Fluidless electronic vibration isolator

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Fluidless Electronic Vibration Isolator

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List of Symbols

TVA	Tuned Vibration absorber
ω	Frequency
$K(i\omega)$	Dynamic stiffness
B_r	Damping coefficient of the rubber
RTM	Rubber to metal mount
X	The amplitude of the vibration
k_r	Stiffness of the rubber
C_i	Compliance of the upper chamber
C_b	Compliance of the lower chamber
A_p	Piston area
A_{pt}	The upper piston area
A_{pb}	The lower piston area
M_{b1}	Mass of the lower piston
M_{b2}	TVA mass
K_{b1}	The lower stiffness
K_{b2}	TVA stiffness
B_{b1}	The lower damping
B_{b2}	TVA damping
$\dot{P}$	Effort
$\dot{f}$	Flow
ω_{nc}	Notch frequency
ω_{peak}	Peak frequency
K''	The imaginary stiffness of TVA
$\tan \delta$	The amount of damping in the rubber
K'	Elastic stiffness
$F(t)$	Excitation force induced by the engine
$P_{ER}(t)$	Pressure drop induced by ER fluid

$F_a(t)$	Control force of the piezo-actuator
B	Magnetic field flux density
ℓ	Conductor length
α	Leverage amplifying ration (ℓ_2/ℓ_1)
γ	Voice coil force sensitivity
I	Electronic circuit inductance
R	Electronic circuit resistance
C	Electronic circuit capacitance

Summary

Vibration isolation is an important issue in automotive and aerospace industry. Because of the increasing importance of fuel consumption reduction, there is a huge trend to construct vehicles with lighter bodies and more power intensive engines. But lighter and more power intensive engines tend to deteriorate the vibration isolation properties of the vehicle.

Hydraulic engine mounts are one of the most popular vibration isolators used in industry, to provide cabin noise and vibration isolation. They have been in use in automotive industry since 1962. Their dynamic stiffness is frequency dependent. They provide the best vibration isolation at a frequency called "notch frequency". Significant improvements in vibration isolation using hydraulic engine mounts have been reported in the literature. However due to the complicated fluid structure interaction, their design procedure is somewhat based on trial and error which is a very time consuming and costly method. Because of this trial and error method the manufacturing of fluid mounts is also costly. In addition, as times goes on the dynamic characteristics of the hydraulic engine mount will change. So after a period of operation the engine mount needs to be retuned. The retuning procedure of hydraulic engine mounts is time consuming and costly.

In this work the bond graph modeling technique is employed to design the electro-mechanical equivalent of typical hydraulic engine mount. The objective of this project is to remove the fluid and replace it with a more tunable mechanism. The advantage of the electronic (non-fluid based) engine mount over hydraulic (fluid based) engine mounts is that its dynamic stiffness especially the notch frequency is sensitive to electronic components. This fact helps in avoiding the costly trial and error design procedure because even if the notch frequency of the mount doesn't coincide with the desired value, it can be tuned to its desired value by changing the electronic components. In addition, the design procedure of the electro-mechanical engine mount is much easier than the typical hydraulic mounts.

The proposed design consists of a mechanical part and an electronic part. Different circuits have been examined. The simulation results show that the proposed electronic vibration isolator has very good dynamic properties.

Chapter1 Introduction

1.1 Introduction

Vibration isolation is an important issue in the automotive and the aerospace industry. Engine mounts have been used in these industries to provide cabin noise and vibration isolation, since 1940s.

All engine mounts have to accomplish two basic tasks. Their primary task is to connect the engine to the frame or the fuselage and constrain the relative motion between the engine and the chassis or the fuselage. The second task is that the engine mounts provide cabin noise and vibration isolation.

Since the primary task of the engine mounts is to attach the engine to the frame or fuselage, if the engine mount is too soft, the engine will statically deflect too much. In order to minimize engine static deflection, the engine mounts need to be stiff at low frequencies, while they should have a low dynamic stiffness, at high frequencies, to accomplish the second task. These two requirements are in conflict if we use a constant dynamic stiffness engine mount. The effective engine mount is the one that satisfies these two requirements, simultaneously.

The most common device to reduce the transmission of the engine vibration to the chassis or fuselage is "Fluid Mount" or also called "Hydraulic engine mount". Fluid mounts are composed of two elastomeric chambers separated by a tube which allows the fluid inside the chambers to flow and oscillate between them. Normally the top chamber is stiffer than the lower one. Top side of the fluid mount is connected to the engine and the vibration of the engine goes to the chassis through the engine mount. Vibration of the engine pushes the fluid inside the upper chamber to go through the tube and go to the lower chamber and vice versa. Going through this tube will cause a

pressure drop. The dynamic stiffness of fluid mounts (the ratio of the input force divided by the relative displacement between the engine and the chassis (F_{in}/X_{in})) gets its minimum value at a frequency called “notch frequency”. It is at this frequency that the optimum cabin noise and vibration isolation is provided. In order to provide cabin or fuselage vibration isolation at a certain frequency, the fluid mount notch frequency should be as close as possible to that frequency.

Notch frequency and notch depth are sensitive to any changes in the fluid mount parameters. Due to the manufacturing tolerances, it is very difficult to get the desired notch frequency value at the first manufacturing pass. Even if the fluid mount notch frequency could be placed at its optimum frequency, due to environmental reasons the fluid mount parameters can change; so does the notch frequency. The process of redesigning or retuning the fluid mount notch frequency is costly and time consuming.

The main objective of this research is to replace the fluid with a magnetic and electronic system which can make the engine mount more controllable and easier tunable. This new design makes it possible to change the characteristics of the fluid mount after manufacturing without redesigning it.

1.2 Scope of the Research

The main focus of this research is to develop a semi-active fluidless electronic vibration isolator. The notch frequency of the proposed electronic vibration isolator design will be tunable whereas in the typical passive fluid mounts, notch can not be tuned in real-time.

The concept of fluidless electronics engine mount has been studied in this thesis in chapter 4. The electronic vibration isolator has the advantage of being easily tunable without redesigning the engine mount. So the very costly and time consuming procedure of redesigning and remanufacturing of a typical fluid mount would be avoided. Different designs for electronic vibration isolator are devised.

Beside the fluidless electronic engine mount, a portion of this thesis will be also devoted to design a semi-active multiple notch fluid mount. The semi-active fluid mount that was considered in this thesis is a semi-active hydraulic engine mount plus a Tuned Vibration Absorber (TVA). The mathematical model of this new engine mount is presented in chapter 3 and its characteristics and advantages have been pointed out.

One of the reasons why semi-active fluid mounts are considered in this thesis is the fact that the notch frequency of the semi-active engine mounts can be moved whereas in passive fluid mounts, the notch frequency is stationary. Additionally, the new semi-active fluid mount provides two notches instead of one. So, it can be used in applications where there are two input disturbance signals. If the two notches are tuned to coincide with the two input disturbances, the best vibration isolation would be provided.

The main objectives of this research can be summarized as follows:

- Providing a comprehensive understanding of the semi-active engine mounts.
- Studying different semi-active vibration isolation schemes and figuring out their advantages and disadvantages.
- Studying the application of tuned vibration absorbers on the dynamic properties of an engine mount.
- Developing the electro-mechanical equivalent of the hydro-mechanical engine mount.
- Developing and examining different designs for the electronic engine mount.
- Examining the feasibility of the proposed designs.
- Constructing an experimental setup for electronic engine mounts and verifying the simulation results via experiment.

1.3 Report Outline

Chapter 2

The literature survey is presented in this chapter. The literature review starts by introducing the concept of vibration isolation and in particular the application of engine mounts in providing vibration isolation. Different types of engine mounts are introduced and their advantages and disadvantages are discussed in this chapter.

Chapter 3

A novel design for a semi-active engine mount is presented in chapter 3. The proposed design is based on adding a tuned vibration absorber (TVA) to a typical hydraulic mount. The effect of adding the TVA is investigated in this chapter.

Chapter 4

The concept of electronic fluidless vibration isolator has been studied in this chapter. The electro-mechanical equivalent of the hydro-mechanical engine mount is developed. The dynamic properties of the electronic vibration isolators with different designs have been examined in this chapter.

Chapter 5

The simulation results presented in chapter 4 needs to be verified via experiment. The experimental results for the fluidless electronic vibration isolator are presented in chapter 5.

Chapter 6

The main achievements of the project are summarized in chapter 6 and recommendations for future plans are presented.

Chapter2 Literature Review

2.1 Introduction

Vibration isolation is an important issue in automotive and aerospace industry. Automotive companies compete to design more comfortable vehicles with lower cabin noise and vibration. Additionally, fuel consumption reduction is another important criterion in designing the new vehicles. In order to reduce fuel consumption, the weight of the vehicle is kept as low as possible (Foumani, Khajepour et al. 2003). But lighter and therefore more flexible vehicle chassis along with more power-intensive engines tend to deteriorate the vibration isolation properties of the vehicle (Yu, Naganathan et al. 2001).

The goals of vibration isolation can be different. In some cases, vibration isolation is provided to a component being itself a source of vibration. Sometimes, isolation of a vibration sensitive object from steady or transient vibration sources can be the aim. For example one of the very popular cases is the internal combustion engine of a vehicle. The engine produces undesirable dynamic inputs to the occupant's compartment, but it has to be protected from the intense vibration induced by the road as well. So the transmissibility of vibratory forces or displacements both from the base to the object and vice versa would be of interest (Rivin 2003).

Usually, the car engine or airplane engine is connected to the vehicle chassis or aircraft fuselage via three to four engine mounts.

2.2 Commercial vibration isolation technologies

There are different types of vibration isolators available in the market. Generally, all engine mounts can be categorized into three main groups; namely, passive, semi-active and active engine mounts.

The most popular vibration isolators are listed below:

Passive:

- *Hard Mounts*
- *Passive elastomeric mounts*
- *Wire mesh mounts*
- *Passive fluid mounts*

Semi-active:

- *Semi-active fluid mounts using damping control*
- *Semi-active fluid mounts using stiffness control*
- *Semi-active dry friction engine mount*

Active mounts:

- *Active hybrid engine mount*

The performance of engine mounts is usually represented in terms of their “dynamic stiffness”. Dynamic stiffness is the ratio of the applied force to the engine side of the mount and the relative displacement between the two sides of the mount ($K_{dyn} = \frac{F_{in}}{X_{in}}$).

In order to study the performance of different engine mounts the dynamic characteristics of the ideal engine mount must be investigated.

2.3 Ideal engine mount

The ultimate goal of all engine mount designers is to reach the ideal engine mount design.

In order to identify the ideal engine mount dynamic properties, the characteristics of the existing vibration sources must be identified, in advance.

Usually there exist different sources of vibration in vehicles. For example in the case of an automobile with internal combustion engine, there are two basic vibration sources. The first source of vibration is the disturbances caused by the engine which contains two subparts: the firing pulse induced by the explosion of the fuel in the cylinder, and the inertia force and torque caused by the rotating and reciprocating parts. The second source of vibration is induced by the uneven road and will result in annoying shakes of the vehicle (Foumani, Khajepour et al. 2003).

The nature of the aforementioned sources of vibration are different. The disturbances caused by the road and those induced by the low engine speed (near idle), will result in low frequency and high-amplitude vibration; while the engine induced disturbances (high RPMs) will result in high frequency and low-amplitude vibration.

To isolate the cabin from the engine induced vibrations (relatively high frequency range), low stiffness and low damping is needed. To prevent the engine from bouncing in the low frequency range (induced by uneven road or engine operating near idle speed), the mounting system needs to be 'Hard' which means having high stiffness and damping (Clark 1985; Yu, Naganathan et al. 2001) (see Figure 2.1).

Figure 2.1 Ideal mount dynamic stiffness (Yu, Naganathan et al. 2001)

Beside the aforementioned dynamic properties, ideal engine mount needs to satisfy the following criteria:

- ✓ cost effectiveness
- ✓ long durability
- ✓ broad temperature range
- ✓ easy maintenance
- ✓ stable isolation characteristics
- ✓ to have low set or drift (permanent deformations)

The performance of different engine mounts can be explained by comparing their characteristics to the ideal engine mount characteristics.

The characteristics of the aforementioned isolation technologies are compared to the ideal engine mount characteristics in Table 2-1.

Chapter 2: Literature review

Table 2.1. Comparison between different mounting technologies

<i>Ideal Engine mount</i>	High Dynamic stiffness at low frequencies (1)	Low Dynamic Stiffness at high frequencies (2)	cost effectiveness (3)	long durability (4)	broad temperature range (5)	easy maintenance (6)	stable isolation characteristics (7)	to have low set or drift (permanent deformations) (8)	Controllability (of isolation properties) (9)
<i>Hard Mount</i>	☒	☒	✓	✓	✓	✓	✓	✓	☒
<i>Passive elastomeric mounts</i>	☒	☒	✓	✓	☒	✓	✓	☒	☒
<i>Wire mesh mounts</i>	☒	☒	✓	✓	✓	✓	✓	✓	☒
<i>Passive fluid mounts</i>	✓	Only at the notch frequency	✓	✓	✓	✓	☒	✓	☒
<i>Semi-active fluid mounts using damping control</i>	✓	✓	☒	☒	✓	☒	✓	✓	✓
<i>Semi-active fluid mounts using stiffness control</i>	✓	✓	☒	☒	✓	☒	✓	✓	✓
<i>Semi-active dry friction engine mount</i>	✓	✓	☒	☒	✓	☒	☒	✓	✓
<i>Active hybrid engine mount</i>	✓	✓	☒	☒	✓	☒	✓	✓	✓

Strongly

As Table 2.1 illustrates, active engine mounts provide best vibration isolation. Although active engine mounts can provide very good vibration isolation for many applications, but it should be noted that, active vibration isolation technology would add more cost and complexity to the system. Since for many typical applications it is neither practical nor economic to use active vibration isolators, passive and semi-active engine mounts have been widely used in industry.

As Table 2.1 illustrates, passive/semi-active fluid mounts can provide good vibration isolation while they are cost effective. The main focus of this project is to develop a new isolation technology based on the passive fluid mount model. Beside this objective, a portion of this thesis has been devoted to develop a new semi-active fluid mount design to enhance the isolation characteristics of a typical passive fluid mount.

In the coming sections of this chapter different isolation technologies listed in Table 2.1 are studied in more details. Interested reader is encouraged to go through this chapter to find out more details about different types of isolation technologies.

2.4 Classification of vibration isolators

Generally, all vibration isolators can be categorized into three main groups; namely, passive, semi active and active vibration isolators.

2.4.1 Passive vibration isolators

Hard mounts, elastomeric mounts, wire-mesh mounts and passive fluid mounts are examples of the most common passive vibration isolators.

2.4.1.1 Hard mounts

Hard mounts are simple structures which connect the engine to the chassis or fuselage. They have a low level of flexibility. Although, hard mounts are not effective in preventing the vibration from going through the structure, they have been used in industry because they can last the life of the vehicle.

Figure 2.2 Example of hard attachment structure(DePriest 2001)

2.4.2 Elastomeric engine mounts

Elastomeric or rubber engine mounts have been used in industry since 1930s. They are cost-effective and they can be designed to bear high temperature. Their maintenance is also easy. Rubber to metal bonded elastomeric engine mounts are supposed to have longer life (Yu, Naganathan et al. 2001).

The elastomeric mounts can be represented by Voigt model which consists of a spring and a damper.

Figure 2.3 Mechanical model for elastomeric mount(Yu, Naganathan et al. 2001)

According to Voigt model for rubber ($K(i\omega) = k_r + i\omega b_r$), the dynamic stiffness of an elastomeric engine mount will slightly increase with frequency due to the damping effect (Singh, Kim et al. 1992).

Figure 2.4 Dynamic stiffness of elastomeric mount (Yu, Naganathan et al. 2001)

Singh et al (Singh, Kim et al. 1992) have verified the Voigt model for rubber using experiment. Their result is shown in Figure 2.5.

Figure 2.5 $K(i\omega)$ of rubber simulated by using a hydro-mechanical mount without fluid: —, experiment; ---, Voigt model (Singh, Kim et al. 1992)

As Figure 2.5 illustrates, the dynamic stiffness of an elastomeric engine mount is slightly frequency dependant. So it implies if the elastomeric engine mount has high stiffness and high damping, it will yield a low shake level at low frequencies but the performance at high frequencies will be poor, while a low stiffness and low damping elastomeric engine mount will yield low vibration and noise level at relatively high frequencies but high shake level will be induced at low frequencies.

Although the elastomeric engine mounts have been used in industry successfully, new light cars with powerful engines need more improved mounting systems.

These requirements can be met using improved elastomeric engine mounts or using other types of mounting systems. The elastomeric engine mounts can be improved using specific nonlinear characteristics to obtain nearly constant natural frequency in a broad weight-load range. Materials with high internal damping as well as materials with highly amplitude dependent damping and stiffness can also be used.

There are some life limiting aspects for any elastomeric engine mount. The most important life limiting aspects are drift, creep or set. These are permanent deformations in the rubber material. Drift and creep are caused by shear forces while set is caused by the compressive forces.

The most important advantages of using the elastomeric engine mounts are as follows:

- 1- They have the lowest cost in comparison with other mounting systems
- 2- They are more flexible than hard mounts so they are more efficient in vibration isolation.
- 3- They can be easily designed to have different characteristics
- 4- They have been in use in industry for a long time and their efficiency have been proven.

2.4.3 Wire mesh mounts

In some special situations elastomeric engine mounts can not be used. For example in very high or very low temperature environment. In these situations wire mesh mounts can be used instead of elastomeric engine mounts. They are made of curled formed compressed stainless steel wire (DePriest 2001). They can be used in a wider range of temperature rather than elastomeric engine mounts.

Figure 2.6 Example of wire mesh mount and attachment system(DePriest 2001)

However, wire-mesh mounts are susceptible to permanent deformations such as ‘Set’ or ‘Drift’ and they are usually unidirectional.

Softer engine mounts will provide better vibration isolation; but by softening the elastomeric mounts, new problems such as gross engine displacements over uneven roads and poor mount durability will be aroused. So a compromise between engine isolation and engine bounce is inevitable.

Passive fluid mounts are developed to end the compromise between engine isolation and engine bounce (Patrick E. Corcoran and Gerd-Heinz Ticks 1984).

2.4.4 Passive fluid mounts

In 1962, Richard Rasmussen of General Motors patented a fluid mount in order to increase the damping of the mounting system. The primary reason for using fluid mounts was to increase the damping. Damping coefficient of 10 N.s/m are possible if the peak damping is developed at less than 10 Hz. Providing the peak at higher frequencies would result in a lower damping (Muzechuk 1984).

Figure 2.7 shows the configuration of the early fluid mount patented by R.E. Rasmussen.

Figure 2.7 Early hydraulic engine mount patent R.E.Rasmussen-General Motors June 16, 1964(Muzechuk 1984)

Fluid mounts are the most popular vibration isolator devices. They have been widely used in automotive and aerospace industry.

A typical passive fluid mount consists of two elastomeric chambers that are connected to each other through a tube or a hose called “inertia track”. The two chambers are filled with a fluid which can be water with anti-freezing mixture or ethylene glycol with anti-freezing mixture (Flower 1985; Singh, Kim et al. 1992). In the passive fluid mounts the upper chamber rubber has two major functions, first to carry the static engine load and, secondly, to act as a piston to pump the fluid through the inertia track (Flower 1985). Figure 2.8 shows the configuration of a typical passive 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 coined “notch frequency”.

Figure 2.8 A typical passive fluid mount and its physical model (Vahdati 2005)

The dynamic stiffness of fluid mounts is strongly frequency dependent. Figure 2.9 shows the dynamic stiffness of a typical passive fluid mount versus frequency.

Figure 2.9 Dynamic stiffness of a typical fluid mount Vs frequency (Vahdati and Ahmadian 2003)

Notice 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.

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 (Vahdati 2005). But at the cruise speed, there are multiple engine disturbance inputs of different amplitudes and frequencies. But since a passive fluid mount design has only one notch, the notch can only be tuned to coincide with only one of the disturbance inputs and therefore other disturbance inputs are transmitted to the cabin.

Passive vibration isolation has been widely used in industry because of its inherently stable nature. It is also inexpensive to implement. However in many cases, passive techniques for vibration isolation and disturbance rejection are insufficient, impractical and less flexible or carry a large weight penalty. These disadvantages together with the development of control theory and the availability of high speed computers have made the active\Semi- active vibration isolation a very attractive area for research (Xu 1994).

2.5 Semi-Active vibration isolation

Although passive vibration isolators have been used in industry for a long time and they can provide good vibration isolation in many typical applications, in order to resolve the vibration problems more effectively, semi active vibration isolators can be used.

The basic idea of semi-active vibration isolation is to dissipate the vibration energy of the system by changing the parameters of the engine mount. Since the semi-active vibration isolation is based on damping the energy of the system, the instability problems will be avoided.

There are two main parameters that can be changed to dissipate the vibration energy: elastic stiffness and damping. Damping control is usually preferred because it is easier than changing the stiffness. Actually the idea of semi-active vibration control is based

on the ability to modify the magnitude of transmitted forces through the coupling devices (Graf and Shoureshi 1988).

Significant improvement in ride comfort using semi-active vibration isolators has been reported in the literature. Figure 2.10 shows the transmissibility improvements using semi-active fluid mount (Shoureshi, Graf et al. 1986).

Figure 2.10 Transmissibility improvements with semi-active hydraulic mounts(Shoureshi, Graf et al. 1986)

Different kinds of semi-active vibration isolators have been found in the literature. The effect of changing different parameters of the fluid mounts on the dynamic stiffness has been studied by some researchers. Intelligent materials like Electro-Rheological fluids (ER fluids) and Magneto-Rheological fluids (MR fluids) have been also used in semi-active vibration isolation technology.

Foumani and khajepour have introduced a new high-performance semi-active engine mount (Foumani, Khajepour et al. 2004). They have conducted a sensitivity analysis to find out the most effective parameter for semi-active control. Sensitivity analysis showed that changing the length of the inertia track and the decoupler area has significant effect on the dynamic performance of the fluid mount. These two parameters have been used to tune the mount dynamic properties in this work. Figure

2.11 shows the configuration of the inertia track and the decoupler area of the proposed design.

Figure 2.11 A design for changing both the inertia track and the decoupler area (Foumani, Khajepour et al. 2004)

Figure 2.12 shows the dynamic stiffness of the proposed semi-active fluid mount.

Figure 2.12 Comparison between the adaptive and passive mounts in high frequency (Foumani, Khajepour et al. 2004)

As Figure 2.12 illustrates, the dynamic stiffness of the proposed semi-active fluid mount has been significantly improved over the typical passive fluid mounts, at high frequencies.

Some of the semi-active control schemes are based on stiffness control instead of damping control.

Vahdati and Ahmadian (Vahdati and Ahmadian 2003) have devised a new semi-active fluid mount based on stiffness control. The proposed design is based on varying the lower chamber compliance in order to control the dynamic stiffness characteristics of the fluid mount. Since the bottom chamber compliance is not coupled to any other parameter, it is a good parameter to change. In this paper the lower chamber compliance has been replaced with a piston connected to a spring, a damper and a voice coil. The voice coil is connected to a circuit. By changing the capacitance in the circuit, the lower chamber compliance can be controlled. So, the capacitance can be used to control the compliance of the lower chamber and as a result the notch location.

The configuration of the proposed semi-active scheme for changing the bottom chamber compliance is shown in Figure 2.13.

Figure 2.13 Variable bottom chamber volume stiffness fluid mount design (Vahdati and Ahmadian 2003)

The effectiveness of this new design has been verified via simulation. According to the simulation results it is possible to vary the notch frequency as much as 4 to 5 Hz by changing the capacitance.

M.Lorenz et al. (Lorenz, Heimann et al. 2006) have presented a novel design for a semi-active engine mount based on damping control instead of stiffness control. The proposed design is based upon using dry friction to control the damping of the system. The advantage of using dry friction damper instead of hydraulic damper is that dry friction damping is limited to a maximum value which means that a friction damper does not allow the motion to be blocked. Additionally, by separating the two pieces which create friction, the damping can be cancelled wherever it is not needed. An on-off switch has been used in this work to turn the damper on and off. For example under good road condition the engine mount acts as a soft spring and its performance will be better than a classical fluid mount. The controller consists of an electromagnetic actuator which controls the normal force which acts on the contact surface and as a result controls the damping of the mount. Experimental results have corroborated the superior performance of the proposed engine mount over the typical passive fluid mounts.

The original mount is a passive fluid mount which consists of a hydraulic damping mechanism. The hydraulic damper has been replaced by the dry-friction mechanism while other parts of the original mount have been kept.

However, using dry-friction damping, the geometry of the damper will change over time because of wear. So after a while the mechanical properties of the system will be changed. In order to resolve this problem the inductance of the coil in the actuator has been used as a sensor which shows the amount of wear. Then this feedback data is used to control the applied force and as a result the dynamic characteristics of the system. In general, they have used a very simple control policy. They activate the damper whenever the engine oscillations reach a predetermined value.

The configuration of the proposed dry-friction mount is shown in Figure 2.14.

Figure 2.14 Schematic of the electromagnetic actuator (left) and photo of the prototype friction damper (right)(Lorenz, Heimann et al. 2006).

Using this design good vibration isolation would be achieved. But the problem is that after a period of operation the friction pad needs to be changed because of wear.

ER and MR fluids can also be implemented in mounting applications for damping control. Their fast response to changes in the electrical field and magnetic field made them very useful. Nearly most of the semi-active engine mounts that have been found in the literature are of hydraulic type. Some of them use ER and MR fluids and some use conventional fluids.

Hong et al. (Hong and Choi 2005) have devised a mixed-type magneto-rheological semi-active fluid mount. They have considered the non dimensional formulation of Bingham plastic flow and applied it to vibration control of structural system subjected to external excitation. The structural system which has been used in their work consists of a vibrating mass, a semi-active MR fluid mount and a passive rubber mount. The MR mount is installed on the beam and acts as a semi-active actuator. The beam itself seats on two elastomeric mounts (see Figure 2.15).

Figure 2.15 Structural system incorporated with MR fluid mount(Hong and Choi 2005)

They have used the linear quadratic Gaussian (LQG) controller to control the MR fluid properties and therefore to control the dynamic properties of the engine mount. They have done some experiments to evaluate the damping force of the MR mount with respect to field intensity. Figure 2.16 shows the configuration of the proposed MR mount.

Figure 2.16 Configuration of MR fluid mount(Hong and Choi 2005)

An electromagnetic exciter is used in order to excite the MR mount. Two accelerometers are attached to the mass and the beam to get feedback signals. Velocity

signals have been obtained using integrator circuit of the charge amplifier. A force transducer is also installed between the rubber mount and the base plate in order to measure the transmitted force. The amplitude of excitation force is set as 1N. The frequency range is between 5-80 Hz. The experimental results showed that the acceleration level of 1st, 2nd and 3rd modes are mitigated effectively by controlling of the MR fluid damping.

Beside semi-active vibration isolation technology, active vibration isolation techniques have been also studied by many researchers. Developments in the field of the intelligent materials such as piezo-electric materials, ER and MR fluids along with the improved active control strategy techniques have made the active vibration isolation technology, a very attractive area for researchers.

2.6 Active vibration isolation

Active vibration isolation technology is based on using active energy sources in order to minimize the transmitted force to the occupant's compartment. An active engine mounts usually consists of a passive mount, a force generating actuator, a structural vibration sensor and an electronic controller. The function of the passive part is to make the system safe in the case of actuator failure. The actuators should provide dynamic forces which are reasonable in magnitude and fast in response to cancel unwanted vibration. These actuators create forces that directly oppose imbalance forces. The sensors provide control signal for the actuators. They are usually force or acceleration sensors. Figure 2.17 shows the simplified mechanical model of a typical active engine mount.

Figure 2.17 Simplified mechanical model of a typical active engine mount (DePriest 2001)

Active engine mounts can provide zero dynamic stiffness at desired frequencies while having high static stiffness. Active engine mounts usually use a full feedback control policy. Different types of controllers can be used in active vibration isolation. They can use both feedback and feed forward controllers.

Different kinds of actuators such as piezo-electric actuators, MR and ER fluids have been used in active vibration isolation technology. ER and MR fluids have been introduced in sections 2.3.1 and 2.3.2, respectively.

Piezo-electric actuators have been widely used in active vibration isolation, especially for high frequency applications.

ER Semi-active engine mounts which change rheological characteristics according to the applied electrical field have a very fast response and low energy consumption. They are effective in resolving noise and vibration problems in low frequency range. On the other hand, active engine mounts which make use of piezo-actuators provide good vibration isolation in the high-frequency low amplitude range, because the displacement induced in the piezo-ceramic materials is small. In order to resolve this problem, Choi et al (Choi, Choi et al. 1996) have used ER fluid and piezo-actuators simultaneously in an active fluid mount design. This type of engine mount is "hybrid engine mount". The proposed engine mount has high performance over all frequency ranges. Figure 2.18 shows the configuration of the proposed engine mount.

Figure 2.18 Configuration of the proposed engine mount (Choi, Choi et al. 1996)

As Figure 2.18 illustrates, the upper and lower chambers are connected to each other via some electrodes. Both chambers are filled with ER fluid. The upper chamber is bounded by a metal-rubber frame. The lower chamber is separated from the air by a diaphragm. The piezo-actuator is installed below the steel case and provides the engine mount with active force.

Different kinds of controllers have been used to control the ER fluid actuator and piezo-electric actuator in this work. The Neuro-controller has been used for the ER fluid and the H_∞ controller is used for the piezo-actuator. Figures 2-19, 2-20, 2-21 and 2-22 illustrates the advantage of using the hybrid mount, in terms of force transmissibility.

Figure 2.19 Transmissibility using the ER fluid only (Choi, Choi et al. 1996)

Figure 2.20 Transmissibility using Piezoactuator (Choi, Choi et al. 1996)

Figure 2.21 Robust stability by μ (Choi, Choi et al. 1996)

Figure 2.22 Transmissibility using both the ER fluid and the piezoactuator (Choi, Choi et al. 1996)

As Figure 2.22 illustrates, using the hybrid engine mount has effectively reduced the transmitted vibration.

One of the problems which are associated with the typical elastomeric engine mounts and nearly most of the passive or semi-active engine mounts is their limited effective frequency range in comparison with the broad engine frequency range.

Olsson (Olsson 2006) have presented an active vibration isolation scheme which makes use of the large frequency band feedback. Figure 2.23 shows the schematic of the principle of operation of the active engine mount.

Figure 2.23 Schematic picture of the principle of operation of the active engine mount (Olsson 2006)

Feed forward control strategy is capable of removing the disturbance effects very effectively if the gain of the controller and error path combined matches the forward path gain exactly and the combination of controller and error path is exactly 180° out of phase with each other. However feed forward control scheme has its own shortcomings. For example if the stated parameters are not exact, the cancellation will not be perfect and in severe mismatch cases the feed forward controller actually worsen the response of the system. In this work a new feed forward scheme based on using Kalman estimators has been proposed. The proposed scheme has resolved the aforementioned shortcomings of the feed forward control schemes. They have found their method to be significantly effective in isolating the disturbances (See Figure 2.24).

Figure 2.24 Acceleration at four different locations on the chassis (Olsson 2006)

According to the results presented in this section, it can be concluded that active vibration isolators, can effectively reduce the transmitted vibration.

Although active engine mounts can provide very good vibration isolation for many applications, but it should be noted that, active vibration isolation technology would add more cost and complexity to the system. Since for many typical applications it is neither practical nor economic to use active vibration isolators, passive and semi-active engine mounts have been used widely.

The focus of this project is on passive and semi-active engine mounts. Since fluid mounts are used in industry for years, they have been chosen as a base to develop new passive and semi-active engine mounts in this project.

Passive fluid mounts have been briefly introduced in section 2.2.4. A more comprehensive study of different types of fluid mounts will be represented.

2.7 Comprehensive study of fluid mounts

The main focus of this project is to design the electro-mechanical equivalent of a typical passive fluid mount. So, fluid mounts are studied more extensively in this section.

Fluid mounts are more tunable than other types of passive engine mounts (Yu, Naganathan et al. 2001). Significant improvements in cabin noise and vibration isolation, using passive fluid mounts have been reported in the literature. Properly tuned fluid mounts not only provide cabin noise and vibration isolation, but also they prevent the engine from bouncing too much, near idle. This is a good point regarding to the stresses applied to the engine parts (Patrick E. Corcoran and Gerd-Heinz Ticks 1984).

In general, passive fluid mounts have been categorized into two main groups; namely, Single-pumper passive fluid mounts and Double-pumper passive fluid mounts.

In single pumper fluid mounts (see Figure 2.8) only one of the chambers pumps the fluid while in double pumper fluid mounts both chambers pump or move the fluid (Vahdati 1998).

Figure 2.25 shows the configuration of a passive double pumper fluid mount.

Figure 2.25 Configuration of a passive double-pumper fluid mount (Vahdati 1998).

Figure 2.26 shows a simplified model for Single and double pumper mounts.

Figure 2.26 A simplified model of a single and a double pumper fluid mount(Vahdati 1998)

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 less probable in double-pumper fluid mounts. In a double-pumper engine 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 (Vahdati 1998). Single pumper fluid mounts can not be easily pressurized due to low stiffness of the 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 easily pressurized.

Vahdati (Vahdati 1998) has introduced the following two complex stiffnesses for fluid mounts:

$$K_t^* = \frac{F_t}{X} \quad K_{in}^* = \frac{F_{in}}{X}$$

Usually the transmitted complex stiffness K_t^* is used for fluid mounts but it has been shown that for most of the applications K_t^* and K_{in}^* are nearly the same.

As Figure 2.9 illustrates, the dynamic stiffness of fluid mounts is strongly frequency dependent. Dynamic stiffness of fluid mounts is amplitude dependent as well (See Figure 2.27). A highly elastic support spring will reduce the amplitude dependency of the dynamic characteristics of the fluid mount (Patrick E. Corcoran and Ticks 1984).

Figure 2.27 Dynamic stiffness of a simple fluid mount (Patrick E. Corcoran and Ticks 1984)

As Figure 2.27 illustrates, the dynamic stiffness of fluid mounts is slightly amplitude dependant.

Another categorization of fluid mounts is based on the way in which the upper and lower chambers are connected to each other. They can be connected via an external bleed or an internal bleed as shown in Figure 2.28 (Shoureshi and Graf 1986).

Figure 2.28 Hydraulic mounts with: a) External Bleed b) Internal Bleed (Shoureshi and Graf 1986)

Fluid mounts with internal bleed are categorized into the following groups: fluid mounts with simple orifice, fluid mounts with inertia track and fluid mount with inertia track and decoupler.

Fluid mounts with inertia track provide high damping and as therefore effective vibration suppression at low frequencies. They provide the best vibration isolation at the notch frequency; but as Figure 2.9 shows, the dynamic stiffness of fluid mounts increases dramatically after the notch frequency. So the fluid mounts provide good vibration isolation at low frequencies while they show poor performance (even worse than elastomeric mounts) at high frequencies. In order to resolve this problem an additional device called “Decoupler” has been introduced to fluid mounts. In fact, using decoupler makes the dynamic characteristics of the fluid mount more amplitude dependent. The decoupler is simply a disk made of rubber or metal which floats in its cage and acts as a mechanical switch. It allows the fluid to move relatively freely between the two chambers or prevent it from flowing freely. The effect of the decoupler specifications on the dynamic stiffness diagram has been shown in Figure 2.29.

Figure 2.29 Dynamic stiffness of hydraulic mount with decoupler (Patrick E. Corcoran and Ticks 1984)

As Figure 2.29 illustrates, using the decoupler has improved the dynamic stiffness of the fluid mount at high frequencies.

Fluid vaporization is one of the problems associated with fluid mounts. One of the remedies to avoid fluid vaporization is to pressurize the mount. An initially pressurized fluid mount is shown in Figure 2.30. This design also has the advantage of providing a constant installed height independent of load (Marjoram 1985). Furthermore, it provides a means for tuning the mount without need to any hardware change. Figure 2.30 shows the configuration of the proposed engine mount.

Figure 2.30 Pressurized hydraulic mount with inertia track and amplitude decoupler (Marjoram 1985)

To apply the initial pressure to the fluid mount, an additional chamber called "air chamber" is used. The effect of this initial pressure comes into the dynamic characteristics of the mount via the bottom compliance term. However this kind of fluid mount needs the additional sealing for the air chamber and it would be larger in size.

Careful design of this kind of engine mount would result in load variation insensitive dynamic characteristics (see Figure 2.31).

Figure 2.31 Dynamic stiffness and loss angle for new mount showing the effect of load variation (Marjoram 1985)

Bernuchon (Bernuchon 1984) has introduced a new type of engine mounts and compared its dynamic characteristics with a rubber mount and a typical fluid mount. Figure 2.32 represents the configuration of the proposed fluid mount.

Figure 2.32 Basic lay-out of a hydraulic mount (Bernuchon 1984)

This type of fluid mount is like typical fluid mounts except having an additional membrane (marked by 8 in Figure 2.32). This membrane together with the separator plate (marked by 9) gives interesting characteristics in the acoustic frequencies range. The efficiency of this mount has been examined via some experiments. Figure 2.33 shows the dynamic stiffness of three different types of mounts. In Figure 2.33, (1) represents the rubber mount, (2) represents the typical fluid mount and (3) stands for the new type fluid mount.

Figure 2.33 Influence of construction parameters-low frequency dynamic rate (Bernuchon 1984)

It can be seen from this figure that the rubber mount shows high stiffening with frequency. Typical fluid mount also shows high stiffening. It is because no particular arrangement has been taken to improve the characteristics in the acoustic range. The new type fluid mount shows significantly improved dynamic characteristics at high frequency due to the more sophisticated components used for the connections between chambers.

In order to choose a hydraulic mount for a specific application the following parameters must be known or measured (Patrick E. Corcoran and Gerd-Heinz Ticks 1984):

- 1- Static and dynamic loads which will be applied to the mount.
- 2- Dynamic properties of existing mounts in vertical and shear directions.
- 3- Engine resonance frequency.
- 4- Position of the roll axis relative to the mount position.
- 5- Engine side amplitude at idle conditions.

The design of the fluid mount should be compatible with the automotive assembly. The concerned issues in the fluid mount design and manufacturing phases are:

- 1- component fits and assembly sequence
- 2- Sealing technique
- 3- Securing the assembly
- 4- Ease of molding
- 5- Fluid fill procedure

The patented "tri-function" diaphragm has been introduced by Muzechuk (Muzechuk 1984). This method provides a design which satisfies the manufacturing concerns (see Figure 2.34).

Figure 2.34 Assembly and sealing(Muzechuk 1984)

Different kinds of fluid mounts have been introduced in this section. In order to analyze the dynamic characteristics of fluid mounts, different methods have been used in the literature.

2.7.1.1 Fluid mount analysis techniques

In order to analyze a typical fluid mount, the comprehensive mechanical model of the mount is needed. Different modeling techniques for modeling the fluid mounts have been used in the literature. Lumped parameter modeling, finite element analysis and bond graph modeling technique are the most common modeling techniques for fluid mounts.

Singh et al (Singh, Kim et al. 1992) have conducted a linear analysis for fluid mounts with decoupler. They have used the lumped parameter modeling technique to investigate the dynamic characteristics of the fluid mount. The physical model of the passive fluid mount analyzed in their study is shown in Figure 2.35.

Figure 2.35 Hydro-mechanical mount example case: a) Physical system with a free decoupler and lumping scheme; b) Physical system with a fixed decoupler (Singh, Kim et al. 1992).

The fluid inside the chambers is water and anti-freezing mixture. In some fluid mounts ethylene glycol is used to fill the chambers. One of the critical factors is the filling of the fluid. Experimental results showed that very small under fill conditions produced large adverse effects on the mount dynamic properties (Muzechuk 1984).

Singh et al (Singh, Kim et al. 1992) have considered the fluid mount linear in the first step. The Linear Time Invariant (LTI) model of the mount is used to model the mount over the range of 1-250 Hz.

Figure 2.36 illustrates the lumped parameter model of the fluid mount.

Figure 2.36 Mathematical model schematic with a free decoupler (Muzechuk 1984)

The proposed lumped parameter model includes six control volumes and three lumped masses. Using Voigt model and applying the continuity arguments to the compliant volume chambers and doing some simplifications, the following equation has been derived:

$$\bar{F} = F'(t) - F(t) = k_r \bar{x} + A_r (\bar{p}_{11} - p_{atm}), \quad \bar{p}_{11} = \bar{p}_{12} = \bar{p}_{21} = \bar{p}_{22} = \bar{p}$$

$$\bar{q}_i = \bar{q}_d = 0,$$

$$A_r \bar{x} = A_r [x'(t) - x(t)] = [C_{11} + C_{12} + C_{21} + C_{22}] \bar{p},$$

$$A_r \bar{x}_1 = [C_{12} + C_{21} + C_{22}] \bar{p}, \quad A_r \bar{x}_2 = C_{22} \bar{p}$$

Since the dynamic characteristics of the fluid mounts is strongly frequency dependent, Singh et al (Singh, Kim et al. 1992) have separated their study into two different parts: Low frequency analysis and high frequency analysis.

In low frequency range (below 120 Hz), the impedances of the lumped fluid masses are negligible so the model will be reduced to a two degree of freedom model. By finding the appropriate equations and transferring them to the Laplace domain it has been shown that the fluid mount is a feedback system and "the fluid part acts as a passive controller" (Singh, Kim et al. 1992). They have used the Routh stability criterion to show that the system is absolutely stable. The Nyquist analysis and root locus methods also have confirmed the stability of the system.

In high frequency range (over 120 Hz), since the fluid inertia force is transmitted to the fixed frame, the dynamic stiffness of the mount increases. So the hydraulic mounts show poor performance at high frequency range (even worse than the rubber mounts).

The experimental analysis showed that the simulation results are in good agreement with the experimental results at low frequency range. Figure 2.37 shows the experimental setup used to examine the fluid mount dynamic characteristics.

Figure 2.37 Schematic of the experiment set-up (Singh, Kim et al. 1992).

Since the proposed LTI model by Singh et al (Singh, Kim et al. 1992) is not valid over high frequency range and for non-linear cases, Kim and Singh (Kim and Singh 1995) have published another paper which explains the non-linear properties of engine mounts. The high frequency and non-linear dynamic characteristics of fluid mounts have been studied in this work, both experimentally and analytically. A new lumped parameter model for fluid mounts based on simulating the decoupler and the inertia track dynamics has been proposed. The quasi-linear analysis method has been used for high frequency analysis. Figure 2.38 shows the proposed lumped parameter model for the fluid mount.

Figure 2.38 The lumped parameter model of the regular hydraulic mount (Kim and Singh 1995)

In this model the fluid impedances for the upper and lower chamber inertance are neglected. The authors of this paper have conducted some experiments. They have measured and compared the impulse response of rubber mount, fluid mount with decoupler and inertia track and fluid mount with inertia track.

Figure 2.39 The simulated impulse response of the vehicle model for $\dot{x}_e(0) = 0.1 \text{ m/s}$. a) ----, Rubber mount; hydraulic mount ($\Delta_d = 0.7 \text{ mm}$). b) Inertia track mount. (Kim and Singh 1995)

It has been concluded that although the response of a fluid mount shows a higher first peak than that of the rubber mount but its response goes down to zero faster. So, fluid mount is more effective than rubber mount. The fluid mount with inertia track is even more effective than a fluid mount with decoupler and inertia track in shock absorption. They have proposed a new adaptive engine mount which is designed based on the results of the experiment. The proposed engine mount consists of two parts: the mechanical actuation system and an electronic controller. It can work as either "hard" or "soft" state.

Figure 2.40 The operation and arrangement of the proposed adaptive hydraulic mount (Kim and Singh 1995)

The black box diagram of this adaptive engine mount is shown in Figure 2.41.

Figure 2.41 The block diagram for the fluid system of the adaptive mount (Kim and Singh 1995)

The adaptive engine mount yields high damping at low frequencies while it exhibits sufficiently low dynamic stiffness at medium frequencies. It acts as a soft spring at high frequencies.

Beside lumped parameter models which have been used in many researches to derive the equations of motion of fluid mounts, finite element analysis method has been also used to analyze the dynamic characteristics of fluid mounts.

The dynamic behavior of two types of fluid mounts have been investigated by Christopherson and Jazar (Christopherson and Jazar 2006). They have analyzed and compared the dynamic performance of two different designs of passive fluid mounts; namely, the floating-decoupler fluid mount and the direct-decoupler fluid mount (see Figures 2.42, 2.43, 2.44 and 2.45) using finite element analysis method.

Figure 2.42 Direct decoupler fluid mount (Christopherson and Jazar 2006)

Figure 2.43 Floating decoupler fluid mount (Christopherson and Jazar 2006)

Figure 2.44 Illustration of a floating-decoupler-type fluid mount (Christopherson and Jazar 2006)

Figure 2.45 Illustration of a direct-decoupler-type fluid mount (Christopherson and Jazar 2006)

Christopherson and Jazar (Christopherson and Jazar 2006) have employed the finite element modeling technique for parameter identification of the two fluid mount designs.

Finite element modeling technique can effectively describe the behavior of the rubber components of the fluid mount. The constitutive equations used in finite element modeling are of crucial importance. The three term Mooney-Rivlin model have been used in this work to describe the rubber behavior.

Since the fluid mount consists of multiple materials in contact with one another, the importance of proper element selection is evident.

The major part of the fluid mount volume is made from nearly incompressible rubber; so using the hyperelastic constitutive models is necessary. Since the metallic parts are not expected to experience excessive deformations, the element selection for these parts is not of great concern. In this work a quadratic element has been used for the metallic parts to ensure the mesh continuity with the hyperelastic domains. In order to obtain a higher quality mesh, shell elements have been used to describe thin rubber elements intended to exhibit large deformations.

The finite element mesh for the upper and lower chambers of a direct decoupler fluid mount has been shown in Figures 2.46 and 2.47.

Figure 2.46 Upper chamber finite element mesh (direct decoupler mount) (Christopherson and Jazar 2006).

Figure 2.47 Lower chamber finite element mesh (direct decoupler mount) (Christopherson and Jazar 2006).

After mesh selection the boundary conditions must be applied. The boundary conditions applied to the upper and lower chambers are illustrated in Figures 2.48 and 2.49, respectively.

Figure 2.48 Upper chamber boundary conditions (direct decoupler mount) (Christopherson and Jazar 2006).

Figure 2.49 Lower chamber boundary conditions (direct decoupler mount) (Christopherson and Jazar 2006).

The finite element analysis has been used for fluid mount parameter identification in this work. The load-deflection relationship for the upper chamber structure of the direct-decoupler mount has been used to verify the finite element model presented.

Figure 2.50 Force-deflection relationship (direct-decoupler mount) (Christopherson and Jazar 2006).

Since the proposed finite element model is sufficiently accurate for this study (see Figure 2.50), it can be used to identify the fluid mount parameters. For example, Figure 2.51 shows the upper chamber volume-pressure relationship. The upper chamber volumetric stiffness can be found using this plot.

Figure 2.51 Upper chamber volume-pressure relationship (direct decoupler mount) (Christopherson and Jazar 2006).

Using the proposed finite element model, all fluid mount parameters can be identified. The identified parameters can be used to simulate the fluid mount dynamic stiffness.

Figure 2.52 Dynamic stiffness comparison plot (Christopherson and Jazar 2006).

Simulation results showed that the direct-decoupler fluid mount exhibits the lowest transmissibility in low-frequency domains whereas the floating decoupler fluid mount behaves better as excitation frequencies increase.

Shangguan and Lu (Shangguan and Lu 2004) have presented a fully coupled fluid-structure interaction (FSI) analysis for fluid mounts. They have used finite element modeling technique to model the fluid mount. The required parameters for the simulation of fluid mounts are stress versus strain relations for rubber material and the fluid physical parameters and the geometry of the mount. Using this method greatly reduces the design time and cost. The authors have conducted some experiments to verify their FEA results. However, they have focused on simulating the behavior of the fluid mount at low frequency (1-50 Hz) and large amplitude (1 mm) excitations. A comparison between performances of 3 typical mounts is presented in this work. These three types of mounts are: a fluid mount with floating decoupler, a fluid mount with fixed decoupler and a rubber mount which is a typical fluid mount whose fluid has been drained out.

The lower chamber is made of thin rubber. So the pressure in the lower chamber is nearly equal to atmosphere pressure. Shangguan and Lu (Shangguan and Lu 2004) have conducted a pressure analysis. They have assumed that the pressure in the lower

chamber is equal to atmosphere pressure and the pressure in the upper chamber is equally distributed in the whole chamber. Their results are shown in Figure 2.53.

Figure 2.53 Pressure Vs time for the upper chamber (Shangguan and Lu 2004)

They have also found the temperature history of the fluid in the chambers for periodic excitation ($x(t) = 3 \sin(30\pi t)$ mm).

Figure 2.54 Fluid temperature Vs time (Shangguan and Lu 2004)

Figure 2.55 shows the load-displacement relationship for static loading of a typical fluid mount and its rubber in three orthogonal directions.

Figure 2.55 Force Vs displacement for a RS and a hydraulic engine mount (Shangguan and Lu 2004)

It can be concluded from this figure that the static performance of a fluid mount is dominated by the characteristics of its rubber.

Finite element analysis is found to be an accurate and highly effective method in parameter identification to aid in the design procedure of fluid mounts; but due to the inherent complexity of fluid-structure interaction in fluid mounts, simulating their dynamic behavior using finite element modeling technique will be time consuming.

Fluid mounts consist of two different energy domains; namely, the mechanical energy domain and the hydraulic energy domain.

Since Bond-Graph modeling technique (Karnopp, Margolis et al. 1990) is a powerful tool to model multi-energy-domain systems, this technique has been widely used to model the typical fluid mounts. It is worth mentioning that finite element models can still be used for parameter identification purposes. Then these parameters can be used in the bond graph model to derive the dynamic behavior of the fluid mount.

The bond graph model of a fluid mount with simple orifice and a fluid mount with inertia track have been presented by Flower (Flower 1985). He has used the bond graph model of the fluid mount to investigate its dynamic characteristics. The physical models and bond graph models of the fluid mount with inertia track and fluid mount with simple orifice are shown in Figures 2.56 and 2.57, respectively.

Figure 2.56 Physical model of a fluid mount with inertia track and its bond graph model (Flower 1985).

Figure 2.57 Physical model of a fluid mount with simple orifice and its bond graph model (Flower 1985).

Bond graph modeling technique makes use of a unique notation for different energy domains. So this technique can describe the dynamic behavior of multi energy domain systems, such as fluid mounts, very effectively.

The bond graph model of the fluid mount can be used to derive its state space equations directly. Some standard rules for deriving the state space equations from the bond graph model have been introduced by Karnopp et al (Karnopp, Margolis et al. 1990). There are also some commercial softwares such as CAMPG and 20-sim which can be used to derive the state space equations of the system from the bond graph model.

Flower (Flower 1985) have used the bond graph modeling technique to investigate the dynamic behavior of the three types of engine mounts. Simulation results presented in this work have shown that the fluid mount with inertia track provides better vibration isolation than the rubber mount and the fluid mount with simple orifice.

Vahdati and ahmadian (Vahdati and Ahmadian 2003) have presented the bond graph model of a passive fluid mount. The state space equations of the fluid mount have been derived.

Figure 2.58 Typical single-pumper fluid mount, its physical model and its bond graph model (Vahdati and Ahmadian 2003).

The objective of their work was to convert the passive fluid mount to a semi-active fluid mount by varying one of the fluid mount parameters. It has been shown that the easiest parameter to vary is the lower chamber compliance. A new design for the semi-active fluid mount has been presented. The configuration of the proposed

variable lower chamber stiffness fluid mount and its bond graph model are shown in Figures 2.59 and 2.60, respectively.

Figure 2.59 Variable lower chamber stiffness fluid mount design (Vahdati and Ahmadian 2003).

Figure 2.60 Bond graph model of the variable lower chamber stiffness fluid mount (Vahdati and Ahmadian 2003).

AS Figure 2.59 illustrates, the lower chamber rubber of the typical passive fluid mount is replaced with a piston supported by a spring and a damper. A fabric diaphragm (with high volumetric stiffness) is placed on the top of the piston, sealing the bottom fluid mount chamber. It has been assumed that it is possible to vary the lower chamber stiffness K . The state space equations of motion have been derived from the bond

graph model. The dynamic stiffness of the fluid mount for different values of K have been simulated (see Figure 2.61).

Figure 2.61 Dynamic stiffness of the variable lower chamber stiffness fluid mount as K varies (Vahdati and Ahmadian 2003).

Bond graph modeling technique can also describe the dynamic behavior of active engine mounts. For example the dynamic behavior of the hybrid engine mount presented in Figure 2.18 has been simulated using bond graph modeling technique. Figure 2.62 shows the bond graph model of the hybrid engine mount shown in Figure 2.18.

Figure 2.62 Bond graph model of the hybrid engine mount (Choi, Choi et al. 1996).

The bond graph model shown in Figure 2.62 has been used to derive the state space equations of motion of the hybrid engine mount and simulating its dynamic behavior.

According to the capability of the bond graph modeling technique in modeling multi energy domain systems, this method has been employed in this work to model the fluid mount.

2.8 Objectives

As mentioned before, fluid mounts have been used in industry for years, and they can provide good vibration isolation. But, beside the advantages of using fluid mounts there are some disadvantages associated with them. The most important disadvantages of fluid mounts are:

- 1- The design procedure of fluid mounts is based on trial and error which is a time consuming and expensive method.
- 2- As time goes on, the dynamic characteristics of the engine as well as the fluid mount properties will change; so the mount needs to be retuned after a period of operation. The retuning procedure of fluid mounts is costly because it needs dispatching the mount and redesigning its internal parts.
- 3- Fluid mounts perform similar to the ideal engine mount, up to the notch frequency; but, beyond the notch frequency, the dynamic stiffness of the fluid mounts increases dramatically. So, at high frequencies, fluid mounts perform even worse than the elastomeric engine mounts.

The main objective of this project is to develop a new engine mount design to reach the ideal engine mount dynamic characteristics while, resolving the aforementioned shortcomings.

Two ways to reach the ideal engine mount dynamic stiffness have been undertaken in this project:

- 1- Creating multiple notches
- 2- Creating a variable notch

Fluid mounts provide the best vibration isolation at the notch frequency. By comparing the dynamic stiffness of a fluid mount with an ideal engine mount, it can be concluded that at the notch frequency the dynamic stiffness of the fluid mount is very close to that of the ideal engine mount. So, by creating multiple notches, the dynamic stiffness of the fluid mount is similar to the ideal engine mount dynamic stiffness, at more points (see Figure 2.63).

Figure 2.63 The dynamic stiffness of a multiple notch engine mount

A portion of this thesis has been devoted on developing a multiple notch fluid mount design. Although creating multiple notches would improve the dynamic characteristics of the fluid mount, the other shortcomings have not been resolved, yet.

As mentioned before, one of the shortcomings of fluid mounts is that their design procedure is based on trial and error. Since some of the fluid mount key parameters are coupled to one another, the design procedure of fluid mounts is complicated. For

example, both of the vertical stiffness and volumetric stiffness of the upper chamber depend on the upper chamber rubber geometry and material properties; so any change in the upper chamber rubber properties would change the vertical stiffness and the volumetric stiffness of the upper chamber, simultaneously. So it is highly desirable to develop a design for engine mounts to decouple the key parameters.

Additionally, since fluid losses through the inertia track and fluid structure interaction between the fluid inside the mount and the rubber elements play an important role in making the fluid mount design complicated, it makes all the sense to remove the fluid and replace it with a more controllable mechanism.

The main focus of this project is to design a fluidless electronic vibration isolator to satisfy the aforementioned criteria, while providing good vibration isolation.

A comprehensive literature survey was conducted to find out if there is any fluidless electronic vibration isolator, but none were found.

Chapter3 Application of Tuned Vibration Absorber (TVA) in fluid mounts.

3.1 Introduction

In chapter 2, different types of fluid mounts were introduced and their dynamic properties were investigated.

It was concluded in chapter 2 that more notch frequencies are needed to provide the best cabin noise & vibration isolation.

For example in the case of the aircraft engine mounts, if two fluid mount notch frequencies could exist, one of them can be tuned to coincide with the N1 frequency (engine low speed shaft imbalance frequency) and the other to coincide with N2 frequency (engine high speed shaft imbalance frequency) and the vibration isolation would be more effective.

In this chapter, a new fluid mount design with two notch frequencies will be proposed. The mathematical model and simulation results of the proposed design will be presented.

A Literature survey was conducted to find out if there is any double or multiple notch fluid mount design documented. Vahdati (Vahdati 2005) has introduced a design for a double notch fluid mount as shown in Figure 3.1.

With the design of Figure 3.1, 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 3.1, re-tuning of the notch frequencies requires re-machining of the inertia tracks, or change in rubber modulus, which is very costly.

Figure 3.1 Physical model of the double notch fluid mount (Vahdati 2005)

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 (Shangguan and Lu 2004; Ahn, Kim et al. 2005); but most of these methods normally involve experimental data for optimization which is again not desirable.

Here in this chapter, a new single pumper fluid mount design is proposed which can provide two notches and the new notch is easily tunable and can be placed at any desired frequency without any major redesign of the fluid mount.

Tuned Vibration Absorber (TVA) plus a fluid mount is used to create two fluid mount notches. The bond graph model and simulation results of the new fluid mount design are presented. The effect of changing the parameters of TVA on the location and depth of the notches are studied as well.

3.2 The New Fluid Mount Design

Vahdati and Ahmadian (Vahdati and Ahmadian 2004), have introduced a new fluid mount design concept where the lower chamber soft rubber diaphragm of Figure 2.8 was replaced by a piston, a spring, and a damper. The piston/spring system acts like a fluid accumulator. Here in this chapter, the same design is used but a TVA is added to the piston as shown in Figure 3.2. The piston will be covered with a diaphragm (not shown) to minimize fluid leakage from the bottom fluid chamber to the air chamber. The fluid mount design shown in Figure 3.2, without the TVA, is alike the typical single pumper fluid mount except that the bottom chamber rubber diaphragm is replaced by a piston, a diaphragm covering the piston (not shown), and a linear spring. The linear spring can be a metal spring or a rubber spring. In either case, the metal or the rubber spring will have some hysteresis damping, so here in this chapter, it is assumed that the spring acts like a spring and a viscous damper.

3.2.1 Physical model of the fluid mount with the TVA

In order to represent the physical model of the new fluid mount design, it is needed to deal with two different energy domains: namely mechanical and hydraulic energy domains. As mentioned in chapter 2, bond graph modeling technique enables us to deal with multi energy domains very efficiently. This method has been employed to model the fluid mount plus the TVA design.

Figure 3.2 New Fluid Mount Design

Finding the physical model of Figure 3.2 is the first step to develop its graph model. The physical model should be simple enough to be solvable and also should accurately retain the main characteristics of the real system. The simplified physical model of the fluid mount with the TVA of Figure 3.2 is represented in Figure 3.3.

Figure 3.3 Physical model of a fluid mount plus a TVA

The bond graph model of Figure 3.3 is shown in Figure 3.4.

Figure 3.4 Bond graph model of Figure 3.3.

State space equations derived from the bond graph model of Figure 3.3 are as follows:

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

$$\dot{q}_6 = A_{pt} V_{in} - \frac{p_9}{I_o}$$

$$\dot{p}_9 = \frac{q_6}{C_t} - R_f \frac{p_9}{I_f} - \frac{q_{11}}{C_b}$$

$$\dot{q}_{11} = \frac{p_9}{I_f} - A_{pb} \frac{p_{14}}{M_{pb1}}$$

$$\dot{p}_{14} = A_{pb} \frac{q_{11}}{C_b} - K_{b1} q_{15} - B_{b1} \frac{p_{14}}{M_{pb1}} - \left[B_{b2} \left[\frac{p_{14}}{M_{pb1}} - \frac{p_{21}}{M_{pb2}} \right] + K_{b2} q_{20} \right]$$

$$\dot{q}_{15} = \frac{p_{14}}{M_{pb1}}$$

$$\dot{q}_{20} = \frac{p_{14}}{M_{pb1}} - \frac{p_{21}}{M_{pb2}}$$

$$\dot{p}_{21} = B_{b2} \left[\frac{p_{14}}{M_{pb1}} - \frac{p_{21}}{M_{pb2}} \right] + K_{b2} q_{20} \quad (3-1a-h)$$

In the above state space equations, q_2 , q_6 , q_{11} , q_{15} , and q_{20} , are the generalized displacement variables, and p_9 , p_{14} , and p_{21} are the momentum variables. The states are defined as follows:

q_2	Relative displacement across spring K_r
q_6	Volume change in the top fluid chamber
q_{11}	Volume change in the bottom fluid chamber
q_{15}	Relative displacement across spring K_{b1}
q_{20}	Relative displacement across the TVA
p_9	Inertia track fluid momentum
p_{14}	Bottom fluid chamber piston momentum (momentum of mass M_{b1})
p_{21}	TVA mass momentum (momentum of mass M_{b2})

The transmitted force, the effort at bond 1, is equal to:

$$F_{in} = K_r q_2 + B_r V_{in} + A_{pt} \frac{q_6}{C_t} \quad (3-2)$$

Dynamic stiffness (F_{in}/X_{in}) of this system can be obtained by the use of Laplace transformation and is shown below:

$$K_{dyn} = K_r + B_r S + \frac{A_{pt}^2}{C_t} \frac{1}{\left[1 + \frac{1}{S^2 C_t I_f \left[1 + \frac{R_f}{S I_f} + \frac{1}{S I_f \left[1 + \frac{A_{pb}^2}{S^2 M_{pb1} C_b \left[1 + \frac{K_{b1}}{S^2 M_{pb1}} + \frac{B_{b1}}{S M_{pb1}} + A \right]} \right]} \right]} \right]} \quad (3-3a)$$

Where the parameter A is equal to:

$$A = \frac{B_{b2} \left[\frac{B_{b2}}{SM_{pb1}} + \frac{K_{b2}}{SM_{pb1}} \right]}{SM_{pb2} \left[1 + \frac{B_{b2}}{SM_{pb2}} + \frac{K_{b2}}{SM_{pb2}} \right]} + \frac{K_{b2}}{S^2 M_{pb1}} - \frac{K_{b2} \left[\frac{B_{b2}}{SM_{pb1}} + \frac{K_{b2}}{SM_{pb1}} \right]}{S^2 M_{pb1} \left[1 + \frac{B_{b2}}{SM_{pb2}} + \frac{K_{b2}}{SM_{pb2}} \right]} \quad (3-3b)$$

It is important to mention that Eq. (3-3a) should be reduced to the dynamic stiffness equation of a fluid mount without TVA if $M_{pb2} \rightarrow 0$. It is the limit case of this equation and can be used to check its correctness. By finding the limit of K_{dyn} when $M_{pb2} \rightarrow 0$, the dynamic stiffness, Eq. (3-3a), approaches:

$$K_{dyn} = K_r + B_r S + \frac{A_{pt}^2}{C_t} \frac{S^2 + S \frac{R_f}{I_f} + \frac{K_b}{I_f}}{S^2 + S \frac{R_f}{I_f} + \frac{1/C_b + 1/C_t}{I_f}} \quad (3-4)$$

This expression is exactly the dynamic stiffness of the classical passive single pumper fluid mount. To do MATLAB simulations, it is needed to assume some baseline values for all of the fluid mount parameters. The baseline fluid mount parameters are defined as follows:

A_{pt}	Piston area, 9E-3 m ²
K_r	Rubber stiffness, 2.05E6 N/m
Tanδ	For low damped mid range elastomers, G''/G' is 0.04.
Tanδ	For low damped low shear modulus elastomers, G''/G' is 0.02.
B_r	Hysteresis damping of rubber, 522.0 N.s/m (Tanδ=0.04 assumed)
C_t	Top chamber rubber volumetric compliance, 1/1.1E11, m ⁵ /N
C_b	Bottom chamber fluid volumetric compliance, 1/2.1E14, m ⁵ /N
R_f	Fluid flow resistance of the inertia track, 6.4E6 N.s/m ⁵
K_{b1}	Stiffness of the spring K_{b1} , 876 N/m

B_{b1}	Equivalent hysteresis damping of the spring, 0.11 N.s/m (Tan δ =0.02)
A_{pb}	Piston area for the bottom chamber, 6.45E-4 m ²
M_{pb1}	Bottom chamber piston mass, 45.3 grams
M_{pb2}	TVA mass, 100 grams
K_{b2}	TVA stiffness, 876 N/m
B_{b2}	TVA equivalent hysteresis damping, 0.186 N.s/m (Tan δ =0.02)
ρ	Fluid density, 1,770 kg/m ³
L_i	Inertia track length, 0.177 m
d_i	Inertia track diameter, 0.0216 m
X_m	Maximum input displacement, ± 0.0381 mm (± 1.5 mils)

The fluid inertia of the inertia track (I_f) can be calculated as follows.

$$I_f = \frac{4 \times \rho \times L_i}{\pi \times d_i^2} \quad (3-5)$$

3.3 Passive fluid mount (without the TVA)

The dynamic stiffness of a fluid mount without the TVA is derived in Eq. (3-4) and is shown in Figure 3.5. As it can be seen from Figure 3.5, the dynamic stiffness of a typical fluid mount without the TVA has one notch and one peak. Figure 3.6 shows the linear plot of dynamic stiffness versus frequency, zoomed to notch frequency, and it can be seen that the notch frequency is about 25.8 Hz.

To explain why and under what conditions the notch and peak frequencies occur, the bottom fluid chamber piston displacement relative to the input displacement given to

the fluid mount is studied. The fluid flow rate through the inertia track versus input flow rate is also studied. Figures 3.7 and 3.8 show the phase difference between the input and the bottom chamber piston displacements and the phase difference between the flow rate through the inertia track and the input flow rate versus frequency, respectively. Figure 3.7 shows that at the notch frequency, the input and the piston displacements are in phase. Meaning, when compressive input displacement is applied to the top chamber, the top chamber is pressurized. At the notch frequency, the bottom chamber piston moves in phase with the input displacement; therefore, creating low pressure region in the bottom chamber, pulling the top chamber fluid downwards. So no fluid lock up takes place at the top chamber. But at the peak frequency, the input and piston displacements are out of phase. At the peak frequency, the top chamber fluid can no longer flow to the bottom chamber since the piston in the bottom chamber is forcing fluid to the top chamber. The fluid is trapped in the top chamber and as a result the dynamic stiffness increases to a very high value at the peak frequency.

Figure 3.5 Dynamic stiffness of a typical fluid mount - semi log plot

Figure 3.6 Dynamic stiffness of a typical fluid mount (zoomed to notch) - Linear Plot

Figure 3.7 Phase between input displacement and lower chamber piston displacement versus frequency

Figure 3.8 Phase between flow rate through the inertia track and input flow rate versus frequency

Figure 3.8 indicates the same conclusion; meaning at the peak frequency, the bottom chamber fluid is moving to the top chamber; therefore, making the top chamber appear very stiff. Meaning, the fluid of the top chamber has no where to go but only bulge or expand the rubber.

Looking at the fluid mount design of Figure 2.8, if one actually removes the bottom chamber rubber diaphragm and replaces it with a piston and a rubber (or metal) spring, we get the design of Figure 3.2, (assuming TVA is removed). With such a design, it is important to make sure that the vibration amplitude of the bottom chamber piston is not too large or else such a design will be impractical.

With no TVA, the vibration amplitude of the bottom chamber piston versus frequency is shown in Figure 3.9.

Figure 3.9 Fluid mount without TVA - Amplitude of the piston versus frequency

As it can be seen from Figure 3.9, the maximum displacement of the bottom chamber piston is ± 25 mm at the peak frequency, which is reasonable and acceptable.

3.4 Passive fluid mounts with a TVA

Adding a TVA to the bottom chamber piston, leads to the design of Figure 3.2. A typical dynamic stiffness of a fluid mount plus a TVA is shown in Figure 3.10. It can be concluded that the dynamic stiffness versus frequency corresponding to the fluid mount plus the TVA has two notch frequencies whereas the typical passive fluid mount dynamic stiffness has only one notch frequency. As we explained earlier, since there are multiple engine disturbance inputs of different amplitudes and frequencies, it is very desirable to have a fluid mount with more than one notch. Commercially available passive fluid mounts have only one notch, and the notch can only be

designed to coincide with only one of the disturbance inputs but with the new design proposed in this chapter, two engine disturbance inputs can now be isolated.

Figure 3.10 Dynamic stiffness of a fluid mount plus TVA

Before analyzing the effect of TVA parameters on the dynamic stiffness of the new fluid mount design, it is necessary to make sure that the amplitudes of the bottom chamber piston and TVA mass are reasonable and practical. Figure 3.11 shows the amplitude of the bottom chamber piston and the amplitude of the TVA mass, respectively.

Figure 3.11 Amplitude of the bottom chamber piston and amplitude of the TVA mass

As Figure 3.11 illustrates, the maximum amplitude of the bottom chamber piston is less than 19 mm and the maximum amplitude of the TVA mass is around 30 mm. These values show that the fluid mount plus TVA is good for applications with very low input displacement amplitudes such as aerospace applications. Note that X_{in} was assumed to be equal to 0.0381 mm (typical jet engine dynamic displacement).

3.4.1 Parametric Studies

TVA is composed of 3 main parts: Mass, stiffness and damping. The location and the depth of the new notch depend on these parameters. So the next step is to find the effect of changing each of these parameters on the depth and the location of the new notch.

TVA as an individual vibrating system has its own natural frequency (ω_l). Fluid mount without TVA has a notch frequency " ω_{nc} " and a peak frequency " ω_{peak} ".

The natural frequency of the TVA could be placed below the fluid mount notch frequency, or in between ω_{nc} and ω_{peak} or above ω_{peak} . In order to find the effect of changing TVA parameters on the location and depth of the new notch, each parameter has been swept over different regions such that the natural frequency of the TVA, namely ω_l , is in the following regions.

- 1- Below ω_{nc} (**Region I**)
- 2- Between ω_{nc} and ω_{peak} (**Region II**)
- 3- Above ω_{peak} (**Region III**)

By doing simulation with Matlab, it is found that the effect of damping is widening the notch region. The TVA, in general, is made of a mass bonded to a rubber piece. So the rubber stiffness of TVA includes damping. The imaginary stiffness of the TVA, K'' is expressed in Eq.(3-6).

$$K'' = \tan \delta \cdot K' \quad (3-6)$$

Where, K' is the elastic stiffness and $\tan \delta$ represents the amount of damping in the rubber. So for a given elastomer (or a given $\tan \delta$) K'' is not controllable. So, only change in stiffness and mass of the TVA has been considered. From practical viewpoint, by considering that there are some space and weight constraints, the range in which we can change the mass is limited. But the TVA stiffness can be changed widely. However, it is necessary to have reasonable values for the mass.

The damping of the TVA is due to the hysteresis of the rubber or spring and the friction between the fluid mount housing and the piston. By lubricating the piston the friction damping can be reduced to a negligible amount. The hysteresis damping of the TVA is considered in the simulation results presented in this thesis.

3.4.1.1 Effect of the natural frequency of the TVA on the dynamic stiffness of the fluid mount plus TVA

In this section, the natural frequency of the TVA has been swept over different regions and its effect on the dynamic stiffness of the hydraulic mount has been studied.

3.4.1.1.1 Sweeping the natural frequency of the TVA in the region 1

According to simulation results, the notch frequency of the hydraulic engine mount without TVA is equal to 25.8 Hz and its peak frequency is 54.9 Hz. So for region 1, the natural frequency of the TVA should be held below 25.8 Hz.

The natural frequency of the TVA is calculated as follows.

$$\omega_{ntva} = \frac{1}{2\pi} \sqrt{\frac{K_{b2}}{M_{b2}}} \text{ (Hz)} \quad (3-7)$$

In order to study the effect of adding a TVA to the fluid mount at region 1, three different TVA natural frequencies are examined.

$$\underline{\omega_{ntva} = 10 \text{ Hz}}$$

By choosing the TVA stiffness (K_{b2}) equal to 394.78 N/m and the TVA mass (M_{b2}) equal to 0.1 Kg , the natural frequency of TVA will be 10 Hz .

$$\omega_{ntva} = 20 \text{ Hz}$$

By choosing the TVA stiffness (K_{b2}) equal to $1.5791e3 \text{ N/m}$ and the TVA mass (M_{b2}) equal to 0.1 Kg , the natural frequency of TVA will be 20 Hz .

$$\omega_{ntva} = 25 \text{ Hz}$$

By choosing the TVA stiffness (K_{b2}) equal to $2.4674e3 \text{ N/m}$ and the TVA mass (M_{b2}) equal to 0.1 Kg , the natural frequency of TVA will be 25 Hz . The simulation result is shown in Figure 3.12.

Figure 3.12 Effect of the natural frequency of the TVA on the dynamic stiffness of the fluid mount plus TVA (Region 1)

Figure 3.12 illustrates that increasing the natural frequency of the TVA will increase both of the notch frequencies. By comparing the dynamic stiffness of the fluid mount

plus TVA and the fluid mount without TVA it can be concluded that, the first notch is due to adding the TVA to the mount. It can be implied from Figure 3.12 that when the natural frequency of the TVA is close to the notch frequency of the fluid mount without TVA, the effect of the TVA on the location of the original notch frequency is more significant.

3.4.1.1.2 Sweeping the natural frequency of the TVA in the region 2

In order to study the effect of adding the TVA to the fluid mount at region 2, three different TVA natural frequencies are examined.

$$\underline{\omega_{ntva} = 30 \text{ Hz}}$$

By choosing the TVA stiffness (K_{b2}) equal to $3.5531e3 \text{ N/m}$ and the TVA mass (M_{b2}) equal to 0.1 Kg , the natural frequency of TVA will be 30 Hz .

$$\underline{\omega_{ntva} = 40 \text{ Hz}}$$

By choosing the TVA stiffness (K_{b2}) equal to $6.3165e3 \text{ N/m}$ and the TVA mass (M_{b2}) equal to 0.1 Kg , the natural frequency of TVA will be 40 Hz .

$$\underline{\omega_{ntva} = 50 \text{ Hz}}$$

By choosing the TVA stiffness (K_{b2}) equal to $9.8696e3 \text{ N/m}$ and the TVA mass (M_{b2}) equal to 0.1 Kg , the natural frequency of TVA will be 50 Hz .

The simulation results are shown in Figure 3.13. As it can be concluded from Figure 3.13, in region 2, the new notch is wider than the new notch in region 1.

Figure 3.13 Effect of the natural frequency of the TVA on the dynamic stiffness of the fluid mount plus TVA (Region 2)

Figure 3.13 illustrates that, increasing the natural frequency of the TVA in region 2 will result in increasing the second notch frequency. The first notch is not affected too much. Additionally, it can be seen in Figure 3.13 that increasing the natural frequency of the TVA would decrease the notch depth. It means that at very high frequencies, adding the TVA to the fluid mount is not so effective. It is due to the hysteresis damping of the rubber which connects the TVA to the lower piston. The equivalent damping of the rubber is defined by Eq.(3-8):

$$B_{b2} = \frac{\tan \delta K_{b2}}{\omega_{ntva}} \quad (3-8)$$

Where K_{b2} is defined as:

$$K_{b2} = \omega_{ntva}^2 \times M_{b1} \quad (3-9)$$

So, at very high frequencies, the dynamic stiffness increases and the notch depth will decrease.

3.4.1.1.3 Sweeping the natural frequency of the TVA over the region 3

In order to study the effect of adding the TVA to the fluid mount at region 3, three different TVA natural frequencies are examined.

$$\underline{\omega_{ntva} = 60 \text{ Hz}}$$

By choosing the TVA stiffness (K_{b2}) equal to $1.4212e4 \text{ N/m}$ and the TVA mass (M_{b2}) equal to 0.1 Kg , the natural frequency of TVA will be 60 Hz .

$$\underline{\omega_{ntva} = 70 \text{ Hz}}$$

By choosing the TVA stiffness (K_{b2}) equal to $1.9344e4 \text{ N/m}$ and the TVA mass (M_{b2}) equal to 0.1 Kg , the natural frequency of TVA will be 70 Hz .

$$\underline{\omega_{ntva} = 80 \text{ Hz}}$$

By choosing the TVA stiffness (K_{b2}) equal to $2.5266e4 \text{ N/m}$ and the TVA mass (M_{b2}) equal to 0.1 Kg , the natural frequency of TVA will be 80 Hz .

The simulation results are shown in Figure 3.14.

Figure 3.14 Effect of the natural frequency of the TVA on the dynamic stiffness of the fluid mount plus TVA (Region 3)

Figure 3.14 illustrates that increasing the natural frequency of the TVA in region 3 will increase the second notch frequency while the first notch frequency moves slightly.

As it can be seen in Figure 3.14, adding the TVA at high frequencies is not efficient because the new notch created by adding the TVA is not deep.

In order to compare the effect of changing the TVA natural frequency on the dynamic stiffness of the fluid mount in regions 1, 2 and 3, Figure 3.15 has been plotted.

Figure 3.15 Effect of the natural frequency of the TVA on the dynamic stiffness of the fluid mount plus TVA (Overall)

It can be concluded that increasing the natural frequency of the TVA will result in increasing the new notch frequency. Additionally, the deepest TVA notch can be achieved when the natural frequency of the TVA is close to the notch frequency of the fluid mount.

It is worth mentioning that the TVA mount can be easily converted to a semi-active mount. It can be implied from Figures 3-12, 3-13 and 3-14 that, changing the stiffness of the TVA (Which results in changing the TVA natural frequency) changes the location of the new notch. It means that by the use of a variable stiffness TVA, the location of the new notch can be controlled semi-actively.

3.5 Conclusions

A new design for a hydraulic engine mount was introduced in this chapter. The proposed hydraulic engine mount is a typical single pumper passive fluid mount where the bottom chamber rubber diaphragm is replaced with a piston and a rubber spring and a tuned vibration absorber (TVA). The dynamic stiffness characteristics of the new hydraulic engine mount have been investigated. It was shown that the addition of a TVA to the fluid mount results in two notches in the dynamic stiffness of the hydraulic mount. The location of the new notch increases with an increase in the natural frequency of the TVA.

Changing the TVA parameters (TVA mass and stiffness) is much easier than changing the fluid mount parameters (Rubber stiffness, volumetric stiffness, inertia track, fluid density). So, adding the TVA makes the hydraulic mount more tunable. Additionally, having two notch frequencies instead of one would result in better vibration isolation. For example, in the case of an aircraft, one of the notches can be tuned to coincide with the N1 frequency (engine low speed shaft imbalance frequency) and the other notch can be tuned to coincide with N2 frequency (engine high speed shaft imbalance frequency) and so the vibration isolation would be improved.

Simulation results have shown that, TVA is more effective at low frequencies than the high frequencies. Additionally, in order to keep the amplitude of the lower chamber piston and the TVA mass low, the input displacement to the mount should be low, meaning this new design is more effective for aerospace applications than automotive applications.

Chapter4 : A novel design for fluidless electronic vibration isolator

In chapter 2, fluid mounts were introduced and their characteristics, advantages and disadvantages were investigated.

As mentioned in chapter 2, there are three main pitfalls associated with typical fluid mounts:

- 1- The design procedure of fluid mounts is somewhat based on trial and error which is a time consuming and expensive method due to non-existence of very accurate analytical models .
- 2- As time goes on, the dynamic characteristics of the engine as well as the fluid mount properties will change; so the mount needs to be retuned after a period of operation. The retuning procedure of fluid mounts is costly because it needs disassembling the mount and redesigning its internal parts.
- 3- The dynamic performance of fluid mounts is similar to the ideal engine mount dynamic behavior, up to the notch frequency; but, beyond the notch frequency, the dynamic stiffness of the fluid mounts increases dramatically. So, at high frequencies, fluid mounts perform even worse than the elastomeric engine mounts.

The nonlinearities induced by the fluid structure interaction (between the fluid inside the mount and the mount structure) and fluid losses through the inertia track along with the coupled key parameters are responsible for the major part of the fluid mount design complexity. So it makes all the sense to remove the fluid and replace it with a more tunable mechanism.

This chapter is devoted to a feasibility study of electronic type vibration isolators replacing passive fluid mounts. If we could replace the hydraulic mechanism with an electronic one, the retuning of the engine mount would be facilitated. Furthermore, the trial and error design procedure would be avoided with an electronic engine mount. The new design will also improve the high frequency properties of the engine mount.

4.1 Electro-mechanical equivalent of a typical fluid mount

Typical fluid mounts consist of two energy domains; namely, the mechanical energy domain and the hydraulic energy domain.

The basic idea of electronic vibration isolator is to develop the electro-mechanical equivalent of a typical fluid mount.

Bond graph modelling technique utilizes a unique notation for different energy domains. For example a C element in a bond graph model may be representative of a mechanical compliance in the mechanical energy domain, or a capacitance in the electronic energy domain. So bond graph modelling technique can be used to find the equivalent of a system in another energy domain. In this work, bond graph modelling technique is used to find the electro-mechanical equivalent of the hydraulic type engine mount.

Figure 4.1 shows the bond graph model of a typical fluid mount. In the electro-mechanical vibration isolator, the hydro-mechanical components must be replaced by their electro-mechanical counterparts.

Figure 4.1 Bond graph model of a typical fluid mount.

Since the primary task of all engine mounts is to connect the engine to the chassis and constraint their relative motion, the upper chamber elastomeric component must be retained in the electro-mechanical vibration isolator.

In the case of fluid mounts the effective piston area (A_p) performs as a transformer and converts the mechanical energy into the hydraulic energy and vice versa. More specifically, the transformer (A_p) converts the mechanical force to the hydraulic pressure and the mechanical velocity to the hydraulic flow and vice versa.

It can be inferred from the above discussion that, in the case of the electronic vibration isolator, an electro-mechanical transformer is needed to convert the mechanical energy to the electronic energy and vice versa.

There are two basic ways to convert the electrical energy to the mechanical power; they are represented by movable core inductor and movable plate capacitor (Karnopp 1984).

The exact electro-mechanical counterpart of the hydro-mechanical transformer does not exist. But with some minor modifications in the bond graph model, an electro-mechanical gyrator can be used instead of the electro-mechanical transformer.

Voice coils are the most common electro-mechanical gyrators. They can convert the mechanical force to the electrical current and the mechanical velocity to the electrical potential (Voltage), and vice versa. Note that an electro-mechanical transformer, if exist, converts the mechanical velocity to the electrical current and the mechanical force to the electrical potential (Voltage), and vice versa.

Voice coils are used mainly in loudspeakers. Figure 4.2 shows a loudspeaker in details.

Figure 4.2 The voice coil used in a loudspeaker

Voice coils are made of two main parts, namely: a permanent magnet and a coil. The relative movement of the permanent magnet and the coil induces voltage in the coil.

The working principle of the voice coils can be described based upon the Lorentz force expression. The Lorentz force expression says that the force applied on a conductor of length ℓ carrying current i in a magnetic field of flux density B is:

$$F = B\ell i \quad (4-1)$$

If the conductor is moving relative to the magnetic field while the vectors F , B and i are perpendicular to each other, the corresponding voltage induced in the conductor is:

$$e = B\ell V \quad (4-2)$$

So the power balance is preserved:

$$FV = ei \quad (4-3)$$

The voice coil acts as a gyrator which converts the mechanical force into the electrical current and vice versa. The relationship between the force induced by the voice coil and the current in the coil is as follows:

$$F = \gamma i \quad (4-4)$$

Where, in Eq (4-4), γ is the force sensitivity of the voice coil with the units of N/Amp . It is worth mentioning that, the linear relationship between the voice coil force and current (F and i) is valid until the voice coil saturation point. Beyond the voice coil saturation point, increasing the current would not result in any increase in the generated force.

Figure 4.3 Relationship between the voice coil force (F) and the voice coil current (i) for a real voice coil

The effect of the voice coil saturation point has been considered in the simulation results presented in this work.

Although the main function of the voice coils is to convert the electrical energy and mechanical energy to each other, since the coil is made of wire, it has some resistance and inductance, as well. The voice coil inherent resistance and inductance have been considered in the simulations presented in this chapter.

A thorough market survey was conducted to find the best available voice coils in the market. The criterion to accept or reject a voice coil is its "Force Sensitivity" which has the units of N/Amp and the voice coil inherent resistance and inductance which must be as low as possible.

The specifications of some of the commercial voice coils which have been found are listed in Table 4.1.

Table 4.1 Voice coil specifications

ITEM	DC Resistance Ω	Inductance mH	Net Weight of the coil (grams)	Force sensitivity N / Amp
LA30-43- 000A	2.6	2.9	726	31.14
LA25-42- 000A	2.4	2.5	540	21.35
NCC08-34- 250-2X	6.36	4.21	3405	57.78

The electrical counterpart of the volumetric stiffness, damping and fluid inertia are capacitance, resistance and inductance, respectively.

Using the aforementioned electro-mechanical counterparts of the hydro-mechanical engine mount components and the bond graph model of the typical fluid mount (Figure 4.1), the fluidless electronic engine mount can be designed.

4.2 Electronic vibration isolator-Design One

Figure 4.4 shows the configuration of the primary design for the electronic vibration isolator. This physical model is based upon the bond graph model of a typical fluid mount.

Figure 4.4 Simplified physical model of the proposed electronic engine mount- first design

The bond graph model of Figure 4.4 is shown in Figure 4.5.

Figure 4.5 Bond graph model of the electronic engine mount shown in Figure 4.4

The state-space equations derived from the above bond graph model are as follows:

$$\dot{q}_3 = V_m \quad (4-5)$$

$$\dot{p}_7 = \gamma V_m - R_1 \frac{p_7}{I_1} - \frac{q_9}{C_t} \quad (4-6)$$

$$\dot{q}_9 = \frac{p_7}{I_1} - \frac{p_{12}}{I_f} \quad (4-7)$$

$$\dot{q}_{11} = \frac{p_{12}}{I_f} \quad (4-8)$$

$$\dot{p}_{12} = \frac{q_9}{C_t} - \frac{q_{11}}{C_b} - R_f \frac{p_{12}}{I_f} \quad (4-9)$$

The transmitted force is:

$$F_{in} = B_r V_{in} + K_r q_3 + \gamma \frac{p_7}{I_1} \quad (4-10)$$

The matrix form of the state-space equations is:

$$\begin{bmatrix} \dot{q}_3 \\ \dot{p}_7 \\ \dot{q}_9 \\ \dot{q}_{11} \\ \dot{p}_{12} \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ 0 & -\frac{R_l}{I_1} & -\frac{1}{C_t} & 0 & 0 \\ 0 & \frac{1}{I_1} & 0 & 0 & -\frac{1}{I_f} \\ 0 & 0 & 0 & 0 & \frac{1}{I_f} \\ 0 & 0 & \frac{1}{C_t} & -\frac{1}{C_b} & -\frac{R_f}{I_f} \end{bmatrix} \begin{bmatrix} q_3 \\ p_7 \\ q_9 \\ q_{11} \\ p_{12} \end{bmatrix} + \begin{bmatrix} 1 \\ \gamma \\ 0 \\ 0 \\ 0 \end{bmatrix} [V_{in}] \quad (4-11)$$

$$F_{in} = \begin{bmatrix} K_r & \frac{\gamma}{I_1} & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} q_3 \\ p_7 \\ q_9 \\ q_{11} \\ p_{12} \end{bmatrix} + [B_r] V_{in} \quad (4-12)$$

$\dot{q}$ and $\dot{p}$ are representatives of the flow and effort in bond graph terms, respectively. Flow, in the mechanical energy domain represents the velocity, while in the electrical energy domain, it represents the current. Effort in the mechanical energy domain stands for force, but in the electrical energy domain it represents the voltage.

So:

$\dot{q}_3$: Velocity of the engine side of the engine mount.

$\dot{p}_7$: Voltage induced in the inductance I_1 .

$\dot{q}_9$: Current induced in the resistance R_1 .

$\dot{q}_{11}$: Current induced in the capacitance C_b .

$\dot{p}_{12}$: Voltage induced in the inductance I_f .

The following baseline parameters have been used for doing the MATLAB simulations (see Table 4.2).

Table 4.2 Baseline engine mount parameters for simulation

Symbol	Description	Value	Dimension
K_r	Rubber vertical stiffness	2.05e6	N/m
B_r	Hysteresis damping of the rubber	10	$N.s/m$
γ	Force sensitivity of the voice coil	31.14	N/Amp
C_t	Capacitance (Counterpart of the upper chamber compliance)	18	μF
I_f	Inductance (Counterpart of the Inertia of the fluid in the inertia track)	0.125	mH
C_b	Capacitance (Counterpart of the lower chamber compliance)	100	μF
R_f	Resistance (Counterpart of the Inertia track resistance)	0.01	Ω
F_{max}	Maximum force the voice coil can generate	445	N
X_{in}	Input displacement	0.3	mm

Simulation results presented in this chapter are done using MATLAB. The MATLAB program which is used to simulate the dynamic stiffness of the electronic vibration isolator is represented in appendix I.

Figure 4.6 shows the dynamic stiffness of the electronic vibration isolator with design one, for three different voice coils.

Figure 4.6 The dynamic stiffness of the electronic engine mount (Design I)

As Figure 4.6 illustrates, although the dynamic stiffness of the proposed electronic vibration isolator is similar to the typical fluid mounts dynamic stiffness, the notch of the electronic vibration isolator dynamic stiffness is not deep enough to provide good vibration isolation. The voice coil specifications of “LA30-43-000A” have been used.

Figure 4.7 illustrates the effect of voice coil parameters on the dynamic stiffness of the electronic vibration isolator.

Figure 4.7 Effect of Voice coil resistance, inductance and force sensitivity on the dynamic stiffness of the electronic engine mount

As Figure 4.7 reveals, the gyrator module has a significant effect on the notch depth. But choices for the gyrator module are limited to the voice coils available in the market. Unfortunately, the force sensitivity (Gyrator module) of the available voice coils is not high enough to provide a deep notch with design one.

It can be inferred from Figure 4.7 that, with a high force sensitivity voice coil a reasonably deep notch can be achieved. Additionally, Figure 4.7 illustrates the adverse effect of the voice coil inherent resistance and inductance on the dynamic stiffness of the electronic vibration isolator. It means that decreasing the voice coil resistance and inductance would improve the dynamic characteristics of the vibration isolator.

Due to the force sensitivity limitation of the available voice coils in the market and voice coil inherent resistance and inductance, the design shown in Figure 4.4 would not be able to provide acceptable dynamic stiffness properties using available voice coils.

There are two remedies for this problem. The first remedy is to wait for voice coil suppliers to develop high force sensitivity voice coils; and the second remedy is to change the design to improve the dynamic stiffness characteristics of the electronic vibration isolator using the available voice coils. The second remedy has been undertaken in this work.

4.3 Electronic vibration isolator-Design two

Figure 4.8 shows the configuration of the electronic vibration isolator with design two.

Figure 4.8 Simplified physical model of an electronic engine mount-plan 2

The mechanical part of this design is the same as design one; but the circuit has been simplified. The objective of design two is to investigate the effect of the circuit parameters on the dynamic stiffness of the electronic vibration isolator.

The bond graph model of design two is shown in Figure 4.9.

Figure 4.9 Bond graph model of the electronic engine mount-plan 2

The state space equations derived from the above bond graph model are:

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

$$\dot{p}_7 = \gamma V_{in} - R \frac{p_7}{I} - \frac{q_8}{C} \quad (4-14)$$

$$\dot{q}_8 = \frac{p_7}{I} \quad (4-15)$$

The matrix form of state space equations is:

$$\begin{bmatrix} \dot{q}_3 \\ \dot{p}_7 \\ \dot{q}_8 \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 \\ 0 & -\frac{R}{I} & -\frac{1}{C} \\ 0 & \frac{1}{I} & 0 \end{bmatrix} \begin{bmatrix} q_3 \\ p_7 \\ q_8 \end{bmatrix} + \begin{bmatrix} 1 \\ \gamma \\ 0 \end{bmatrix} V_{in} \quad (4-16)$$

$$F_{in} = \begin{bmatrix} K_r & \frac{\gamma}{I} & 0 \end{bmatrix} \begin{bmatrix} q_3 \\ p_7 \\ q_8 \end{bmatrix} + [B_r] V_{in} \quad (4-17)$$

Simulation results for design two are shown in Figure 4.10. In this Figure the effect of the capacitance on the dynamic stiffness of the electronic vibration isolator is investigated. Note that the resistance and inductance of the circuit are the voice coil inherent resistance and inductance.

Figure 4.10 Dynamic stiffness of the proposed electronic engine mount for different capacitance values.

As Figure 4.10 illustrates, using the simplified circuit (design two), the notch has been retained. Additionally, increasing the capacitance of the circuit would increase the notch frequency and the notch depth. Although, the notch depth can be increased by increasing the capacitance of the circuit, the amount of this increase is not acceptable yet.

4.4 Electronic vibration isolator-Design three

As mentioned before, a high force sensitivity voice coil can provide a reasonably deep notch. Since none of the voice coils found in the market has high enough force

sensitivity, one remedy is to use two voice coils in parallel to improve the dynamic stiffness properties of the vibration isolator.

The simplified physical model of the proposed design is illustrated in Figure 4.11.

Figure 4.11 The simplified physical model of a double voice coil-Design III

Figure 4.12 shows the bond graph model of the design two.

Figure 4.12 The bond graph model of the proposed electronic engine mount-Design III

The state space equations derived from the above bond graph model in matrix form are:

$$\begin{bmatrix} \dot{q}_3 \\ \dot{p}_6 \\ \dot{q}_8 \\ \dot{p}_{11} \\ \dot{q}_{13} \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ 0 & -\frac{R}{I} & -\frac{1}{C} & 0 & 0 \\ 0 & \frac{1}{I} & 0 & 0 & 0 \\ 0 & 0 & 0 & -\frac{R}{I} & -\frac{1}{C} \\ 0 & 0 & 0 & \frac{1}{I} & 0 \end{bmatrix} \begin{bmatrix} q_3 \\ p_6 \\ q_8 \\ p_{11} \\ q_{13} \end{bmatrix} + \begin{bmatrix} 1 \\ \gamma \\ 0 \\ \gamma \\ 0 \end{bmatrix} V_{in} \quad (4-18)$$

$$F_{in} = \begin{bmatrix} K_r & \frac{\gamma}{I} & 0 & \frac{\gamma}{I} & 0 \end{bmatrix} \begin{bmatrix} q_3 \\ p_6 \\ q_8 \\ p_{11} \\ q_{13} \end{bmatrix} + [B_r] V_{in} \quad (4-19)$$

The simulation result is shown in Figure 4.13.

Figure 4.13 Dynamic stiffness of the double voice coil electronic engine mount

Figure 4.13 shows that using the two voice coil system would improve the notch depth of the dynamic stiffness of the electronic vibration isolator. But it should be noted that this increase in the notch depth is achieved at the expense of increasing the cost and weight of the vibration isolator.

4.5 Design Four

Analyzing the state space equations of a typical fluid mount shows that one of the most important parameters which affects the notch depth is the transformer gain. The transformer gain for a fluid mount is the ratio of piston area to the inertia track area (A_p/A_t). In electronic vibration isolator, a gyrator has been used instead of the transformer. In the case of the electronic vibration isolator, due to the force sensitivity limitation of the commercial voice coils, the notch depth is not satisfactory.

One remedy is to add an electronic transformer in the circuit but the very high inherent resistance of the electronic transformers tends to deteriorate the notch depth. The other remedy would be to amplify the effect of the voice coil force sensitivity using a mechanical transformer.

Design four is the most detailed design presented in this chapter. In previous designs since the objective was to figure out the feasibility of the electronic vibration isolator concept, the mass of the voice coil, upper plate mass and the friction in the pivots have been assumed to be negligible. Since design four will be used to fabricate the experimental test apparatus (In Chapter 5), it should be more detailed. In design four the effect of the voice coil mass and the upper plate mass are considered. It has been assumed that by lubricating the pivots the friction can be kept at a negligible level.

Figure 4.14 shows the configuration of the proposed design.

Figure 4.14 The physical model of the electronic Vibration isolator with mechanical transformer

Figure 4.15 shows the bond graph of Figure 4.14.

Figure 4.15 The bond graph model of Figure 4.20

As Figure 4.15 illustrates, the voice coil mass and the upper plate mass have added two derivative causalities to the bond graph model of the electronic vibration isolator. When there are derivative causalities in the bond graph model, derivation of the state-space equations will become more complicated.

The state space equations derived from the bond graph model in matrix form are:

$$\dot{q}_3 = V_m \quad (4-20)$$

$$\dot{p}_{11} = \gamma \frac{\ell_2}{\ell_1} V_m - R_1 \frac{p_{11}}{I_1} - \frac{q_{13}}{C_t} \quad (4-21)$$

$$\dot{q}_{13} = \frac{p_{11}}{I_1} - \frac{p_{16}}{I_f} \quad (4-22)$$

$$\dot{q}_{15} = \frac{p_{16}}{I_f} \quad (4-23)$$

$$\dot{p}_{16} = \frac{q_{13}}{C_t} - R_f \frac{p_{16}}{I_f} - \frac{q_{15}}{C_b} \quad (4-24)$$

The transmitted force is:

$$F_m = B_r V_m + K_r q_3 + M_t \dot{f}_2 + \frac{\ell_2}{\ell_1} \left[M_b \dot{f}_7 + \gamma \frac{p_{11}}{I_1} \right] \quad (4-25)$$

Where $\dot{f}_2 = \dot{V}_m$ and $\dot{f}_7 = \dot{f}_6 = \frac{\ell_2}{\ell_1} \dot{f}_5 = \frac{\ell_2}{\ell_1} \dot{V}_m$.

So,

$$\begin{aligned} f_m &= B_r V_m + K_r q_3 + M_t \dot{V}_m + \frac{\ell_2}{\ell_1} \left[M_b \frac{\ell_2}{\ell_1} \dot{V}_m + \gamma \frac{p_{11}}{I_1} \right] \\ &= B_r V_m + K_r q_3 + \frac{\ell_2}{\ell_1} \gamma \frac{p_{11}}{I_1} + \left[M_t \dot{V}_m + M_b \left(\frac{\ell_2}{\ell_1} \right)^2 \dot{V}_m \right] \end{aligned} \quad (4-26)$$

The above state space equations illustrate that, the voice coil mass and the upper plate mass only appear in the equation of F_m .

Since there is the $\dot{V}_{in}$ term in the F_{in} equation, the standard method can not be used to simulate the dynamic stiffness of this design. The dynamic stiffness of the electronic vibration isolator with design 5 can be derived using Laplace transformation.

The Laplace transform of the state space equations of the system are:

$$Q_3 = X_m \quad (4-27)$$

$$P_{11} = \left(\frac{\ell_2}{\ell_1} \right) \gamma X_m - R_1 \frac{P_{11}}{sI_1} - \frac{Q_{13}}{sC_t} \quad (4-28)$$

$$Q_{13} = \frac{P_{11}}{sI_1} - \frac{P_{16}}{sI_f} \quad (4-29)$$

$$Q_{15} = \frac{P_{16}}{sI_f} \quad (4-30)$$

$$P_{16} = \frac{Q_{13}}{sC_t} - \frac{Q_{15}}{sC_b} - R_f \frac{P_{16}}{sI_f} \quad (4-31)$$

The Laplace transform of F_{in} is:

$$F_{in} = B_r s X_m + K_r Q_3 + \frac{\ell_2}{\ell_1} \gamma \frac{P_{11}}{I_1} + M_t s^2 X_m + M_b \left(\frac{\ell_2}{\ell_1} \right)^2 s^2 X_m \quad (4-32)$$

$$\text{Eqs (4-39) and (4-40)} \quad \Rightarrow \quad P_{16} = \frac{Q_{13}}{sC_t \left[1 + \frac{1}{s^2 I_f C_b} + \frac{R_f}{sI_f} \right]} \quad (4-33)$$

Eqs (4-38) and (4-42)

$$\Rightarrow Q_{13} = \frac{P_{11}}{sI_1 \left[1 + 1 / \left[s^2 I_f C_i \left[1 + \frac{1}{s^2 I_f C_b} + \frac{R_f}{sI_f} \right] \right] \right]} \quad (4-34)$$

Eqs (4-37) and (4-43)

$$\Rightarrow P_{11} = \frac{\left(\frac{\ell_2}{\ell_1} \right) \gamma X_m}{\left[1 + R_1/sI_1 + 1 / \left[s^2 I_1 C_i \left[1 + 1 / s^2 I_f C_i \left[1 + \frac{1}{s^2 I_f C_b} + \frac{R_f}{sI_f} \right] \right] \right] \right]} \quad (4-35)$$

By Substituting Eqs (4-35) and (4-34) in Eq (4-32):

$$F_m = B_r s X_m + K_r X_m + \frac{\left(\frac{\ell_2}{\ell_1} \right)^2 \gamma^2}{I_1 \left[1 + R_1/sI_1 + 1 / \left[s^2 I_1 C_i \left[1 + 1 / s^2 I_f C_i \left[1 + \frac{1}{s^2 I_f C_b} + \frac{R_f}{sI_f} \right] \right] \right] \right]} X_m$$

$$+ M_i s^2 X_m + M_b \left(\frac{\ell_2}{\ell_1} \right)^2 s^2 X_m \quad (4-36)$$

By putting $s = j\omega$:

$$K^* = \frac{F_{in}}{X_{in}} = B_r j\omega + K_r + \frac{\left(\frac{\ell_2}{\ell_1}\right)^2 \gamma^2}{I_1 \left[1 + R_1/(j\omega I_1) - 1/\left[\omega^2 I_1 C_t \left[1 - 1/\left[\omega^2 I_f C_t \left[1 - \frac{1}{\omega^2 I_f C_b} + \frac{R_f}{j\omega I_f} \right] \right] \right] \right] \right]} - M_t \omega^2 - M_b \left(\frac{\ell_2}{\ell_1}\right)^2 \omega^2 \quad (4-37)$$

Note that K^* is a complex number. The dynamic stiffness of the electronic vibration isolator with design four is:

$$K_{dyn} = \sqrt{((Real(K^*))^2 + (imag(K^*))^2)} \quad (4-38)$$

The dynamic stiffness of the electronic vibration isolator with design four for different values of $\frac{\ell_2}{\ell_1}$ is shown in Figure 4.16. Note that the values of M_t and M_b has been kept very low to see the effect of adding the mechanical transformer.

Figure 4.16 Dynamic stiffness of the electronic vibration isolator shown in Figure 4.14

Figure 4.17 Dynamic stiffness of the electronic vibration isolator shown in Figure 4.14 (Showing the notch depth)

Figure 4.17 shows the notch region of the electronic vibration isolator shown in Figure 4.14 with $\frac{\ell_2}{\ell_1} = 4$. As Figure 4.17 illustrates, the notch depth is significantly improved over the previous designs.

It can be implied from Figure 4.17 that using the mechanical transformer has resolved the force sensitivity limitation problem of the voice coils.

4.5.1 Effect of the voice coil mass and the upper plate mass

In this section the real values of the voice coil mass and the upper plate mass have been used to examine their effects on the dynamic stiffness of the electronic vibration isolator.

A commercial voice coil is purchased to fabricate the test apparatus; so, the mass of the voice coil is known.

Since in practice, the vibration isolator will be connected directly to the vibratory object (e.g. engine), the upper plate can be removed. So the mass of the upper plate is not of much interest.

According to the technical specifications of the voice coil (LA30-43-000A), $M_b = 0.726 \text{ Kg}$. Figure 4.18 shows the dynamic stiffness of the electronic vibration isolator (design four) with $M_b = 0.726 \text{ Kg}$ and $M_b = 0 \text{ Kg}$.

Figure 4.18.Effect of the voice coil mass on the dynamic stiffness of the electronic vibration isolator.

As Figure 4.18 illustrates, considering the voice coil mass would affect the notch depth and notch location, significantly.

Note that when the voice coil mass is considered there are two sources to provide the notch phenomenon. One is the resonance of the voice coil mass (M_b) and the K_r , which act as a TVA, and the second is the notch caused by the interface of the mechanical system and the electronic circuit.

The contribution of the electronic circuit in the dynamic stiffness of the electronic vibration isolator when the voice coil mass is considered is investigated in Figure 4.19.

Figure 4.19 Dynamic stiffness of the electronic vibration isolator, with and without the circuit.

It can be implied from Figure 4.19 that with the current design the effect of the voice coil mass (M_b) on the notch is dominant.

In order to increase the contribution of the electronic circuit in the dynamic stiffness of the electronic vibration isolator, the circuit design must be modified.

Circuit parameters have been widely changed to find the appropriate parameters to increase the contribution of the electronic circuit in the dynamic stiffness of the electronic vibration isolator. The following parameters have been used for simulation.

Table 4.3 Baseline circuit parameters

C_t	C_b	γ	I_1	R_1	I_f
$500 \mu F$	$700 \mu F$	31.14 N/Amp	2.9 mH	2.6Ω	10 mH

Figure 4.20 shows the contribution of the electronic circuit with new parameters in the dynamic stiffness of the electronic vibration isolator.

Figure 4-20 Dynamic stiffness of the electronic vibration isolator with and without the circuit

As Figure 4.20 shows, with the new circuit parameters the effect of the electronic circuit on the dynamic stiffness of the electronic vibration isolator has been significantly increased.

It can be implied from Figure 4.20 that the electronic vibration isolator with design four can provide a reasonably deep notch.

4.6 Sensitivity analysis for designs 1 and 5

The effect of changing the circuit parameters on the dynamic behavior of the electronic vibration isolator is studied in this section.

4.6.1 Sensitivity analysis for the electronic vibration isolator with Design 1

At first step the effect of changing the circuit parameters in the first design will be studied.

The parameters of interest in the first design are C_t , I_f , C_b , R_f . Figure 4.21 illustrates the effect of changing C_t on the dynamic stiffness of the electronic vibration isolator with the Design one.

Figure 4.21 Effect of C_t on the dynamic stiffness of the electronic engine mount (Design 1)

As Figure 4.21 illustrates, increasing C_t results in decreasing the notch frequency and notch depth. The effect of C_t on the notch frequency is dominant.

The effect of changing I_f on the dynamic stiffness of the electronic vibration isolator is shown in Figure 4.22.

Figure 4.22 Effect of I_f on the dynamic stiffness of the electronic engine mount (Design 1)

As Figure 4.23 shows, increasing I_f would decrease the notch frequency and notch dept.

The effect of C_b on the dynamic stiffness of the electronic vibration isolator is shown in Figure 4.23.

Figure 4.23 Effect of C_b on the dynamic stiffness of the electronic engine mount (Design 1)

As Figure 4.23 illustrates, the effect of C_b on the dynamic stiffness is the same as C_t .

Meaning, increasing C_b would decrease the notch frequency and notch depth.

Finally the effect of R_f has been investigated. Figure 4.24 shows the effect of R_f on the dynamic stiffness of the electronic vibration isolator with design one.

Figure 4.24 Effect of R_f on the dynamic stiffness of the electronic engine mount (Design 1)

As illustrated by Figure 4.24, increasing R_f would decrease the notch depth while it does not affect the notch frequency so much.

4.6.2 Sensitivity analysis for the electronic vibration isolator with Design 4

The effect of circuit parameters on the dynamic stiffness of the electronic vibration isolator with design four will be studied in this section. It should be noted that all of the simulation results presented in this section are for $\frac{\ell_2}{\ell_1} = 4$.

The effect of the voice coil mass on the dynamic stiffness of the electronic vibration isolator with design four has been investigated in section 4.6. In this section the sensitivity analysis has been conducted for the electronic vibration isolator of design four, with and without considering the voice coil mass.

4.6.2.1 Electronic vibration isolator with a light-weight voice coil

In this section the effect of the circuit parameters on the dynamic stiffness of the electronic vibration isolator with a light-weight voice coil have been investigated.

Figure 4.25 shows the effect of C_t on the dynamic stiffness of the electronic vibration isolator.

Figure 4.25 Effect of C_t on the dynamic stiffness of the electronic vibration isolator (Design 4 - light-weight voice coil)

As Figure 4.25 illustrates, increasing C_t would decrease the notch frequency and the notch depth. It can be implied from Figure 4.25 that C_t is an appropriate parameter to control the notch frequency of the electronic vibration isolator.

Figure 4.26 shows the effect of I_f on the dynamic stiffness of the electronic vibration isolator.

Figure 4.26 Effect of I_f on the dynamic stiffness of the electronic engine mount (Design 4-light-weight voice coil)

As Figure 4.26 illustrates, I_f does not affect the notch frequency and the notch depth significantly; but, increasing I_f would decrease the peak frequency and the dynamic stiffness at peak frequency.

The effect of C_b on the dynamic stiffness of the electronic vibration isolator is shown in Figure 4.27.

Figure 4.27 Effect of C_b on the dynamic stiffness of the electronic vibration isolator (Design 4 light-weight voice coil)

As Figure 4.27 shows, increasing C_b decreases the notch frequency and the notch depth. It can be implied from Figure 4.27 that C_b is also an appropriate parameter to control the notch frequency of the electronic vibration isolator.

The effect of R_f on the dynamic stiffness of the electronic vibration isolator is shown in Figure 4.28.

Figure 4.28 Effect of R_f on the dynamic stiffness of the electronic vibration isolator (Design 4 light-weight voice coil)

It can be Implied form Figure 4.28 that, increasing R_f would decrease the notch depth; but it does not affect the notch frequency.

4.6.2.2 Electronic vibration isolator with a heavy voice coil

In this section the effect of the circuit parameters on the dynamic stiffness of the electronic vibration isolator with a heavy voice coil have been investigated.

The coil assembly mass of a commercial voice coil (LA43-30-000A) has been used for the simulation results in this section. For this particular voice coil, the mass of the coil assembly is $M_b = 0.726 \text{ Kg}$. Since the permanent magnet is fixed, its mass does not play any role in the dynamics of the system.

The circuit parameters of Table 4.4 have been used in this section.

Figure 4.29 shows the effect of C_t on the dynamic stiffness of the electronic vibration isolator.

Figure 4.29 Effect of C_t on the dynamic stiffness of the electronic vibration isolator (Design 4-heavy voice coil)

Figure 4.29 shows that, there are two notches in the dynamic stiffness diagram of the electronic vibration isolator, when the mass of the coil assembly is considered. The first notch is not deep enough to provide good vibration isolation.

As Figure 4.29 illustrates, the effect of C_t on the second notch is dominant. Increasing C_t would decrease the second notch frequency. The effect of C_t on the notch depth is negligible.

Figure 4.30 shows the effect of I_f on the dynamic stiffness of the electronic vibration isolator.

Figure 4.30 Effect of I_f on the dynamic stiffness of the electronic Vibration isolator (Design 4-heavy voice coil)

As Figure 4.30 illustrates, increasing I_f would decrease the second notch frequency. The effect of I_f on the first notch is negligible.

The effect of I_f on the dynamic stiffness of the electronic vibration isolator is significant. But it should be noted that all inductances have some inherent resistance as well. This inherent resistance increases dramatically with the inductance of the inductor and it tends to deteriorate the dynamic behavior of the electronic vibration isolator. So I_f is not an appropriate parameter to control the notch frequency of the electronic vibration isolator.

Figure 4.31 shows the effect of C_b on the dynamic stiffness of the electronic vibration isolator.

Figure 4.31 Effect of C_b on the dynamic stiffness of the electronic vibration isolator (Design 4-heavy voice coil)

As Figure 4.31 illustrates, increasing C_b would decrease the second notch frequency and increase the notch depth of the second notch. It has an adverse effect on the first notch depth.

Figure 4.32 shows the effect of R_f on the dynamic stiffness of the electronic vibration isolator.

Figure 4.32 Effect of R_f on the dynamic stiffness of the electronic vibration isolator (Design 5-heavy voice coil)

As Figure 4.32 shows R_f has an adverse effect on both notch depths.

According to the aforementioned simulation results, the electronic vibration isolator with design four provides acceptable vibration isolation.

It can be implied from Figures 4.25, 4.27, 4.29 and 4.31 that the proposed electronic vibration isolator is easily tunable.

Comparing the dynamic stiffness of the electronic vibration isolator (Figure 4.17) and the dynamic stiffness of the ideal engine mount it can be implied that the electronic vibration isolator performs similar to the ideal engine mount at low frequencies while its performance at high frequencies is even worse than the elastomeric vibration isolators. As mentioned in chapter two, this problem also existed in typical fluid mounts.

4.7 Resolving the poor performance at high frequencies problem in electronic vibration isolators with light-weight voice coil

In the case of the electronic vibration isolator with light-weight voice coil, the poor performance problem at high frequencies can be resolved. The remedy is to cut off the circuit when the performance of the electronic vibration isolator is worse than an elastomeric vibration isolator. With an open circuit the electronic vibration isolator (with a light-weight voice coil) will perform as a simple elastomeric vibration isolator. An on-off switch can be used to turn the circuit on and off at desired frequency (see Figure 4.33). Note that if the coil assembly is heavy, the dynamic stiffness of the electronic vibration isolator will dramatically increase even if the circuit is open. So the proposed solution will only work when the coil assembly mass is low.

Figure 4.33 Electronic vibration isolator with the on-off switch.

Figure 4.34 shows the dynamic stiffness of the electronic vibration isolator with the on-off switch.

Figure 4.34 Dynamic stiffness of the electronic vibration isolator with the on-off switch.

As Figure 4.34 shows, the high frequency performance of the electronic vibration isolator with a light weight voice coil is significantly improved using the On-Off switch.

4.8 Design five (extension of design four)

In chapter 3, a novel design for a double notch fluid mount was introduced. The design was based on adding a tuned vibration absorber (TVA) to the typical fluid mount.

As mentioned before, bond graph modeling technique can be used to find the equivalent of a system in the new energy domain; in this case we aim to find the electro-mechanical equivalent of the double notch fluid mount presented in chapter 3.

The bond graph model of the TVA mount is used just as a hint to design the appropriate circuit for the electronic vibration isolator. Actually, the exact electro-mechanical equivalent of a fluid mount may not exist.

The bond graph model of the TVA mount is shown in Figure 3.4. As Figure 3.4 illustrates the TVA mount model consists of two transformers. Since the electrical transformers usually have very high resistance and this resistance usually tends to deteriorate the dynamic stiffness properties of the mount, it would be desirable to ignore the second transformer. Note that the first transformer is replaced by the voice coil.

Figure 4.35 shows the physical model corresponding to design 5.

Figure 4.35 The physical model of the electronic vibration isolator (Design 5)

The bond graph model of the design 5 is shown in Figure 4.36.

Figure 4.36 The bond graph model of Figure 4.14

The state space equations derived from the bond graph model are as follows:

$$\dot{q}_2 = V_m \quad (4-39)$$

$$\dot{p}_8 = \gamma V_m - R_1 \frac{p_8}{I_1} - \frac{q_{10}}{C_1} \quad (4-40)$$

$$\dot{q}_{10} = \frac{p_8}{I_1} - \frac{p_{13}}{I_2} \quad (4-41)$$

$$q_{12} = \frac{p_{13}}{I_2} \quad (4-42)$$

$$\dot{p}_{13} = \frac{q_{10}}{C_1} - \frac{q_{12}}{C_2} - \frac{q_{16}}{C_3} \quad (4-43)$$

$$\dot{p}_{15} = \frac{q_{16}}{C_3} \quad (4-44)$$

$$\dot{q}_{16} = \frac{p_{13}}{I_2} - \frac{p_{15}}{I_3} \quad (4-45)$$

The matrix form of the state space equations is:

$$\begin{bmatrix} \dot{q}_2 \\ \dot{p}_8 \\ \dot{q}_{10} \\ \dot{q}_{12} \\ \dot{p}_{13} \\ \dot{p}_{15} \\ \dot{q}_{16} \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & -\frac{R}{I_1} & -\frac{1}{C_1} & 0 & 0 & 0 & 0 \\ 0 & \frac{1}{I_1} & 0 & 0 & -\frac{1}{I_2} & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{1}{I_2} & 0 & 0 \\ 0 & 0 & \frac{1}{C_1} & -\frac{1}{C_2} & 0 & 0 & -\frac{1}{C_3} \\ 0 & 0 & 0 & 0 & 0 & 0 & \frac{1}{C_3} \\ 0 & 0 & 0 & 0 & \frac{1}{I_2} & -\frac{1}{I_3} & 0 \end{bmatrix} \begin{bmatrix} q_2 \\ p_8 \\ q_{10} \\ q_{12} \\ p_{13} \\ p_{15} \\ q_{16} \end{bmatrix} + \begin{bmatrix} 1 \\ \alpha\gamma \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} V_{in} \quad (4-46)$$

$$F_{in} = \begin{bmatrix} K_r & \frac{\alpha\gamma}{I_1} & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} q_2 \\ p_8 \\ q_{10} \\ q_{12} \\ p_{13} \\ p_{15} \\ q_{16} \end{bmatrix} + [B_r] V_{in} \quad (4-47)$$

Table 4.4 shows the baseline parameters for the simulation.

Table 4.4 Baseline parameters for MATLAB simulation

K_r	B_r	γ	I_1	R
$2.05 \text{ e6 } \text{N/m}$	10 N.s/m	31.14 N/Amp	2.9 mH	2.6Ω
C_1	C_2	I_2	I_3	C_3
$200 \mu\text{F}$	$500 \mu\text{F}$	3 mH	1 mH	$400 \mu\text{F}$

Figure 4.37 shows the dynamic stiffness of the multiple-notch electronic vibration isolator.

Figure 4.37 the dynamic stiffness of the multiple notch electronic vibration isolator
Obviously there exist three notches in the dynamic stiffness of design 5. As it was discussed in chapter 4, having more notches will improve the vibration isolation characteristics.

4.9 Summary and Conclusions of chapter 4

In chapter 4 the electro-mechanical equivalent of a typical passive fluid mount was developed. It has been shown that the proposed electro-mechanical vibration isolator has similar dynamic stiffness characteristics as the typical passive fluid mounts.

Five different designs for the electronic vibration isolator were developed and the dynamic stiffness of the vibration isolator for each design was investigated.

Simulation results for designs one and two showed that due to the force sensitivity limitation of the available voice coils in the market, the notch depth would not be deep enough to provide good vibration isolation at the notch frequency.

The effect of using multiple voice coils on the dynamic stiffness of the electronic vibration isolation was studied in design three. Simulation results corroborated that using multiple voice coils in parallel, would enhance the dynamic characteristics of the electronic vibration isolator. But since this improvement is at the expense of adding more cost and weight to the vibration isolator, other designs were considered.

In design four a mechanical transformer (leverage) was added to amplify the effect of the voice coil force sensitivity and as a result enhance the dynamic stiffness of the electronic vibration isolator. Simulation results showed that using design four the notch depth would be satisfactory. In design four the mass of the voice coil has been also considered. Simulation results showed that when the voice coil mass is considered, one additional notch will appear in the dynamic stiffness diagram of the vibration isolator. This notch is caused by the resonance between the voice coil mass and the vertical stiffness (K_v). So the contribution of the electronic assembly on the dynamic stiffness will be negligible. In order to increase the contribution of the electronic assembly in the dynamic stiffness of the electronic vibration isolator, the circuit needs to be modified. It has been shown that using a modified circuit a new notch caused by the electronic assembly appears. The new notch is easily tunable by varying the electronic circuit parameters.

One of the advantages of the electronic vibration isolator is that its dynamic properties can be easily modified by changing the electronic circuit parameters, whereas in the case of passive fluid mounts this procedure was costly and time consuming. The electro-mechanical equivalent of the TVA mount which was introduced in chapter 3 was investigated in design five, to see whether it would improve the dynamic characteristics of the electronic vibration isolator. Simulation results showed that multiple notches will appear in the dynamic stiffness of the electronic vibration isolator using this design.

Finally, a sensitivity analysis for designs one and four was conducted to investigate the effect of the circuit parameters on the dynamic stiffness of the electronic vibration isolator.

4.10 Advantages of the electronic vibration isolator over the typical passive fluid mounts

Since in the proposed electronic vibration isolator the key parameters ($K_r, \ell_2/\ell_1, C_t, C_b, I_f$) are decoupled, its design procedure is much easier than that of the typical passive fluid mounts.

It can be implied from the sensitivity analysis section that the proposed electronic vibration isolator is easily tunable. For instance, since the effect of the capacitance of the circuit on the notch frequency is significant (see Figures 4.21, 4.23, 4.25, 4.27, 4.29 and 4.31), using a variable capacitor, the notch location can be controlled in real time, by changing the capacitance of the capacitor. So the retuning procedure of the electronic vibration isolator is very easy, whereas in the case of passive fluid mounts this procedure was costly and time consuming.

Finally, it has been shown that using an on-off switch in the electronic circuit would significantly enhance the dynamic stiffness of the electronic vibration isolator with a light-weight voice coil. So the overall dynamic performance of the electronic vibration isolator with a light-weight voice coil is improved over the typical passive fluid mounts.

Although the simulation results are satisfactory, they need to be validated via experiments.

Chapter5 Experimental validation of the electronic vibration isolator design

In chapter 4, five different designs for the electronic vibration isolator were proposed. Simulation results have shown that the electronic vibration isolator with design four can provide good vibration isolation.

The simulation results presented in chapter 4 need to be validated via experiment. In order to verify the simulation results for design four, a test fixture has been designed and fabricated. The design of Figure 4.14 was modified and four springs were added around the voice coil to keep the coil and the magnet concentric. The circuit was also simplified. With these modifications the theoretical dynamic stiffness of the electronic vibration isolator is as follows:

$$K^* = \frac{F_{in}}{X_{in}} = B_r \quad j\omega + (K_r + \alpha^2 K_b) + \frac{\left(\frac{\ell_2}{\ell_1}\right)^2 \gamma^2}{I_1 - 1/\omega^2 C - \frac{R_1 j}{\omega}} - M_t \omega^2 - M_b \left(\frac{\ell_2}{\ell_1}\right)^2 \omega^2 \quad (5-1)$$

The modified design of Figure 4.14 was fabricated and shown in Figure 5.1.

Figure 5.1 Photo of the electronic vibration isolator.

The fluidless electronic vibration isolator was tested using MTS 810- 100 KN test machine.

Design of Figure 5.1 is composed of two different subparts: the electronic part and the mechanical part. The mechanical leverage amplifies the effect of the electronic part. Theoretically the electronic part (the voice coil plus the circuit) acts as a variable stiffness.

The specifications of the electronic voice coil was provided by the supplier, but in order to make sure that the nominal values of the voice coil parameters are the same as the real values, we managed to derive the experimental value of the voice coil parameters.

5.1 Parameter Identification

The voice coil by itself was tested using MTS 810. Figure 5.2 shows the simplified physical model of the voice coil assembly plus the electronic circuit.

Figure 5.2 Simplified physical model of the voice coil assembly plus the circuit

5.1.1 Voice coil inherent resistance and inductance

I_1 and R_1 are the inherent inductance and the inherent resistance of the voice coil, respectively. The nominal values of I_1 and R_1 (provided by the voice coil manufacturer) are 2.9mH and 2.6Ω , respectively. But the experimental value of R_1 is 3.5Ω which has been measured using an Ohm-meter. Although the value of I_1 might be also different from its nominal value, since we did not have any equipment to measure the experimental value of I_1 , the nominal value is used in simulations. It should be noted that the effect of I_1 on the dynamic stiffness of the electronic vibration isolator is negligible, so its exact value is not of much interest.

5.1.2 Voice coil force sensitivity

The most important voice coil parameter is the force sensitivity. The nominal value (provided by the voice coil manufacturer) of the voice coil force sensitivity is

31.14 N/Amp . Due to the importance of this parameter on the dynamic stiffness of the electronic vibration isolator, we decided to measure this parameter experimentally.

In order to measure the experimental value of the voice coil force sensitivity, two fixtures were fabricated to hold the two parts of the voice coil (The permanent magnet and the coil assembly). The test was conducted using MTS 810 test machine. Figure 5.3 shows the photo of the voice coil assembly in the test machine.

Figure 5.3 Voice coil assembly in the MTS 810 testing machine

In order to measure the force sensitivity of the voice coil, the theoretical dynamic stiffness of the voice coil assembly was derived for different values of force sensitivity (γ). Then the experimental dynamic stiffness was compared with the simulation results. A very simple circuit has been used for the experiment. The circuit includes the inherent resistance and inductance of the coil and an electronic capacitor. The values of I_1 , R_1 and C used for the simulation are 2.9 mH , 3.5Ω and $18000 \mu\text{F}$, respectively. Figure 5.4 shows the dynamic stiffness of the voice coil assembly for three different force sensitivities and the experimental dynamic stiffness of the voice coil assembly.

Figure 5.4 Dynamic stiffness of the voice coil assembly (simulation and experiment)

As Figure 5.4 implies, the force sensitivity of the voice coil is very close to the value given by the manufacturer (31.14 N/Amp).

5.1.3 Test rig equivalent stiffness

In order to measure the equivalent stiffness of the electronic vibration isolator, the circuit has been cut and the electronic vibration isolator was tested using MTS 810 at 0.2 Hz . Figures 5.5 and 5.6 show the input displacement to the electronic vibration isolator and the measured force (By the load cell) versus frequency, respectively.

Figure 5.5 Input displacement versus frequency at 0.2 Hz

Figure 5.6 Measured force versus frequency at 0.2 Hz

By dividing the average amplitude of Figure 5.6 to the average amplitude of Figure 5.5 the average value of the electronic vibration isolator equivalent stiffness can be derived. Using the experimental results presented in Figures 5.5 and 5.6, the equivalent stiffness of the electronic vibration isolator is 0.301 kN/mm .

5.1.4 Coil assembly mass

As mentioned in section 4.5.1, the mass of the coil assembly has a significant effect on the electronic vibration isolator dynamic stiffness. In the fabricated test rig the mass of the coil assembly plus the plate holding the coil is approximately 1 Kg .

5.2 Electronic vibration isolator test results

In this section the experimental dynamic stiffness properties of the electronic vibration isolator will be presented.

The electronic vibration isolator was tested using MTS 810 test machine. Figure 5.7 shows the electronic vibration isolator in the test machine.

Figure 5.7 Electronic vibration isolator in MTS 810

In order to find out the effect of the electronic circuit on the dynamic stiffness of the electronic vibration isolator, three different tests were conducted.

5.2.1 Test one: The electronic vibration isolator without the electronic circuit

The first test was conducted for the electronic vibration isolator without the electronic circuit.

Figure 5.8 shows the experimental and theoretical dynamic stiffness of the electronic vibration isolator without the electronic circuit.

Figure 5.8 Dynamic stiffness without circuit

Since direct measurement of the equivalent damping of the electronic vibration isolator (B_r) is difficult, the simulation results has been presented for three different values of B_r . As Figure 5.8 implies, the equivalent damping of the electronic vibration isolator is around 6000 N.s/m . The B_r represents damping of the spring and the friction at the joints.

For the second test a simple electronic circuit has been used to investigate the effect of the electronic circuit parameters on the dynamic stiffness of the electronic vibration isolator.

5.2.2 Test 2: Electronic vibration isolator with electronic circuit ($C = 10000 \mu\text{F}$)

In this section the experimental results for the electronic vibration isolator with the electronic circuit has been presented.

The electronic circuit is composed of I_1 , R_1 (the voice coil inherent inductance and resistance) and C (capacitance). The values of I_1 and R_1 are 2.9 mH and 3.5Ω , respectively. The value of C is equal to $10000 \mu\text{F}$.

Figure 5.9 shows the dynamic stiffness of the electronic vibration isolator with the electronic circuit, both experimentally and theoretically.

Figure 5.9 Dynamic stiffness of the electronic vibration isolator with the electronic circuit ($C = 10000 \mu F$, Experiment and simulation)

As Figure 5.9 illustrates, the electronic vibration isolator with the aforementioned circuit parameters can provide a very shallow notch. Although the notch is not so deep, the existence of the notch proves the feasibility of the electronic vibration isolator concept.

The simulation results for three different damping values are presented in Figure 5.9. It can be implied from Figure 5.9 that, the equivalent damping of the electronic vibration isolator is around $6000 N.s/m$ which is in agreement with the conclusion of Figure 5.8. Figure 5.9 also implies that by decreasing the equivalent damping of the electronic vibration isolator, a deeper notch would be achieved.

5.2.3 Test 3: Electronic vibration isolator with electronic circuit ($C = 4400 \mu F$)

The third test is conducted to investigate the effect of the circuit capacitance of the dynamic stiffness of the electronic vibration isolator.

The values of I_1 and R_1 are $2.9mH$ and 3.5Ω (the same as previous). The value of circuit capacitance (C) is $4400 \mu F$.

Figure 5.10 shows the dynamic stiffness of the electronic vibration isolator with the above circuit parameters. The simulation results for three different values of B_r are also presented in Figure 5.10.

Figure 5.10 Dynamic stiffness of the electronic vibration isolator with the electronic circuit ($C = 4400 \mu F$, Experiment and simulation)

As Figure 5.10 illustrates, with the aforementioned circuit parameters, a very shallow notch appears in the dynamic stiffness of the electronic vibration isolator. It can be inferred from Figure 5.10 that it is due to the high equivalent damping of the electronic vibration isolator that the notch is too shallow. Figure 5.10 also implies that the equivalent damping of the electronic vibration isolator is around 6000 N s/m .

By comparing Figure 5.10 with Figures 5.9 and 5.8, it can be seen that the dynamic stiffness of the electronic vibration isolator is variable by changing the circuit parameters (see Figure 5.11).

Figure 5.11 Dynamic stiffness of the electronic vibration isolator for different circuits.

Additionally, by comparing Figures 5.10 and 5.9, it is implied that increasing the capacitance of the electronic circuit has decreased the notch frequency which is the same

5.3 Summary and conclusions of chapter 5

Chapter 5 was devoted on experimentally investigating the feasibility of the concept of electronic vibration isolator.

In order to compare the experimental and simulation results, parameter identification was conducted to find out the real values of the electronic vibration isolator parameters. These parameters have been used to simulate the dynamic stiffness of the electronic vibration isolator.

Three different tests were conducted to investigate the dynamic characteristics of the electronic vibration isolator.

In the first test, the electronic circuit has been turned off and the vibration isolator was tested without the electronic circuit. Experimental results showed that no notch appears in the dynamic stiffness of the electronic vibration isolator without the electronic circuit. The experimental result has been compared to the simulation results for three different values of B_r (equivalent damping). It has been concluded that the equivalent damping of the electronic vibration isolator is around 6000 N.s/m .

In the second and third tests the electronic vibration isolator was tested for two different values of the electronic circuit capacitance. The second test was conducted for $C = 10000 \mu\text{F}$. The third test was conducted for $C = 4400 \mu\text{F}$.

Experimental results showed that with $C = 10000 \mu\text{F}$ a very shallow notch appears in the dynamic stiffness of the electronic vibration isolator. Comparison between the experimental and simulation results imply that by decreasing the equivalent damping of the system the notch depth can be improved.

Experimental results of tests one, two and three have corroborated that the dynamic characteristics of the electronic vibration isolator can be varied easily by changing the electronic circuit parameters.

By comparing the simulation and experimental results for tests one, two and three it can be concluded that, the notch depth can be improved by decreasing the equivalent damping of the electronic vibration isolator. We did our best to minimize the equivalent damping of the electronic vibration isolator but it seems that there is still undesirable damping in the system. There are two main sources of damping in the electronic vibration isolator. A portion of damping comes from the hysteresis damping of the springs. The other source of damping is the friction in the joints. The friction damping can be minimized by lubricating the joints.

Although the notch provided by the electronic vibration isolator is not deep enough to provide good vibration isolation, the existence of a shallow notch proves the feasibility of the electronic vibration isolator concept. Future students can work on minimizing the damping of the system. By decreasing the equivalent damping of the electronic vibration isolator, a deeper notch would be achieved.

Chapter6 : Conclusions

The thesis was started by looking at semi-active engine mounts. Semi-active engine mounts are more effective than passive engine mounts in reducing cabin noise and vibration.

It is believed that the best cabin noise and vibration isolation is achievable at the notch frequency. Typical passive fluid mounts have just one notch frequency. According to the importance of the notch frequency, having multiple notch frequencies would be very desirable. A new semi-active engine mount based on the use of tuned vibration absorbers (TVA) was proposed. The simulation results, shown in chapter 4, showed that this design provides two notches in the dynamic stiffness of the engine mount and adding the TVA will make the engine mount more tunable because the TVA mass and stiffness have significant effect on the notch frequency and depth. The effects of changing the TVA parameters were also studied.

Fluid mounts can provide good vibration isolation for many typical applications. They have been used in industry for years. However there are some shortcomings associated with passive fluid mounts.

The design procedure of fluid mounts is complicated and it is somewhat based on trial and error. This procedure is costly and time consuming.

Additionally, as time goes on, the dynamic characteristics of the engine as well as the fluid mount properties will change; so the mount needs to be retuned after a period of operation. The retuning procedure of fluid mounts is costly because it needs disassembling the mount and redesigning its internal parts.

Since the complicated fluid structure interaction in fluid mounts is responsible for the major part of the complicated design procedure of them, it makes all the sense to remove the fluid and replace it with a more tunable mechanism.

The main objective of this project was to design an easily tunable fluidless electronic engine mount. The feasibility of fluidless electronic engine mount was examined in chapter 4.

Five different designs for the fluidless electronic vibration isolator were proposed in chapter 4. Simulation results have corroborated that, design four can provide good vibration isolation.

In order to verify the mathematical simulation results presented in chapter 4 experimentally, a test fixture based on design four of chapter 4 were fabricated. The experimental results for the electronic vibration isolator were presented in chapter 5.

Experimental results presented in chapter 5 have shown that the dynamic stiffness of the electronic vibration isolator can be changed by changing the electronic circuit parameters. However, because of the high damping of the test fixture, the notch of the dynamic stiffness is not deep enough. We did our best to minimize the damping of the electronic vibration isolator, but there is still undesirable damping in the system. By comparing the simulation and experimental results, it has been shown that by decreasing the overall damping of the electronic vibration isolator, the notch depth would be improved.

It should be noted that in the case of typical fluid mounts, much effort is devoted to minimize the damping in the fluid mount. The damping sources in fluid mounts are mainly the fluid losses through the inertia track and the hysteresis damping of the rubber.

Experimental results presented in chapter 5 have corroborated that the dynamic characteristics of the electronic vibration isolator can be changed easily by changing the electronic circuit parameters such as the capacitance. It is a big improvement over typical passive fluid mounts where changing the dynamic characteristics of them after manufacturing is costly.

Additionally, since the fluid has been removed in the electronic vibration isolator, its design procedure is easier.

One of the nice surprises or outcomes of this project was that the device we designed can be not only used as an electronic vibration isolator, but as a variable stiffness device. This was demonstrated in Figure 5.11.

Future students can work on minimizing the damping in the electronic vibration isolator. Different electronic circuit designs can also be considered to improve the dynamic characteristics of the electronic vibration isolator.

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