed passive, semi-active and active engine mount technologies are given in Chapter 2 and their odds and ads are shown. But in summary, it can be concluded that, a pure elastomeric engine mount cannot satisfy all the above three mentioned requirements simultaneously. But a passive hydraulic (or also called a fluid) mount, to some degree, can. A passive fluid mount is stiff at low frequencies; therefore, it can firmly support engine weight. At high frequencies near the fluid mount notch frequency, it can provide low dynamic stiffness, but most of passive fluid mounts do not provide damping at large engine motions and the low dynamic stiffness is only good for a narrow frequency range. Some passive fluid mount designs that do include a decoupler do provide high damping for large engine motions but normally such fluid mount designs do not have a deep notch (low dynamic stiffness).

Another problem with passive fluid mounts is that they are very difficult to tune and several manufacturing passes are needed to tune the passive fluid mount to a specific

notch frequency and provide low dynamic stiffness at a particular frequency. Fluid mount tuning and re-tuning manufacturing cost is an issue.

To remedy the above two problems, namely a fluid mount design that can be easily tuned and can provide wide or variable notch frequency is needed. Some researchers have resorted to semi-active or active fluid mounts. Many of the documented semi-active and active fluid mount designs provide few functionalities but not all that we would like to have and provide.

Here in this research project, the aim is to develop a fluid mount design or designs that can have several modes of operations. The fluid mount notch frequency is very easy to tune and there is no compromise between the three mentioned mount design requirements.

To reduce cabin noise and vibration due to engine vibrations, the engine mounts need to act as vibration isolators, and to control the engine large motions due to shock, the engine mounts need to turn into dampers. In the event of sensor or actuator malfunction, the engine mounts need to still perform passively. And in the event that power needs to be reduced, the actuator can be turned off, and the engine mounts can be turn into semi-active mounts.

So here in this thesis we have introduced a new engine mounting design that could act as an active mount, a semi-active mount, a passive mount, and a damper as and when needed. Filtered-X LMS algorithm is used to control the mount. Energy harvesting device is added to the fluid mount design to make it self-contained.

1.2 The new proposed fluid mount designs

Previous studies reveal that in a single pumper fluid mount, the bottom chamber volumetric stiffness is the easiest parameter to vary in real-time to control the notch frequency location [1]. Figure 1-1 demonstrates the overall design concept. This design consists of a piston 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. In this design, it is assumed that the stiffness of the spring K is able to vary and it changes the volumetric stiffness of the bottom chamber and consequently changes the notch frequency location.

Figure 1-1 Variable spring rate fluid mount design [1]

According to this idea, here in this thesis three different designs were investigated to change the lower chamber volumetric stiffness of the single pumper fluid mount.

- 1) **Design 1:** Multiple notch semi-active fluid mount by using a single pumper fluid mount in conjunction with two PZT beams, acting as two variable stiffness devices.
- 2) **Design 2:** Semi-active fluid mount by using a single pumper fluid mount in conjunction with an electromechanical tuned vibration absorber (ETVA),

acting as a variable stiffness device. Design 2 is also developed to self-contained semi-active fluid mount by adding an energy harvester device to the mounting system.

- 3) Design 3:** Active fluid mount by using a single pumper fluid mount in conjunction with an electromagnetic actuator.

The combination of Design 2 and 3 creates a multi-functional fluid mount which can act in active, semi-active and passive isolation mode, and it can be converted to a damper as and when needed. The following sections explain the configuration of the proposed mounting systems briefly. It should be noted that through this report, the aforementioned three designs are addressed as Design 1, Design 2 and Design 3.

1.2.1 Design 1: Multiple Notch Semi-Active Fluid Mount with Two PZT beams

Figure 1-2 shows schematic of the multiple notch semi-active fluid mount design. It consists of a conventional passive fluid mount plus two 3-layer PZT cantilever beams and two pistons which are used to create two controllable notches. 3-layer PZT cantilever beams act as variable stiffness devices. They are shunted to the capacitive circuits and their stiffness can be controlled by varying shunt capacitance condition. Each piston and its corresponding 3-layer PZT beam act as an adaptive tuned vibration absorber and they can go to resonance at a certain frequency and create a notch.

For the turbofan engine applications where the structure is affected by two disturbances ($N1$ and $N2$), the first piston and its 3-layer PZT beam properties are

selected such that the first notch frequency of the mount coincides with the frequency corresponds to the fan speed ($N1$). Similarly, the second piston and its 3-layer PZT beam properties are selected such that, the second notch frequency of the mount coincides with the frequency corresponds to the high-pressure compressor speed ($N2$). According to literature review [2], the current commercially available technology of the aircraft engines makes the balancing of the engine low-speed-shaft ($N1$) possible after engine assembly; thus the vibration due to unbalance of the shaft can be minimized and fine tuning is fair enough for to make the first notch, coincide with the ($N1$) speed. However, this is not possible for the high-speed shaft ($N2$) to be corrected by engine balancing process. So an acceptable range of tuning is required to tune the second notch. This semi-active fluid mount design (Design 1) resolves this problem by tuning the notch locations through capacitance shunt variation in a wide frequency range. This semi-active fluid mount design can also be developed into an active fluid mount by applying voltage across the PZT layers. Moreover, by adding an energy harvester device to the mounting system, it can be developed into a self-contained fluid mount and its power requirement for the sensors and actuator, which is using to adjust the capacitance shunt values can be harvested from the engine vibration.

Figure 1-2 Multiple-notch semi-active fluid mount design (Design 1)

1.2.2 Design 2: Semi-Active Fluid Mount with an Electromechanical Tuned Vibration Absorber

PZT beams are very brittle and they can only sustain low deflections; so, Design 1 is suitable for turbofan aerospace applications, where the input disturbances happen at high frequencies but very low displacements; also two disturbance frequencies ($N1$ and $N2$) happen very far from each other [2] and they should be controlled independently. However, for applications such as piston engine aircraft applications and automotive applications where there exist one significant input disturbance (firing pulse) and it happens at low frequency (80 Hz) [3] but higher displacement (± 0.1 to ± 0.7 mm), another semi-active fluid mount (Design 2) is designed and shown in Figure 1-3. In Design 2, the lower chamber of a conventional single pumper fluid mount is replaced by a variable stiffness electromechanical tuned vibration absorber (ETVA) system. Volumetric stiffness of the bottom chamber is varied by varying the shunt capacitance, connected to the voice coil, used in ETVA.

Design 2 is developed into a self-contained fluid mount by adding an energy harvester design to the mounting system. Power requirement for the sensors and controller, which are using to adjust the capacitance shunt value, is harvested by the energy-harvesting device (See Figure 1-4). This research will explain how an energy-harvesting device uses undesirable vibration of the engine in order to supply the required power of the mount's components.

Figure 1-3 Semi-active fluid mount design using an ETVA (Design 2)

Figure 1-4 Self-contained semi-active fluid mount design using an ETVA and an energy harvester device (Design 2)

1.2.3 Design 3: Active Fluid Mount with an Electromagnetic Actuator

Another focus of this thesis is on the design of an active fluid mount, addressed as Design 3 (Figure 1-5). This design is the development of Design 2, while a current is applied to the electromagnetic actuator (voice coil), used in the ETVA. The amplitude and phase of the actuator current are controlled such that in a wide range of frequencies, the transmitted force from the engine to the structure eliminates completely.

Since the material properties of the active fluid mount's components would vary by temperature or ageing of the system and the dynamic characteristics of the mount would change as time goes on, an adaptive controller is required. Here, in this thesis an adaptive controller based on Filtered-X LMS algorithm is used to control the actuator force. The secondary path transfer function, which is derived according to the bond graph model of the active fluid mount, guarantees the stability of the LMS algorithm. The active fluid mount parameters are selected such that it is suitable for both piston engine aircraft applications and automotive applications.

As it was explained, the combination of Design 2 and 3 creates a fluid mount with multi-functional capability (passive isolator, semi-active isolator, active isolator and damper); because, when no current is applied to the voice coil, the mount acts as a passive fluid mount. In the case that a variable capacitance circuit is connected to the voice coil, it works as a semi-active mount (refer to Design 2) and finally if a current is applied to the voice coil and it is controlled by a controller, an active fluid mount is created (refer to Design 3); and in the conditions which the cruise speed coincides with one of the engine's natural frequencies or a large shock is present, a high value

capacitance can be shunted to the coil such that the dynamic stiffness of the mount increases and the mount acts as a damper.

Figure 1-5 Active fluid mount design (Design 3)

1.3 Research Scope

The scope of current research can be summarized as:

- Proposing a semi-active multiple-notch single-pumper fluid mount by adding two triple-layer PZT cantilever beams to the conventional passive fluid mount (Design 1).
- Deriving the bond graph model of a multiple layer PZT beam.
- Proposing a semi-active single-pumper fluid mount by adding an electromechanical tuned vibration absorber (ETVA) system and a ladder capacitive circuit to the conventional passive fluid mount (Design 2).
- Fabricating an experiment test rig to justify the capability of the ETVA system in creating a variable stiffness apparatus.

- Developing a suitable control policy to adjust notch frequency of the proposed semi-active fluid mount designs according to the input disturbance frequency variation.
- Proposing a self-powered self-contained semi-active fluid mount by adding an energy harvester device to the semi-active mount design.
- Developing the semi-active model to an active model (Design 3).
- Investigating the application of an adaptive controller using the Filtered-X LMS algorithm in order to cancel out the engine's disturbance vibration in a wide range of frequencies.

1.4 Thesis outline

Chapter 2

This chapter is the literature review. It starts with an introduction to vibration isolation concept for engine mounts applications. Different types of engine mounts are introduced and their advantages and disadvantages are discussed. Since the focus of this research is on the fluid mounts, Chapter 2 is continued by a careful discussion of the various types of fluid mounts such as single-pumpers and double-pumpers. Various types of semi-active and active fluid mount systems are discussed as well. The chapter finishes by a conclusion, which shows the deficiencies of the existing engine mount designs and the remedy of this research study to resolve all these deficiencies.

Chapter 3

Chapter 3 starts with an introduction to single-pumper fluid mounts. The physical and mathematical models of a single-pumper fluid mount are presented. Thereafter, the physical and mathematical modeling of the Design 1, consisting of a single pumper

fluid mount and two PZT beams are presented according to the simple single-pumper fluid mount.

In order to investigate the capability of PZT beams in creating tunable multiple notches, the bond graph modeling technique is employed to derive state-space equations of the new mounting system (Design 1), because Design 1 is a multi-energy domain system (mechanical, hydraulic and electrical) and the bond graph modeling technique would be an efficient method to derive state-space equations of such systems. So, Chapter 3 is continued with the bond graph model of a multi-layer PZT beam. Thereafter, the bond graph model of whole system is defined and the state-space equations of the system are derived. The feasibility of Design 1 is investigated by doing MATLAB simulations and the application of capacitance shunt circuit in tuning notch frequency locations of Design 1 is shown for a turbofan engine application. Chapter 3 is ended with some practical considerations and a conclusion which shows pros and cons of Design 1.

Chapter 4

In Chapter 4, Design 2 is presented. First of all, an electromechanical tuned vibration absorber (ETVA) is introduced consisting of a voice coil and a single degree of freedom mass spring system. The natural frequency of the ETVA is varied and tuned by addition of an electronic circuit (RLC circuit). The design, mathematical model, and experimental data of the ETVA is presented and the effect of changing RLC parameters in varying natural frequency of the tuned vibration absorber (TVA) is discussed. Thereafter, the ETVA is used in conjunction with a passive single pumper fluid mount to create a variable notch frequency semi active fluid mount (Design 2).

The mathematical model of the Design 2 is then presented and the simulation results for an automotive application are shown.

Finally, the real-time control strategy to adjust the notch frequency of the Design 2 to the desired value is explained and the energy harvesting device is added to Design 2 in order to supply the required power of mount's controller from the engine vibration. Chapter 4 ends with a conclusion, which shows the capability and advantage of the proposed self-contained self-powered electromechanical fluid mounting design (Design 2).

Chapter 5

In this chapter, an active fluid mount design (Design 3) is proposed. The bond graph model of Design 3 is derived and according to the state-space equations of the system, the required actuator force, stroke, frequency bandwidth and size are investigated. An adaptive control strategy, namely Filtered-X LMS, is used in order to control the actuator force in real time. The primary and secondary path transfer functions of the Filtered-X LMS algorithm are derived according to the dynamic model of Design 3. Chapter 5 is finished by the simulation results and a conclusion.

Chapter 6

This is the conclusion chapter. The main outcomes and contributions of the research are highlighted and some suggestions for the future research are presented. The thesis ends with the references and appendices.

Chapter2 Literature Review

2.1 Introduction

In the past decades, the functional characteristics of engine mounting systems have been investigated carefully in both automotive and aerospace industries. Generally, engine mounting systems have three principal functions: (1) firmly support engine weight, (2) isolate structure (cabin or fuselage) from the engine vibrations, and (3) control and limit engine motion when large vibration inputs or shocks or engine resonances are present. In order to improve these functional characteristics of an engine mount, first of all, it is needed to understand the characteristics of vibration forces, created by the engine and the performance requirement of the engine mounting system, which can isolate these forces.

In aerospace applications, most turbofan engines have two rotating spools $N1$ and $N2$ which are sources of vibration in the fuselage. These rotating spools are created by the fan and the high pressure compressor rotating. For an example, a typical business jet has the fan speed ($N1$) of 175 Hz and the high pressure compressor speed ($N2$) of 433 Hz at cruise altitude [2]. The current commercially available technology of the aircraft engines makes the balancing of the engine low-speed-shaft ($N1$) possible after engine assembly; thus the vibration due to unbalance of the fan and its shaft can be minimized to some extent for ($N1$). However, this is not easily achievable for the high-speed shaft ($N2$), since high pressure compressor is not easily accessible. In summary, $N1$ and $N2$ frequencies can be considered as disturbances in aerospace applications. As the forces transmitted to the fuselage are proportional to the stiffness and damping of the mounts to isolate the vibration created at $N1$ frequency by the fan and at $N2$ frequency by the

high pressure compressor, a mount with low damping and low elastic stiffness at these two frequencies is required.

The frequency range and sources of vibration in automotive applications are different from the aerospace applications. In automotive applications, for an internal combustion engine, there exist two basic dynamic disturbances: the firing pulse due to the explosion of the fuel in the cylinders, and the inertia forces and torques generated by the rotating and reciprocating parts (piston, connecting rod, and crank). The frequency of these unbalance disturbances depends on the number of cylinders in the engine and are correlated to the engine speed. Generally, at low engine speeds (near idle), the engine disturbances cause the entire vehicle to shake. At higher speeds, when the engine disturbance force coincides with an acoustic resonance of the passenger compartment, a booming sound is created inside the vehicle compartment [3]. To isolate the vibration created by the engine imbalance disturbances at high frequencies (80 Hz), low damping and low elastic stiffness is required as the forces transmitted to the structure are proportional to the stiffness and damping of the mounts [4]. In order to get low transmissibility, the stiffness of the engine mount should be selected such that the natural frequency of the mounting system in a certain direction becomes below the engine disturbance frequency to avoid excitation of the engine during normal driving conditions. Therefore, the ratio of the frequency of the engine disturbance over the natural frequency of the mounting system should be kept as high as possible. This means that to obtain low transmissibility the engine mount stiffness coefficient should be as low as possible. To get low transmissibility at high frequencies, a low damping mount is also desirable. Reducing engine mount stiffness and damping has a drawback. If the elastic stiffness of an engine mount is too low, the engine mounting system cannot perform well in the shock excitation situations that may be caused by

uneven roads, sudden acceleration or braking. Low stiffness will lead to large static engine displacement and damage the engine components. So from this point of view, high stiffness and high damping are required to minimize the engine motion and absorb engine shake and resonance.

From the above discussions, it is concluded that a good engine mounting system is the one that support the engine statically, prevent engine bounce from shock excitation, and isolate cabin from vibration caused by engine disturbances in the complete speed range. To provide the best cabin noise and vibration reduction, the engine mounts are required to be “soft” (low elastic stiffness and low damping). To prevent engine bounce in the low frequency range, engine mounts should be “hard” (high elastic stiffness and high damping). The dynamic stiffness (ratio of the applied force to the engine side of the mount and the relative displacement between the two sides of the mount, F_{in}/X_{in}) of an ideal mount is shown in Figure 2-1 [5-6] which shows that the dynamic stiffness and damping of an engine mount should be frequency dependent. Since lower frequencies usually lead to larger amplitude of displacement and higher frequencies lead to smaller amplitude of displacement, the mounting system with an amplitude-dependent characteristic can also meet this requirement.

Figure 2-1 Dynamic stiffness of ideal mount [3]

Therefore the optimization of the engine mount dynamic stiffness is the goal in designing a good engine mounting system. Many studies have been conducted in this area and various types of engine mounts have been proposed. Before we proceed to our research, a literature review on the previous vibration isolator systems and understanding their limitations is required. The following sections are dedicated to the demonstration of various types of engine mounts as follow:

- Passive (non-fluid) mounts
 - Elastomeric Mounts
- Passive fluid mounts
 - Hydraulic mounts with one or multiple inertia tracks
 - Hydraulic mounts with inertia track and decoupler
- Semi-active Mounts
 - Electro-Rheological fluid mounts
 - Magneto-Rheological fluid mounts
 - Piezoelectric fluid mounts
 - Shape memory alloy engine mounts
- Active Mounts
 - Electromagnetic fluid mounts
 - Piezoelectric fluid mounts

2.2 Passive (Non-Fluid) Mounts

2.2.1 Elastomeric Mounts

Elastomeric (or rubber) mounts have been used to isolate vehicle structure from engine vibration since the 1930's [7]. Since then, many research studies have been done on

the material and structural design of the elastomeric mounts to improve their performance [6, 8]. The discussion of detailed structural designs of elastomeric mounts is found in the literature [9-10].

Elastomeric mounts can be designed for the necessary elastic stiffness rate characteristics in all directions for proper vibration isolation. They are compact, cost-effective, and maintenance free. Rubber to metal bonded elastomeric mounts are known to provide more consistent performance and longer life [10]. Elastomeric mounts can be represented by the familiar Voigt model which consists of a spring and a viscous damper as shown in Figure 2-2 [4]. The dynamic stiffness of an elastomeric mount is greater at higher frequencies than lower frequencies due to elastomeric material properties being frequency dependent and it is shown in Figure 2-3. This characteristic makes it difficult to design a mounting system that satisfies the design requirement of Figure 2-1 (ideal mount). A high stiffness or high damping elastomeric mounts can yield a low shake level at low frequency, but its performance at high frequencies will be poor. On the other hand, low stiffness and low damping yields low noise levels, but it induces a high shake level at low frequencies because of the shock excitation. The solution is to obtain a compromise between engine isolation and engine bounce.

Figure 2-2 Mechanical model for elastomeric mount [4]

Figure 2-3 Dynamic stiffness of elastomeric mount [4]

Due to this limitation of the elastomeric mounts, the study of engine mounting systems was conducted on passive hydraulic (fluid) mounts technology, which resolves this limitation of elastomeric mounts to some extent.

2.3 Passive Fluid (Hydraulic) Mounts

The 1st fluid mount patent dates back to 1945. In recent years, hydraulic mounts have become commercially popular [11]; because, they have provided better performance than elastomeric mounts, which have caused significant developments in vehicle designs to smaller and lighter ones. Various types of hydraulic mounts have been developed for the automotive and aerospace applications. It has been reported that in automotive applications, significant improvements in ride comfort and reduced noise levels has been achieved by using hydraulic mounts compared with the conventional elastomeric mounts [12]. For instance, Bernuchon [12] and Corcoron and Ticks [13] reported an improvement in the ride comfort and reduction of 5 dB noise levels .

All hydraulic mounts reported in the literature are conceptually similar but different in details of their design. Two typical hydraulic mounts, often used by industry are:

- Hydraulic mount with inertia track
- Hydraulic mount with inertia track and decoupler

The following sections explain the design of these two types in more details.

2.3.1 Passive Fluid Mounts with Inertia Track

A commonly used fluid mount, in the automotive and the aerospace industry to provide cabin noise and vibration isolation is a “passive fluid mount with inertia track”. There are two types of fluid mounts with inertia track designs:

- Single-Pumper [14]
- Double-pumper [15]

A typical passive single-pumper fluid mount consists of two chambers that are connected to each other through a tube or a hose called “inertia track”. The two chambers are filled with an incompressible fluid. Figure 2-4 shows the configuration of a typical passive single-pumper fluid mount. As the fluid mount is subjected to a sinusoidal displacement, the fluid will oscillate between the two fluid chambers (upper and lower chambers). At some specific frequency, the fluid goes to resonance. This frequency is called “notch frequency”.

Figure 2-4 A typical passive single-pumper fluid mount and its physical model [16]

The dynamic stiffness of fluid mounts is strongly frequency dependent. Figure 2-5 shows a typical dynamic stiffness (F_{in}/X_{in}) of a passive single-pumper fluid mount versus frequency.

Figure 2-5 Typical dynamic stiffness of a fluid mount versus frequency [16]

It should be noticed that the dynamic stiffness of the fluid mount gets its minimum value at the “notch frequency”. It is at this frequency that the best cabin noise and vibration isolation is achieved. Usually fluid mounts are designed such that their notch

frequency coincides with the frequency of the longest period of constant speed of the vehicle or aircraft.

For example, in the case of an aircraft, the fluid mount notch frequency is usually designed such that it coincides with the frequency of the cruise speed. Otherwise, vibration isolation would not be optimum [16]. However at the cruise speed, there are multiple disturbance inputs of different amplitudes and frequencies (engine, turbine and fan disturbances) and a passive fluid mount design has only one notch. Therefore, the notch can only be designed to coincide with only one of the disturbance inputs and other disturbance inputs can create noise in the cabin. So, it is highly desirable to have more than one notch frequency. For example, in the case of the aerospace applications, if the fluid mount could have two notches, one of them could be tuned to coincide with the N_1 frequency (engine low speed shaft imbalance frequency) and the other to coincide with the N_2 frequency (engine high speed shaft imbalance frequency) and the cabin noise and vibration isolation would be more effective.

Literature survey on this area was conducted to find out if there is any double or multiple notch frequency passive fluid mount design documented. Vahdati [16] has introduced a design for double notch fluid mount as shown in Figure 2-6. With the design of Figure 2-6, the mount will have one more notch frequency which is desirable but the location of the new notch is not easily tunable after manufacturing. In the design of Figure 2-6, re-tuning of the notch frequencies requires re-machining of the inertia tracks, or change in rubber modulus, which is very costly. Therefore, it doesn't resolve the retuning problem of single-pumper fluid mounts, however the second problem, requiring multiple notches has been resolved to some extent.

Figure 2-6 Physical model of the double notch fluid mount [16]

The limitation of the existing single-pumper fluid mount designs lead us to design a new type of single-pumper fluid mount which has compact envelope and has multiple notch frequencies which are easily tunable. On the next chapter (Chapter 3), a new fluid mount design will be presented and the mathematical and physical model of this multiple notch frequency fluid mount will be explained in more details.

In general, placing the notch frequency at the right location is not easy. Due to manufacturing tolerances, the notch frequency usually does not coincide with the desired value in the first manufacturing pass and as a result fluid mounts need to be redesigned and re-tuned several times. This procedure is time consuming and costly. Some researchers have proposed design methods to avoid this expensive trial and error procedure [17-18] but most of these methods normally involve experimental data for optimization which is again not desirable. Therefore, the current study was also focused on this matter in order to propose a new design which not only has multiple notches but also resolves retuning limitations.

Another type of passive fluid mount which had already designed is double-pumper fluid mounts. A detailed mechanical model for a double-pumper fluid mount is represented by Vahdati [15]. It was concluded that the proposed model of double-pumper is good for both linear and non linear simulations.

Figure 2-7 A double pumper fluid mount [15]

Figure 2-8 shows a very simplified model for single and double pumper mounts. It is worth mentioning that, double-pumper fluid mounts are more commonly used in aerospace applications because they are safer than single pumper fluid mounts. The fluid cavitation is also less probable in double-pumper fluid mounts. In a double-pumper fluid mount the top and bottom plates are connected. If one of the elastomers fails the other one is still connected. So double-pumper fluid mounts are fail-safe [15]. A comparison between single-pumper and double-pumper fluid mounts shows that single pumper fluid mounts cannot be easily pressurized due to low stiffness of lower chamber; so in single pumper mounts the fluid cavitation is very probable. In the case of double-pumper fluid mount both chambers have high stiffness so they can be pressurized easily.

Figure 2-8 A simplified model of a single and a double pumper fluid mount [15]

Overall, hydraulic mounts with simple inertia track can be tuned to have high damping at the frequency corresponding to shock excitation that can be used to reduce shock level. Usually the dynamic stiffness of a hydraulic mount with an inertia track is greater than that of a comparable elastomeric mount, except at the notch frequency. Therefore, a comparison between passive hydraulic mounts and pure elastomeric mounts reveals that, hydraulic mounts have better isolation performance only at notch frequency, but at other frequencies, they are worse than pure elastomeric mounts.

2.3.2 Passive Fluid Mounts with Inertia Track and Decoupler

As it was explained in the previous section, using a fluid mount without decoupler does not have good performance at high frequencies; but a decoupler can resolve this limitation. A decoupler functions as an amplitude limited floating piston. It makes hydraulic mount amplitude dependent at low amplitude displacement. It allows the mount to behave like an elastomeric mount to provide good vibration isolation at large displacement. It also allows the hydraulic mount to have a high stiffness in the case of shock excitation. A simple hydraulic mount with direct decoupler is shown in Figure 2-9. Figure 2-10 shows a hydraulic mount with floating decoupler.

Figure 2-9 Hydraulic mount with direct decoupler [19]

Figure 2-10 Hydraulic mount with floating decoupler [20]

The hydraulic mount with decoupler seems to have desired dynamic performance and is used in many vehicles. For the large input amplitude, the top compliance pumping action will cause the decoupler to bottom on its seats, terminating the flow of liquid around it and sending the flow through the inertia track. In this case, at extreme low frequency, fluid passes freely from the top chamber to the bottom chamber through the inertia track. Since the bulge stiffness of the rubber below is very low, the static stiffness of the hydraulic mount is approximately equal to the static stiffness of the top elastomeric section. As the frequency increases from 10 Hz [21] the fluid traveling through the inertia track creates additional damping, so that the dynamic stiffness

increases until a maximum value is reached (called the resonance frequency). Above the resonance frequency, the inertia track essentially closes off. When this occurs, the stiffness of the mount is approximately equal to the sum of the elastomeric stiffness and the volumetric stiffness. At high frequency, the dynamic stiffness of the hydraulic mount will be higher compared to the elastomeric mount. For low amplitude, fluid travels around the decoupler instead of through the inertia track and causes the mount to behave like an elastomeric mount. Although a decoupler adds some advantages to conventional hydraulic mounts, modeling of the hydraulic mount is essential in order to understand the dynamic response of the mount system. This type of hydraulic mount has been modeled as one-degree to multi-degree linear and nonlinear systems by many researchers [5, 8, 12]. The proposed models depend on the mount structure, analytical method used and purpose of the analysis. A typical example of the mechanical model for a hydraulic mount is shown in Figure 2-11 as reported by Miller [22]. In this model, a mass M , representing the fluid in the flow path of the mount, and a spring K_2 , representing the volumetric compliance of the rubber chambers have been added to the elastomeric model.

For detailed structural design and analysis, mass of the fluid in the mount must be considered, and the mounting system is usually modeled as a multi-degree nonlinear system. A lumped parameter non-linear mathematical model of hydraulic mounts is also used by simulating its decoupler switching mechanism and inertia track dynamics [8, 21, 23].

Figure 2-11 Mechanical model for hydraulic mount [22]

J. E. Colgate et al. [20] represented two nonlinear models of hydraulic mounts which includes floating piston (decoupler). Moreover, the frequency response of hydraulic mount to the sinusoidal and composite excitation was presented. Their paper gives a very good guideline to define all hydraulic mount parameter experimentally.

Another detailed discussion of the modeling of hydraulic mount is given by Flower. He used a linear model to analyze the dynamic behavior of the hydraulic. In his model, the decoupler was either always open or always closed [24]. Kim and Singh [21] have illustrated the applicability of such modeling techniques for limited frequency domains thereby introducing the limitation of the linear modeling techniques in that they never consider the true decoupler switching mechanism.). The effect of decoupler was studied by Corcoran and Ticks [13]. The damping effect of simple orifice and inertia track was compared by Flower [24]. By conducting parametric performance studies, it is possible to achieve optimal tuning of the hydraulic mount for specific engine vibration and noise control. Some effort has been made to carry out the optimization design of structural parameters [25-26]. It has also been shown that there exists an optimum set of mount design parameters for a specific engine mount. Recently, J. Christopherson and G. Nakhaie Jazar [19] gave a linear and non linear modeling of

two types of hydraulic engine mounts. Their results give a good guideline for selecting the parameters of the hydraulic mounts according to both linear and nonlinear models.

Some recent studies have been devoted to find methods which can be used to optimize the design of the passive-type hydraulic mount by way of using some of the aforementioned nonlinear modeling techniques. In addition to the analytical modeling approaches utilized by the researchers Shangguan and Lu [27] have illustrated the applicability of nonlinear finite elements in conjunction with fluid–structure interaction to the analysis of hydraulic mounts .

From the above discussion, it can be concluded that the liner and nonlinear modeling of the hydraulic mounts have already studied carefully and the functionality of the hydraulic mounts with frequency-dependent and amplitude-dependent characteristics which offer lower dynamics stiffness for vibration and noise isolation and high damping for shock excitation control, were considered.

However, by looking carefully through the dynamic stiffness characteristics of various types of passive mounting systems, it can be concluded that non-fluid passive mounts are suitable to isolate cabin from engine disturbances at the low frequency. Moreover, the passive fluid mount are designed such that, they can provide minimum dynamic stiffness at only one frequency (notch frequency). Therefore, it is highly desirable to create a passive or a semi-active fluid mount which is able to minimize dynamic stiffness at more than one frequency (multiple notch frequency). Moreover, it has the capability to retune its notch frequency without remanufacturing pass. These two

improvements have been considered in our study for the fluid mount design (See Chapters 3 and 4).

Beside passive engine mounts, semi-active and active fluid mounts have been studied before and so many designs have been reported in the literature. Semi-active and active mounting systems minimize the dynamic stiffness of the mount at the desired disturbance frequency and consequently isolate cabin from engine vibration. Various type of semi-active and active engine mount designs which have already been created, are explained in the following sections.

2.4 Semi-Active Fluid Mounts

Generally, an engine mount with a semi-active control mechanism is called a semi-active engine mount. Semi-active control can change the dynamic response of a system through controlling parameters of that system. The controlled parameters for a semi-active engine mount system can be stiffness, damping, inertia track size or fluid characteristics. Therefore, the proposed semi-active designs can be classified into four main groups:

- 1) Control both rubber stiffness and damping parameters
- 2) Control inertia track size
- 3) Control volumetric stiffness parameter
- 4) Control fluid characteristics
 - Electro-rheological (ER) fluid filled mount
 - Magneto-rheological (MR) fluid filled mount

2.4.1 Semi-Active Mounts with both Stiffness and Damping Parameters Controlled

A typical example of both stiffness and damping controlled engine mount system was reported by Shoureshi and Graf [26]. Their proposed semi-active hydraulic engine mount system was equipped with an external bleed and the system resonant frequency was controlled through the change of the system equivalent spring constant and damping by changing the fluid injection pressure and the restriction of orifices, respectively.

2.4.2 Semi-Active Mounts with Inertia Track Size Controlled

Another example of a semi-active engine mounts was introduced by Thanh et al. [28]. They proposed a hydraulic engine mount with the controllable inertia track. The notch and resonant frequencies of the mount were varied by the variation of the area of the inertia track. However, changing the geometry of the inertia track has a significant drawback on the flow resistance and in consequence damping characteristics of the mount; but it was ignored in their work. Zhang et al. [29] proposed a fluid mount with multiple inertia tracks. A precise fluid flow modeling was developed in their work and the non-linearity of the fluid in inertia track was taken into account; however, they also skipped the relation between the fluid resistance and fluid inertia. Since the isolation characteristics of the mount can be affected by varying the inertia track size due to the flow loss variation; inertia track size variation may not be so applicable for applications where the fluid mount is going to be used as an isolator.

2.4.3 Semi-Active Mounts with Volumetric Stiffness Parameters Controlled

An example of controlling the volumetric stiffness parameter of the mounting system was introduced by Vahdati and Ahmadian [30]. They introduced a variable bottom chamber volumetric stiffness fluid mount design, using a pinion and rack gear system. Their proposed design resolved the notch frequency tunability problem of a conventional passive fluid mount; however, their design envelope was a bit large.

2.4.4 Semi-Active Mounts with Fluid Characteristics Controlled

Beside, changing the geometrical properties of a passive fluid mount to create a semi-active fluid mount, many studies have been conducted on semi-active fluid mounts which are filled with controllable fluids such as magneto-rheological (MR) fluid or electro-rheological (ER) fluid but majority of these studies have been focused on control of damping characteristic in fluid mounts rather than on stiffness control.

2.4.4.1 Electro-Rheological (ER) Fluid Semi-Active Mounts

Electro-Rheological fluids (ER fluid) have drawn much attention in the semi-active control of engine mounts because of their rapid change in apparent viscosity properties when an electric field is applied. These properties were used to control the damping of the semi-active engine mounting systems. Ushijima [31] and Petek [32] used the ER fluid characteristic to increase damping at resonance and reduce the transmissibility for shock excitation. Duclos [33] proposed an ER fluid filled mount with multiple

inertia elements to obtain low notch stiffness by tuning a combination of ER fluid valve to isolate idle shake. Hong et al. [34-35] proposed some design for semi-active ER and MR mounts. Hong et al. [34], developed a semi-active ER fluid which operates in flow mode and is able to sustain a mass of 200 kg. Later on in another paper [35], they presented the performance of an ER mount working in squeeze mode. Hong et al. [36] have also carried a comparison between a squeeze mode and a flow mode ER mount concluding that the squeeze flow mount performs better at low levels of excitation. Several more references address some semi-active fluid mount designs using ER-fluid; however, due to high voltage requirement and other challenges which arise in ER fluids in high temperature, ER fluid mounts were put aside and recently more studies have been focused on semi-active mounts using MR fluid.

2.4.4.2 Magneto-Rheological (MR) Fluid Semi-Active Mounts

Seug-Bok Choi et al. [37] proposed a semi-active magnetorheological (MR) fluid mount and active piezoceramics mount to control the vibration of flexible structures. In their design, the fluid mount damping at the resonant frequency of the structure is controlled not the stiffness. Arzanpour et. al. [38] designed a magnetorheological (MR) bushing which provides variable damping and stiffness. However, the design envelope is large and it is mainly suitable for applications when damping needs to be controlled rather than stiffness. Ciocanel et al [39] proposed a mixed mode (flow and squeeze) magnetorheological (MR) fluid mount which performs well for the excitations with high amplitudes and low frequencies but in their research, change of damping of the fluid mount was the focus. So many other studies have been reported on semi-active MR fluid mounts [39] and most of them focused on changing the

damping properties of the engine mount by MR fluid medium.

In summary, the dynamic responses of the most semi-active fluid mount systems are very sensitive to the system parameters. Research work show that only through extensive parametric studies, it is possible to achieve the optimal turning of the mounting system. Almost all reported work show that the semi-active engine mounting systems are used mainly to improve the system performance at the low frequency range. For improving the high frequency vibration isolation, active control techniques should be employed [3]. However, in this thesis, some attempts have been made to resolve the frequency limitation of semi-active fluid mounts and they are explained in more detail. The semi-active fluid mount which is presented in here, changes the stiffness characteristics of the passive fluid mount to control the dynamic stiffness while most of the previous research studies focused on changing the damping characteristics of the mount and they are mostly improved the functionality of the mount in shock excitations. But the semi-active design in this research is mostly focused on improvement of the isolation characteristics of the engine mount. Moreover, here in this thesis, the design of the proposed semi-active fluid mount is developed to a self-contained self-powered mount and it would distinguish the proposed design from the current available semi-active mounts. The self-contained capability makes this semi-active fluid mount suitable for application which wiring and electrical power supply to the sensors and the controller are issue.

2.4.4.3 Active engine Mounts

As it was explained in section 2.1, engine mounting systems should be designed such that satisfy three contradictory functions: engine support, vibration isolation, and limit

engine motion. These conflicting requirements have promoted researchers for designing active mounting system which has a dynamic stiffness similar to Figure 2-1 (ideal dynamic stiffness) and limiting engine motion (due to shocks) was not the focus, particularly at high frequencies.

In active mounting systems, a counteracting dynamic force is created by one or more actuators in order to suppress the transmitted disturbance force from the engine to the chassis or fuselage. A typical active mounting system is shown in Figure 2-12. It consists of a passive mount (elastomeric or hydraulic), force generating actuator, structural vibration sensors and an electronic controller. The passive mounts are used to support the engine in the case of actuator failure. The controlled signal is provided to the actuator's controller by a structural vibration sensor which is mainly a force sensor or an accelerometer. This signal adjusts the amplitude and phase of the force generated by the actuator to minimize the measured sensor signal, so as to reduce the vibration. The controller can be either a feedback or feed forward type [40]. The active vibration control is typically implemented with the closed-loop controller utilizing on-line sensor measurement.

Figure 2-12 Structure of conventional active mounting system [41]

Active mounting systems can be categorized according to the type of actuators, which are used to suppress the transmitted force or according to the type of controllers which are used. In the last decade, different types of active engine mounts have been developed. Different types of actuators, which commonly used in active mounting systems, are:

- Pneumatic actuators
- Electromagnetic actuators
- Servo-hydraulic actuators
- Piezoceramics actuators
- MR-elastomers

Ozaki [42] proposed an active mounting system that used a pneumatic actuator. Hagino et al. [43] developed an active mounting system which used an electromagnetic actuator to generate the dynamic force which minimizes engine disturbance and reduces the frame vibration at engine idle speeds. In their design, the

actuator was placed parallel with conventional elastomeric engine mount and fixed on the front portion of the chassis cross member.

Muller et al. [44] proposed another type of active engine mount, which is a combination of a classical hydraulic mount, and an electromagnetic actuator to isolate high frequency vibration. Servo-hydraulic actuators are also used for that purpose [45]. Other electromagnetic type active mounting systems were developed by other researchers [40, 46-47].

More recently, active engine mounts with piezoceramic actuators were also developed to isolate vehicle vibration [48]. Halden Wanger and Klose [49] and Ushijima and Jumakawa [50] developed active mounting systems which are activated by piezoelectric actuators. Piezo-actuators have a very high-speed response, but their displacement range is very small and requires a suitable mechanism to increase the amplitude. This would add more cost to these types of active mounting systems and commercially, they are not desirable [51].

From the control strategy point of view, active mounting systems can be classified into two main categories: feedback and feed-forward controllers [40]. In the case of active mounting systems, feed-forward controllers have better performance than feedback controllers; Feedback controllers require precise modeling of the dynamic characteristics of the system to be controlled. But the material and geometrical properties of an active mounting system could change by aging or environmental effects such as temperature; so feedback control scheme would not be so reliable for mounting applications. A Feed-forward controller has an adaptive nature and resolves

this problem [40]. That's why so many studies have been done on active mounting system which has a feed-forward controller. The most common types of active mounting controllers are designed based upon adaptive algorithm such as LMS or Filtered-X LMS.

Nakaji et al. [47] developed an active mounting system which used electromagnetic actuator and was controlled based on Filtered-X LMS algorithm. Lee. et al. [40] also proposed an active mounting system which is also controlled by Filtered-X LMS algorithm. In their paper the experimental and theoretical results were compared and the success of such control strategy in active engine mounts was verified. A. j. Hillis et al. [52] developed similar active engine mount by using an electromagnetic actuator underneath of the decoupler of a conventional hydraulic mount and compared the functionality of Filtered-X LMS and Er-MSCI. They concluded that Er-MCSI strategy offers significant computational advantages as it does not require cancellation path modeling. It was shown that both Filtered-X LMS and Er-MSCI have similar performance and reduce chassis vibration by 50-90%. Recently, Bo-Ha Lee and Chong-Won Lee [41] developed another feed-forward control for an electromagnetic type active mount system which reduced the cost of previous designs while improved the performance of the mount.

In summary, in all the developed active engine mounting systems, the common goal is to design a mount which is very stiff at low frequencies and is tuned to be very soft at high frequencies. Moreover, the mounting systems have been designed such that they are able to perform both actively and passively. However current thesis aims to present an active fluid mount by using an electromagnetic actuator and capacitive circuit such

that it is able to act as a passive mount, semi-active mount and an active mount. Moreover, the design has the capability to be converted to a damper as and when needed. The multi-functional capability of the proposed design (passive, semi-active, active and damper) distinguishes the current design from the previous active engine mounting designs.

2.5 Concluding Remarks

After exhaustive literature review, the following deficiencies were noted:

- Only 1 multiple notch fluid mount design was documented.
- Variable notch semi-active fluid mount designs were documented but majority of these designs had a drawback and that was an increase in damping occurring when notch location was varied.
- Majority of the semi-active and active engine mount designs were not self-contained.
- For semi-active or active engine mount previous designs, no mechanism or control strategy was developed to handle shock.
- Many semi-active and active engine mount designs needed a source of power but power requirements were high.

Therefore, to address all these deficiencies, the current study will be focused on development of semi-active and active fluid mounts as below:

- Create a multiple notch frequency semi-active fluid mounts by using smart materials such as piezoelectric materials in a novel combination with a beam in

conventional fluid mounts and providing a capability of tuning notch frequency and dynamic stiffness of fluid mounts.

- Create a semi-active fluid mounts by using an electromagnetic devices such as voice coil to tune the notch frequency locations of the conventional passive fluid mount.
- Develop the semi-active fluid mount to a self-contained fluid mount design which has tunable notch frequency while it does not require any external power to run its controller and sensors.
- Create an active fluid mount which has a multi-functional capability and can act as a passive, a semi-active and an active mount and it can convert to a damper when is needed.

Chapter3 Application of Multi-layer PZT Beams in Semi-Active Fluid Mounts (Design 1)

3.1 Introduction

An engine mount is a device that is placed in between the engine and the fuselage or the car frame to isolate cabin from engine vibrations, to control engine shock, and to control cabin noise and engine resonance. Fluid mounts are tuned for a specific application so that their notch frequency coincides with the disturbance frequency, by selecting the proper parameters for the mount. Additionally, the fluid mount parameters are selected such that the notch is as deep (close to zero) as possible (See Figure 3-2) and the resonance peak is kept as low as possible. The notch depth and the resonance peak represent opposing requirements for the selection of fluid mount parameters in a sense that lowering one will result in increasing the other.

For the aerospace fixed wing applications where the engine has more than one disturbance frequency; it is highly desirable to have more than one notch frequency to coincide with the disturbance frequencies of the engine. In the case of the turbofan aircraft engine mounts, two notches could exist; one of them can be tuned to coincide with the $N1$ frequency (engine low speed shaft imbalance frequency-“fan speed”) and the other to coincide with the $N2$ frequency (engine high speed shaft imbalance frequency-“high-pressure compressor speed”). However, current passive fluid mount designs have two limitations. First of all, due to manufacturing tolerances, the notch frequency of the conventional passive fluid mount may not coincide with the desired value and a time consuming and costly redesign procedure is required to retune the

mount. Moreover, a conventional passive fluid mount only provides vibration isolation at one frequency (notch frequency) and not several frequencies.

According to the literature review, some research has been conducted to avoid the expensive redesign procedure [18, 53]. Although the proposed optimization methods may resolve retuning limitation of the current passive fluid mounts, they do not result in a fluid mount design that can isolate a structure from multiple input disturbances.

As it was explained in Chapter 2, a literature survey was conducted to find if there are any multiple notch fluid mount designs. A double notch fluid mount design was first proposed by Vahdati [16] and the design was shown in Figure 2-6. With his design, the fluid mount has two notch frequencies which is desirable but there are two disadvantages to this design. The first disadvantage is the fact that, the two notch frequencies are not easily tunable after manufacturing. The second disadvantage is that the design envelope of this design will be large. Therefore, the current study is focused on developing a novel multiple notch fluid mount design which is compact and the notches are easily tunable. This new design consists of a conventional single-pumper fluid mount plus two 3-layer PZT cantilever beams which act as two tunable tuned vibration absorbers; creating multiple notches.

Before we approach the new design, we first investigate the modeling of conventional single-pumper fluid mounts and its constitutive equations. The notch frequency equation which depends on the fluid mount parameters (the top 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) will be found and the effect of each parameter on the notch frequency will

be studied to understand which parameters are easier to vary and which ones are not. In the following section, all the fluid mount equations are explained in more detail.

3.2 Single-pumper Fluid Mounts

Fluid-filled (or fluid) mounts have been referred to in many publications by many different names, such as hydraulic mounts [8, 13], hydroelastic mounts [54] or Fluidlastic® mounts [55]. There are two types of fluid mounts, double pumper fluid mounts [16] and single pumper fluid mounts [14].

The focus of this research will be on single pumper fluid mounts since it is widely used in the automotive and the fixed wing aerospace applications.

Figure 3-1 A typical single pumper fluid mount and its physical model [14]

A passive fluid mount, as shown in Figure 3-1, 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 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, 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 3-2 shows a typical dynamic stiffness of a passive fluid mount versus frequency.

Figure 3-2 Dynamic stiffness of a typical fluid mount versus frequency

Figure 3-3 Bond graph model of the single pumper fluid mount of Figure 3-2 [14]

Since the fluid mount configuration is a multi-energy domain system (mechanical, hydraulic,...), bond graph modeling technique [56] is utilized to derive the constitutive equations of the system. Figure 3-3 shows the bond graph model of the Figure 3-2.

Most of the elastomers which are commercially used in the design of fluid mounts have very low damping ratios; so the volumetric damping of the rubber in the bulge direction has been considered negligible in the bond graph model of Figure 3-3. From the bond graph model, the following state space equations can be derived:

$$\dot{q}_3 = V_{in} - V_0 \quad (3-1)$$

$$\dot{q}_6 = A_p (V_{in} - V_0) - \frac{p_{10}}{I_{10}} \quad (3-2)$$

$$\dot{q}_9 = \frac{p_{10}}{I_{10}} \quad (3-3)$$

$$\dot{p}_{10} = \frac{q_6}{C_6} - \frac{q_9}{C_9} - R_8 \frac{p_{10}}{I_{10}} \quad (3-4)$$

In the above state-space equations, q_3 , q_6 , and q_9 are the generalized displacement variables, and p_{10} is 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 the transmitted force (effort in bond 1 of Figure 3-3) is measured, the following input force will be given:

$$F_{in} = \frac{q_3}{C_3} + R_2 V_{in} + A_p \frac{q_6}{C_6} \quad (3-5)$$

If the Laplace transformation is applied to all the state space and output equations, the following dynamic stiffness (defined as $K^* = F_{in}/X_{in}$) equation will be derived:

$$K^* = K'_r + \frac{K''_r}{\omega} S + A_p^2 K'_{vt} \frac{S^2 + \frac{R_o}{I_f} S + \frac{K'_{vb}}{I_f}}{S^2 + \frac{R_o}{I_f} S + \frac{K'_{vt} + K'_{vb}}{I_f}} \quad (3-6)$$

In the above equations, the symbols are defined as:

V_0 and V_{in}	Velocities across the mount, m/s
A_p	Effective piston 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 $j\omega$

The fluid inertia is equal to:

$$I_f = \frac{\rho L}{A_t} \quad (3-7)$$

Where ρ , L and A_t are:

ρ	Fluid density, kg/m ³
L	Inertia track length, m
A_t	Cross section area of the inertia track, m ²

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:

$$f_{notch} = \frac{1}{2\pi} \sqrt{\frac{A_p^2 K_{vt} K_{vb} + K_r (K_{vt} + K_{vb})}{I_f (K_r + A_p^2 K_{vt})}} \quad (3-8)$$

Equation (3-8) 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 . These parameters that were 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.

3.3 Fluid Mount Parameters

3.3.1 Piston Area

If one considers the fluid mount design of Figure 3-1, the conical shaped rubber component not only acts like a spring 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.

3.3.1.1 Vertical (or axial) Stiffness and Damping

The conical shaped rubber component, shown in Figure 3-1, has stiffness and damping in the axial direction, labeled as K'_r and K''_r ; respectively.

3.3.1.2 Top and Bottom Chamber Volumetric Stiffnesses

There are two volumetric stiffnesses, one for the top chamber rubber component and one for the bottom chamber rubber component. When the fluid pressure in a cavity increases, the rubber is forced to bulge out. So the change in cavity pressure divided by the change in cavity volume is defined as the volumetric stiffness. The rubber volumetric stiffness equation can be described as,

$$K_v = \frac{\Delta P}{\Delta V} \quad (3-9)$$

The fluid, in the chambers, in general can be considered incompressible and its volumetric stiffness is very high as compared to rubber volumetric stiffness; so, its compressibility can be ignored.

3.3.1.3 Inertia Track Length, Diameter, and Fluid Density

The fluid in the inertia track has inertia and it can be calculated using Equation (3-7) and it can be varied if the track dimensions such as length, diameter, or fluid density are changed.

3.3.1.4 Parameters easiest to vary

To be able to tune the notch frequency to the desired location, we need to be able to vary some of the fluid mount parameters without the need to redesign the fluid mount. Let's discuss which parameters are the easiest ones to change.

It is impractical to vary piston area, axial stiffness, or top chamber volumetric stiffness in real-time since all these parameters are coupled. Firstly, the rubber is a passive

uncontrollable element so changing its properties is not possible, but even if we were able to develop a special controllable rubber material such as Magnetorheological elastomers (MREs), since the piston area, axial stiffness, and top chamber volume stiffness are all coupled, a change in one affect the others; therefore, the tuning of the notch frequency becomes a difficult task.

It may appear that the easiest parameters to vary in real time for notch frequency placement are the inertia track length, the track diameter, or the fluid density since these parameters are decoupled and it does not involve rubber redesign. But to vary inertia track length and diameter, one needs to develop a controllable inertia track design and even one was able to do so, we need to bear in mind that any change to the track dimensions directly impacts the flow losses [29].

Currently, it is a common practice by most fluid mount manufacturers to fine-tune the notch by changing inertia track length, diameter, or fluid. But varying these parameters can cause problems with the notch depth since flow losses can increase (Figure 3-3). Changing fluid is also not always a practical thing to do since changing fluid changes fluid viscosity and that can impact notch depth. However, in the cases that fluid mount is used as a vibration reduction device it would be useful to increase the damping of the fluid mount in a specified frequency range by changing the apparent viscosity of the fluid.

3.3.1.5 Bottom Chamber Volumetric Stiffness

From all of the fluid mount parameters; it is believed 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 fluid mount parameter and it can be readily varied with the design that will be described soon.

In the new design, the flexibility of two 3-layer PZT cantilever beams, shunted to the capacitive circuits, are used to change the bottom chamber volumetric stiffness of the mount and in consequence, create two notches which can be tuned to coincide with the multiple engine disturbance frequencies. In the following sections, the physical model, the bond graph model and mathematical simulation of the new fluid mount will be presented and the effect of changing the capacitance shunt condition on the location of notches will be investigated.

3.4 Physical Model of a Fluid Mount with Two 3-layer PZT Cantilever Beams

The current research aims to design a single-pumper fluid mount, which is shown in Figure 3-4. Generally, a single pumper fluid mount consists of two fluid chambers, which are connected together by an inertia track. In order to change the bottom volumetric stiffness of the lower chamber, the lower elastomeric chamber is replaced with two springs, two pistons and two 3-layer PZT cantilever beams. The two lower pistons are supported by two springs, and by two 3-layer PZT cantilever beams. In order to seal the bottom fluid chamber, fabric diaphragms are attached on the top of the lower pistons. When the fluid enters the bottom fluid chamber, the increase in the fluid pressure forces the diaphragms and the pistons to displace and the piston displacements cause the springs and the 3-layer PZT cantilever beams to move. Basically, the fabric diaphragms transfer power from the bottom fluid chamber to the

lower pistons, springs, and the 3-layer PZT cantilever beams without impacting the volumetric stiffness of the bottom fluid chamber.

Figure 3-4 Double-notch semi-active fluid mount design

Figure 3-5 shows the physical model of the fluid mount with two 3-layer PZT beams (Figure 3-4).

Figure 3-5 Simplified physical model of the fluid mount with two 3-layer PZT beams

In order to derive the constitutive equations of the system, it is required to deal with three different energy domains: mechanical, electrical and hydraulic. Bond graph modeling technique is an efficient method in modeling multi energy domains [56].

According to the literature review, the bond graph model of a multi-layer PZT beam has never been developed by other researchers. Therefore, before modeling the system of Figure 3-5, it is necessary to first derive the bond graph model of a multi-layer PZT beam. The next section illustrates the bond graph model of a multi-layer PZT beam and thereafter the bond graph model of the whole system of Figure 3-5.

3.5 Bond Graph Model of a Multi-layer Piezoelectric Cantilever Beam

The bond graph model of a uniform single layer cantilever beam was derived by ref. [57]. This model was generated based on the Bernoulli-Euler beam model (See Appendix A). Similarly, for the multi-layer PZT cantilever beam, we start our modeling by taking the assumptions of the Bernoulli-Euler model. Dynamic partial differential equation of the multi-layer cantilever beam that describes the motion of a purely mechanical cantilever beam, can be derived from the equilibrium of forces and moments on an infinitesimal element of the beam as,

$$\frac{\partial^2}{\partial x^2} \left(YI \frac{\partial^2 u}{\partial x^2} \right) + \rho A \left(\frac{\partial^2 u}{\partial t^2} \right) = f(x, t) \quad (3-10)$$

Where $f(x, t)$ is the force per unit length applied to the beam, ρ and A are the beam's density and cross sectional area, YI is the averaged rigidity term defined from the neutral axis. In the case that cantilever beam consists of several piezoelectric layers (Figure 3-6), an electromechanical term that represents the piezoelectric effect, should

be added to Equation (3-10); considering 1D piezoelectric equations, derived by Ikeda [58],

$$T_1^i = \frac{S_1^i}{s_{11}^{E_i}} - \frac{d_{31}^i}{s_{11}^{E_i}} E_3^i \quad (3-11)$$

$$D_3^i = d_{31}^i T_1^i + \varepsilon_{33}^{T_i} E_3^i \quad (3-12)$$

Where s_{11}^E denotes the compliance, at constant electric field, and ε_{33}^T dielectric permittivity, at constant stress, for each layer. S_i and T_i are the strain and stress, in the 1 direction (x-axis). D_3 is displacement vector and E_3 is the electric field in the direction 3 (z-axis).

Equation (3-11) shows the relationship between stress and strain (first term in equation), plus an additional stress related to the applied electric field (second term in equation). In the equilibrium of forces in Equation (3-10), an additional bending moment that is created by the PZT stress, needs to be considered. Therefore, the new equilibrium, Equation (3-13) is derived from Equation (3-10) as:

$$\frac{\partial^2}{\partial x^2} \left(YI \frac{\partial^2 u}{\partial x^2} \right) + \mu \left(\frac{\partial^2 u}{\partial t^2} \right) = f(x, t) + \frac{\partial^2}{\partial x^2} (M_v) \quad (3-13)$$

Where μ is mass per length and “ M_v ” is the bending moment related to transversal forces appearing due to the piezoelectric effect and it is defined as:

$$M_v = w \int_z z T_1^i dz = \sum_{i=1}^m \int_{h_{io}}^{h_{iu}} - \frac{z d_{31}^i}{h_i s_{11}^{E_i}} V(x, t) dz = M_v^* V(x, t) \quad (3-14)$$

Where h_{iu} and h_{io} are the relative distance from the neutral axis to the upper and to the lower sides of the “ i ” layer; respectively, and $M_v^* = \sum_{i=1}^m \int_{h_{io}}^{h_{iu}} - \frac{z d_{31}^i}{h_i s_{11}^{E_i}} dz$.

Figure 3-6 Generic multi-layer cantilever beam

In order to solve Equation (3-13), some parameters such as the neutral axis ($\bar{z}$), flexural rigidity (YI) and total mass per length (μ) must be calculated. Figure 3-7 describes the schematic of a segment of a multi-layer beam with its neutral axis.

Figure 3-7 Derivation of the neutral axis position - schematics of a multilayered beam segment

The equation to define the neutral axis and the averaged flexural rigidity of the multi-layer beam (Figure 3-7) were described by Ballas et al. [59] as:

$$\bar{z} = \frac{\sum_{i=1}^m \frac{w_i}{s_{11,i}} h_i^2 - 2 \sum_{i=1}^m \frac{w_i}{s_{11,i}} h_i \sum_{j=1}^i h_j}{2 \sum_{i=1}^m \frac{w_i}{s_{11,i}} h_i} \quad (3-15)$$

$$YI = \frac{1}{3} \sum_{i=1}^m \frac{w_i}{s_{11,i}} \left[3h_i \left(\bar{z} - \sum_{j=1}^i h_j \right) \left(\bar{z} - \sum_{j=1}^{i-1} h_j \right) + h_i^3 \right] \quad (3-16)$$

Where the parameter w_i represents the width of each layer. For each shunt condition (open, short, shunted), the elastic compliance ($s_{11,i}$) is defined as [60] :

$$\text{Short circuit condition: } s_{11,i} = s_{11,i}^E \quad (3-17a)$$

$$\text{Open circuit condition: } s_{11,i} = s_{11,i}^E - \frac{d_{31}^2}{\varepsilon} \quad (3-17b)$$

$$\text{Capacitive shunt circuit condition: } s_{11,i} = s_{11,i}^E - \frac{d_{31}^2 w_i l_i}{h_i (C_p + C_s)} \quad (3-17c)$$

Where $s_{11,i}$ is the elastic compliance of the i th layer (m^2/N). For the non-piezoelectric elastic layers, it is equal to $1/E_i$. $s_{11,i}^E$ is the elastic compliance of the i^{th} layer at constant electric field. C_p is the inherent capacitance of the PZT layer and C_s for each layer is equal to the shunted capacitance. The parameters d_{31} , ε , s_{11} and C_p are provided by the PZT manufacturer. The parameter μ in Equation (3-13) is defined as:

$$\mu \equiv \frac{m}{l} = \sum_{i=1}^n w_i h_i \rho_i \quad (3-18)$$

In order to have a precise model, a dissipation force contribution should also be considered in Equation (3-13). To account for air damping and beam's material hysteretic damping, a damping term would be added to the Equation (3-13). One possible choice to model this damping contribution is to assign a viscous contribution (velocity dependence) proportional to the rate of strain in the beam as is shown in Equation (3-19), with a constant B called damping constant. This damping constant is also called Kelvin-Voigt damping [61] :

$$\frac{\partial^2}{\partial x^2} \left(YI \frac{\partial^2 u}{\partial x^2} \right) + \frac{\partial^3}{\partial x^2 \partial t} \left(BI \frac{\partial^2 u}{\partial x^2} \right) + \mu \left(\frac{\partial^2 u}{\partial t^2} \right) = f(x,t) + \frac{\partial^2}{\partial x^2} (M_v) \quad (3-19)$$

The first term of the right side of the Equation (3-19), $f(x,t)$ represents an external force per unit length applied to the beam and the second term, $\frac{\partial^2}{\partial x^2} (M_v)$ represents a force per unit length due to the external voltage applied to the PZT layers.

To solve the partial differential governing Equation (3-19), first of all the solution of the unforced governing equation is found by using separation of variables $u(x, t) = W(x)T(t)$ and considering the beam's boundary conditions.

$$W(0) = 0, \frac{\partial^3 W(l)}{\partial x^3} = 0 \quad (3-20)$$

$$W'(0) = 0, \frac{\partial^2 W(l)}{\partial x^2} = 0 \quad (3-21)$$

$$YIW^{(4)}(x)T(t) + BIW^{(4)}(x)T'(t) + \mu W(x)T''(t) = 0 \quad x \neq l \quad (3-22)$$

$$YIW^{(4)}(x)T(t) + BIW^{(4)}(x)T'(t) + \mu W(x)T''(t) = F_{in}e^{j\gamma t} \quad x = l \quad (3-23)$$

Spatial solutions $W_n(x)$ and natural frequencies are defined as:

$$\frac{-\mu W(x)}{W^{(4)}(x)} = \frac{YIT(t) + BIT'(t)}{T''(t)} = \frac{-1}{\omega_n^2} \Rightarrow \begin{cases} YI\omega_n^2 T(t) + BI\omega_n^2 T'(t) + T''(t) = 0 \\ \mu\omega_n^2 W(x) - W^{(4)}(x) = 0 \end{cases} \quad (3-24)$$

The above equations and boundary conditions results in the following mode shapes:

$$\left\{ \begin{aligned} W_n(x) &= A_n \left[\sin(\beta_n x) - \sinh(\beta_n x) - \frac{\sin(\beta_n l) + \sinh(\beta_n l)}{\cos(\beta_n l) + \cosh(\beta_n l)} (\cos(\beta_n x) - \cosh(\beta_n x)) \right] \\ n &= 1, 2, 3, \dots \\ \beta_1 l &= 1.8751, \beta_2 l = 4.6941, \beta_3 l = 7.8548, \dots \end{aligned} \right. \quad (3-25)$$

The natural frequencies are calculated as follows:

$$\omega_n = \frac{(\beta_n l)^2}{l^2} \sqrt{\frac{YI}{\mu}} \quad (3-26)$$

where YI and μ are found according to the Equations (3-16) and (3-18); respectively.

To find the temporal part $T_n(t)$, the forced Equation (3-19) needs to be solved. Since there are two forces per unit length terms in Equation (3-19), superposition method

would be used to solve for $T_n(t)$. We proceed to solve for temporal part by first considering $f(x, t)$ in Equation (3-19), and then for a voltage applied term ($\frac{\partial^2}{\partial x^2}(M_v)$).

If the force, $f(x, t)$ is assumed to be applied at tip of the beam as,

$$f(x, t) = F_{in}(t)\delta(x - l) \quad (3-27)$$

Equation (3-19) becomes

$$\frac{\partial^2}{\partial x^2} \left(YI \frac{\partial^2 u}{\partial x^2} \right) + \frac{\partial^3}{\partial x^2 \partial t} \left(BI \frac{\partial^2 u}{\partial x^2} \right) + \mu \left(\frac{\partial^2 u}{\partial t^2} \right) = F_{in}(t)\delta(x - l) \quad (3-28)$$

Thus by multiplying both sides of equation by $W_n(x)$ and integrating along the length of the beam, Equation (3-29) is derived as,

$$\begin{aligned} \int_0^l W_n(x) \frac{\partial^2}{\partial x^2} \left(YI \frac{\partial^2 u}{\partial x^2} \right) dx + \int_0^l W_n(x) BI \frac{\partial^3}{\partial x^2 \partial t} \left(\frac{\partial^2 u}{\partial x^2} \right) dx \\ + \int_0^l W_n(x) \mu \left(\frac{\partial^2 u}{\partial t^2} \right) dx = \int_0^l W_n(x) F_{in}(t) \delta(x - l) dx \end{aligned} \quad (3-29)$$

Using orthonormality and orthogonality properties of the mode shapes [62], Equation (3-29) is reduced to the following equation,

$$\begin{aligned} \left(\mu \int_0^l W_n^2(x) dx \right) \ddot{T}_n(t) + \omega_n^2 \left(\mu \int_0^l W_n^2(x) dx \right) T_n(t) + \\ 2\zeta_n \omega_n \left(\mu \int_0^l W_n^2(x) dx \right) \dot{T}_n(t) = F_{in}(t) W_n(l) \quad \zeta_n = \frac{B}{2Y} \omega_n \end{aligned} \quad (3-30)$$

Karnopp et al. [57] developed the bond graph model of a cantilever beam according to the Equation (3-30) for the case where a force is applied to an isotropic mechanical cantilever beam. However, for the multi-layer PZT beam, an additional mechanical bending moment, which occurs due to the electromechanical PZT effect and an applied voltage, needs to be considered (see Equation (3-19)). Assuming that the voltage is only a function of time and not x and the same voltage ($V_{in}(t)$) is applied to

all PZT layers, the spatial voltage function can be defined as [61]:

$$V(x, t) = V_{in}(t)(H(x) - H(x - l)) \quad (3-31)$$

Substitution of Equation (3-31) into Equation (3-19) results in the following equation:

$$\frac{\partial^2}{\partial x^2} \left(YI \frac{\partial^2 u}{\partial x^2} \right) + \frac{\partial^3}{\partial x^2 \partial t} \left(BI \frac{\partial^2 u}{\partial x^2} \right) + \mu \left(\frac{\partial^2 u}{\partial t^2} \right) = M_v^* \left(\frac{\partial}{\partial x} [\delta(x) - \delta(x - L)] \right) V_{in}(t) \quad (3-32)$$

The homogenous solution of Equation (3-32) is identical to Equation (3-25). For the temporal solution, similarly both sides of equation are multiplied by $W_n(x)$ and integrated along the length of the beam. Again, orthonormality and orthogonality properties of the mode shapes are used and the following equation is obtained:

$$\begin{aligned} & \left(\mu \int_0^l W_n^2(x) dx \right) \ddot{T}_n(t) + \omega_n^2 \left(\mu \int_0^l W_n^2(x) dx \right) T_n(t) + \\ & 2\zeta_n \omega_n \left(\mu \int_0^l W_n^2(x) dx \right) \dot{T}_n(t) = M_v^* V_{in}(t) \frac{\partial}{\partial x} (W_n(l)) \quad \zeta_n = \frac{B}{2Y} \omega_n \end{aligned} \quad (3-33)$$

Notice, in Equation (3-33), $M_v^* V_{in}(t)$ has the unit of the moment. Therefore the procedure explained by Karnopp et al. [57] for the applied moment is used to derive the bond graph model of a multi-layer PZT beam. The PZT layers inherent capacitance with a total capacitance equals to Equation (3-34) and it is completely described by Brufau-Penella et al.[61].

$$C_e = \sum_{i=1}^m \left(-d_{31}^{2i} \frac{wl}{h_i s_{11}^{Ei}} + \frac{\epsilon_{33}^{Ti} wl}{h_i} \right) \quad (3-34)$$

Figure 3-8 Bond graph model of a PZT multi-layer beam

By considering Equations (3-30), (3-33) and (3-34) and using the principles of bond graph modeling of a conventional cantilever beam [57] (See Appendix A), the above bond graph, (Figure 3-8) is derived for a Multi-layer PZT cantilever beam, where the modal masses, stiffnesses, and dampings are;

$$I_n = \left(\mu \int_0^l W_n^2(x) dx \right) \quad (3-35)$$

$$C_n = \frac{1}{\omega_n^2 \left(\mu \int_0^l W_n^2(x) dx \right)} \quad (3-36)$$

$$R_n = 2\zeta_n \omega_n \times I_n \quad (3-37)$$

and S_{e1} is equal to the applied force at the tip of the beam ($f_{in}(t)$) and S_{e2} is the applied voltage ($e_{in}(t)$). The transformers between the applied force and 1-junctions are the value of the mode shape functions (Equation (3-25)) at the beam's tip ($x=L$), and the

transformer between the applied voltage and 1-junctions are the value of derivative of the mode shape functions at the beam's tip ($x=L$) times M^*_ν . The compliance labeled C_e in Figure 3-8 is equal to the equivalent capacitance of the multi-layer PZT beam (Equation (3-34)).

It should be noted that, in order to verify the derived bond graph model of the PZT beam, the geometrical and material properties of a commercially available 3-layer cantilever beam were inserted to the MATLAB and the natural frequencies of the beam were derived and compared to the catalog data. The comparison shows very precise accuracy of the bond graph model.

In the case when a variable capacitance is shunted to the PZT layers, the shunted capacitance's value needs to be added to the capacitance of the beam; also its contribution in flexural rigidity of the beam needs to be considered according to Equations (3-16) and (3-17).

According to Equation (3-17), for the different shunt conditions (open, short, capacitive shunt), the elastic compliance of the i^{th} layer ($s_{11,i}$) varies and in consequence the flexural rigidity of the beam can be controlled by changing the ratio of capacitive shunted circuit (C_s) to the inherent capacitance of PZT layer from the open circuit ($C_s/C_p=10^{-4}$) to the short ($C_s/C_p=10^4$) conditions. Charnegie [60] proved that by changing the capacitive shunt condition from short circuit to open circuit the first natural frequency of the beam varies by 7.5%. We would take the advantage of this fact in the proposed design, which will be explained in more details in the next section.

3.6 Modeling of the Proposed Semi-Active Fluid Mount

Since the bond graph model of the PZT beam has been derived, the overall bond graph model of our design (Figure 3-5) can be developed. Here it is assumed that one end of the mount is fixed ($V_0 = 0$) and other end of the mount is subjected to an input velocity (V_{in}). Figure 3-9 shows the bond graph model of a conventional single-pumper fluid mount (Figure 3-3) plus the bond graph model of the two 3-layer cantilever PZT beams (Figure 3-8).

It should be noticed that in Figure 3-9, the first 3 modes of the 3-layer PZT beam models are kept, since the natural frequencies of the higher modes are not in the frequency range of our application. Moreover, keeping 3 first modes gives precise natural frequency results.

Here in this design, the PZT layers are not used as actuators, so there is no applied voltage to the PZT layers. So, the bond 2 in Figure 3-8, which is connected to an applied voltage, is removed in Figure 3-9. However, for active fluid mounts, one would be able to create an active mount with controlling the applied voltage to the PZT layers and in consequence control the displacement of the lower pistons.

Figure 3-9 Bond graph model of Figure 3-5 (Variable bottom chamber volumetric stiffness fluid mount design with two controllable notches)

The state-space equations, obtained from the bond graph model of Figure 3-9 are as follows:

$$\dot{q}_2 = V_{in} \quad (3-38)$$

$$\dot{q}_6 = V_{in} \times TF_{4-5} - \frac{p_8}{I_8} \quad (3-39)$$

$$\dot{p}_8 = \frac{q_6}{C_6} - \frac{p_8}{I_8} R_9 - \frac{q_{11}}{C_{11}} \quad (3-40)$$

$$\dot{q}_{11} = \frac{p_8}{I_8} - \frac{p_{14}}{I_{14}} \frac{1}{TF_{12-13}} - \frac{p_{49}}{I_{49}} \frac{1}{TF_{45-46}} \quad (3-41)$$

$$\dot{p}_{14} = \frac{q_{11}}{C_{11}} \frac{1}{TF_{12-13}} - \frac{p_{14}}{I_{14}} R_{15} - \frac{q_{16}}{C_{16}} - \frac{q_{18}}{C_{18}} \quad (3-42)$$

$$\dot{q}_{16} = \frac{p_{14}}{I_{14}} \quad (3-43)$$

$$\dot{q}_{18} = \frac{p_{14}}{I_{14}} - \frac{p_{27}}{I_{27}} \frac{1}{TF_{20-21}} - \frac{p_{30}}{I_{30}} \frac{1}{TF_{22-23}} - \frac{p_{33}}{I_{33}} \frac{1}{TF_{24-25}} \quad (3-44)$$

$$\dot{p}_{27} = \frac{q_{18}}{C_{18}} \frac{1}{TF_{20-21}} + \frac{q_{37}}{C_{37}} \frac{1}{TF_{43-44}} - \frac{p_{27}}{I_{27}} R_{26} - \frac{q_{28}}{C_{28}} \quad (3-45)$$

$$\dot{q}_{28} = \frac{p_{27}}{I_{27}} \quad (3-46)$$

$$\dot{p}_{30} = \frac{q_{18}}{C_{18}} \frac{1}{TF_{22-23}} + \frac{q_{37}}{C_{37}} \frac{1}{TF_{41-42}} - \frac{p_{30}}{I_{30}} R_{29} - \frac{q_{31}}{C_{31}} \quad (3-47)$$

$$\dot{q}_{31} = \frac{p_{30}}{I_{30}} \quad (3-48)$$

$$\dot{p}_{33} = \frac{q_{18}}{C_{18}} \frac{1}{TF_{24-25}} + \frac{q_{37}}{C_{37}} \frac{1}{TF_{39-40}} - \frac{p_{33}}{I_{33}} R_{32} - \frac{q_{34}}{C_{34}} \quad (3-49)$$

$$\dot{q}_{34} = \frac{p_{33}}{I_{33}} \quad (3-50)$$

$$\dot{q}_{37} = -\frac{p_{27}}{I_{27}} \frac{1}{TF_{43-44}} - \frac{p_{30}}{I_{30}} \frac{1}{TF_{41-42}} - \frac{p_{33}}{I_{33}} \frac{1}{TF_{39-40}} \quad (3-51)$$

$$\dot{p}_{49} = \frac{q_{11}}{C_{11}} \frac{1}{TF_{45-46}} - \frac{p_{49}}{I_{49}} R_{48} - \frac{q_{47}}{C_{47}} - \frac{q_{51}}{C_{51}} \quad (3-52)$$

$$\dot{q}_{47} = \frac{p_{49}}{I_{49}} \quad (3-53)$$

$$\dot{q}_{51} = \frac{p_{49}}{I_{49}} - \frac{p_{65}}{I_{65}} \frac{1}{TF_{57-58}} - \frac{p_{62}}{I_{62}} \frac{1}{TF_{55-56}} - \frac{p_{59}}{I_{59}} \frac{1}{TF_{53-54}} \quad (3-54)$$

$$\dot{p}_{65} = \frac{q_{51}}{C_{51}} \frac{1}{TF_{57-58}} + \frac{q_{75}}{C_{75}} \frac{1}{TF_{72-73}} - \frac{p_{65}}{I_{65}} R_{66} - \frac{q_{67}}{C_{67}} \quad (3-55)$$

$$\dot{q}_{67} = \frac{p_{65}}{I_{65}} \quad (3-56)$$

$$\dot{p}_{62} = \frac{q_{51}}{C_{51}} \frac{1}{TF_{55-56}} + \frac{q_{75}}{C_{75}} \frac{1}{TF_{70-71}} - \frac{p_{62}}{I_{62}} R_{63} - \frac{q_{64}}{C_{64}} \quad (3-57)$$

$$\dot{q}_{64} = \frac{p_{62}}{I_{62}} \quad (3-58)$$

$$\dot{p}_{59} = \frac{q_{51}}{C_{51}} \frac{1}{TF_{53-54}} + \frac{q_{75}}{C_{75}} \frac{1}{TF_{68-69}} - \frac{p_{59}}{I_{59}} R_{61} - \frac{q_{60}}{C_{60}} \quad (3-59)$$

$$\dot{q}_{60} = \frac{p_{59}}{I_{59}} \quad (3-60)$$

$$\dot{q}_{75} = -\frac{p_{65}}{I_{65}} \frac{1}{TF_{72-73}} - \frac{p_{62}}{I_{62}} \frac{1}{TF_{70-71}} - \frac{p_{59}}{I_{59}} \frac{1}{TF_{68-69}} \quad (3-61)$$

In the above state-space equations $q_2, q_6, q_{11}, q_{16}, q_{18}, q_{28}, q_{31}, q_{34}, q_{37}, q_{47}, q_{51}, q_{67}, q_{64}, q_{60}$ and q_{75} are the generalized displacement variables, and $p_9, p_{14}, p_{27}, p_{30}, p_{33}, p_{49}, p_{65}, p_{62}$ and p_{59} are the momentum variable. The state-space variables are defined as:

q_2 Relative motion across the mount

q_6 Top chamber change in volume

q_{11}	Bottom chamber change in volume
q_{16}	Displacement of lower piston, labeled piston 1 in Figure 3-5
q_{18}	Relative displacement between PZT beam tip (Beam 1) and lower piston, labeled piston 1 in Figure 3-5 (≈ 0)
q_{28}	First mode modal displacement of the PZT beam 1
q_{31}	Second mode modal displacement of the PZT beam 1
q_{34}	Third mode modal displacement of the PZT beam 1
q_{37}	Capacitance charge of the PZT beam 1
q_{47}	Displacement of lower piston, labeled piston 2 in Figure 3-5
q_{51}	Relative displacement between PZT beam tip (Beam 2) and lower piston, labeled piston 2 in Figure 3-5 (≈ 0)
q_{67}	First mode modal displacement of the PZT beam 2
q_{64}	Second mode modal displacement of the PZT beam 2
q_{60}	Third mode modal displacement of the PZT beam 2
q_{75}	Capacitance charge of the PZT beam 2
p_9	Integral of the pressure drop in the inertia track
p_{14}	Momentum of lower piston, labeled piston 1 in Figure 3-5
p_{27}	First mode modal momentum of lower PZT beam, labeled PZT Beam 1 in Figure 3-5
p_{30}	Second mode modal momentum of lower PZT beam, labeled PZT Beam 1 in Figure 3-5
p_{33}	Third mode modal momentum of lower PZT beam, labeled PZT Beam 1 in Figure 3-5
p_{49}	Momentum of lower piston, labeled piston 2 in Figure 3-5
p_{65}	First mode modal momentum of lower PZT beam, labeled Beam 2 in

Figure 3-5

- p_{62}

Second mode modal momentum of lower PZT beam, labeled PZT Beam 2 in Figure 3-5
- p_{59}

Third mode modal momentum of lower PZT beam, labeled PZT Beam 2 in Figure 3-5

The transmitted force to the chassis or fuselage is equal to the effort at bond 1 in Figure 3-9;

$$F_{in} = \frac{q_2}{C_2} + V_{in} \times R_3 + \frac{q_6}{C_6} T_{4-5}$$

(3-62)

Dynamic stiffness of the proposed mount can be obtained by taking the Laplace transformation of all state-space equations and determining F_{in}/X_{in} ratio.

3.7 Simulation Results for a Turbofan Engine Application

In order to do MATLAB simulation and find out the effect of 3-layer PZT beams, the capacitive shunt conditions and fluid mount parameters on the notch frequencies and the dynamic stiffness of the proposed system, it is required to define certain values for all of the parameters of the fluid mount. The Initial values (baseline parameters) for the fluid mount are presented in Tables 3-1.

Table 3-1 Fluid mount parameters

Fluid mount parameters	Unit	Parameters value
Real component of the vertical stiffness ($K_r' = \frac{1}{C_2}$)	$\frac{N}{m}$	2.05e6
Imaginary component of the vertical stiffness ($K_r'' = R_3\omega$)	$\frac{N}{m}$	19.6e4
Rubber damping (R_3)	$\frac{N.s}{m}$	522.03
Effective Piston Area (A_{pt}), T_{4-5}	m^2	0.0061

Inertia track area (A_m)	m^2	0.000366
Inertia track fluid inertia (I_8)	$\frac{N.s^2}{m^5}$	4.83e4
Inertia track flow resistance (R_9)	$\frac{N.s}{m^5}$	12.8e5
Top chamber volumetric stiffness ($\frac{1}{C_6}$)	$\frac{N}{m^5}$	1.1e11
Compressibility of the fluid ($\frac{1}{C_{12}}$)	$\frac{N}{m^5}$	2.1e14
Stiffness of support spring ($\frac{1}{K_1}$)	$\frac{N}{m}$	2000
Damping coefficient support spring, b (R_{15})	$\frac{N.s}{m}$	0
Stiffness of support spring ($\frac{1}{K_2}$)	$\frac{N}{m}$	4000
Damping coefficient support spring, b (R_{20})	$\frac{N.s}{m}$	0
Mass of piston 1, mb_1	Kg	0.169
Mass of piston 2, mb_2	Kg	0.0102
Area of piston 1, $Apb1$	m^2	0.0013
Area of piston 2, $Apb2$	m^2	0.0013

Fluid mount parameters, assumed here for the simulation, come from reference [14]. In most fluid mounts, the rubber used need to have a very low $\tan\delta$; therefore the imaginary component of the vertical stiffness of the rubber which is indicative of rubber damping is assumed very low. In addition, in the chambers, the fluid is considered incompressible. Incompressibility would be acceptable since the volumetric stiffness of the fluid is very high in comparison to the rubber volumetric stiffness; so, its compressibility can be ignored. Moreover, since in commercial fluid mounts the working fluid is insensitive to temperature, it is assumed that the fluid density remains constant during a wide range of temperatures.

In order to define the Initial values (baseline parameters) for the 3-layer PZT cantilever beams, application of the mount for a turbofan engine was considered.

According to the literature review, most turbofan engines, used in aerospace applications, have two rotating spools $N1$ and $N2$ which are sources of vibration in the fuselage. These rotating spools are created by the fan and the high pressure compressor rotating. For an example, business jet applications have the fan speed ($N1$) of 175 Hz and the high pressure compressor speed ($N2$) of 433 Hz at cruise altitude [2]. The current commercially available technology of the aircraft engines makes the balancing of the engine low-speed-shaft ($N1$) possible after engine assembly; thus the vibration due to unbalance of the fan and its shaft can be minimized and a very fine tuning is desirable for $N1$. However, this is not easily achievable for the high-speed shaft ($N2$). The proposed fluid mount design (Figure 3-5) resolves the multiple disturbances isolation problem and also it provides an easy tuning process. So, to do MATLAB simulation, the 3-layer PZT cantilever beam parameters are selected such that two notches of the fluid mount coincide with the $N1$ and $N2$ frequencies for a business jet. Tables 3-2 and 3-3 illustrate the parameters of the 3-layer PZT beams placed under pistons 1 and 2; respectively.

Table 3-2 3-layer PZT beam parameters (PZT Beam 1)

Parameter	Description	Unit	Value
w_1	Width of the lower layer (PZT)	mm	20
w_2	Width of the middle layer (Steel)	mm	20
w_3	Width of the upper layer (PZT)	mm	20
h_1	Thickness of the lower layer (PZT)	mm	0.28
h_2	Thickness of the middle layer (steel)	mm	0.28
h_3	Thickness of the upper layer (PZT)	mm	0.28
l_1	Length of the lower layer (PZT)	mm	50

l_2	Length of the middle layer (Steel)	mm	50
l_3	Length of the upper layer (PZT)	mm	50
ρ_1	Density of the lower layer (PZT)	Kg.m ⁻³	7750
ρ_2	Density of the middle layer (Steel)	Kg.m ⁻³	7550
ρ_3	Density of the upper layer (PZT)	Kg.m ⁻³	7750
$s_{11,1}$	Elastic compliance of the upper layer (PZT)	m ² N ⁻¹	1/(6.2e10)
$s_{11,2}$	Elastic compliance of the middle layer (Steel)	m ² N ⁻¹	1/(21e10)
$s_{11,3}$	Elastic compliance of the upper layer (PZT)	m ² N ⁻¹	1/(6.2e10)
$s_{11,1}^E$	Elastic compliance of the upper layer (PZT) at constant electric field	m ² N ⁻¹	1/(6.2e10)
$s_{11,2}^E$	Elastic compliance of middle layer (Steel) at constant electric field	m ² N ⁻¹	1/(21e10)
$s_{11,3}^E$	Elastic compliance of upper layer (PZT) at constant electric field	m ² N ⁻¹	1/(6.2e10)
d_{31}^1	Strain constant of upper layer (PZT)	mV ⁻¹	-320*10e-12
d_{31}^2	Strain constant of middle layer (Steel)	mV ⁻¹	0
d_{31}^3	Strain constant of upper layer (PZT)	mV ⁻¹	-320*10e-12
ϵ_{33}^{T1}	Permittivity of upper layer (PZT)	-	3800*8.9e-11

Table 3-3 3-layer PZT beam parameters (PZT Beam 2)

Parameter	Description	Unit	Value
w_1	Width of the lower layer (PZT)	mm	20

w_2	Width of the middle layer (Steel)	mm	20
w_3	Width of the upper layer (PZT)	mm	20
h_1	Thickness of the lower layer (PZT)	mm	0.35
h_2	Thickness of the middle layer (steel)	mm	0.35
h_3	Thickness of the upper layer (PZT)	mm	0.35
l_1	Length of the lower layer (PZT)	mm	50
l_2	Length of the middle layer (Steel)	mm	50
l_3	Length of the upper layer (PZT)	mm	50
ρ_1	Density of the lower layer (PZT)	Kg.m ⁻³	7750
ρ_2	Density of the middle layer (Steel)	Kg.m ⁻³	7550
ρ_3	Density of the upper layer (PZT)	Kg.m ⁻³	7750
$s_{11,1}$	Elastic compliance of the upper layer (PZT)	m ² N ⁻¹	1/(6.2e10)
$s_{11,2}$	Elastic compliance of the middle layer (Steel)	m ² N ⁻¹	1/(21e10)
$s_{11,3}$	Elastic compliance of the upper layer (PZT)	m ² N ⁻¹	1/(6.2e10)
$s_{11,1}^E$	Elastic compliance of the upper layer (PZT) at constant electric field	m ² N ⁻¹	1/(6.2e10)
$s_{11,2}^E$	Elastic compliance of middle layer (Steel) at constant electric field	m ² N ⁻¹	1/(21e10)
$s_{11,3}^E$	Elastic compliance of upper layer (PZT) at constant electric field	m ² N ⁻¹	1/(6.2e10)
d_{31}^1	Strain constant of upper layer (PZT)	mV ⁻¹	-320*10e-12
d_{31}^2	Strain constant of middle layer (Steel)	mV ⁻¹	0
d_{31}^3	Strain constant of upper layer (PZT)	mV ⁻¹	-320*10e-12
ϵ_{33}^{T1}	Permittivity of upper layer (PZT)	-	3800*8.9e-11

3.7.1 Simulation Results of Dynamic Stiffness versus Frequency for a Turbofan Engine Application

In this section the simulation results of dynamic stiffness versus frequency for a turbofan engine used in business jets, are presented. The state-space equations (3-38 to 3-61) and the parameters defined in Table 3-1, 3-2 and 3-3 were used in MATLAB program. Figure 3-10 shows the dynamic stiffness of the proposed mount versus frequency for the short circuit, capacitive shunt circuit and open circuit condition. Figures 3-11 and 3-12 show the close up view of the dynamic stiffness diagram around the lower ($N1$) and the upper ($N2$) notch frequency respectively.

Figure 3-10 Dynamic stiffness versus frequency of the proposed semi-active fluid mount (short circuit, capacitive shunt circuit and open circuit condition)

Figure 3-11 Fine tuning of first notch frequency to the *N1* engine speed versus frequency

Figure 3-12 Tuning second notch frequency to the *N2* engine speed versus frequency

As is shown in the above figures, by adjusting the capacitance shunt circuit conditions, applied across the piezoelectric layers, the effective elastic modulus of the PZT layer changes and hence the overall stiffness of the beam structure can be adjusted or tuned to a desired value. Therefore, in the case that two 3-layer PZT beams are connected to two pistons in the fluid mount structure, two notches are created which are tunable and

each of them can be tuned to the desired value. The tuning range of the lower notch ($N1$ frequency) is less than the tuning range of the higher notch ($N2$ frequency). As it was explained, the range of fine tuning for the $N1$ frequency is adequate; since $N1$ frequency corresponds to the engine low-speed shaft and at this frequency the engine is mostly balanced through the manufacturing process. However, balancing engine at the high-speed shaft ($N2$) can not be done through the engine manufacturing pass and the new mount design gives 32 Hz of tuning for the second notch by changing capacitive shunt condition. The control strategy for the capacitive tuning will be explained in Chapter 4.

For aerospace applications, the input displacement (X_{in}) is in the range of few mils [63]. By assuming the input displacement (X_{in}) equals to 1.5 mils (3.81×10^{-5} (m)), the transmitted force from the engine to the structure (F_{in}) can be defined. Figures 3-14 and 3-15 show the close up view of the transmitted force versus frequency (Figure 3-13), around the lower notch and the upper notch; respectively. As are shown in the figures, the transmitted force has reduced significantly. It should also be noticed that, the transmitted force around the lower notch frequency, even in a 10 Hz frequency range, is still very low (20 N), so if the fine tuning (15 Hz) around the lower notch does not make the lower notch exactly coincide with the low-engine speed, the transmitted force is still in an acceptable range. However, for the upper notch frequency, it would not be an issue, since the tuning range is wide enough (32 Hz) and the capacitive tuning can place the upper notch to coincide with the high-speed engine frequency.

Figure 3-13 Transmitted force versus frequency of the proposed semi-active fluid mount (short circuit, capacitive shunt circuit and open circuit condition)

Figure 3-14 Transmitted force versus frequency around the upper notch frequency (N_2)

Figure 3-15 Transmitted force versus frequency around the lower notch frequency ($N1$)

In summary, the following procedure can be followed to select the mount parameters for any application.

1. Select mount parameters and 3-layer PZT beams parameters such that the first notch frequency of the mount coincides with the frequency corresponding to the $N1$ engine speed and the second notch frequency to the frequency corresponding to the $N2$ engine speed.
2. By varying the capacitive shunt condition, tune the two notch frequencies to the desired location. It should be noted that, since the second notch frequency occurs at the high frequencies, the capacitive shunt variation can give a very wide tuning range.

3.7.2 Simulation Results of the Lower Pistons' Displacement versus Frequency for a Turbofan Engine Application

Since PZT sheets are very brittle and they are not able to sustain high deflections, the displacement of the lower pistons should also be plotted. Therefore this limitation was taken into account in our application for selection of proper PZT layers. Figure 3-16 shows the ratio of the piston displacement over the input displacement (X_{in}) versus frequency for piston 1 and Figure 3-17 shows the ratio of the piston displacement over the input displacement (X_{in}) versus frequency for piston 2.

Figure 3-16 Ratio of piston displacement over X_{in} versus frequency (Piston 1)

Figure 3-17 Ratio of piston displacement over X_{in} versus frequency (Piston 2)

Figure 3-18 Displacement of piston 1 versus frequency around $N/1$ frequency

Figure 3-19 Displacement of piston 1 versus frequency around $N2$ frequency

Figure 3-20 Displacement of piston 2 versus frequency around $N1$ frequency

Figure 3-21 Displacement of piston 2 versus frequency around $N1$ frequency

With an input displacement (X_{in}) of ± 1.5 mils ($\pm 3.81 \times 10^{-5}$ (m)), piston 1 and the tip of the corresponding 3-layer PZT beam (beam 1) have the displacement of $\pm 270 \mu\text{m}$ to $\pm 410 \mu\text{m}$ around the lower notch ($N1$) and the displacement of $\pm 135 \mu\text{m}$ to $\pm 145 \mu\text{m}$ around the upper notch ($N2$). (See Figure 3-18 and 3-19 respectively). Similarly, the displacement of piston 2 and the tip of the corresponding 3-layer PZT beam (beam 2) is simulated and is shown in Figures 3-20 and 3-21. Piston 2 has the displacement of $\pm 21 \mu\text{m}$ to $\pm 23 \mu\text{m}$ around the lower notch ($N1$) and the displacement of $\pm 380 \mu\text{m}$ to $\pm 540 \mu\text{m}$ around the upper notch ($N2$). (See Figures 3-20 and 3-21 respectively). As it was expected, piston 1 has higher displacement at $N1$ frequency and its displacement plays more important role on creating first notch ($N1$). Similarly, piston two has larger deflection at $N2$ frequency and its displacement has a visible role on creating second notch ($N2$); Thus, by changing capacitive shunt condition across PZT layers of the beam 1, the $N1$ notch frequency can be tuned and by changing capacitive shunt condition across PZT layers of the beam 2, the $N2$ notch frequency can be tuned to the disturbance frequency.

All the plotted displacements (Figures 3-18 to 3-21) around both notch frequencies, are in the range of maximum deflection of the selected PZT layers. So the PZT beams can be used safely. If there is an application where the PZT beam displacements are very large, one can limit the motion of the PZT beam using an elastomeric crash stop. Some practical issues should be taken into account and they will be explained in the next section.

3.8 Practical Consideration

When the design of Figure 3-4 is used, one has to be careful not to stress the three-layer PZT beams too much or else it would break or shatter. When a sinusoidal input displacement is applied to the fluid mount, the dynamic pressure, in each chamber, changes sinusoidally. So in order to make sure no fluid cavitations occurring in the fluid chambers due to the input displacement, the fluid chambers are pre-pressurized to some static pressure. This static pressure causes the fabric diaphragm to displace the piston and compress the spring underneath the pistons. The PZT beams need to be installed and connected to the pistons after the pistons are in static equilibrium (after pre-pressurization) and not before.

Another practical issue to bear in mind is that most PZT multi-layer beams can only withstand small dynamic motions. So the design of Figure 3-4 is more suitable for applications with small input displacements such as aerospace applications rather than automotive applications with large input displacements. In the next chapter, a semi-active fluid mount design will be developed which is suitable for automotive applications.

3.9 Conclusion

A new semi-active single-pumper fluid mount was described in this chapter. Two 3-layer PZT beams were used to vary the lower chamber volumetric stiffness and thus the locations of the notch frequencies. By adding a variable capacitance to the PZT layers, the proposed design provides not only the ability to tune the fluid mount after manufacturing but also the ability to tune the notch frequencies in the field; thus, providing the minimum cabin noise and vibration. This new design reduces the notch frequency tuning cycle time, and eliminates fluid mount redesigns and re-manufacturing processes; therefore, saving time and reducing cost. The proposed design is highly desirable for the aerospace applications where there exist multiple engine disturbance inputs of various frequencies. In summary Design 1 can act in

- 1) Passive mode
- 2) Semi-active mode
- 3) Semi-active mode plus and energy harvesting device
- 4) Active mode

In the case that capacitive shunt circuits across the PZT beams are not controlled and no voltage applied to the PZT layers, the PZT beams play the role of passive stiffnesses and the mount performs in the passive mode. When the capacitive shunt circuits across the PZT layers are controlled, the mount acts as a semi-active fluid mount. Moreover, if one desires, an energy harvesting device can be added to the semi-active mounting system, in order to convert the vibrational energy of the engine to the suitable electrical power which is used to run capacitive circuit controllers and sensors; thus creating a self-powered semi-active mount which acts independently while no external wiring is required. Self-powered capability will be developed similar

to the one which will be explained in the last section of chapter 4. Finally, the mount has ability to convert to the active mode by applying the voltage to the PZT beams.

Chapter 4 Design of an Electromechanical Tuned Vibration

Absorber and its Application in Semi-Active Fluid Mount

Design (Design 2)

4.1 Introduction

As it was explained in the previous chapter, PZT materials are brittle and they can sustain low deflections, therefore the multiple-notch frequency semi-active fluid mount design, proposed in Chapter 3 would be an adequate isolator for the aerospace applications, with low input disturbance displacements ($\pm 3.8 \times 10^{-5}$ mm). However, for automotive applications, where the input displacement to the engine mount is large (± 0.1 mm to ± 1 mm), the proposed semi-active fluid mount design (Design 1) can not be used safely and another semi-active fluid mount design is needed to be developed. Here in this chapter a semi-active fluid mount design is proposed for the automotive applications. This design is combination of a conventional fluid mount and an adaptive electromechanical tuned vibration absorber (ETVA).

An adaptive Electromechanical Tuned Vibration Absorber (ETVA) is a semi-active vibration control device which is tunable and it has the ability of reducing undesired vibration at any desired frequency.

Here in this chapter, an electromechanical tuned vibration absorber (ETVA) is introduced consisting of a voice coil and a single degree of freedom mass spring system. The natural frequency of the ETVA is varied and tuned by addition of an electronic circuit (RLC circuit). The design, mathematical model, and experimental

data of the ETVA is presented and thereafter, the effect of changing RLC parameters in varying natural frequency of the tuned vibration absorber (TVA) is discussed. The ETVA is used in conjunction with a passive single pumper fluid mount to create a variable notch frequency semi active fluid mount. The mathematical model and simulation results of the passive single pumper fluid mount plus the ETVA is then presented and discussed. Thereafter, the control strategy for the notch frequency tuning is presented and an energy harvesting device is added to the system. The energy harvesting device supplies the required power of controller and the sensors; thus the design becomes a self-contained semi-active fluid mount which can work independently (without any external power). This capability would add a significant advantage since self-contained fluid mount is suitable for the applications that wiring to the mount and power supply are important issues.

4.2 Literature Review on Adaptive Tuned Vibration Absorbers

A Tuned Vibration Absorber (TVA) is a common device, which is connected to a primary system to achieve significant attenuation in the vibration of the primary system subject to harmonic disturbance inputs. A passive TVA consists of a mass-spring-damper assembly attached to a primary system that is being acted upon by an external tonal excitation.

In general, TVAs are either tuned to the natural frequency of the primary system or tuned to the frequency of the disturbance input. When external excitation frequency is unknown or it is time-varying or the primary system parameters change, off-tuning of the TVA can occur. To resolve this off-tuning problem, many adaptive tuned vibration

absorbers (ATVA) have been developed to tune the natural frequency of the TVA to any desired frequency as and when needed.

To create an ATVA, either mass, stiffness, or damping is varied but literature review indicates that little work has been done in the area of mass variation and majority of the work done on ATVAs has been on variation of stiffness or damping. So ATVAs can be classified to two main groups:

- ATVA with variable stiffness
- ATVA with variable damping

4.2.1 ATVA with Variable Stiffness

The most common ATVAs have been developed using variable stiffness devices [66 to 81]. In the last two decades, variable stiffness devices were typically developed by using:

- Smart Materials
 - Shape Memory Alloys (SMA) [64-66]
 - Magneto-rheological Elastomers (MRE) [67-69]
 - Piezoelectric [70-74]
- Mechanical Means
 - Helical Spring [75]
 - Flexible Beam [76-78]
 - Electromagnetic Devices [79]

4.2.1.1 Variable Stiffness ATVAs Using Shape Memory Alloys (SMA)

Many researches have been done on variable stiffness devices which work upon shape memory alloys (SMA) characteristics [64-66]. Williams et al. [66] used three SMAs and steel wires configured as cantilever beams with a concentrated mass at the end to create an adaptive tuned vibration absorber (AVTA). The reported change in natural frequency ($\frac{\omega_{\max} - \omega_{\min}}{\omega_{\min}} * 100$) for their design was 31.5%. They succeeded to create an adaptive tuned vibration absorber with 31.5% frequency range variation; however, the response time of SMAs are not as quick as other smart materials.

4.2.1.2 Variable Stiffness ATVAs Using Magneto-Rheological Elastomers (MRE)

Generally, Magnetorheological elastomers (MREs) are solid polymers with dispersed polarizable particles. Application of a magnetic field to the MRE causes a change in stiffness. The elastomer is cured in a magnetic field, causing the magnetic particles to align in chains and remain aligned after the magnetic field is removed [67]. Zhou [68] experimented a single degree of freedom (SDOF) system with MRE. Zhou found that the natural frequency of his system could be varied from 1397.6 to 1773.5 rad/s meaning by 26.9%.

In another experiment, Albanese et al. [69] tested silicone mixed with several different percentages of iron particle concentrations. They concluded that at 35% iron content, as much as 400% change in frequency can be achieved when a magnetic field is applied. The response time of the MREs is much better than SMAs but when magnetic

field is applied to MREs, the damping is also significantly increased which makes the ATVA not very effective.

4.2.1.3 Variable Stiffness ATVAs Using Piezoelectric Device

Piezoelectric devices are another type of variable stiffness devices which have been studied and used in ATVAs. Stiffness of piezoelectric materials can be varied by connecting them to a capacitive shunt circuit. This and other methods of shunting piezoelectrics for vibration control were reviewed by Lesieutre [70]. Richard et al. [71], Clark [72], Ramaratnam et al. [73] and Davis and Lesieutre [74] used capacitive shunt circuit to vary the stiffness of the piezoelectric ATVA. Davis and Lesieutre reported 7.9% change in ATVA natural frequency. piezoelectric ATVAs are brittle, cannot sustain high displacements and the change in natural frequency of piezoelectric ATVAs is less than 10%.

4.2.1.4 Variable Stiffness ATVAs Using Shape Mechanical Means

Some mechanical concepts for creating an ATVA have been discussed in the literature. One is a vibration absorber that consists of a mass attached to a helical spring with a spring collar dividing the spring into two parts. The spring collar isolates part of the spring from the rest of the absorber and the number of coils used in the absorber can be changed by rotating the spring [75]. Another one is a vibration absorber which consists of a flexible cantilever beam attached by a mass at its free end. By varying the effective length of the beam, the absorber frequency can be varied [76]. Bonello et al. [77] introduced an active tuned vibration absorber by using a

pinned-pinned beam as the stiffness apparatus and the stiffness was changed by changing the curvature of the beam using an actuator.

The other proposed mechanical variable stiffness system consists of two leaf springs which are connected in opposition to each other and a stepper motor is used to increase the separation distance between the two springs. In this concept, the authors reported a change of natural frequency by 687% [78]. This design provides a wide frequency variation, but it requires an actuator and a controller which add more cost and complexity to the system.

4.2.1.5 Variable Stiffness ATVAs Using Electromagnetic Devices

Some research has been conducted on active electromagnetic vibration absorbers (ETVA). J. Liu et al. [79] designed an electromagnet vibration absorber (EMVA), whose stiffness is tuned actively. Their design consists of a clamped-clamped aluminum beam and a permanent magnet that is embedded in the center of the beam and placed between two poles of a C-shape electromagnet. By varying the current of the electromagnet, stiffness of the EMVA is adjusted instantaneously such that the absorber frequency is tuned. The drawbacks of their design are that it requires an electrical power and a controller which again add more cost and complexity to the system. Table 4-1 summarizes the literature review results for different variable stiffness devices that have been used to create adaptive tuned vibration absorbers.

Table 4-1 Summary of the developed active and adaptive tuned vibration absorbers

Article	Year	Device	$\frac{\omega_{\max} - \omega_{\min}}{\omega_{\min}} * 100$
Williams et al. [66]	2002	SMA	31.5 %
Albanese et al. [69]	2003	MRE (35% Fe)	400%
Lesieutre [70]	2000	Piezoelectric (Capacitive shunt)	7.9%
Walsh et al. [78]	1992	Leaf Spring	687%

4.2.2 ATVA with Variable Damping

Some researchers have used damping to shift the natural frequency of the TVAs and broaden the isolation frequency range. The following studies have used damping to create ATVAs.

- Smart Fluids
 - Electrorheological (ER) Damper
 - Magnetorheological (MR) Damper [80]

4.2.2.1 Variable Damping ATVAs Using Smart Fluids

Recently, many studies have been conducted on ATVAs using magneto-rheological (MR) and electro-rheological (ER) fluids. For an example, J.H Koo et. al. [80] developed a MR fluid TVA (MRTVA) which is able to reduce resonance peak and provide good isolation around the primary structure’s natural frequency. One of the

drawbacks of using damping to change the natural frequency of the TVA is that the percentage change in natural frequency is small.

This chapter intends to present an ATVA which is acting like a “vibration neutralizer”. An electromechanical tuned vibration absorber (ETVA) is presented which is easily tunable through an electrical circuit and its natural frequency changes in a wide frequency range. The proposed design is a semi-active TVA and does not need complicated controller. So, it has the capability of frequency tuning while it does not have the complexity of active TVAs. The developed design is explained in more detail in the following section.

4.3 Electromechanical Tuned Vibration Absorber

In this section the developed adaptive electromechanical tuned vibration absorber (ETVA) is explained in details. The ETVA consists of a mass, a spring, a voice coil and a RLC electrical circuit. The RLC circuit is shunted to the voice coil and is used to change the natural frequency of the TVA. A RLC circuit consists of a resistor, an inductor, and a capacitor element in series. Here, it is not intended to use all the three electrical elements simultaneously but one at a time. Meaning, the coil will be connected to either a variable resistor, or to a variable inductor, or to a variable capacitor. Figure 4-1 shows a picture of the developed ETVA which is fixed to a dynamic testing machine.

Figure 4-1 Adaptive Electromechanical Tuned Vibration Absorber (ETVA)

4.4 Physical Model of Electromechanical Tuned Vibration Absorber

The schematic physical model of the ETVA is shown in Figure 4-2. The math model consists of a mass (M) representing the mass of moving coil plus the mass of the steel disk, a spring (K) representing the total spring rate of the four springs, and a damper (C) representing the overall damping of the 4 springs. Figure 4-2 indicates that the moving coil is connected to a variable resistor, or an inductor, or a capacitor to change the natural frequency of the TVA. As is shown, permanent magnet and one end of the spring-damper are fixed to the rigid frame. Mass-spring-voice coil system is stimulated by a sinusoidal acceleration which is applied to the rigid frame. Two accelerometers are connected to the mass (M) and the rigid frame. The accelerometer which is connected to the mass (M) measures output acceleration (a_{out}) and another one which is connected to the rigid frame measures input acceleration (a_{in}).

Figure 4-2 Physical model of ETVA

Voice coil consists of a wire winding (coil) and a magnet. The coil can be modeled as an inductor and a resistor, so when a RLC circuit is added to the coil, the model of the coil plus the RLC circuit can be as follows:

Figure 4-3 Coil model plus the RLC circuit

Since the ETVA is a multi energy domain system (electrical, mechanical and magnetic), bond graph modeling technique was used to derive the constitutive equations of the system. Figure 4-4 shows the bond graph model of Figure 4-2.

Figure 4-4 Bond graph model of Figure 4-2

The following state-space equations of the system can be derived from the bond graph model:

$$\dot{q}_5 = Sf_1 - \frac{p_9}{I_9} \quad (4-1)$$

$$\dot{p}_9 = (Sf_1 - \frac{p_9}{I_9})R_4 + \frac{q_5}{C_5} + GY_{10-11} \frac{p_{12}}{I_{12}} \quad (4-2)$$

$$\dot{p}_{12} = GY_{10-11}(Sf_1 - \frac{p_9}{I_9}) - \frac{p_{12}}{I_{12}}R_{13} - \frac{q_{14}}{C_{14}} \quad (4-3)$$

$$\dot{q}_{14} = \frac{p_{12}}{I_{12}} \quad (4-4)$$

In the above state-space equations, q_5 and q_{14} are the generalized displacement variables; p_9 and p_{12} are the momentum variables and they can be defined as below:

- q_5 Relative displacement between rigid frame (shown as MTS frame) and Mass (M)
- q_{14} Induced electrical charge in the Voice coil-RLC circuit
- p_9 Mass momentum
- p_{12} Flux linkage variable (λ) which is equal to time integral of the induced

voltage in voice coil).

By taking Laplace transform of Equation (4-1) to (4-4), the ratio of mass acceleration (a_{out}) over the input acceleration (a_{in}) is derived as:

$$\frac{\frac{\dot{p}_9(s)}{sX_{in}(s)}}{\frac{a_{out}(s)}{a_{in}(s)}} = \frac{(sC_5R_4 + 1)(I_{12}C_{14}s^2 + C_{14}R_{13}s + 1) + s^2C_{14}C_5GY_{10-11}^2}{(I_9C_5s^2 + C_5R_4s + 1)(I_{12}C_{14}s^2 + C_{14}R_{13}s + 1) + s^2C_{14}C_5GY_{10-11}^2} \quad (4-5)$$

To simulate the design of Figure 4-2 in MATLAB, the following parameters were plugged into State-Space Equations (4-1) to (4-5).

K_{sp}	Stiffness of the spring, ($C_5 = 1/K_{sp}$), 509.94 N/m
R_{sp}	Damping of the spring, ($R_4 = R_{sp}$), 17 N.s/m
$M + M_{coil}$	Disk Mass (M) plus Coil mass ($I_9 = M + M_{coil}$), 1.44+0.204=1.644 Kg
α	Voice coil force sensitivity ($GY_{10-11} = \alpha$), 21.1 N/Amp
L	Voice Coil Inductance ($I_{12} = L$), 17.5 mH, 70 mH, 140 mH, 210 mH
R	Voice Coil Resistance ($R_{13} = R$), 9 ohms, 18 ohms, 36 ohms, 1000 ohms
C	Shunted Capacitance ($C_{14} = C$), 0, 1000 μf , 3300 μf

A parametric study was done to evaluate the effect of changing resistance, inductance

and capacitance on $\frac{a_{out}(s)}{a_{in}(s)}$ ratio. The parametric study defines how variation of

electrical components affects the natural frequency of the TVA and among variable resistor, variable inductor and variable capacitor, which one is more viable to use as a variable component in shunted electrical circuit. The following sections describe the simulation results.

4.5 Effect of a Shunted Variable Resistance (R) on Natural Frequency of the ETVA

First of all it was assumed that a variable resistance is connected to the coil. Therefore, the coil model plus the electrical circuit (Figure 4-3) includes the coil resistance (R_{coil}), coil inductance (I_{coil}), and a variable resistance (R_v). In bond graph model (Figure 4-4), R_{13} represents the combination of coil resistance and the variable resistance ($R_{coil} + R_v$). I_{12} defines coil inductance (I_{coil}) and C_{14} is assigned equals to $10e6 \mu f$ to eliminate the capacitance from the model.

Figure 4-5 shows how changing shunted resistance affects the natural frequency of the ETVA. When the shunted resistance (R_v) varies from very low (short circuit voice coil) to very high (open circuit voice coil), the natural frequency of the ETVA and the peak value of (a_{out}/a_{in}) increase. It can be verified by the bond graph modeling principles [57]. According to the bond graph model (Figure 4-4), when the shunted resistance (R_v) increases, the induced current ($i_{induced}$ =flow in bond 11) in coil-RLC circuit decreases; so the applied force ($F = GY_{10-11} * i_{induced}$ = force in bond 10) from voice coil to the SDOF mass-spring system decreases. Ultimately, when the shunted resistance goes to very high value, the induced current goes to zero and no force will apply from voice coil to the mass (M). In this case, the whole system reduces to a SDOF mass spring system. So natural frequency of the ETVA approaches to natural frequency of the SDOF system (2.8 Hz for the selected parameter in here).

On the contrary, when the coil's wires are short circuit and no shunted resistance (R_v) is added to the coil, the induced current in coil-RLC circuit will be high; therefore the applied force ($F = GY_{10-11} * i_{induced}$ = force in bond 10) from the voice coil to the mass

(M) is noticeable. This force intends to fight with the motion of SDOF system. Therefore whole ETVA apparently becomes rigid body and mass (M) follows the excited motion (a_{out}/a_{in} approximately 1). For the applications that the peak value of the (a_{out}/a_{in}) is going to be used; higher resistance can give higher (a_{out}/a_{in}) peak value.

Figure 4-5 Ratio of the mass acceleration to the MTS input acceleration (a_{out}/a_{in}) versus frequency (a variable resistance is shunted to the voice coil)

4.5.1 Voice coil with Superconducting Wires

The above simulation result (Figure 4-5) was shown for an ETVA including a voice coil with the coil resistance (R_{coil}) of 9 Ohms. However, if the coil is manufactured from very low resistance wires (approximately zero) such as superconductors which has a very low resistance in room temperature, the natural frequency of the ETVA would change in a very wide frequency range.

Figure 4-6 shows, when the coil of ETVA is made of a very low resistance (1 Ohms) wires, the natural frequency of the ETVA can be shifted by approximately 8 times from 19 Hz to 2.8 Hz when a variable resistance is shunted to it. According to bond graph model of the ETVA (Figure 4-4), when the coil resistance is very low and the coil's wires are short circuit, R_{13} is eliminated from the model; then just an inductance (I_{12}) is bonded to the gyrator (GY_{10-11}).

Figure 4-6 Ratio of the mass acceleration over the MTS input acceleration (a_{out}/a_{in}) versus frequency (for very low resistance voice coil and high resistance voice coil)

Based upon bond graph modeling principles, gyrator (GY) essentially interchanges the roles of effort (force) and flow (velocity). So, in the bond graph model, when an inductance (I) is bonded to a gyrator (GY), it acts like a capacitance (C) which is equivalent to a compliance in a mechanical system (See Figure 4-7).

$$\begin{aligned}
 \frac{1}{\rightarrow} \xrightarrow{\alpha} \ddot{Y} \xrightarrow{2} C &\equiv \frac{1}{\rightarrow} I & I_1 &= \alpha^2 C_2 \\
 \frac{1}{\rightarrow} \xrightarrow{\alpha} \ddot{Y} \xrightarrow{2} I &\equiv \frac{1}{\rightarrow} C & C_1 &= \alpha^2 I_2 \\
 \frac{1}{\rightarrow} \xrightarrow{\alpha} \ddot{Y} \xrightarrow{2} R &\equiv \frac{1}{\rightarrow} R & R_1 &= \frac{\alpha^2}{R_2}
 \end{aligned}$$

Figure 4-7 Bond graph principle [56]

Therefore when the low resistance coil is short circuit, a compliance is added up to the SDOF; so, the total stiffness of the ETVA increases and in consequence the natural frequency of the ETVA.

On the contrary, when a very high resistance is shunted to the voice coil, the induced current in coil-RLC circuit decreases and the contribution of voice coil in mass-spring motion is negligible, so the total ETVA acts like a SDOF system and its natural frequency would approach to the natural frequency of the SDOF mass-spring system. Generally, for the applications which the disturbance frequency is changed in a large frequency range, this type of ETVA would be quit applicable.

Literature review shows that scientists are looking for ways for superconductors to function at room temperature without resistance (loss of energy). Right now, superconductors with zero resistance are available but the cooling system with liquid nitrogen is required and commercially they are not suitable for the voice coil's wire. Once the technology of superconductors in room temperature is developed, the creation of an adaptive ETVA with a very wide frequency range would be possible.

4.6 Effect of a Shunted Variable Inductance (L) on Natural Frequency of the ETVA

Another parametric study was done on a shunted variable inductance. Here, the coil model plus the electrical circuit (Figure 4-3) consists of the coil resistance (R_{coil}), coil inductance (I_{coil}) and a variable inductance (I_v). In bond graph model (Figure 4-4), I_{12} represents the coil inductance plus the variable inductance ($I_{coil}+I_v$) and R_{13} shows the coil resistance. C_{14} is $10e6 \mu f$ (Actually bond 14 is eliminated).

Simulation result for the ETVA with the shunted variable inductance is shown in Figure 4-8. Changing inductance from 17.5 (mH) to 210 (mH) changes the natural frequency of the ETVA by 50%. Similar to the resistance case, here is also the simulation results can be verified by its bond graph model. According to the bond graph modeling principles, gyrator (GY) interchanges the roles of effort (force) and flow (velocity). So, in the bond graph model, when an inductance (I) is bonded to a gyrator (GY), it acts like a capacitance (C) which is equivalent to a compliance in a mechanical system. Therefore, by changing the shunted inductance, the compliance of the TVA varies and in consequence its natural frequency changes. However, it should be noticed that an inductance has some inherent resistance and when the inductivity is changed, the inherent resistance of the inductor is also varied simultaneously; so, the shunted inductance is not controllable independently. Therefore, creating an ETVA by using inductance would not be a suitable choice.

Figure 4-8 Ratio of the mass acceleration to the MTS input acceleration (a_{out}/a_{in}) versus frequency (Variable inductance)

4.7 Effect of a Shunted Variable Capacitance (C) on Natural Frequency of the ETVA

The last parametric study was done on connecting a variable capacitance to the voice coil. The coil model plus the electrical circuit (Figure 4-3) consists of the coil resistance (R_{coil}), coil inductance (I_{coil}) and a variable capacitance (C_v). In bond graph model (Figure 4-4), R_{I3} represents the coil resistance (R_{coil}), I_{I2} defines coil inductance (I_{coil}) and C_{I4} shows the variable capacitance (C_v).

Figure 4-9 Ratio of the mass acceleration to the MTS input acceleration (a_{out}/a_{in}) versus frequency (Variable capacitance)

In the MATLAB simulations, the shunted capacitance in voice coil electrical circuit was changed from 0 (very low capacitance) which is equivalent to open circuit to 1000 μf , 3300 μf and very high capacitance which is equivalent to short circuit. Figure 4-9 shows the ratio of the mass acceleration to input acceleration (a_{out}/a_{in}) versus frequency as shunted capacitance is changed. This result can be justified by the bond graph model as well. As it was explained for the variable inductance, gyrator (GY) interchanges the roles of effort (force) and flow (velocity). So, when a capacitor (C) is connected to a Gyrator (GY), it plays the role of an inductor and it is equivalent to a mass in mechanical system. Therefore, when the capacitance is varied, it seems the apparent mass of the TVA is changed and it changes the natural frequency of the TVA. For the current selected parameters, when the shunted capacitance is changed from very low value (≈ 0) to 3300 μf , natural frequency of the TVA changes by 66%, from 2.5 Hz to 1.5 Hz, which translates $\frac{\omega_{max} - \omega_{min}}{\omega_{min}} = 0.66$. As is seen in Figure 4-8

changing capacitance not only changes the natural frequency of the TVA but also the peak value of the (a_{out}/a_{in}) at resonance frequency.

The peak value reduction happens because of the springs' damping, also coil resistance. So, in order to have less reduction, all damping and resistance should be kept as low as possible. If the damping of the springs and resistance of the coil are kept very low, Equation 4-5 reduces to

$$\frac{\dot{p}_9(s)}{sX_{in}(s)} = \frac{a_{out}(s)}{a_{in}(s)} = \frac{(I_{12}C_{14} + C_{14}C_5GY_{10-11}^2)s^2 + 1}{(I_9C_5I_{12}C_{14})s^4 + (C_{14}C_5GY_{10-11}^2 + I_9C_5 + I_{12}C_{14})s^2 + 1} \quad (4-6)$$

The natural frequency of the ETVA is derived by taking derivative of Equation (4-6) and setting it to zero,

$$\begin{aligned} \frac{d}{ds} \left(\frac{a_{out}(s)}{a_{in}(s)} \right) &= 0 \\ \Rightarrow \frac{d}{ds} \left(\frac{(I_{12}C_{14} + C_{14}C_5GY_{10-11}^2)s^2 + 1}{(I_9C_5I_{12}C_{14})s^4 + (C_{14}C_5GY_{10-11}^2 + I_9C_5 + I_{12}C_{14})s^2 + 1} \right) &= 0 \end{aligned} \quad (4-7)$$

$$s^2 = \frac{-1 \pm \sqrt{(-C_5GY_{10-11}^2 / I_{12})}}{(I_{12}C_{14} + C_{14}C_5GY_{10-11}^2)} = \frac{1}{C_{14}} \frac{-1 \pm \sqrt{(-C_5GY_{10-11}^2 / I_{12})}}{(I_{12} + C_5GY_{10-11}^2)} \quad (4-8)$$

$$\left\| \frac{a_{out}(i\omega)}{a_{in}(i\omega)} \right\|_{\omega=\text{resonance frequency}} = \left(1 - \frac{I_9C_5}{I_{12}C_{14}} \left(1 + \sqrt{\frac{C_5GY_{10-11}^2}{I_{12}}} \right)^{-1} \right)^{-1/2} \quad (4-9)$$

Equation (4-8) and Equation (4-9) verify the MATLAB simulation results. Equations (4-8) and (4-9) show that increasing capacitance (C_{14}) not only decreases the natural frequency of the ETVA but also the peak amplitude of the (a_{out}/a_{in}) at resonance frequency. It was mentioned earlier that damping and capacitance both impact the resonance peak amplitude, but it is important to mention that if the coil resistance is

high, when capacitance is increased, the peak amplitude drops more aggressively than without any damping.

In summary, a comparison between Figure 4-5, Figure 4-8 and Figure 4-9 shows that the variable capacitance has more effect on changing natural frequency of the TVA than changing resistance or inductance. Moreover, in compare to other tunable TVA devices, which was discussed in the previous section, this device provides more tunability frequency range ($\approx 60\%$) with a simple electrical circuit. The parametric study and simulation results show that creating an ETVA by using a variable capacitor would be the best viable choice. However, when the technology of superconductive wires in room temperature will be improved, using a voice coil made of superconductive wires and adding a variable resistance, would create the best adaptive ETVA which is tunable in a large frequency range.

4.8 Experimental Verification

In order to justify the capability of the proposed ETVA, an experimental test rig was developed. Here in section, the ability of a variable capacitance in creating an adaptive ETVA was examined and compared with simulation results. Similar experiment can be done for an ETVA with variable inductance and variable resistance.

As is shown in Figure 4-10, the designed apparatus consists of 4 axial stainless steel springs ($K_{eq}=509$ N/m, $R_{eq}=20$ N.s/m), a steel disk ($M=1.44$ Kg), a voice coil ($M_{coil}=0.204$ Kg, $L=17.5$ mH, $R=9$ Ohms, $\alpha=21.2$ N/amp), and an electrical board consisting various capacitors. A sinusoidal driving displacement is applied to the system by the dynamic testing machine (MTS810 machine). Two piezoelectric

accelerometers which connected to the steel disk and the lower plate of test rig measure the mass acceleration (a_{out}) and MTS input acceleration (a_{in}) respectively. The capacitance which is shunted to the coil was changed in a range, similar to the simulation (from 0 (very low capacitance) which is equivalent to open circuit to 1000 μf , 3300 μf and very high capacitance which is equivalent to short circuit). Practically, there are two possibilities for variable capacitance circuit. Using commercially available variable capacitors or design a customized arrays of capacitors from the available constant capacitors on a circuit board which covers the required capacitance range. However, the size of capacitors in the array should be chosen such that a full range of tuning (from very low (open circuit) to high (short circuit) could be observed with a good accuracy. Here in this experiment an array of capacitors was put on circuit board.

Experimental results are plotted in Figure 4-11. A comparison between Figure 4-8 and Figure 4-11 reveals that the experimental data and theoretical simulation data are very close. It verifies the viability of the proposed ETVA design. Similarly, ETVA with variable inductance and variable resistance can be tested. Since the viability of the proposed ETVA is justified, it can be used in semi-active fluid mount design to change the notch frequency location.

Figure 4-10 Experimental test rig of ETVA

Figure 4-11 Experimental data for the ratio of mass acceleration to the MTS input acceleration (a_{out}/a_{in}) versus frequency

4.9 New Semi-Active Fluid Mount Design

As it was explained in Chapter 3, practically, in passive fluid mounts, the notch frequency location changes due to tolerances on all the fluid mount dimensions, material property variations, and variation in elastomer molding processes and would not coincide with the designed notch frequency location. These variations would cause

remufacturing costs. Therefore, it is desirable to design a semi-active fluid mount whose notch frequency is easily tunable at any desired location after the manufacturing pass. Moreover, in automotive applications, for an internal combustion engine, there exist two basic dynamic disturbances: primary disturbances (the inertia forces and torques caused by the rotating and reciprocating parts such as piston, connecting rod, and crank) and secondary disturbances (the firing pulses due to the explosion of the fuel in the cylinders) [3]. The engine speed of 4-cylinder car passengers in the cruise condition (highway driving condition) varies from one engine manufacturer to the other, but it is usually in the range of 2000 rpm to 2500 rpm. For example, for a typical 4-cylinder car with the engine speed of 2400 rpm (40 Hz) in the cruise condition (highway driving condition), the rotating and reciprocating parts of the engine such as the piston, the connecting rod, and the crank rotate at 40 Hz but the cylinders fire at twice the speed of the engine, namely 80 Hz. Engine manufacturers make great attempts to balance the rotating and reciprocating parts of an engine in the manufacturing pass and minimize the vibration induced due to the primary disturbances. However, the vibration created by the firing of the cylinders can affect the vehicle compartments and it is highly desirable to design the fluid mount such that it eliminates the effect of secondary disturbances (firing pulses). In summary, the mount parameters variations in manufacturing pass and the secondary disturbances have conducted the researchers to develop semi-active fluid mounts, based upon changing one of the fluid mount parameters. As it was explained in the literature review (Chapter 2), most of the previous studies on semi-active fluid mount designs, focused on damping characteristics of the mount and the isolation characteristics of the mount have not been developed. Moreover, the previous proposed designs do not have a wide notch frequency variation range. In this chapter, we aim to propose a multi-

functional fluid mount which can act as a passive, semi-active and active isolator and can be converted to a damper as and when needed. In this chapter the semi-active mode of this multi-functional fluid mount is developed and in Chapter 5, the active isolator mode and the damper mode of the multi-functional fluid mount will be developed.

In Chapter 3, it was shown that among all fluid mount parameters, some parameters are easier to vary than others. It was proved that the easiest fluid mount parameter to vary to get a variable notch frequency fluid mount is the bottom chamber volumetric stiffness. In Chapter 3, the application of PZT beams in changing bottom chamber volumetric stiffness of the fluid mount was studied. Here in this chapter, the application of the proposed adaptive electromechanical tuned vibration absorber (ETVA) system in changing the bottom chamber volumetric stiffness of a single pumper fluid mount will be presented and the performance of a new variable notch frequency fluid mount is determined.

4.10 Physical Model of the New Semi-Active Fluid Mount Design

Figure 4-12 shows the physical model of the new semi active fluid mount design where the lower chamber of a conventional single pumper fluid mount is replaced by the proposed ETVA system.

Figure 4-12 shows, in the bottom fluid chamber, a piston is supported by a spring. A fabric diaphragm (with high volumetric stiffness) is placed on the top of the piston, sealing the bottom fluid chamber. As fluid enters the bottom chamber, it moves the piston (cup). When the piston moves, it moves the coil in and out of the permanent

magnet. As coil moves up and down in permanent magnet, current generates in coil's copper wire. The coil has N turns of pure copper wire. The coil wire has some inductance, but a variable capacitance is also added to the coil's electrical circuit, for the purpose of varying natural frequency of the ETVA. Since the coil has N turns of wire, there will be some electrical resistance in the wire. With most fluid mount designs, great efforts are made to minimize flow losses and damping; so, here again ideally not only we would like to have very low inertia track flow losses, and rubber damping, but also we would like to have very small resistance in the coil wires and the entire electrical system. In this design, volumetric stiffness of the bottom chamber can be varied by varying the shunted capacitance which is connected to voice coil.

Figure 4-12 Variable bottom chamber volumetric stiffness fluid mount design

Figure 4-13 Bond graph model of Figure 4-12

The bond graph model of Figure 4-12 was developed and is shown in Figure 4-13.

From the bond graph model (Figure 4-13), the following state-space equations can be obtained:

$$\dot{q}_3 = V_{in} \quad (4-13)$$

$$\dot{q}_6 = A_p V_{in} - \frac{P_{10}}{I_{10}} \quad (4-14)$$

$$\dot{p}_{10} = \frac{q_6}{C_6} - R_8 \frac{p_{10}}{I_{10}} - \frac{q_{11}}{C_{11}} \quad (4-15)$$

$$\dot{q}_{11} = \frac{p_{10}}{I_{10}} - A_{pb} \frac{p_{14}}{I_{14}} \quad (4-16)$$

$$\dot{p}_{14} = A_{pb} \frac{q_{11}}{C_{11}} - \frac{q_{15}}{C_{15}} - R_{16} \frac{p_{14}}{I_{14}} - GY_{17-18} \frac{p_{19}}{I_{19}} \quad (4-17)$$

$$\dot{q}_{15} = \frac{p_{14}}{I_{14}} \quad (4-18)$$

$$\dot{p}_{19} = GY_{17-18} \frac{p_{14}}{I_{14}} - R_{20} \frac{p_{19}}{I_{19}} - \frac{q_{21}}{C_{21}} \quad (4-19)$$

$$\dot{q}_{21} = \frac{p_{19}}{I_{19}} \quad (4-20)$$

In the above state-space equations, q_3 , q_6 , q_{11} , q_{15} and q_{21} are the generalized displacement variables; p_{10} , p_{14} and p_{19} are the momentum variables. The state-space variables are defined as:

q_3 Relative motion across the mount

- q_6 Top chamber change in volume
- q_{11} Bottom chamber change in volume
- q_{15} Lower piston displacement
- q_{21} Induced electrical charge in the Voice coil-capacitor circuit
- p_{10} Time integral of the pressure drop in the inertia track (Pressure momentum)
- p_{14} Lower piston momentum
- p_{19} Flux linkage variable (λ) (Time integral of induced voltage in the voice coil-capacitor circuit)

The input force (effort in bond 1) is equal to:

$$F_{in} = \frac{q_3}{C_3} + R_2 V_{in} + A_p \frac{q_6}{C_6} \tag{4-21}$$

To simulate input force (F_{in}) versus frequency in the design of Figure 4-13, MATLAB Program and the above State-Space equations were used. The following parameters were used for the MATLAB simulation.

Table 4-2 Fluid mount and ETVA parameters used in MATLAB simulation

Symbol	Description	value	Unit
$K'_r = 1/C_3$	Real component of the vertical stiffness	1.95e5	N/m
K''_r	Imaginary component of the vertical stiffness	41.6	N/m
A_p	Effective Piston area	0.0038	m ²
A_m	Cross sectional area of the mass (cup) m	0.0038	m ²
$I_f = I_{10}$	Inertia track fluid inertia	1.5607e4	N-S ² /m ⁵
$R_o = R_8$	Inertia track flow resistance	1.0115e5	N-S/m ⁵
$K_{vt} = 1/C_6$	Top Chamber Volumetric or bulge Stiffness	2.177e10	N/m ⁵
$K_{vb} = 1/C_{11}$	Compressibility of the fluid	2.1e14	N/m ⁵
$K = 1/C_{15}$	Stiffness of support spring K	5256	N/m
$M = I_{14}$	piston mass+ coil mass	0.282	kg

$b = R_{16}$	Spring Damping	3	N.s/m
$R_{coil} = R_{20}$	Resistance of the copper wire wrapped around the coil	4.8	ohms
$I_{coil} = I_{19}$	Coil Inductance	3.4	mH
α	Voice coil force sensitivity	30.25	N/amp
$C_v = C_{21}$	Shunted Capacitance,	varied	μf

In the MATLAB simulations, the variable capacitance parameter, C_{21} , was varied from very low (open circuit) to $300 \mu f$. Figure 4-14 shows the dynamic stiffness of the new fluid mount design versus frequency as the element C_{21} is varied. The simulation shows that the notch frequency moves about 20 Hz (approximately 25 %) and generally it is enough for fine tuning of the notch frequency of a fluid mount.

Figure 4-14 Dynamic stiffness as variable capacitance parameter is varied from open circuit to $300 \mu f$

Figure 4-15 shows that as the coil resistance (R_{20}) decreases, the notch will be deeper for the same variation in the capacitance. In order to get a deep notch, the coil

resistance, R_{20} , should be kept as low as possible. Moreover, if a coil is made of low resistance wires (low R_{20}), the range of notch variation of the mount through changing the capacitive shunt circuit will be wider. As is shown in Figure 4-15, for the coil resistance of 0.5 Ohms, the notch can be varied from 80 Hz in open circuit to 40 Hz when 1100 μf capacitance is shunted (50% variation).

Overall, it was shown that how the ETVA was used in conjunction with a passive single pumper fluid mount and creates a variable notch frequency semi active fluid mount whose notch frequency is easily tunable and can be moved about 25% from its original notch frequency. Figure 4-13 to 4-15 give a guideline how the ETVA parameters should be selected to get the desired notch frequency range variation.

Figure 4-15 Dynamic stiffness at different coil resistance as C21 is 1000 μf

4.11 Real-time Notch Tuning Control Implementation

The previous sections introduced the capability of a capacitive shunt circuit in changing the bottom chamber volumetric stiffness of the fluid mount and in consequence creating a variable notch frequency fluid mount. It was shown that if the capacitive shunt circuit changes such a way that the notch frequency of the fluid mounts coincides with the input disturbance frequency; the minimum force would be transmitted from the engine to the structure.

The following sections are aimed to explain a practical control strategy and an appropriate electrical circuit design which can be used to change the shunt capacitance of the semi-active mount according to the input disturbance frequency variation.

4.11.1 Tuning Procedure

Generally, the tuning of the shunt capacitance of the proposed semi-active fluid mount, involves four main steps:

1. Select a proper mount parameters (such as pressure etc.) or structure acceleration or velocity which can be used to estimate the frequency of input disturbance.
2. Select a method to extract the disturbance frequency from the measured parameter.
3. Define the required shunt capacitance which is able to move the notch frequency of the fluid mount such that it coincides with the input disturbance frequency.

4. Define a control strategy to adjust the calculated required shunt capacitance in the designed electrical circuit.

The following sections explain these four steps in more details.

4.11.2 Disturbance Frequency Measurement

In order to estimate the frequency of disturbance input, two important questions should be answered.

- 1) What is the most suitable sensor to measure the input disturbance?
- 2) How the frequency content of the input disturbance can be extracted from the measured data?

Consider the input disturbance frequency resulting from engine vibration, is represented as:

$$X_{in} = X_{max} \sin(2\pi ft) \quad (4-22)$$

Where X_{max} is the amplitude and f is the frequency of input disturbance. There are three possibilities to measure this disturbance. It can be measured by a displacement sensor, a velocity sensor or an accelerometer. Since the ultimate goal of the proposed semi-active design is creating a self-powered self-contained semi-active fluid mount system, all the sensors and controller should be selected such that they require very low power. A review on available sensors reveals that, ultra low power accelerometers are commercially available. So, an accelerometer is selected for the current work which would help us, when the design is developed to a self-contained one. The input measurable data to the accelerometer can be represented as:

$$a_{in}(t) = -X_{max} (2\pi f)^2 \sin(2\pi ft) \quad (4-23)$$

while the output from the accelerometer is in the form of charge or voltage and it has a magnitude proportional to the magnitude of the input data and the frequency equals to the disturbance frequency (f). The accelerometer selected for this research [81] has the output in the form of voltage and it can be represented as:

$$v_{out}(t) = \alpha_{accelerometer} X_{max} (2\pi f)^2 \sin(2\pi f t) \quad (4-24)$$

While, $\alpha_{accelerometer}$ is a constant coefficient and it is provided by the accelerometer's manufacturer. The question remains is that how to extract the frequency content of the disturbance from the measured data (Equation (4-24)).

Consider the integral of the Equation (4-24) as:

$$Y_{out}(t) = \int v_{out}(t) dt = -\frac{\alpha_{accelerometer} X_{max} (2\pi f)^2}{2\pi f} \cos(2\pi f t) + C \quad (4-25)$$

Where, C is a constant, depending upon the initial acceleration. If Y_{out} is passed through a high-pass filter, this constant would be removed and it becomes:

$$Y_{outfiltered}(t) = -\frac{\alpha_{accelerometer} X_{max} (2\pi f)^2}{\omega} \cos(2\pi f t) \quad (4-26)$$

The RMS value of Equations (4-24) and (4-26) are $\frac{\alpha_{accelerometer} X_{max} (2\pi f)^2}{\sqrt{2}}$ and

$\frac{\alpha_{accelerometer} X_{max} 2\pi f}{\sqrt{2}}$ respectively. So the ratio of RMS value of the accelerometer's

output voltage to the RMS value of filtered integral of the accelerometer's output voltage multiplied by the gain of $1/2\pi$, gives the frequency of the input disturbance as:

$$f = \frac{1}{2\pi} \frac{RMS(v_{out}(t))}{RMS(Y_{outfiltered}(t))} = \frac{1}{2\pi} \frac{\frac{\alpha_{accelerometer} X_{max} (2\pi f)^2}{\sqrt{2}}}{\frac{\alpha_{accelerometer} X_{max} (2\pi f)}{\sqrt{2}}} \quad (4-27)$$

Assuming that the disturbance frequency is defined as above, now it is required to determine the proper value of the shunt capacitance which makes the notch frequency of the mount coincide with this measured input disturbance frequency.

4.11.3 Variable Capacitance Shunt Circuit

As was shown in the simulation results in section 4-10, for the known disturbance frequency within the operational frequency of the proposed semi-active mount (between short circuit frequency and open circuit frequency), there exists a corresponding shunt capacitance that makes the notch frequency of the mount coincide with the disturbance frequency. The procedure to define the corresponding shunt capacitance from the state-space equations of the mount can be summarized as below:

- Take Laplace transform of the state-space Equations (4-13) to (4-20) and Equation (4-21).
- Put all damping parameters (R_s) equals to zero. It should be noticed that, the damping parameters just affect the depth of the fluid mount's notch not its location. Therefore, they can be neglected while we are going to determine the notch frequency location.
- Put the Laplace transformed of Equation (4-21) (transmitted force) equal to zero to derive the notch frequency location as a function of fluid mount parameters. It should be noticed that by assuming all damping equals to zero, the transmitted force at the notch frequency would be zero, so it would be an acceptable assumption.
- Put f_{notch} in Equation (4-28), equals to the disturbance frequency (f) which is known.

- Define the required shunt capacitance which satisfies the equality of both sides of Equation (4-28). (See Appendix B)

4.11.3.1 Required Shunt Capacitance

Considering the above procedure, the corresponding shunt capacitance value (C_{21}) for the indicated disturbance frequency is derived as a function of the fluid mount parameters and input disturbance frequency as:

$$C_{21} = \frac{-(2\pi \times f_{notch})^5 \gamma_4 + (2\pi \times f_{notch})^3 \gamma_5 - (2\pi \times f_{notch}) \gamma_6}{-(2\pi \times f_{notch})^7 \gamma_1 + (2\pi \times f_{notch})^5 \gamma_2 - (2\pi \times f_{notch})^3 \gamma_3} \quad (4-28)$$

Where f_{notch} is equal to the measured disturbance frequency (f) and γ_1 to γ_6 are the functions of mount parameters ($C_3, C_6, I_{10}, \dots$, etc.) and are defined as:

$$\gamma_1 = (C_6 + A_p^2 C_3)(I_{10} I_{19} I_{14} C_{15} C_{11}) \quad (4-29)$$

$$\gamma_2 = (C_6 + A_p^2 C_3)(I_{10} C_{11} I_{19} + I_{10} I_{19} C_{15} A_{pb}^2 + I_{19} I_{14} C_{15} + I_{10} C_{11} GY_{17-18}^2 C_{15}) + (I_{19} I_{14} C_{15} C_{11}) \quad (4-30)$$

$$\gamma_3 = (C_6 + A_p^2 C_3)(GY_{17-18}^2 C_{15} + I_{19}) + (C_{11} GY_{17-18}^2 C_{15} + C_{11} I_{19} + I_{19} C_{15} A_{pb}^2) \quad (4-31)$$

$$\gamma_4 = (C_6 + A_p^2 C_3) I_{10} C_{11} I_{14} C_{15} \quad (4-32)$$

$$\gamma_5 = (C_6 + A_p^2 C_3)(I_{10} C_{11} + I_{10} C_{15} A_{pb}^2 + I_{14} C_{15}) + (C_{11} I_{14} C_{15}) \quad (4-33)$$

$$\gamma_6 = (C_6 + A_p^2 C_3) + (C_{11} + C_{15} A_{pb}^2) \quad (4-34)$$

Readers can refer to Appendix B for the expanded equations which were used to derive Equation (4-28).

4.11.3.2 Ladder Shunt Capacitance Circuit Design

The required shunt capacitance value which places the notch frequency of the semi-active fluid mount in any frequency between open circuit notch frequency condition

($f_{Open\ Circuit}$) and short circuit notch frequency condition ($f_{Short\ Circuit}$) was defined by Equation (4-28). Now, it is required to define an electrical circuit and a suitable control strategy to change the capacitance shunt condition to the required value as the disturbance frequency is changed.

Practically, there are three possibilities to create an adjustable capacitance shunt circuit:

- 1) Using the commercially available variable capacitance devices.
- 2) Using commercially available programmable capacitors.
- 3) Designing our own customized ladder circuit of discrete capacitors wired in parallel.

For the current application, the shunt capacitance changes in a large range (from $18\ \mu f$ to $300\ \mu f$). A review, on all the above possibilities shows that the commercially variable capacitors have a limited capacitance range variation. Programmable capacitors are also not suitable since they are complex and they have high internal resistance which highly affects the depth of dynamic stiffness at the notch frequency. Therefore, among the above three possibilities, a ladder circuit of discrete capacitors which are wired in parallel would be the best viable choice. Figure 4-16 shows the overall physical model of the ladder capacitive circuit which is shunted to the coil.

Figure 4-16 Physical model of the ladder capacitive circuit which is shunted to the coil

Figure 4-17 General configuration of the discretized frequency bandwidth between open circuit and short circuit condition

In order to define a suitable control strategy which can select the proper number of parallel capacitors whose sum are the required capacitance value (C_{2I}) to tune the notch frequency (f_{notch}) to the estimated disturbance frequency (f), the frequency tuning band of the shunted voice coil ($f_{Short\ Circuit}$ to $f_{Open\ Circuit}$) should be discretized into a finite number of intervals. In the following section, the overall configuration of the capacitive ladder circuit and its suitable control strategy are explained in more detail.

Assuming, the frequency interval between short circuit shunt condition ($f_{Short\ Circuit}$) and open circuit shunt condition ($f_{Open\ Circuit}$) is discretized into the finite number of equally-spaced intervals (e.g: 2 Hz) and the corresponding shunt capacitance for the frequencies at the border of each interval had been defined according to Equation (4-28). Therefore, the required shunt capacitance is calculated for the finite number of

frequencies, which are the border frequencies of each interval. Figure 4-17 shows the general configuration of the discretized frequency bandwidth between open circuit and short circuit condition.

Assuming the open circuit shunt condition which corresponds to a very small shunt capacitance; if capacitor C_1 in Figure 4-17 is selected small and all switches are kept open, the notch frequency of the semi-active mount will happen at 79.8 Hz ($f_{Open\ Circuit}$). Thereafter, consider the first switch (S_1) in Figure 4-17 is closed; so the overall capacitance of the shunt circuit changes to the summation of C_1 and C_2 and the notch frequency of the semi-active fluid mount happens at 78 Hz. (It should be noticed that capacitance value are already calculated properly according to Equation (4-28) for the corresponding notch frequencies). Similarly, closing both switches S_1 and S_2 changes the net shunt capacitance and put the notch frequency of the semi-active mount at 76 Hz. Finally, closing all of the switches changes the net shunt capacitance to a high value (summation of all capacitances) and it would be a good approximate of the short circuit condition and the notch frequency occurs at 62 Hz ($f_{short\ circuit}$).

Now it is required to define a suitable control strategy to open up and close switches in the ladder circuit according to the change in the input disturbance frequency. Christopher [82] suggested a control strategy for this ladder circuit and it is summarized in the following paragraphs.

The switches shown in Figure 4-17 can be selected such that they operate in response to a prescribed voltage from the controller [82]. However it should be noticed that, the switches should not add any additional resistance into the shunt circuit because it would highly affect the depth of dynamic stiffness at the notch frequency. Relays can

be viable choice for this matter. Commonly, there are two types of relays, electromechanical relays and solid-state relays (SSR). Electromechanical relays work upon the small current which flows through a coil to create a magnetic field and pulls one switch contact against another. However in SSRs the coil and moving switch contact are replaced with transistors and the value of current in emitter, collector and base of the transistor defines the close and open status of the switch. In order to use SSRs for the current switching application, a relay driving circuit is required. The relay driving circuit allows a small electrical current to base of the transistor to the control the status of the switch. Figure 4-18 shows the schematic of the relay driver circuit.

Figure 4-18 Circuit diagram for the relay driver circuit

Adequate control voltage, V_c , switches the transistor and lets the relay drive voltage, V_+ , to close the relay switch. It should be noticed that the resistor, R , is used to limit the current flowing to the transistor. Moreover, the controlling voltage V_c can be much smaller than V_+ . Therefore using SSRs and using voltage divider circuit would be a suitable method for switching capacitive ladder circuit into the various shunt capacitance condition. Now it is desirable to define a control strategy for applying the control voltage, V_c , to the relay driver circuit.

The voltage divider circuit can be used to control the relay driver circuits by using only a single voltage source (V_c). As it is clear from its name, the voltage divider circuit divides an input voltage by using two or more resistors in series. So, by putting the relay driver circuits (Figure 4-18) in between of the resistors of a voltage divider circuit (R_1 through R_9 , Figure 4-19), changing the single control voltage determines which relays turn to “on” condition and which ones should remain in “off” state. Figure 4-19 shows the schematic of the controlling circuit used to change the capacitive shunt value of the semi-active fluid mount.

Figure 4-19 Capacitive shunt controlling circuit

In Figure 4-19, $V_{control}$ represents the control voltage using to turn required switches “on” or “off”. Variable resistors, R_1 to R_9 are adjusted such that when the discretized control voltage, $V_{control}$ changes from 1 v to 9 v by an interval of 1 v, the corresponding switches Q_1 to Q_9 turns on respectively. For an example, for $V_c = 1$ v, Q_1 turns “on” while other Q s are still in the “off” state. Once the control voltage, V_c , increases to 2v both Q_1 and Q_2 turn “on” and the corresponding capacitors C_1 and C_2 come to the capacitive circuit. Finally, when the control voltage, $V_{control}$, is adjusted to 9v, all the switches are in “on” state and the capacitive shunt circuit is an equivalent of short circuit shunt condition. In order to have proper “on” and “off” states for each

transistor, the input control voltage, $V_{control}$, to the voltage divider circuit should be an integer value. Figure 4-20 illustrates the relation between control voltage value, $V_{control}$, capacitive shunt condition and the notch frequency location of the semi-active fluid mount.

Figure 4-20 Relation between control voltage value, $V_{control}$, capacitive shunt condition and the notch frequency location of the semi-active fluid mount

Table 4-3 represents the discretized tuning frequency which is equal to the disturbance frequency, the required shunt capacitances, and the corresponding control voltages needed to discretely tune the notch frequency of the semi-active fluid mount from open to short circuit frequency ($f_{Open\ Circuit}$ to $f_{Short\ Circuit}$).

Table 4-3 Discretized tuning frequency, shunt capacitance and control voltage

Tuning frequency [Hz]	Total shunt capacitance, C_{tot} [μf]	Ladder leg capacitance, C_i [μf]	Control voltage, V_c [Volts]
$f_{Open\ Circuit}=79.8$	1e-3	1e-3	0
$f_8=78$	18	18	1
$f_7=76$	47	29	2
$f_6=74$	77	30	3
$f_5=72$	109	32	4
$f_4=70$	142	33	5
$f_3=68$	177	35	6
$f_2=66$	215	38	7
$f_1=64$	255	40	8
$f_{Short\ Circuit}=62$	298	43	9

It should be noticed that each tuning frequency has its own corresponding shunt capacitance value and the control voltage. Therefore, all the shunt capacitors in the capacitive ladder circuit and the related control voltage should be chosen in such a way that the notch frequency will place in the required place once the disturbance frequency is changed. The control strategy which adjusts the control voltage, V_C , according to the disturbance frequency can be represented as (See Figure 4-21):

$$V_C = \frac{V_{C\ max}}{f_{Open\ Circuit} - f_{Short\ Circuit}} [f_{Open\ Circuit} - f_{disturbance}] + 0.5 \tag{4-35}$$

Where $f_{Open\ Circuit}$ and $f_{Short\ Circuit}$ are notch frequencies of the semi-active fluid mount in open circuit and short circuit condition respectively and they can be defined experimentally for the proposed mount. $f_{disturbance}$ is the frequency of the disturbance input which was already explained how to measure and analyze it from the accelerometer data (See section 4-11-2). The fraction value of 0.5 is added to the equation just to make sure that the calculated control voltage, V_C , is an integer number.

V_{Cmax} is the maximum integer value of the control voltage, and for the current work, it is assigned equal to 9 v, because the maximum generated voltage by the energy harvester device , which will later added to this design, is 9 v.

Figure 4-21 Calculate control voltage, V_c , according to input disturbance

4.11.4 Self-powered Self-Contained Semi-Active Fluid Mount

The previous sections introduced the design and control strategy of the semi-active fluid mount. This section aims to develop a semi-active fluid mount which is self-powered and self-contained. As it was reviewed in Chapter 2, all semi-active mount designs need a controller and some sensors in order to adjust the fluid mount's notch frequency to the desired frequency location. These requirements translate to a need for an external power which should be supplied to the mount to run the controller and all the measurement sensors. Power supply requirement adds a disadvantage to the semi-active fluid mount designs. Since supplying external power means the mount can not work independently and the semi-active design would be more costly and complicated than the passive ones; therefore, it is highly desirable to design a semi-active fluid mount which uses an energy harvester device to convert the vibration energy of the engine into the appropriate form of electrical power which can be supplied to the mount and run the controller and all the sensors.

Shashank priya and Daniel Inman [83] gave a comprehensive literature review on the different types of energy harvesting devices available on the market. All important energy harvesting devices such as piezoelectric, thermoelectric, MEMS-based devices, and electromagnetic energy harvesters were reviewed by [83] and for each type, the maximum available harvesting power was defined.

In order to select an appropriate energy harvester, for our semi-active fluid mount design, the following criteria were considered.

- 1) All the components of the semi-active mount design, which required external power, were defined (ex. : sensors, controller and so on)

- 2) Maximum required electrical power, required by each component was calculated.
- 3) Frequency operation range of the semi-active fluid mount design was defined.
- 4) Design envelope, for the energy harvester and the sensors & sensor wiring, was estimated.

Table 4-4 summarizes the required power supply for the components of current semi-active fluid mount design;

Table 4-4 Required power supply for the components of semi-active fluid mount design

Component Description	Supply voltage [Volts]	Supply current [Amps]	Supply power [Watts]
Ultra low power MEMS accelerometer [81]	2.5	250E-6	625E-6
Ultra low power Digital Signal Processor (DSP) [84]	0.25	1E-3	250E-6
Solid-State Relay (SSR)	3.6	Depends on coil resistance	Depends on coil resistance
Voltage control divider circuit	9	Depends on variable resistors	Depends on variable resistors
Miniature Step-Down DC-DC converter	0.25, 2.5, 3.6	-	-

Frequency operating range of the selected energy harvester device should cover the frequency range of short circuit frequency ($f_{Short\ Circuit}$) through open circuit frequency ($f_{Open\ Circuit}$) of the proposed semi-active fluid mount (62 Hz to 80 Hz); and finally, the dimension of the energy harvester should be such that it can be installed in a 6cm by 6cm by 6cm design envelope.

The search for the commercially available energy harvester devices reveals that there exists an energy harvester device which satisfies all the power requirements, operating

frequency range, and size dimensions mentioned in the previous page [85]. Figure 4-23 shows the characteristics of this energy harvester device as the power output versus vibration frequency curves for different vibration acceleration inputs.

Figure 4-22 Power output versus vibration frequency

For automotive applications, the amplitude of input disturbance is anywhere in the range of ± 0.1 mm to ± 1 mm. Here it was assumed ± 0.2 mm. It means that in the frequency operating range from 62 Hz through 80 Hz, the maximum vibration acceleration is from 3.09 gRMS to 5.15 gRMS an they are far below maximum total vibration that the device can sustain (10 gRMS). Therefore, the selected device can be used safely to provide an output voltage up to 10v in the operating frequency range, to supply the required power for the capacitive shunt controlling circuit, the ultra low power accelerometer sensor, and ultra low power DSP.

Figure 4-24 illustrates the physical model of the self-contained semi active fluid mount design where an energy harvester device is placed between the lower chamber and the upper chamber of a conventional single pumper fluid mount. The wiring from the

accelerometer and the energy harvester device is done such that the overall mount design envelope becomes very compact and completely independent from any external source. Figure 4-25 shows the overall configuration of the electrical circuit and the connections that will be necessary for the energy harvester device, accelerometer, DSP, and the capacitive shunt controlling circuit.

Figure 4-23 Schematic of the new semi active fluid mount design

Figure 4-24 Electrical circuit configuration of the self-contained semi-active fluid mount

4.12 Conclusion

An electromechanical tuned vibration absorber (ETVA) was introduced. It includes a voice coil, a single degree of freedom mass spring system and an electronic RLC circuit. The electronic circuit (RLC circuit) was added to the voice coil in series and it was used to tune the natural frequency of the ETVA. The mathematical model of the ETVA was defined and parametric studies were done to evaluate the effect of variable resistance, variable inductance and variable capacitance in tuning the natural frequency of the ETVA. Parametric studies showed that if the technology of superconductive wires at room temperature is one day improved; using a voice coil made of a very low resistance superconductive wires and a variable resistance, would create the best adaptive ETVA which is tunable in a large frequency range (more than 900%). However at this point in time among existing commercially available electrical components, it was shown that the variable capacitance has more effect on changing natural frequency of the TVA than changing resistance or inductance. Natural frequency of the ETVA can be changed by 66% ($\frac{\omega_{\max} - \omega_{\min}}{\omega_{\min}} = 0.66$) when the shunted capacitance condition is varied from open circuit to short circuit.

The viability of the proposed ETVA design was verified by developing an experimental test rig. The experimental data and theoretical simulation data were compared and it was shown that they are quite similar. Experimental data reveals that a variable capacitance is able to change the natural frequency of the ETVA in a wide frequency range. Thereafter, the application of the ETVA in creating a variable notch frequency fluid mount design was studied. Simulation results showed that indeed it is possible to vary notch frequency location by approximately 25% with the change of

shunted capacitance of the ETVA. As the capacitance is changed, the effective volumetric stiffness of the bottom chamber changes; and so does the notch frequency. Therefore in situations which notch frequency location changes due to tolerances on all the fluid mount dimensions, material property variations, and variation in elastomer molding processes, the notch frequency can be easily tuned to any desired location and it reduces remanufacturing cost and the fluid mount notch frequency tuning cycle time. A practical control scheme and an appropriate electrical circuit design were introduced in order to change the shunt capacitance of the semi-active mount according to the input disturbance frequency variation.

Finally, the application of an energy harvesting device in creating a self-contained semi-active fluid mount was evaluated and important criteria in selecting such a device for automotive applications were listed. The Chapter 4 was concluded by the physical model of proposed self-contained semi-active fluid mount design.

Chapter 5 Design and Control of an Active Fluid Mount using an Electromagnetic Actuator (Design 3)

5.1 Introduction

Engines are one of the most significant sources of vibration and noise in aerospace and automotive applications; therefore the isolation of engine vibration in a wide range of frequencies is highly desirable. As it was explained in the previous chapters, in the last two decades, some adaptive (semi-active) hydraulic mounts have been proposed where the working fluid is a controllable fluid such as a MR (Magento-Rheological) or an ER (Electro-Rheological) fluid [34, 36, 39, 86]. In addition, previous chapters of the current thesis also dedicated to two semi-active fluid mount designs and it was shown that the semi-active fluid mounts are able to reduce the vibration and noise to some level but they are not capable of eliminating the structure borne noise and vibration completely in a wide frequency range. To eliminate vibration in a wide frequency range, active fluid mounts have been designed and developed [40-46, 48-50, 52] .

Generally, an active fluid mount is equipped with an actuator and a controller. The controller controls the dynamic force of the actuator in such a way that it would soften the mount and allows the engine to move freely; therefore, reducing transmitted force and structure borne noise and vibration.

Previous studies have developed various types of active fluid mount systems by using different actuators such as pneumatic actuators [42], electromagnetic actuators [40-41, 43-44, 46, 52], servo-hydraulic actuators [45] and piezoelectric actuators [48-50].

Moreover, the performance of various control strategies such as feed-forward and feedback have been investigated on active mounting systems [40-41, 52, 87]. In all the developed active engine mounting systems, the common goal is to design a mount which is very stiff at low frequencies and is tuned to be very soft at high frequencies. Moreover, the mounting systems have been designed such that they are able to perform both actively and passively. However current thesis aims to present an active fluid mount by using an electromagnetic actuator and capacitive circuit such that it is able to act as a passive mount, semi-active mount and an active mount. Moreover, the design has the capability to be converted to a damper as and when needed. The multi-functional capability of the proposed design (passive, semi-active, active and damper) distinguishes the current design from the previous active engine mounting designs, and this multi-functional capability will be explained in more details in the coming sections.

In this chapter, dynamic analysis of the proposed active fluid mount design is conducted by using the bond graph modeling technique [56]. Thereafter, simulation results are shown for automotive applications and the performance of the proposed active mount is evaluated as an isolator and as a damper. Finally, an adaptive controller is proposed based on Filtered-X LMS algorithm and its performance is investigated.

5.2 Active Fluid Mount Design

Generally, engine mounts need to satisfy three design requirements: (1) firmly support engine weight, (2) isolate structure from the engine's noise and vibration [40], and (3) control engine motion when large shocks or engine resonance are present. In addition

to these three criteria which are common for designing all types of engine mounts (passive, semi-active and active), two more design requirements need to be satisfied in the active fluid mount designs. First, they should be designed such that if any abnormal situation occurs and the actuator, the controller or the sensors do not work properly, the active mount should still safely operate as a passive mount. Second, the power consumption, size and weight of the required actuator and its controller should be as low as possible [40].

According to the above design requirements, a new active mount is proposed. Figure 5-1 shows the physical model of the new active fluid mount design where the lower chamber of a conventional single pumper fluid mount [14] is replaced by a piston, a fabric diaphragm, a spring, an electromagnetic actuator, and a capacitive circuit. Figure 5-1 shows, in the bottom fluid chamber, a piston is supported by a spring which supports the static pressure of the fluid. A fabric diaphragm (with high volumetric stiffness) is placed on the top of the piston, sealing the bottom fluid chamber. An electromagnetic actuator (voice coil) is connected to the piston and is able to change the lower chamber dynamic pressure by moving the piston. The input current to the electromagnetic actuator is controlled by a controller such that the actuator force eliminates the transmitted force from the engine to the structure within a wide range of frequencies.

It should be noted that in the proposed design, an electromagnetic actuator is selected rather than other types of commercially available actuators such as hydraulic, electrodynamic and piezo-stacked; because it requires less space in comparison to a hydraulic or an electrodynamic actuator (See Appendix C) and it makes the overall

fluid mount envelope more compact; also a typical electromagnetic actuator has a larger stroke and a force range than a piezo-stacked actuator and can be used safely within the required frequency range.

Moreover, as it was mentioned in the introduction, the main goal of the proposing fluid mount design is its multi-functional capability (passive isolator, semi-active isolator, active isolator and damper) and an electromagnetic actuator is the best option; because, when the input current to the voice coil is in the “OFF” state, the mount acts as a passive fluid mount. In the case that a variable capacitance circuit is connected to the voice coil, it works as a semi-active mount (refer to Chapter 4) and finally if the input current to the voice coil is in the “ON” state and it is controlled by a controller, an active fluid mount is created; and in the conditions which the cruise speed coincides with one of the engine’s natural frequencies or a large shock is present, a high value capacitance can be shunted to the coil such that the dynamic stiffness of the mount increases and the mount acts as a damper.

Figure 5-1 Schematic of the proposed active fluid mount

5.3 Physical Model of the Active Fluid Mount Design

Figure 5-2 is the physical model of Figure 5-1 and in this model the vertical stiffness and damping of the rubber are shown as K_r and B_r ; respectively and the bulge stiffnesses of the upper chamber and the lower chamber are represented as C_{v1} and C_{v2} . K_{spring} and R_{spring} show the stiffness and damping of the supporting spring and C represents the shunted capacitance, used for converting the active mount to a damper.

In order to derive the constitutive equations of the mount and find out the required actuator current, the bond graph model of Figure 5-2 is developed and is shown in Figure 5-3.

Figure 5-2 Mathematical model of the proposed active fluid mount

Figure 5-3 Bond graph model of figure 5-2

From the bond graph model of Figure 5-3, the following state-space equations can be derived:

$$\dot{q}_2 = V_{in} \quad (5-1)$$

$$\dot{q}_6 = A_{pt} V_{in} - \frac{P_8}{I_8} \quad (5-2)$$

$$\dot{p}_8 = \frac{q_6}{C_6} - R_9 \frac{p_8}{I_8} - \frac{q_{11}}{C_{11}} \quad (5-3)$$

$$\dot{q}_{11} = \frac{p_8}{I_8} - A_{pb} \frac{p_{14}}{I_{14}} \quad (5-4)$$

$$\dot{p}_{14} = A_{pb} \frac{q_{11}}{C_{11}} - \frac{q_{15}}{C_{15}} - R_{16} \frac{p_{14}}{I_{14}} + \alpha_{Lower} \frac{p_{19}}{I_{19}} \quad (5-5)$$

$$\dot{q}_{15} = \frac{p_{14}}{I_{14}} \quad (5-6)$$

$$\dot{p}_{19} = \frac{q_{22}}{C_{22}} - \frac{p_{19}}{I_{19}} R_{20} - \alpha_{Lower} \frac{p_{14}}{I_{14}} \quad (5-7)$$

$$\dot{q}_{22} = S f_{23} - \frac{p_{19}}{I_{19}} \quad (5-8)$$

In the above state-space equations, q_2 , q_6 , q_{11} , q_{15} and q_{22} are the generalized displacement variables; p_8 , p_{14} and p_{19} are the momentum variables. The state-space variables are defined as:

- q_2 Relative motion across the mount
- q_6 Top chamber change in volume
- q_{11} Bottom chamber change in volume
- q_{15} Lower piston displacement
- q_{22} Electrical charge in shunted capacitor
- p_8 Time integral of the pressure drop in the inertia track (Pressure momentum)
- p_{14} Lower piston momentum
- p_{19} Flux linkage variable across the voice coil

The transmitted force to the structure (effort at bond 1) is equal to:

$$F_{in} = \frac{q_2}{C_2} + R_3 V_{in} + A_p \frac{q_6}{C_6} \quad (5-9)$$

By taking Laplace transform of the Equations (5-1) to (5-9), the transmitted force (F_{in}) in terms of the engine input displacement (X_{in}) and the input current to the actuator ($I_{actuator}$) can be expressed as:

$$F_{in}(s) = \gamma_1(s)X_{in}(s) + \gamma_2(s)I_{actuator}(s) \quad (5-10)$$

$\gamma_1(s)$ and $\gamma_2(s)$ are the functions of the fluid mount parameters and are defined as:

$$\gamma_1(s) = \left(\frac{1}{C_2} + sR_3 + \frac{A_{pt}^2 s}{\lambda_4(s)} \right) \quad (5-11)$$

$$\gamma_2(s) = \left(A_{pt} \frac{\lambda_2(s)}{\lambda_1(s)\lambda_3(s)\lambda_4(s)} \right) \quad (5-12)$$

And $\lambda_1(s)$ to $\lambda_4(s)$ are defined as:

$$\lambda_1(s) = \left(sC_{11} + \frac{A_{pb}^2}{\left(sI_{14} + \frac{1}{sC_{15}} + R_{16} + \frac{\alpha_{Lower}^2}{\left(sI_{19} + \frac{1}{sC_{22}} + R_{20} \right)} \right)} \right) \quad (5-13)$$

$$\lambda_2(s) = \frac{A_{pb}\alpha_{Lower}}{sC_{22} \left(sI_{19} + \frac{1}{sC_{22}} + R_{20} \right) \left(sI_{14} + \frac{1}{sC_{15}} + R_{16} + \frac{\alpha_{Lower}^2}{\left(sI_{19} + \frac{1}{sC_{22}} + R_{20} \right)} \right)} \quad (5-14)$$

$$\lambda_3(s) = \left(sI_8 + R_9 + \frac{1}{\lambda_1(s)} \right) \quad (5-15)$$

$$\lambda_4(s) = \left(sC_6 + \frac{1}{\lambda_3(s)} \right) \quad (5-16)$$

The required actuator current ($I_{actuator}$) which eliminates the effect of input engine displacement (X_{in}) and makes the transmitted force equal to zero can be derived from

Equation (5-10). Setting the transmitted force (F_{in}) (See Equation (5-10)) equal to zero gives:

$$I_{actuator}(s) = \frac{-\gamma_1(s)}{\gamma_2(s)} X_{in}(s) \quad (5-17)$$

Since both $\gamma_1(s)$ and $\gamma_2(s)$ are functions of the fluid mount parameters, all these parameters need to be defined to get a first estimate of the required actuator current and force and the most viable electromagnetic actuator. In addition to the required actuator current, the lower piston displacement needs to be defined; because it represents the required actuator stroke and it would be an important criterion in actuator selection.

According to the state-space equations (5-1 to 5-8), and Equations (5-9 to 5-16), the required lower piston displacement (See q_{15} in the bond graph of Figure 5-3) can be defined as a function of input engine input displacement ($X_{in}(s)$) and the mount parameters as:

$$q_{15}(s) = \frac{p_{14}(s)}{sI_{14}} = \left(\beta_3(s)\beta_1(s) + \frac{-\gamma_1(s)(\beta_3(s)\beta_2(s) + \beta_4(s))}{\gamma_2(s)} \right) X_{in}(s) \quad (5-18)$$

Where $\gamma_1(s)$ and $\gamma_2(s)$ are defined according to the Equations (5-11) and (5-12) respectively and $\beta_1(s)$ to $\beta_4(s)$ are defined as below,

$$\beta_1(s) = \left(\frac{A_{pi}s}{\lambda_1(s)\lambda_4(s)\lambda_3(s)} \right) \quad (5-19)$$

$$\beta_2(s) = \left(\frac{\lambda_2(s)}{\lambda_1^2(s)\lambda_3(s)} - \frac{\lambda_2(s)Sf_{23}}{\lambda_1^2(s)\lambda_3^2(s)\lambda_4(s)} - \frac{\lambda_2(s)}{\lambda_1(s)} \right) \quad (5-20)$$

$$\beta_3(s) = \frac{A_{pb}}{\left(s^2 I_{14} + \frac{1}{C_{15}} + sR_{16} + \frac{s\alpha_{Lower}^2}{\left(sI_{19} + \frac{1}{sC_{22}} + R_{20} \right)} \right)} \quad (5-21)$$

$$\beta_4(s) = \frac{\alpha_{Lower}}{sC_{22} \left(sI_{19} + \frac{1}{sC_{22}} + R_{20} \right) \left(s^2 I_{14} + \frac{1}{C_{15}} + sR_{16} + \frac{s\alpha_{Lower}^2}{\left(sI_{19} + \frac{1}{sC_{22}} + R_{20} \right)} \right)} \quad (5-22)$$

Amplitude of the engine input displacement, and the operating frequency range of the engine are two other important parameters which play important role in selecting the suitable actuator. In summary, the following steps are usually followed in designing an active fluid mount and selecting a proper actuator:

- a) Define input engine displacement.
- b) Define operating frequency range.
- c) Define the maximum available space for the actuator which should be expressed in terms of maximum dimension and weight.
- d) Define the required force and stroke of the actuator.

In this thesis, the above steps were followed and the performance of the proposed active fluid mount for automotive applications is considered and shown in the following sections.

5.4 Simulation Results of the Active Fluid Mount for Automotive Applications

In automotive applications, for an internal combustion engine, there exist two basic dynamic disturbances: primary disturbances (the inertia forces and torques caused by the rotating and reciprocating parts such as piston, connecting rod, and crank) and secondary disturbances (the firing pulses due to the explosion of the fuel in the cylinders) [3]. The engine speed of 4-cylinder car passengers in the cruise condition (highway driving condition) varies from one engine manufacturer to the other, but it is usually in the range of 2000 rpm to 2500 rpm. For example, for a typical 4-cylinder car with the engine speed of 2400 rpm (40 Hz) in the cruise condition (highway driving condition), the rotating and reciprocating parts of the engine such as the piston, the connecting rod, and the crank rotate at 40 Hz but the cylinders fire at twice the speed of the engine, namely 80 Hz. Engine manufacturers make great attempts to balance the rotating and reciprocating parts of an engine in the manufacturing pass and minimize the vibration induced due to the primary disturbances. However, the vibration created by the firing of the cylinders can affect the vehicle compartments and it is highly desirable to design the fluid mount such that it eliminates the effect of secondary disturbances (firing pulses).

In the proposed active fluid mount design of this chapter, only secondary disturbances are considered and to verify the performance of the proposed active fluid mount, the simulation results of a typical 4-cylinder car with the engine speed of 2400 rpm (80 Hz) in the cruise condition (highway driving condition) are given.

The mount parameters are selected such that the notch frequency of the proposed mount in the passive mode occurs at 80 Hz and in the active mode; the mount is able to eliminate the transmitted force from engine to the structure in the operating frequency interval of 15 Hz to 125 Hz (idle speed till 3750 rpm). Table 5-1 shows the selected parameters. It should be noted that the rubber and the voice coil specifications are selected from the commercially available products.

Table 5-1 Active fluid mount parameters

Symbol	Description	Value	Unit
$K_r = 1/C_2$	Rubber stiffness	1.96e5	N/m
$B_r = R_3$	Rubber damping	41.6	N.s/m
$A_{pt} = TF_{4-5}$	Effective piston area	3.8e-3	m ²
$K_{vt} = 1/C_6$	Top chamber volumetric (bulge stiffness)	2.177e10	N/m ⁵
$I_f = I_8$	Inertia track fluid inertia ($4\rho * L_v / (\pi * d_i^2)$)	1.5607e4	N-S ² /m ⁵
$R_o = R_9$	Inertia track flow resistance	1.011e5	N-S/m ⁵
$K_{vb} = 1/C_{11}$	Compressibility of the fluid	2.1e14	N/m ⁵
$A_{pb} = 1/TF_{12-13}$	Cross sectional area of the lower piston	3.8e-3	m ²
$M = I_{14}$	piston mass+ coil mass	0.282	Kg
$K = 1/C_{15}$	Stiffness of support spring K	5256	N/m
$b = R_{16}$	Spring damping	5	N.s/m
$\alpha_{lower} = GY_{17-18}$	Lower voice coil force sensitivity	30.25	N/amp
$I_{coil} = I_{19}$	Lower coil inductance	3.4	mH
$R_{coil} = R_{20}$	Resistance of the copper wire wrapped around the lower coil	4.8	ohms
$C_{shunt} = C_{22}$	Shunt capacitance in active and passive isolator modes	eliminated	-
	Shunt capacitance in passive damper mode	1000	μf

5.4.1 Dynamic Stiffness versus Frequency of the Active Fluid Mount for Automotive Applications

By inserting the above mount parameters in the state-space Equations (5-1 to 5-8), the fluid mount behavior can be described and the required actuator current and stroke can be evaluated. Figures 5-4 and 5-5 show the amplitude and phase of the dynamic stiffness of the active fluid mount in the passive mode when the lower voice coil is either short circuited or open circuited.

As is shown, even in the passive mode, when the lower voice coil is in open circuit condition (coil wires are not connected) the mount is still able to reduce the transmitted force in the notch frequency (80 Hz); It means the mount parameters are selected such that the notch frequency in the passive mode, is as close as possible to the secondary disturbance frequency at cruise condition and in the case of actuator or sensor failure, the mount is still able to isolate structure from engine vibration. It should be noted that in passive mode, if the coil's wires are short circuited, the voice coil adds high damping to the mounting system and the mount would act as a damper. This characteristic makes the mount suitable for situations a sudden vehicle stop or engine resonance occurring.

Figure 5-4 Magnitude of the dynamic stiffness versus frequency in passive mode

Figure 5-5 Phase of the dynamic stiffness versus frequency in passive mode

Figure 5-6 shows the dynamic stiffness of the mount versus frequency for the both passive and active modes. As is shown, the dynamic stiffness significantly decreases in active mode; thus the transmitted force is eliminated completely in a wide frequency band (15 Hz to 125 Hz).

Figure 5-6 Dynamic stiffness versus frequency for both passive and active condition

It should be noticed that in the current design, the notch frequency of the mount in the passive mode is intentionally designed to occur in the middle of the operating frequency range of the active mode (15 to 125 Hz). However, the previous research studies which were done on active fluid mounts [40-41], the notch frequency was placed, out of the controlled frequency range and this creates two major disadvantages. First, under abnormal situations, or actuator or sensor failure, the mount is not able to isolate the structure from the engine vibration. Moreover, by putting the notch frequency of the mount in the passive mode out of the range of the controlled frequency, the required actuator force would be higher in comparison to the current design. These two issues were carefully considered in our design.

5.4.2 Required Actuator Current versus Frequency of the Active
Fluid Mount for Automotive Applications

Using Equation (5-17), the required actuator current which should be applied to the coil is defined and shown in Figure 5-7. For the engine input displacement (X_{in}) of ± 0.2 mm, the required actuator current in the operating frequency range of 15 Hz to 125 Hz is less than 2 Amperes and it is below the saturation current of the coil wires (2.87 Amps); thus the voice coil will not saturate in the operating frequency range and can be used safely.

Figure 5-7 Magnitude and phase of required actuator current

5.4.3 Required Actuator Stroke versus Frequency of the Active Fluid Mount for Automotive Applications

Figures 5-6 and 5-7 indicated that from the force and current point of views, the selected voice coil can be used safely. The lower piston displacement is the only

remaining parameter which should be considered to make sure that the selected voice coil can handle the displacements in both the passive and the active modes in the range of 15 to 125 Hz. The lower piston displacement in the active mode is about 1 mm which is less than the maximum voice coil stroke. Moreover, Figure 5-8 shows that the lower piston displacement is far below the maximum stroke of the selected voice coil (5.7 mm) in the passive mode too. So even in the case of controller failure, the voice coil would not be damaged and the selected voice coil can be safely used.

Figure 5-8 Lower piston displacement versus frequency in passive condition

It should be noted that usually, force sensitivity (α) defines the capability of an electromagnetic actuator (voice coil). Higher force sensitivity would give higher actuator force; however, large force sensitivity (α) would need bigger coil and heavier permanent magnet. So, the selection of the actuator would be a compromise between the actuator force and stroke versus weight.

5.4.4 Active Fluid Mount in Damper Condition

As it was explained in the introduction, one aim of the proposed active fluid mount design is to act as a damper when needed. To turn the active mount into a damper, two switches (S_1 and S_2) plus a capacitor as shown in Figure 5-9 will be needed. When S_2 is “ON” and S_1 is “OFF”, the mount acts like an active mount and when S_2 is “OFF” and S_1 is “ON”, the mount acts like a damper.

Figure 5-9 Schematic of the voice coil shunted to a capacitance and the controlled current source

According to Equation (5-10), when the input current is in the “OFF” state (switch S_2 is open), the transmitted force is reduced to;

$$F_{in}(s) = \gamma_1(s)X_{in}(s) = \left[\frac{1}{C_2} + sR_3 + \frac{A_{pt}^2 s}{sC_6 + \left(\frac{1}{sI_8 + R_9 + \frac{1}{\lambda_1(s)}} \right)} \right] X_{in}(s) \quad (5-23)$$

$\lambda_1(s)$, defined by Equation (5-13), is a function of capacitance, C_{22} . When C_{22} is increased, $\lambda_1(s)$ decreases, therefore the mount converts to a damper. Figure 5-10 shows the dynamic stiffness versus frequency of the mount in the damper mode versus

passive and active isolator modes. A control strategy should be implemented to control the input current as is defined in Figure 5-8; also a control strategy to close and open the switches S_1 and S_2 .

Figure 5-10 Dynamic stiffness versus frequency for both isolator and damper modes

5.5 Design of an Adaptive Controller

The control strategies for active vibration isolation can be classified into two main categories: feedback and feed-forward. Among these two, feed-forward strategy is more suitable for active vibration and noise cancellation than feedback. However, feed-forward controllers require the precise modeling of the dynamic characteristics of the system which is to be controlled. For the current application (engine vibration isolation), even in the case that the mount and its actuator are modeled precisely; the material properties of the mount's components may change with time due to the

ageing, temperature and other environmental effects. Therefore, the feed-forward controller needs to be modified with an adaptive control strategy to resolve system variation problems. Combination of an adaptive and feed-forward control strategy makes a feed-forward controller enable to adjust itself according to the variations. Among adaptive control strategies, Filtered-X LMS algorithm is widely used for noise and vibration cancellation due to its simplicity and good performance [40, 88-89]. Therefore, it is selected as the adaptive algorithm for the current research.

5.5.1 The Filtered-X Algorithm

Widrow and Hoff originally developed the LMS algorithm. In the current study, narrow-band Filtered-X LMS is used to cancel periodic engine vibration [89]. Figure 5-12 shows the Filtered-X LMS arrangement for the current application. As is shown in Figure 5-12, the engine vibration $X_{in}(t)$ is transmitted to the structure through the primary path dynamics $\gamma_1(s)$ representing the passive component of the engine mount. This results in the primary vibration $d(s)$ at the structure, which is to be cancelled. The cancelling signal $y(s)$ is modified by the secondary path dynamics $\gamma_2(s)$. For this application, the secondary path represents the dynamics of the actuator and the vibration path to the feedback sensor. The error signal $e(t)$ is the resultant transmitted force to the structure. (See Equation (5-10))

The Filtered-X LMS algorithm updates the input current $I_{in}(t)$, which has effect on active mount system through the secondary path dynamics ($\gamma_2(s)$). Current is updated such that the error signal $e(t)$ (transmitted force) goes to zero. The controller update

equation is derived as:

$$W_{k+1} = W_k + 2\mu F_T(n)Z \quad (5-24)$$

Where W_{k+1} is the adaptive controller weight factor, μ is the step size, $F_T(n)$ is the sampled of the overall transmitted force ($F_T(t) = e(t)$) and Z is the sampled of the reference signal $R(n)$ which is filtered through the transfer function $\hat{\gamma}_2(z)$. $\hat{\gamma}_2(z)$ is the estimated version of the discrete time expression of $\gamma_2(s)$.

Figure 5-11 Single frequency active vibration cancellation system using the Filtered-X LMS algorithm [40]

In the Filtered-X LMS algorithm of Figure 5-12, the secondary path estimation ($\hat{\gamma}_2(z)$) is derived online and the transfer function $\hat{\gamma}_2(z)$ is updated in real time with another LMS algorithm which is added to the current Filtered-X LMS configuration but it is not shown in the above figure. Readers can refer to the references [90] for more details. Kue [89] showed the block diagram of active noise cancellation system with on-line secondary-path modeling using adaptive random noise and it can be similarly used for the current application. However, this on-line secondary path estimation is unable to cancel any interference that is uncorrelated with the input engine vibration

$X_{in}(t)$. For instance vibration input from structure to the mount or other uncanceled engine vibrations will be included in the measured error signal. In order to resolve this problem, the LMS algorithm which is used for the estimation of $\gamma_2(z)$ should be replaced by a fast-block LMS (FBLMS) which operates in frequency domain. The general structure of FBLMS is explained by J.J. Shynk [90]. The current work uses the LMS algorithm for the secondary path estimation of the Filtered-X LMS algorithm. The primary path transfer function, $\gamma_1(s)$ and the secondary path transfer function, $\gamma_2(s)$ which are used in the Filtered-X LMS algorithm are defined by Equations (5-11) and (5-12); respectively.

The reference signal $R(t)$ (equals to $X_{in} \sin(\omega t)$) is generated by a tachometer and it has a frequency equal to the engine disturbance frequency (ω), which is required to be cancelled. The error signal $e(t)$ is measured by a load cell which is placed in between the mount and the structure and shows the overall transmitted force from the engine to the structure.

Using configuration of Figure 5-12 and doing some simulations in SIMULINK reveal that, a Filtered-X LMS algorithm with 120 taps, sampling frequency of 5 KHz and step size (μ) of 0.0002 shows the appropriate performance. Figure 5-8 shows the performance of the active fluid mount corresponding to the 30 Hz excitation. As is shown, in the active mode, the transmitted force is almost completely canceled out.

Figure 5-12 Performance of active fluid mount in 30 Hz

As it was explained in section 5.3.5, in order to convert the mount from an isolator to a damper, switch (S_2) should be opened and switch (S_1) should be closed. This task can be done by using SSRs and using voltage divider circuit which can control the close and open state of each switch according the voltage signal, coming from an accelerometer. The accelerometer can be put on the lower part of the mount. Once it gives high voltage which means a shock situation happens, switch (S_1) will be closed and switch (S_2) is opened; thus dynamic stiffness of the mount increases and the engine stays in its place rigidly. On the normal conditions, which accelerometer gives low voltage signal, (S_2) is closed and (S_1) is open. So the mount performs in active mode. Readers can refer to Chapter 4, to see how SSRs and voltage divider circuit can be used to control switches according to the input accelerometer voltage signal.

Table 5-2 Required hardware for the adaptive controller

Hardware		Application
Sensor	Tachometer	To create a periodic square wave with a period equals to the engine vibration frequency ($R(t)$)
	Load cell	To measure the overall transmitted force from mount to the structure ($F_T(t)=e(t)$)
Actuator		An electromagnetic actuator which has a suitable stroke

	and proper maximum continuous force in the operating frequency range
Current generator	To supply required current of the electromagnetic actuator
Digital Signal Processor (DSP)	To implement Filtered-X LMS algorithm
A/D convertor	To convert analog signals ($R(t)$, $e(t)$) to the digital signals which are inserted into the DSP for the adaptive control analysis
D/A convertor	To convert the digital required current into the analog current and supply it to the actuator
Customized passive fluid mount	To place put the actuator under its lower chamber
Solid-state Relays	To control switch S_1 and S_2
Capacitor	To be shunted to the voice coil in shock condition and increase the mount's dynamic stiffness
Accelerometer	To detect shock condition

In order to experimentally verify the performance of the proposed design and its Filtered-X LMS controller, the components listed in Table 5-3, are required. The experimental prototype of the active fluid mount was designed and manufactured by the author (See Figure 5-13). However, due to some technical problems with the dynamic testing machine (MTS810) (See Figure 5-14), the test could not be conducted. In order to conduct the test, a new servo valve with high frequency capability and frequency sweep software need to be purchased. Due to engine mount

forces being small, a new load cell needs to be also purchased. Lack of calibrated load cell for low forces, software, and high frequency servo-valve, forced us to leave the experiment to be conducted by future researchers.

(a) Upper Chamber Rubber

(b) Electromagnetic actuator (Voice coil)

(c) Inertia track

(d) Lower chamber

(e) Lower spring support plate

(f) Plate to connect mount to MTS machine

(g) Lower piston

(h) Fabric diaphragm

Figure 5-13 Active fluid mount components: (a) Upper Chamber Rubber, (b) Electromagnetic actuator (Voice coil), (c) Inertia track, (d) Lower chamber, (e) Lower spring support plate, (f) Plate to connect mount to MTS machine, (g) Lower piston, (h) Fabric diaphragm

Figure 5-14 Experimental set up to evaluate of active mount performance

5.6 Conclusion

In most vibration textbooks, vibration is referred to as small amplitude oscillation, and shock is referred to as large amplitude oscillation. In automotive applications, the car engine experiences both small amplitude vibrations due to the imbalance of the engine rotating parts and cylinder firings, and large amplitude vibrations due to sudden starts and stops of the car, and the car going over bumps and potholes.

To reduce cabin noise and vibration due to engine vibrations, the engine mounts need to act as vibration isolators, and to control the engine large motions due to shock, the engine mounts need to turn into dampers. In the event of sensors and actuator malfunction, the engine mounts need to still perform passively. And in the event that

the need for actuator power becoming too much, the engine mounts can become semi-active to preserve power.

Here in this chapter, we have introduced an engine mounting design that could act as an active mount, a semi-active mount, a passive mount, and a damper as and when needed. This is the novelty of our design.

Bond graph modeling technique was used to model the active mount design and MATLAB was used for simulations. The simulation results for automotive applications were obtained and the performance of the proposed active mount was evaluated. It was shown that the proposed design can eliminate transmitted force from the engine to the structure in a frequency range of 15 Hz to 125 Hz.

Moreover, it was shown that, the proposed active mount system is able to act as a passive, semi-active, or active mount; and in the event of shock, the mount can be converted to a damper mode creating high dynamic stiffness and damping and prevent large engine displacements. Finally, an adaptive controller was proposed based on Filtered-X LMS algorithm and its performance was indicated.

Chapter6 Conclusion and future works

6.1 Conclusion

As it was explained in Chapter 2, the existing fluid mount designs have several deficiencies. Here in this thesis all those deficiencies were addressed. First of all, the feasibility of using two 3-layer PZT beams in conjunction with a single-pumper passive fluid mount to create a multiple notch semi-active fluid mount was studied. Two 3-layer PZT beams were used to vary the lower chamber volumetric stiffness and thus the locations of the notch frequencies of the mount. The bond graph model of a multi-layer PZT beam was derived and the proposed semi-active mount was modeled. It was shown that, by adding a variable capacitance to the PZT layers and changing the capacitive shunt condition from the open circuit to short circuit, the notch frequencies of the mount can be tuned. The simulation results for the business jet turbofan engine applications were presented and it was shown that the first notch frequency can be changed by 15 Hz and the second notch frequency can be changed by 32 Hz. This variation in the notch frequencies is adequate for notch tuning for turbofan engine applications.

The proposed design provides not only the ability to tune the fluid mount after manufacturing but also the ability to tune the notch frequency location in the field; thus, providing the minimum cabin noise and vibration. This new design reduces the notch frequency tuning cycle time, and eliminates fluid mount redesigns and re-manufacturing processes; therefore, saving time and reducing cost. The proposed design is highly desirable for the aerospace applications where there exist multiple

engine disturbance inputs of various frequencies and the cabin should be isolated from those multiple disturbances simultaneously.

This design provides two additional advantages. First, the design can be developed to a self-contained semi-active fluid mount by adding an energy harvesting device to the system to convert the vibrational energy of the engine to the suitable electrical power which is used to run the capacitive circuit controller and sensors. The self-contained semi-active mount acts independently while no external wiring is required. Second, the design has an ability to convert to an active mount by applying voltage across the PZT beams. In Chapter 3, energy harvesting device was not added to the semi-active PZT mount but self-contained capability can be developed and implemented similar to the one which was explained in the last section of Chapter 4.

The simulation results of the two lower pistons' displacement revealed that the proposed design is more suitable for turbofan engine applications than automotive applications since the input displacement of turbofan engines is very low as compared to the automotive engines'. Therefore the thesis was continued by another semi-active fluid mount which is suitable for vibration isolation in both automotive and aerospace applications.

In order to illustrate the new design, in Chapter 4 of the thesis, first of all, an electromechanical tuned vibration absorber (ETVA) was introduced which includes a voice coil, a single degree of freedom mass spring system and an electronic RLC circuit. The electronic circuit (RLC circuit) was added to the voice coil in series and it was used to tune the natural frequency of the ETVA. The mathematical model of the

ETVA was defined and a parametric study was done to evaluate the effect of variable resistance, variable inductance and variable capacitance in tuning the natural frequency of the ETVA. Parametric studies showed that if the technology of superconductive wires in room temperature is improved; using a voice coil with a very low resistance which is made of superconductive wires and a variable resistance would create the best adaptive ETVA which is tunable in a large frequency range (more than 900%). However, among existing commercially available electrical components, it was shown that the variable capacitance is more effective in changing natural frequency of the TVA than changing resistance or inductance. Natural frequency of the ETVA can be changed by 66% ($\frac{\omega_{\max} - \omega_{\min}}{\omega_{\min}} = 0.66$) when the shunted capacitance condition is varied from the open circuit to the short circuit.

The viability of the proposed ETVA design was verified by developing an experimental test rig. The experimental data and theoretical simulation data were compared and it was shown that they are quite similar. Experimental data reveals that a variable capacitance is able to change the natural frequency of the ETVA in a wide frequency range.

Thereafter, the application of the ETVA in creating a variable notch frequency fluid mount design was studied. Simulation results showed that indeed it is possible to vary notch frequency location approximately 25% with the change of shunted capacitance of the ETVA. As the capacitance is changed, the effective volumetric stiffness of the bottom chamber changes; and so does the notch frequency. Therefore in situations which notch frequency location changes due to tolerances on all the fluid mount dimensions, material property variations, and variation in elastomer molding

processes, the notch frequency can be easily tuned in any desired location and it reduces remanufacturing cost and the fluid mount notch frequency tuning cycle time. Moreover, the notch frequency can be tuned according the secondary disturbance variations, creating by firing pulses.

A practical control scheme and an appropriate electrical circuit design were introduced in order to change the shunt capacitance of the semi-active mount according to the input disturbance frequency variation.

Afterwards, the application of an energy harvesting device in creating a self-contained semi-active fluid mount was evaluated, important criteria in selecting such a device for automotive applications were listed, and the physical model of the proposed self-contained semi-active fluid mount design and the electrical circuit configuration were illustrated. The capacitive tuning control strategy and the self-contained capability were developed such that they are suitable to implement on both semi-active fluid mount designs, introduced in Chapters 3 and 4.

An active fluid mount design was presented in Chapter 5 of the thesis. Dynamic analysis of the proposed active fluid mount design was done using bond graph modeling technique and an adaptive controller, based on Filtered-X LMS algorithm was proposed. The simulation results were shown and the performance of the proposed active mount and its controller were evaluated. It was shown that, the proposed active mount system is able to act as a passive, semi-active and active mount.

The application, advantages and disadvantages of three proposed design is summarized in Table 6-1.

Table 6-1 Summary of three proposed fluid mount designs

Design Description	Application	Advantages	Disadvantages
Semi-active fluid mount with two multi-layer cantilever PZT beams	Aerospace (Business jet Turbofan engines)	<ul style="list-style-type: none"> • Isolates structure from multiple disturbances simultaneously • Provides two tunable notch frequencies • Reduces the notch frequency tuning cycle time, • Eliminates fluid mount redesigns and re-manufacturing processes • Can be developed to a self-contained semi-active fluid mount by adding an energy harvesting device to the system 	<ul style="list-style-type: none"> • Is only suitable for aerospace applications, where the range of input displacements is low.
Self-contained semi-active fluid mount with an electromagnetic tuned vibration absorber	Automotive	<ul style="list-style-type: none"> • Provides a variable notch frequency • Reduces the notch frequency tuning cycle time, • Eliminates fluid mount redesigns and re-manufacturing processes • can act as passive, semi-active and active fluid mount • Once, energy harvester voice coil is added, it no external wiring and external power needed 	<ul style="list-style-type: none"> • Coil resistance should be kept low to give a wide range of notch frequency variation • By adding energy harvester voice coil, it would be more expensive than conventional single-pumper passive fluid mount
Active fluid mount with an electromagnetic actuator	Automotive	<ul style="list-style-type: none"> • Provides a variable notch frequency • Reduces the notch frequency tuning cycle time, • Eliminates fluid mount redesigns and re-manufacturing processes • Eliminates transmitted force from engine to the structure in a wide frequency range 	<ul style="list-style-type: none"> • Requires a controller which may add cost

6.2 Contributions

In summary, the contributions of this research can be listed as below:

1. Created a controllable double notch semi-active fluid mount using two 3-layer PZT beams in conjunction with a conventional single-pumper passive fluid mount.
2. Derived the bond graph model of a multi-layer PZT beam. This bond graph model is generic and it can be used by all researchers in this field.
3. Created a new adaptive electromagnetic tuned vibration absorber (ETVA), easily tuned by an electronic circuit.
4. Created a controllable notch frequency semi-active fluid mount using an ETVA in conjunction with a conventional single-pumper passive fluid mount.
5. Developed a practical control scheme and an appropriate electrical circuit to change the shunt capacitance of the semi-active mount according to the input disturbance frequency variation.
6. Developed the semi-active fluid mount design to be a self-contained mount by connecting an energy harvesting device to the mount. The vibration energy of the engine is harvested and controlled to electricity and this electricity is used to supply power to the controller and the sensors.
7. Proposed a multi-functional fluid mount which is able to act as a passive, semi-active and active vibration isolator and act like a damper as and when needed.

6.3 Future Work

Here in this section, the next phases of this research which might be conducted by other researchers are listed.

1. Fabricate the multiple-notch semi-active fluid mount system, presented in Chapter 3. The prototype has been already designed by the author. The prototype should be tested by a MTS dynamic testing machine or a shaker which is able to run in high frequencies for a large number of cycles in order to make sure that the PZT beam are durable and can sustain dynamic loads in real situation. Moreover, the effect of temperature variation on the performance of the mount should be studied.
2. Modify the control strategy presented in chapter 4, for tuning the capacitance shunt value according to the disturbance input variation. The proposed controller is not adaptive and it is mostly suitable for the automotive applications rather than aerospace application; because in aerospace applications, the temperature variation is higher and it may affect the mount parameters significantly. So, it is suggested to study and design an adaptive controller which can tune the mount even in harsh environments.
3. The capability of energy harvesting device in creating required controller and sensor powers was presented. However, it is recommended to test the prototype experimentally.
4. The experimental prototype of the active fluid mount was designed and manufactured by the author. However due to some technical problems of the available dynamic testing machine (MTS810) in the laboratory, the experiment could not been done and it was left for the future researchers to conduct the experimental section.

6.4 Research Contribution

6.4.1 Journal Papers

1. N. Vahdati, S. Heidari, Novel Semi-Active Fluid Mount using a Multi-layer Piezoelectric Beam, Journal of Vibration and Control, 2009 (Accepted for publication)
2. N. Vahdati, S. Heidari, Adaptive Electromechanical Tuned Vibration Absorber and its Application in Semi-Active Fluid Mounts, ASME Journal of Vibration and Acoustics, 2009, (submitted for review).
3. S. Heidari, N. Vahdati, R. Tikani, S. Ziaei-Rad, Development of an Electromagnetic Active Engine Mount, Journal of Vehicle System Dynamics, 2010, (submitted for review).
4. N. Vahdati, S. Heidari, Self-Contained Semi-Active Fluid Mount Design, Journal of Vibration and Control, 2010, (submitted for review).
5. R. Tikani, S. Ziaei-Rad, N. Vahdati, S. Heidari, M. Esfahanian, A Novel Double-Notch Passive Hydraulic Mount Design, International Journal of Vehicle Design, 2009, (submitted for review).
6. R. Tikani, N. Vahdati, S. Ziaei-Rad, S. Heidari, M. Esfahanian, Two-Mode Operation Engine Mount Design for Automotive Application, Journal of Vibration and Control, 2010, (submitted for review).

6.4.2 Conference Papers

1. N. Vahdati, S. Heidari, Bond Graph Model of a Multi-layer Piezoelectric (PZT) Beam and Creation of a Semi-active Hydraulic Mount using a PZT Beam,

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- 2009 ASME Dynamic Systems and Control Conference, Hollywood, California, October 12-14, 2009. (Accepted for publication and presentation).
2. N. Vahdati, S. Heidari, Adaptive Electromechanical Tuned Vibration Absorber and Its Application in Single Pumper Fluid Mounts, 2010 New Challenges in Aerospace Technology and Maintenance Conferance, Suntec Singapore Intl Convention & Exhibition Centre, Singapore, February 1st, 2010. (Accepted for publication and presentation).
 3. N. Vahdati, S. Heidari, Self-contained Active Fluid Mount, 2010 SPIE Smart Structures and Materials+ Nondestructive Evaluation and Health Monitoring, San Diego, California, USA, March 9-10, 2010 (Accepted for publication).

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Appendix A Bond Graph Modeling of the Bernoulli-Euler Beam and Voice coil

In order to use bond graph modeling technique [57] for the investigation of 3-layer PZT cantilever beam flexibility application in controlling notch frequency of fluid mount, the general bond graph modeling of Bernoulli-Euler beam is presented for a uniform 1 layer metal beam. Figure A-1 shows the Bernoulli-Euler beam acted upon by two external forces $F_1(t)$ and $F_2(t)$. It should be noticed that the general case is presented. However, for the current study, the clamped-free beam is used; thus $F_1(t)$ is the vertical reaction force at the clamped side ($x_1=0$) and $F_2(t)$ is the external force acting upon lower piston in fluid mount ($x_1=L$).

Figure A-1 A Bernoulli-Euler beam acting upon external forces [57]

The equation of motion of the uniform Bernoulli-Euler beam with point forcing is:

$$EI \frac{\partial^4 w}{\partial x^4} + \rho A \frac{\partial^2 w}{\partial t^2} = F_1 \delta(x - x_1) + F_2 \delta(x - x_2) \quad (\text{A-1})$$

Appendix A

The force-free boundary conditions are that no shear force and no moment exist at $x=0$ and $x=L$. Therefore, the boundary condition can be written in terms of $w(x,t)$ as:

$$\text{Zero-moment constraint: } \frac{\partial^2 w}{\partial x^2}(0,t) = \frac{\partial^2 w}{\partial x^2}(L,t) = 0 \quad (\text{A-2})$$

$$\text{Zero-shear constraint: } \frac{\partial^3 w}{\partial x^3}(0,t) = \frac{\partial^3 w}{\partial x^3}(L,t) = 0 \quad (\text{A-3})$$

First of all, the solution is separated to the homogenous form of Equation (A-1) and $w(x,t)$ is assumed as:

$$w(x,t) = Y(x)f(t) \quad (\text{A-4})$$

Substituting (A-4) into (A-1) yields:

$$EI \frac{d^4 Y}{dx^4} f + \rho A Y \frac{d^2 f}{dt^2} = 0 \quad (\text{A-5})$$

Dividing by $\rho A Y f$ yields:

$$\frac{EI}{\rho A} \frac{d^4 Y}{dx^4} + \frac{1}{f} \frac{d^2 f}{dt^2} = 0 \quad (\text{A-6})$$

This equation can be satisfied at all x and all t only if both terms equal the same constants. By convention we let the second term equal $-\omega^2$, yielding

$$\frac{d^2 f}{dt^2} + \omega^2 f = 0 \quad (\text{A-7})$$

and

$$\frac{d^4 Y}{dx^4} - \frac{\rho A}{EI} \omega^2 Y = 0 \quad (\text{A-8})$$

or

$$\frac{d^4 Y}{dx^4} - k^4 Y = 0 \quad (\text{A-9})$$

Where

$$k^4 = \frac{\rho A}{EI} \omega^2 \quad (\text{A-10})$$

Equation (A-9) is a total differential equation which, when solved subject to the boundary conditions (A-2) and (A-3), will yield the mode shapes and associated mode frequencies. Using Equation (A-4) in the boundary conditions yields:

$$\frac{d^2 Y}{dx^2}(0) = \frac{d^2 Y}{dx^2}(L) = \frac{d^3 Y}{dx^3}(0) = \frac{d^3 Y}{dx^3}(L) = 0 \quad (\text{A-11})$$

The spatial Equation (A-9) has the general solution

$$Y(x) = A \cosh(kx) + B \sinh(kx) + C \cos(kx) + D \sin(kx) \quad (\text{A-12})$$

By using Equations (A-11) and (A-12), the frequency equation is derived as:

$$\cosh(K_n L) \times \cos(K_n L) = 1 \quad (\text{A-13})$$

And the mode shape functions:

$$Y_n(x) = (\cos(K_n L) - \cosh(K_n L))(\sin(K_n x) + \sinh(K_n x)) - (\sin(K_n L) - \sinh(K_n x))(\cos(K_n x) + \cosh(K_n x)) \quad (\text{A-14})$$

By solving (A-13) numerically for the special values of $K_n L$, and use these in (A-10), the mode frequencies are obtained as below:

$$\omega_n^2 = \frac{EI}{\rho A} \frac{(K_n L)^4}{L^4} \quad (\text{A-15})$$

The force-free boundary conditions permit $\omega_n = K_n = 0$ to be a mode frequency.

Using this information yields:

$$\frac{d^4 Y}{dx^4} = 0 \rightarrow Y = c_1 x^3 + c_2 x^2 + c_3 x + c_4 \quad (\text{A-16})$$

There are two possible solutions to (A-16) that satisfy the boundary conditions. They are $Y = \text{const}$ and $Y = ax + b$. These are called rigid-body modes, and it is convenient to think of them as rigid-body vertical translation of the entire beam and rigid-body rotation of the beam about the centrally located centre of mass. Thus

$$Y_{00} = 1 \quad (\text{A-17})$$

And

$$Y_0 = x - \frac{L}{2} \quad (\text{A-18})$$

The bendings are given by (A-14) and rather complicated functions. The rigid-body modes are orthogonal:

$$\int_0^L \rho A Y_n(x) Y_m(x) dx = 0, n \neq m \quad \text{For } n, m = 0, 1, 2, 3, \dots \quad (\text{A-19})$$

The calculation of the forced response proceeds as:

$$w(x, t) = \sum_{n=0}^{\infty} Y_n(x) \eta_n(t) \quad (\text{A-20})$$

Use this in equation (A-1), and multiply each term by $Y_m(x)$ and integrate with respect to x from $x=0$ to $x=L$. We then use the orthogonality property of the modes and obtain:

$$\left(\int_0^L \rho A Y_n^2 dx \right) \ddot{\eta}_n + \left(\int_0^L \rho A Y_n^2 dx \right) \omega_n^2 \eta_n = F_1 Y_n(x_1) + F_2 Y_n(x_2) \quad (\text{A-21})$$

The first zero-frequency mode yields:

$$\left[\int_0^L \rho A (1)^2 dx \right] \ddot{\eta}_{00} = F_1 + F_2 \Rightarrow m \ddot{\eta}_{00} = F_1 + F_2 \quad (\text{A-22})$$

This simply states that the external forces accelerate the centre of mass of the beam.

The other zero-frequency mode yields:

$$\left[\int_0^L \rho A \left(x - \frac{L}{2} \right)^2 dx \right] \ddot{\eta}_0 = F_1 \left(x_1 - \frac{L}{2} \right) + F_2 \left(x_2 - \frac{L}{2} \right) \quad (\text{A-23})$$

Or

$$J_g \ddot{\eta}_0 = F_1 \left(x_1 - \frac{L}{2} \right) + F_2 \left(x_2 - \frac{L}{2} \right) \quad (\text{A-24})$$

Which simply states that the moment of external forces about the centre of mass produces angular acceleration $\ddot{\eta}_0$, where J_g is the centroidal moment of inertia of the beam. The bond graph for a beam with force-free conditions is shown in Figure A-2.

Figure A-2 Bond graph for a beam with force-free conditions [57]

The rigid-body modes, which will be present when ever force-free boundaries are assumed; appear simply as inertia elements with no associated model stiffness. The first inertia parameter is beam mass m , and the second is the beam moment of inertia J_g . The $-TF-$ elements connected to the rigid-body modes correctly apply the forces and moments to these elements. The modal masses are symbolically,

$$m_n = \int_0^L \rho A Y_n^2 dx, \quad n = 1, 2, 3, \dots \quad (\text{A-25})$$

And the modal stiffnesses are:

$$k_n = m_n \omega_n^2 \quad (\text{A-26})$$

The modes Y_n are given by Equation (A-14) and the frequencies ω_n come from solving Equation (A-13). The non-dimensional form of (A-26) is:

$$\frac{m_n}{m} = \int_0^1 Y_n^2 \left(\frac{x}{L} \right) d \frac{x}{L} \quad (\text{A-27})$$

Therefore, with the above equations and also general bond graph model (Figure (A-2)), all the specific examples can be modeled and the regarding state-space equations can be derived. The external moment was not derived here. However it can also be applied to the structure of figure (A-1). If a moment is applied instead of a force, the associated -TF- element, coming from the moment 0-junction, have the mode shapes, $(\frac{dY_n}{dx})$, evaluated at the location of the moment as their moduli instead of the mode shapes themselves.

Bond Graph Model of a Voice Coil

According to the Lorentz Law, the force on a conductor of length l carrying current i in a magnetic field of flux density B is:

$$F = Bli \quad (\text{A-28})$$

If as is usual, the vector directions of F , B and i are mutually orthogonal, the corresponding voltage e induced in the conductor if it moves with respect to the field with velocity V is:

$$e = BlV \quad (\text{A-29})$$

So that the power balance, $(FV = ei)$ is maintained. Equations (A-28) and (A-29) are represented by a gyrator bond graph element with parameter Bl . Considering the inductance and resistance of the coil wire, the overall bond graph model of a voice coil, with the coil having resistance R and inductance L is shown in Figure A-3.

Figure A-3 Bond graph model of a voice coil

Appendix B Self-contained Semi-active Fluid Mount

Equations:

As it was explained in section 4-10 from the bond graph model (Figure 4-12), the state-space equations of the system were obtained. In order to find out the required capacitive shunt value which makes the notch frequency of the semi-active mount coincide with the disturbance value, first the Laplace transform of the state-space Equations 4-13 to 4-20 and Equation 4-21 were derives as: (assuming zero initial condition)

$$\dot{q}_3 = V_{in} \xrightarrow{\text{Laplace}} sq_3 = sX_{in} \quad (\text{B-1})$$

$$\dot{q}_6 = A_p V_{in} - \frac{P_{10}}{I_{10}} \xrightarrow{\text{Laplace}} sq_6 = A_p sX_{in} - \frac{P_{10}}{I_{10}} \quad (\text{B-2})$$

$$\dot{p}_{10} = \frac{q_6}{C_6} - R_8 \frac{P_{10}}{I_{10}} - \frac{q_{11}}{C_{11}} \xrightarrow{\text{Laplace}} sp_{10} = \frac{q_6}{C_6} - R_8 \frac{P_{10}}{I_{10}} - \frac{q_{11}}{C_{11}} \quad (\text{B-3})$$

$$\dot{q}_{11} = \frac{P_{10}}{I_{10}} - A_{pb} \frac{P_{14}}{I_{14}} \xrightarrow{\text{Laplace}} sq_{11} = \frac{P_{10}}{I_{10}} - A_{pb} \frac{P_{14}}{I_{14}} \quad (\text{B-4})$$

$$\begin{aligned} \dot{p}_{14} &= A_{pb} \frac{q_{11}}{C_{11}} - \frac{q_{15}}{C_{15}} - R_{16} \frac{P_{14}}{I_{14}} - GY_{17-18} \frac{P_{19}}{I_{19}} \xrightarrow{\text{Laplace}} \\ sp_{14} &= A_{pb} \frac{q_{11}}{C_{11}} - \frac{q_{15}}{C_{15}} - R_{16} \frac{P_{14}}{I_{14}} - GY_{17-18} \frac{P_{19}}{I_{19}} \end{aligned} \quad (\text{B-5})$$

$$\dot{q}_{15} = \frac{P_{14}}{I_{14}} \xrightarrow{\text{Laplace}} sq_{15} = \frac{P_{14}}{I_{14}} \quad (\text{B-6})$$

$$\dot{p}_{19} = GY_{17-18} \frac{P_{14}}{I_{14}} - R_{20} \frac{P_{19}}{I_{19}} - \frac{q_{21}}{C_{21}} \xrightarrow{\text{Laplace}} sp_{19} = GY_{17-18} \frac{P_{14}}{I_{14}} - R_{20} \frac{P_{19}}{I_{19}} - \frac{q_{21}}{C_{21}} \quad (\text{B-7})$$

$$\dot{q}_{21} = \frac{P_{19}}{I_{19}} \xrightarrow{\text{Laplace}} sq_{21} = \frac{P_{19}}{I_{19}} \quad (\text{B-8})$$

$$F_{in} = \frac{q_3}{C_3} + R_2 V_{in} + A_p \frac{q_6}{C_6} \xrightarrow{\text{Laplace}} F_{in} = \frac{q_3}{C_3} + R_2 sX_{in} + A_p \frac{q_6}{C_6} \quad (\text{B-9})$$

Since, damping parameters just affect the depth of the fluid mount's notch not its location, they can be neglected while we are going to determine the notch frequency location. Thus by putting all damping parameters (R s) equal to zero in Equations (B-1) to (B-9), the following equations are derived:

$$sq_3 = sX_{in} \quad (\text{B-10})$$

$$sq_6 = A_p sX_{in} - \frac{P_{10}}{I_{10}} \quad (\text{B-11})$$

$$sp_{10} = \frac{q_6}{C_6} - \frac{q_{11}}{C_{11}} \quad (\text{B-12})$$

$$sq_{11} = \frac{P_{10}}{I_{10}} - A_{pb} \frac{P_{14}}{I_{14}} \quad (\text{B-13})$$

$$sp_{14} = A_{pb} \frac{q_{11}}{C_{11}} - \frac{q_{15}}{C_{15}} - GY_{17-18} \frac{P_{19}}{I_{19}} \quad (\text{B-14})$$

$$sq_{15} = \frac{P_{14}}{I_{14}} \quad (\text{B-15})$$

$$sp_{19} = GY_{17-18} \frac{P_{14}}{I_{14}} - \frac{q_{21}}{C_{21}} \quad (\text{B-16})$$

$$sq_{21} = \frac{P_{19}}{I_{19}} \quad (\text{B-17})$$

$$F_{in} = \frac{q_3}{C_3} + R_2 sX_{in} + A_p \frac{q_6}{C_6} \quad (\text{B-18})$$

By putting the Laplace transformed of transmitted force equation equal to zero (See Equation (B-18)) and substituting q_3 and q_6 as the functions of X_{in} and fluid mount

parameters, the notch frequency location is derived as a function of fluid mount parameters and input disturbance frequency as:

$$\begin{aligned}
 & C_{21}s^7((C_6 + A_p^2 C_3)(I_{10}I_{19}I_{14}C_{15}C_{11})) + \\
 & C_{21}s^5 \left(\begin{aligned} & (C_6 + A_p^2 C_3)(I_{10}C_{11}I_{19} + I_{10}I_{19}C_{15}A_{pb}^2 + I_{19}I_{14}C_{15}) \\ & + I_{10}C_{11}GY_{17-18}^2 C_{15}) \\ & + (I_{19}I_{14}C_{15}C_{11}) \end{aligned} \right) + \\
 & C_{21}s^3 \left(\begin{aligned} & (C_6 + A_p^2 C_3)(GY_{17-18}^2 C_{15} + I_{19}) + \\ & (C_{11}GY_{17-18}^2 C_{15} + C_{11}I_{19} + I_{19}C_{15}A_{pb}^2) \end{aligned} \right) \\
 & = \\
 & s^5((C_6 + A_p^2 C_3)(I_{10}C_{11}I_{14}C_{15})) + \\
 & s^3 \left(\begin{aligned} & (C_6 + A_p^2 C_3)(I_{10}C_{11} + I_{10}C_{15}A_{pb}^2 + I_{14}C_{15}) \\ & + (C_{11}I_{14}C_{15}) \end{aligned} \right) + \\
 & s((C_6 + A_p^2 C_3) + s(C_{11} + C_{15}A_{pb}^2))
 \end{aligned}$$

Therefore;

$$C_{21}(s^7 \gamma_1 + s^5 \gamma_2 + s^3 \gamma_3) = s^5 \gamma_4 + s^3 \gamma_5 + s \gamma_6 \quad (\text{B-19})$$

where

$$\gamma_1 = (C_6 + A_p^2 C_3)(I_{10}I_{19}I_{14}C_{15}C_{11}) \quad (\text{B-20})$$

$$\begin{aligned}
 \gamma_2 = & (C_6 + A_p^2 C_3)(I_{10}C_{11}I_{19} + I_{10}I_{19}C_{15}A_{pb}^2 + I_{19}I_{14}C_{15} + I_{10}C_{11}GY_{17-18}^2 C_{15}) \\
 & + (I_{19}I_{14}C_{15}C_{11}) \quad (\text{B-21})
 \end{aligned}$$

$$\gamma_3 = (C_6 + A_p^2 C_3)(GY_{17-18}^2 C_{15} + I_{19}) + (C_{11}GY_{17-18}^2 C_{15} + C_{11}I_{19} + I_{19}C_{15}A_{pb}^2) \quad (\text{B-22})$$

$$\gamma_4 = (C_6 + A_p^2 C_3)I_{10}C_{11}I_{14}C_{15} \quad (\text{B-23})$$

$$\gamma_5 = (C_6 + A_p^2 C_3)(I_{10}C_{11} + I_{10}C_{15}A_{pb}^2 + I_{14}C_{15}) + (C_{11}I_{14}C_{15}) \quad (\text{B-24})$$

$$\gamma_6 = (C_6 + A_p^2 C_3) + (C_{11} + C_{15}A_{pb}^2) \quad (\text{B-25})$$

Appendix C Actuator Review

The purpose of this appendix is to represent various types of actuators and compare their capabilities in order to satisfy the required criteria for the active fluid mount application (Chapter 5).

1. Actuator Selection

There are five important parameters which can be used to describe the attributes and function of an actuator:

1. Type
2. Location
3. Impedance
4. Relative size
5. Bandwidth

According to these criteria, various types of actuators are manufactured or have been tested in laboratories and they can be categorized as below:

- Electromagnetic Actuators
- Piezoelectric Actuators
- Electrorheological and Magnetorheological Actuators
- Hydraulic
- Pneumatic
- Hybrid-combination of two or more technologies
- Magnetostrictive
- Electrostatic

- Shape Memory
- Diamagnetism
- Phase Change
- Thermomechanical
- Electrohydrodynamic

Among above commercially available actuators, electromagnetic actuators, piezoelectric actuators, Magnetostrictive are the most suitable choices for the active mount applications; because their stroke and overall dimension is the order of actuator requirements using in the active mount. The following sections illustrate the characteristics of the commercially available of aforementioned actuator types; also advantages and disadvantages of each type are listed.

2. Electromagnetic Actuators

Technology	Electromagnetic (moving coil)
References	BEI Company http://www.beikimco.com/
Strengths/Weaknesses	<ul style="list-style-type: none">• Very fast operating speeds• Extreme positioning accuracy• Scale ability• High degree of efficiency• Have an upper temperature limit ($\approx 180^{\circ}\text{C}$)• Performance is primarily limited by the properties of the material used in constructing the actuator• Difficult to build small electromagnetic coils• The most devices require perpendicularity between the current conductor and the moving element• Simply on/off control but relatively limited functions usually only lift and not lift states available
Maximum Displacement (Stroke)	LA12-17-000A : 3.81 mm LA10-12-027A : 4.57 mm LA05-30-001Z : 10 mm
Frequency Range (Hz)	> 1000 Hz
Sensitivity	LA12-17-000A : 7.34 N/Amp LA10-12-027A : 5.56 N/Amp LA05-30-001Z : 3.6 N/Amp
Response Time (ms)	<< 2 Theoretical
Relative Speed	Fast
Maximum Efficiency (%)	
Need of Voltage (V)/ Current (A)	Dependent on relative size and principle
Size (Height/Length/Width)	LA12-17-000A : L=35 mm, Q=30 mm LA10-12-027A : L=20.3 mm, Q=25.4 mm LA05-30-001Z : L=43.18 mm, Q=30.48 mm
Weight (gr)	LA12-17-000A : 133 field assembly, 32.6 coil assembly LA10-12-027A : 56.7 field assembly, 13.89 coil assembly LA05-30-001Z : 396 field assembly, 34 coil assembly Depend on the force that is required. Permanent magnet instead of electromagnet increases the weight.
Comments	Magnetic actuators are perhaps among the oldest types of actuators. Magnetic actuation methods offer the possibility of generating repulsive forces in addition to attractive forces.

Technology	Electromagnetic (moving coil)
References	MotiCont company http://www.moticont.com/voice-coil-motor.htm
Strengths/Weaknesses	<ul style="list-style-type: none"> • Very fast operating speeds • Extreme positioning accuracy • Scale ability • High degree of efficiency • Have an upper temperature limit ($\approx 180^{\circ}\text{C}$) • Performance is primarily limited by the properties of the material used in constructing the actuator • Difficult to build small electromagnetic coils • The most devices require perpendicularity between the current conductor and the moving element • Simply on/off control but relatively limited functions usually only lift and not lift states available
Maximum Displacement (Stroke)	LVCM-025-029-01: 15.2 mm LVCM-032-025-01 :12.7 mm LVCM-025-038-01 : 9 mm
Frequency Range (Hz)	> 1000 Hz
Sensitivity	LVCM-025-029-01: 3.75 N/Amp LVCM-032-025-01:3.1 N/Amp LVCM-025-038-01 : 8.5 N/Amp
Response Time (ms)	<< 2 Theoretical
Relative Speed	Fast
Maximum Efficiency (%)	
Need of Voltage (V)/ Current (A)	Dependent on relative size and principle
Size (Height/Length/Width)	LVCM-025-029-01: L=43 mm, Q=25.4 mm LVCM-032-025-01: L=40 mm, Q=31.8 mm LVCM-025-038-01 : L=49 mm, Q=25.4 mm
Weight (gr)	LVCM-025-029-01: 87 field assembly, 27coil assembly LVCM-032-025-01:136 field assembly, 31 coil assembly LVCM-025-038-01: 102field assembly, 26 coil assembly Depend on the force that is required. Permanent magnet instead of electromagnet increases the weight.
Comments	Magnetic actuators are perhaps among the oldest types of actuators. Magnetic actuation methods offer the possibility of generating repulsive forces in addition to attractive forces.
Accuracy	Extremely high

Technology	Electromagnetic (moving coil and moving magnet available)
References	H2Wcompany http://www.h2wtech.com/noncommmdcactu.htm
Strengths/Weaknesses	<ul style="list-style-type: none"> · Small Size, High Force to Weight Ratio · High Accelerations · No Cogging or Commutation · Custom Designs with Quick Turnaround Times · High Flux Density Neodymium Magnets are Used
Maximum Displacement (Stroke)	According to the models Moving coil : 3 mm to 132.1 mm Moving magnet : 2.2 mm to 76.2 mm
Frequency Range (Hz)	> 1000 Hz
Response Time (ms)	<< 2 Theoretical
Relative Speed	Fast
Maximum Efficiency (%)	
Need of Voltage (V)/ Current (A)	Dependent on relative size and principle
Size (Height/Length/Width)	According to the models From L=9.1 mm, Q=9.5 mm to L=50.4 mm, Q=150 mm
Weight (gr)	<p>According to the models Total mass 5.7 to 138</p> <p>Depend on the force that is required. Permanent magnet instead of electromagnet increases the weight.</p>
Comments	Magnetic actuators are perhaps among the oldest types of actuators. Magnetic actuation methods offer the possibility of generating repulsive forces in addition to attractive forces.
Accuracy	Extremely high

2. Piezoelectric Actuators

Technology	Piezoelectric
References	PI company http://www.physikinstrumente.com/en/products/piezo/piezo_selection.php?table=all#bender
Strengths/Weaknesses	<ul style="list-style-type: none">• High stiffness results in isotropic high actuator performance• Easily controlled• Provide fast response• Small dimensions and weight• Simply driven by voltage• Maximum strain under electrical field can approach 0.2%• They can cover a wide range of frequencies• High precession• They can hardly be used at very low frequency and at dc.• Very large force• Very small displacement
Maximum Displacement (Stroke)%	According to the materials Ceramics 0.2, Single Crystal 1.7, Polymer 0.1
Frequency Range (Hz)	> 100 KHz
Response Time (ms)	<< 1
Relative Speed	Fast
Maximum Efficiency (%)	>90
Need of Voltage (V)/ Current (A)	From 100 V to 1000 V for full extension. Very low current use (mA).
Size (Height/Length/Width)	According to the models (Generally small)
Weight (gr)	Low
Price	Depend on the design, from low to very high
Comments	The best known and most used piezoceramics, Lead Zirconate Titanate (PZT), can deliver up to 0.1% strain.
Accuracy	High in closed loop design

Technology	Piezoelectric (Multilayered Stacked)			
References	Adaptronics, Inc http://www.adaptronics.com/products/piezoelectric_actuators/direct_piezo_actuators/index.html			
Strengths/Weaknesses	• High power efficiency • Low drive voltage			
Products Names	DPA40	DPA60	DPA80	PPA40M
Maximum Displacement (Stroke) mm	0.04	0.06	0.08	0.032
Maximum Force (N)	3500	3500	3500	800
Resonance Frequencies (Hz)	9000	7000	5000	2000
Stiffness (N/μm)	87.5	58.3	43.8	25
Frequency Range (Hz)				
Response Time (ms)	0.08	0.1	0.14	0.04
Relative Speed	Very Fast			
Maximum Efficiency (%)				
Need of Voltage (V)/ Current (A)	-20 to 150 V	20 to 150 V	-20 to 150 V	20 to 150 V
Size (Height/Length/Width)	52.5/25.0/25.0	82.5/25.0/25.0	112.5/25.0/25.0	49.0/10.0/8.0
Weight (gr)	155	230	310	25
Price	Depend on the design, from low to very high			
Comments	DPA40, DPA60 and DPA80 have diameter 25.0 mm. PPA40M exist with less displacement.

http://www.physikinstrumente.com/en/pdf/P212_Datasheet.pdf