A Study of Active Engine Mounts

Examensarbete utfört i Reglerteknik
vid Tekniska Högskolan i Linköping
av

Fredrik Jansson & Oskar Johansson

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A STUDY OF ACTIVE ENGINE MOUNTS

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Titel Title	Studie av aktiva motorkuddar A Study of Active Engine Mounts
Författare Author	Fredrik Jansson and Oskar Johansson

Sammanfattning
Abstract
Achieving better NVH (noise, vibration, and harshness) comfort necessitates the use of active technologies when product targets are beyond the scope of traditional passive insulators, absorbers, and dampers. Therefore, a lot of effort is now being put in order to develop various active solutions for vibration control, where the development of actuators is one part.

Active hydraulic engine mounts have shown to be a promising actuator for vibration isolation with the benefits of the commonly used passive hydraulic engine mounts in addition to the active ones. In this thesis, a benchmark of actuators for active vibration control has been carried out. Piezoelectric actuators and electromagnetic actuators are studied further and two methods to estimate parameters for electromagnetic actuators have been developed. A parameterized model of an active hydraulic engine mount valid for frequencies from zero to about 300 Hz has also been developed. Good agreement with experimental data has been achieved.

Nyckelord
Keyword
active engine mount, actuator, active vibration control, electromagnetic actuator model, parameter estimation

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This thesis is the final part of our Master of Science degrees in Applied Physics and Electrical Engineering at Linköping University. It could not have been completed without help from a great number of people. We wish to take this opportunity to express our appreciations for their help throughout this project.

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Fredrik Jansson and Oskar Johansson

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Chapter 1

INTRODUCTION

To achieve better NVH (Noise, Vibration and Harshness) comfort, the development and use of ANVC (Active Noise and Vibration Control) systems is necessary, when goals and visions are beyond the scope of traditional passive insulators, absorbers and dampers.

Consumers demand better ride-comfort in their cars, but use of passive solutions would increase the weight. At the same time, higher safety demands, greener cars and lower fuel consumption demands lower weight of the car. Introducing ANVC systems in automobiles will to some extent solve these contradicting demands. This thesis suggests solutions to this contradiction, and discusses actuators, sensors and active engine mounts. The purpose is to construct a complete model of an active engine mount. In the way to accomplish that a benchmark of the today existing actuator technologies suitable for use in ANVC has been carried out and is presented in Chapter 2. Chapter 3 is short overview of the sensors used today in active vibration control with the intention to introduce the area and should not be seen as a complete benchmark.

This Master of Science thesis work has been carried out at Volvo Car Corporation in cooperation with the Division of Control and Communication, Department of Electrical Engineering, Linköping University.

1.1 Engine Mounts

The first and most obvious role of the engine mount is to support the engine and transmission. Another important role of the engine mount is vibration isolation, to reduce the dynamic force, vibrations, transmitted from the engine to the frame.

The vibrations that the mount has to handle come from two different sources. The engine vibrations that are to be isolated are typically in the region of 30-200 Hz, with amplitudes generally less then 0.3 mm. The other source is the frame that is affected by road surface irregularities via the suspension system. These frequencies are typically in the region of 1 to 30 Hz and have an amplitude greater then 0.3 mm [58].

The ideal dynamic stiffness for an engine mount is viewed in Figure 1.1 [47]. For low frequencies, high damping for shock excitation is needed to prevent engine bounce and give driving stability. For example it is desirable that the engine follows the frame when the car is going over a bump. At higher frequencies, low damping is desirable to isolate low-amplitude engine vibrations caused by engine disturbances.

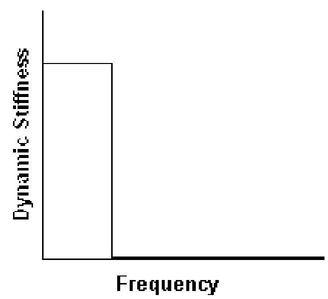

Figure 1.1 Dynamic stiffness of an ideal engine mount

The use of hard rubber would give high stiffness and would be good for providing firm support for the engine and give good driving stability. However, the use of hard rubber enables engine vibrations to be easily transmitted to the chassis. To in a further extend isolate vibrations from the chassis the use of soft rubber is better. Resolving these contradictory needs of driving stability and vibration isolation different solutions have been tried out. Today most manufacturers use passive hydraulic engine mounts that can give a frequency dependent dynamic stiffness. But the increasing demands on the engine mounts are now becoming out of scope for the passive solutions. This makes it interesting to investigate the possibilities with use of active engine mounts.

The difference between a passive and an active mount is that an active mount makes it possible to provide controlled energy to the system. The use of active engine mounts has many benefits. For instance, they can adapt to manufacturing differences and changes during the lifetime of the mount. Structures can be made lighter and parts can be eliminated. It would be interesting to study the effects of removing the engine balance shaft by the use of active parts. Even though there are a number of benefits with an active hydraulic engine mount the main advantage is improved performance in vibration isolation.

1.2 Objective

The engine suspension system is a promising area for application of AVNC. Therefore, control systems have been developed but in many cases actuators, sensors and active engine mounts have been assumed ideal. In reality, the characteristics and also the working principle of the actuators, sensors and the complete active engine mounts have a great impact on control algorithm design and the system performance. The questions that this thesis work aims to answer are:

Which actuator and sensor technologies and principles can be used in AVNC, for example in active engine mounts and what characterises them?

Which are the characteristics of an active engine mount?

Is it possible to crate a model of an active engine mount that are valid for both lower (<30 Hz) and higher frequencies (30 Hz to ~300 Hz)?

1.3 Limitations

The work is limited to actuators dealing with single axes isolation. The focus is on actuators and complete active engine mounts and we will only briefly discuss sensors and their impact on the complete system.

1.4 Approach

To fulfil the objectives the project was divided in three parts:

Part 1: Investigation of existing actuators and sensors for use in active vibration control concerning:

- Technologies.
- Working principles (different implementations of the technologies).
- Technical specification/strengths and weaknesses, e.g. force and frequency ranges, sensitivity, max amplitudes.

Part 2: Characterization and generation of parameterized models of selected actuators:

- Generate parameterized models of selected actuators.
- Identification of parameters.
- Validate the actuator models against experimental data.
- Develop a method for use in specification dependent design.

Part 3: Characterize and generate parameterized model of a complete active hydraulic engine mount:

- Generate parameterized model of a complete active engine mount.
- Validate the complete active engine mount model against experimental data.

In Chapter 2 the benchmark of actuators conduced in part 1 is presented. Chapter 3 is a brief discussion of the most commonly used sensors in AVNC. In Chapter 4 the different actuators are compared and other factors that influence the AVNC system are briefly discussed. Chapter 5 continue the discussions about piezoelectric and electromagnetic actuators. The two are modelled and guidelines for voice coil actuator design from frequency-domain specifications are given. Chapter 6 is the main chapter and present a complete model of an active engine mount. In the last chapter, Chapter 7, conclusions and recommendations are found.

Chapter 2

ACTUATOR TECHNOLOGIES AND PRINCIPLES

One of the objectives is to find an actuator that is suitable to be used in active vibration control. Depending on the application, the actuator requirements can be very different in size, power, energy source, achievable forces, frequencies, displacements etc. An actuator is a device that transforms energy into controllable motion and/or force, which performs useful work on the environment.

An active vibration control system consists of a sensor and an actuator together with a control unit. The development of the systems is often limited by the chosen actuator technology. The usual effects that limit the potential of actuators are restrictions of bandwidth, displacement and placement, according to Hersle and Svensson [4]. Furthermore, difficulties with producing actuators that manage high temperatures and mechanical stress have been confirmed.

In the field of active engine-vibration isolation some principles have been well tested over many years, while research on other principles is just beginning. The development of new actuator principles has been forced by the space industry's high demands on systems and actuators.

Smart materials are materials that can change shape and/or have the ability to affect their characteristics with an applied voltage, change in temperature or magnetic field. Examples of well-known smart materials are shape memory alloys, electrorheological fluids, electrostrictive, magnetostrictive materials and piezoelectric materials. Components that are based on utilization of smart materials have shown to be very useful in active vibration damping systems, both as actuators and sensors.

In the field of active vibration control, piezoelectric ceramics, electrostatic, magnetostrictive, electromagnetic, and hydraulic devices are used as actuators. In both [30] and [39], each of them meets the classification of a fully-active actuator, which is defined as:

"A fully-active actuator is able to supply mechanical power to its system".

The other group of actuators is semi-active, which dissipates energy similar to passive elements. The difference is that semi-active actuators can adjust their passive mechanical properties by a control signal.

In this thesis, an actuator technology is defined as a physical phenomenon to create motion and/or force, and a principle is a realization or an application of a technology. The following sub-chapters deal with promising actuator technologies and principles, selected actuator technologies and principles, and other actuator technologies and principles. The first sub-chapter introduces technologies possible to be used for active vibration control applications. The next sub-chapter deals with the two actuator technologies we have chosen to investigate more. And the third treats actuator technologies that are not yet ready to be used in ANVC-applications. We give some information about different actuator technologies, such as benefits, drawbacks and commercial products. For more detailed information refer to Appendix A.

2.1 Promising Actuator Technologies and Principles

The intention of this sub-chapter is to discuss some actuator technologies and principles that are promising for use in active vibration control. We are told by researchers that some technologies face a very good future and that we have still not seen their actual potential. Several technologies have a good possibility to succeed as actuators for active vibration control. These include electrorheological, electrostatic, electrostrictive, hydraulic, magnetorheological, magnetostrictive, and pneumatic.

2.1.1 Electrorheological and Magnetorheological

In 1947 Willis M. Winslow discovered that the flow resistance of certain fluids increases with field strength when exposed to alternating current electric fields in the order of 4 kV/mm [17]. The response of the electric field of these fluids is very quick, in the region of milliseconds. When an electrorheological fluid is exposed to an electric field it changes the viscosity of the fluid or its flow rate (rheology). Electrorheological fluids consist of non-conducting fluid and micro-sized polarized particles.

This technology is very sensitive to ambient temperatures, separation between fluid and particles, and wear due to abrasion from particles in the fluid. In the worst case every one of them can lead to device failures. The electrorheological fluids that exist today can operate at higher temperatures than their predecessors. These fluids are improving as new objectives are being set by the customer. This technology has a great potential according to some laboratory tests, but still there are some problems with quality of the available fluids and their long-term stability. It is still difficult to reproduce the manufacturing process for electrorheological fluids, because the stability of the electrical and rheological properties of the fluids may change over time. They also have high power consumption and are sensitive to moisture. There are not many commercial products and devices based on this technology despite the fact that it has been known for over 50 years. Electrorheological devices are primarily used in macro scale applications. According to Ushijima, Takano and Kojima [25], a semi-active engine mount based on utilization of the electrorheological fluid has been tested. An electrorheological actuator can be built very simply, because only fluid and electrodes are necessary to create actuation. High bandwidth can be achieved according to Lind, Kallio and Koivo [28].

Magnetorheological fluids operate very much like electrorheological fluids, but their flow rate is instead controlled by the strength of a magnetic field. Magnetorheological dampers are available, commercially, from Lord Corporation based on hydrocarbon, silicone or water fluids.

In both [20] and [26], the electrorheological actuator is tested to improve a passive hydraulic engine mount. In [20] was an active engine mount prototype working with electrorheological fluid or ferrofluid (magnetorheological fluid) tested. Both fluids have shown to be controllable, but there were some problems with resonance in the inertia track due to high viscosity. Therefore, Gennesseaux tried to minimize the amount of particles in the fluid, but this resulted in drastic changes in control. In [26] was an adaptive control system tested, with the result that stiffness and damping could be controlled, but the vibration amplitude affected the characteristics of a controllable damper.

2.1.2 Electrostatic

An electric field and a force emerge between positive and negative charged particles. The property that electrostatic fields arise and disappear rapidly is utilized for very fast operational speed. Through special structures it uses the electrostatic force to create motion.

Electrostatic actuators are not affected by ambient temperatures and are often used in active vibration control. These devices have extremely low current consumption, because of high efficient actuation. They can generate great forces, but the forces are generally limited to very short distances. To preserve a given force for a longer distance, higher voltage is required. A dust particle can, at worst, cause breakdown due to a small air gap. Short stroke is another limitation of linear electrostatic actuators.

Electrostatic devices are widely used in small regions. One simple commercial electrostatic actuator which is used commonly in micro-electromechanical systems (MEMS) is the parallel plate capacitor [32]. The lower plate is fixed, while the upper plate can move. Most electrostatic actuators are still at the research stage.

2.1.3 Electrostrictive

Electrostriction refers to the process in which a material is deformed when it is exposed to an electric field. Commercially available actuators exist, which are based on electrostrictive crystals. They use a stack design in which displacement is a superposition of the strain from several thin crystal layers. Electrostrictive crystals are not polarized like piezoelectric ceramics. The displacement direction depends on the voltage applied: positive or negative. Electrostrictive ceramics produce a strain, which is in the same order as the strain from piezoelectric ceramics. Electrostrictive ceramics and piezoelectric ceramics have different advantages. The electrostrictive ceramics provide better characteristics of hysteresis and creep (slow deformation), but their strain sensitivity to temperature is much higher than for piezoelectric ceramics. Most commercial electrostrictive actuators exist in the micro region. They use materials that are based upon solid solutions of PMN (lead magnesium niobate) and PT (lead titanate).

Swanson [22] has investigated whether electrostrictive actuators can be used in passive hydraulic engine mounts, but they can still be too costly and produce too small displacement outputs for many engine mount applications. It is proved that these devices can be built with high stiffness, high forces and high frequencies (several kHz). They can also be built extremely compact.

2.1.4 Hybrid

Sometimes it is possible to merge two or more technologies together to utilize the advantages of each. Hybrids are used to produce compact devices. Piezoelectric is a common technology in hybrids because a piezoelectric actuator produces very large forces with small displacement. Together with some other technology, such as hydraulic, it is possible to convert force to displacement through special structures. Smart materials such as electrostrictive, magnetostrictive and piezoelectric have proven their usefulness in precision applications, but they are normally not considered for use in actuators that require large linear displacements. Anderson, Linder and Regelbrugge [38] present a hybrid actuator that combines smart materials, specifically piezoelectrics, with a closed hydraulic system. This actuator produces large displacements, without affecting the high force capacity. The net power output is high. The hydraulic system acts as a transmission to convert smart material output to useful mechanical work. "Solid-fluid hybrid" actuation is a common name in articles concerning smart material-hydraulic actuation.

According to Hallinan, Kashani and Bartsch [16], it is possible to create an electrostatically-driven phase change actuator for vibration control, which is capable of generating forces over 300 N with displacements of a few millimeters and extremely rapid response time for pressure (force). This micro-actuator consists of two electrodes with a porous ceramic between them, and a vapor cavity and diaphragm above.

2.1.5 Hydraulic and Pneumatic

Hydraulic and pneumatic devices can establish a counterforce without any energy consumption, and they have damping capabilities. Pneumatic actuators are quite similar to hydraulic actuators, but the big difference is that hydraulic use a fluid and pneumatic use gas or air. These actuators are both often simple devices with few mechanical parts.

Fully-active and semi-active hydraulic actuators exist, they are light in weight in relation to the power they can admit and emit, and they have fast reaction time. Fully-active actuators have been used in commercial products for damping. One application where they have been used is to damp vibrations in helicopters. Valves or pumps control the pressure in hydraulic actuators. In a valve-controlled actuator, there is a servo-valve that controls the flow while the pump works at constant power. In a pump-controlled actuator the flow is changed with the power of the pump. Valve-controlled actuators are quicker, but pump-controlled actuators are more efficient.

This technology is often used in large sizes with high output forces. A benefit is that they can, in special situations, have zero friction and nearly backlash- (recoil-) free power transmission [28]. There are small piston and rod systems with special seals and coatings. For instance, Teflon is widely used. A hydraulic actuator consists of piston, rod systems, metallic bellows and rubber components.

Bormann, Ulbrich, and Abicht [15] have developed a fist-sized hydraulic actuator that can apply forces of several kN with displacements up to ±1 mm and with a frequency range of up to 100 Hz. The objective was to give flexibility to the system together with high radial stiffness; this is accomplished through two annular membranes that are connected to the lower and upper body. Servo-valves control the oil pressures in the two chambers.

Swanson [22] has investigated the suitability of servo-valve hydraulic actuators for active extension to passive hydraulic engine mounts. These devices can produce extremely high forces, but they are costly and have limited bandwidth. They also require high maintenance and a hydraulic power supply. According to Gennesseaux [20], active hydraulic systems were in engine mounts discarded, because of their cost and the need for high pressure.

According to Stein [18], an active control system with a pneumatic spring actuator has been developed to improve vibration in heavy vehicle seats, including agricultural tractors and off-road vehicles.

2.1.6 Magnetostrictive

The magnetostrictive effect comes from a ferromagnetic crystal that changes its shape when subjected to a magnetic field. Magnetostrictive actuators output large forces and have quick dynamic responses, but they have small displacements: typically less than 1 percent of total length. This is the main disadvantage of magnetostrictive actuators. Devices can harness these high forces to create moving mechanisms. They require high magnetic biases for operation, but can operate at low voltages. Magnetostrictive materials are generally very brittle, difficult to manufacture and develop heat during operation, which must be dissipated to prevent damage to the actuator. High ambient temperatures generally decrease the performance, for some, over 400 ºC.

Magnetostrictive materials transform electrical energy to mechanical motion very rapidly via an induced magnetic field from the coil. Other advantages with this technology are that these types of actuators have a long life and can be used in high-frequency and high-precision applications. These actuators are quite complicated both in mechanical and electrical construction. To control the magnetic field, a magnetic coil is required. Bigger magnetostrictive materials appear nonlinear, for which special models must be worked out.

Magnetostrictive materials generally are low weight with high extension, and the characteristics do not change in time. These materials give new possibilities to development of components with high density, rapid reaction time and extremely good precision. These materials have been used successfully in actuators and sensors for vibration control. They have also been used in hearing aids, operated into teeth and in microsurgery.

In [28] it is explained that the actuator is typically composed of a magnetostrictive rod (for example Terfefenol-D) placed inside a coil. According to Gennesseaux [20], magnetostrictive materials are still too costly to be used as actuators in active engine mounts. Therefore, they are limited to be used in military or space applications.

2.2 Selected Actuator Technologies and Principles

One of the objectives was to find the most suitable actuators to be used in active vibration control and implemented in an active engine mount. After studying different technologies the conclusion was that electromagnetic and piezoelectric technologies were the most interesting, because they exert great forces, have linear relation between electrical quantity and force, wide bandwidth, quick responses, and are well investigated. See section 4.1.1 for details on choice, i.e. comparison between different technologies due to different characteristics. One principle of electromagnetic technology and one principle of piezoelectric technology are investigated further in Chapter 5. Existing commercial actuators, from different suppliers, based on different principles of electromagnetic and piezoelectric technologies are presented with technical data in Appendix D.

2.2.1 Electromagnetic

When electric current is flowing through a conducting material an electromagnetic field is produced. The actuation physics is based on the magnetically induced motion caused by the interaction between a coil and a magnet. This technology can generate attractive and repulsive forces, which are proportional to the current in the conductor. Electromagnetic actuators are well investigated and these devices are used in many different applications. Typical examples of principles that use this technology are electromagnetic motors, solenoids and voice coil actuators.

An electromagnetic actuator has very quick operating speed, scale ability, extreme positioning accuracy that is independent of load or velocity and can operate over a wide range of temperatures (up to approximately 180 degrees). The performance of an electromagnetic actuator is primarily limited by the properties of the material used in constructing it. These actuators are highly efficient in converting electrical energy into mechanical.

Electromagnetic actuators also exist in the micro and nano region, but it is complicated to build small electromagnetic coils. A voice coil is the most commonly used linear motor, because of good characteristics such as high force and good displacement. Voice coils are of two types: moving coil and moving magnet. They are well tested and often cheaper than other alternatives. This technology produces the fastest actuators in electro-mechanics according to Compter [55]. To obtain the highest available force it is important that the current conductor and the magnetic field are perpendicular.

Fursdon, Harrison, and Stoten [40] have proposed an active engine mount with an electromagnetic actuator using a self-tuning cancellation algorithm. The engine mount is a combination of a conventional hydraulic engine mount and a voice coil actuator. The actuator is turned on for frequencies over 25 Hz. It generates a force output greater than 40 N over a 25 to 200 Hz frequency range. The coil is attached to a diaphragm, which replaces the decoupler. By controlling the motion of the diaphragm, the pressure in the upper chamber is changed and as a result the net output force from the mount to the chassis and engine is controlled. The active engine mount is capable of reducing road induced engine shake and active cancellation of engine induced chassis vibration.

Swanson [22] concludes that voice coils and solenoids are interesting for use as actuators in active engine mounts. They fulfill the requirements stated for the actuator concerning force, stroke and power. And it is mentioned that it requires a force of 20 N to reduce vibration at 60 Hz. Voice coils are compact, have high frequency bandwidth (up to few kHz) and generate output forces that are both linearly proportional to current and independent of stroke. Solenoids generate nonlinear forces to current and stroke, but they offer higher force outputs in smaller package than voice coils.

2.2.2 Piezoelectric

In 1880, the brothers Pierre and Jacques Curie discovered the piezoelectric effect, when certain crystalline materials (ceramics) are compressed they produce a voltage proportional to the applied pressure. Conversely, when an electric field is applied across the ceramics they mechanically deform. This is known as the indirect piezoelectric effect. Piezoelectric devices are used commonly as both actuators and sensors, with success accordingly to the indirect and direct effects.

Piezoelectric actuators have some advantages such as good resolution, high output force, and quick response to input voltage changes. The energy consumption for keeping a load at a fixed position is very low. The only limiting factor in the positional resolution is power supply noise. Piezoelectric devices have two sources of loss, these are mechanical and electrical.

Piezoelectric actuators still have some problems with small total strain and hysteresis, and they are expensive in comparison to other technologies. Another drawback is that drift and lifetime are not even known by suppliers. Furthermore, there is a problem to obtain durable attachment between the piezoelectric actuator and the structure. Cyanocrylates (super glue) and two-part epoxies have proven useful in many applications. Manufactures have different solutions to specific bonding requirements, such as extreme ambient temperatures, unusual shear stress requirements or the type of metal surface that are to be joined together.

The principles that can be of interest are amplified piezoelectric actuators and multilayered stacks as they have bigger strain than the other principles. In [19] piezoelectric materials are discussed and the prospect of them to be used for impact applications in future automobiles.

According to Gennesseaux [20] a piezoelectric can hardly be used as an actuator in an active engine mount for low frequencies and large amplitude vibrations, because of very limited stroke.

Ushijima and Kumakawa [23] have proposed and tested an active engine mount with piezoelectric ceramic actuators. The piezoelectric actuators have some disadvantages, such as very small displacement and large temperature dependency, but they have significant advantages too, such as large output forces and high speed of response. To provide enough displacement required for the active mount the actuators are built with alternating ceramic layers and electrodes along with a hydraulic multiplication mechanism. The engine mount is limited to idling vibration control, but the high speed of response gives it a significant potential even in higher frequency regions of booming noise and road noise. In [24] an active engine mount is introduced, for large amplitude idling vibrations working with piezoelectric actuator. It is quite similar to the one discussed in [23]. The engine mount is able to absorb large amplitude vibration, such as the idling vibration of a two-cylinder engine.

2.3 Other Actuator Technologies

This sub-chapter deals with remaining technologies that so far are used for other purposes than active vibration control. Diamagnetism and electrohydrodynamic are referred to only in the Appendix A.

2.3.1 Electrochemical

Electrochemical technology transforms electrical energy into mechanical energy and is based on using the electrolysis field to build up a gas pressure of an aqueous solution. Electrochemical actuators use forces that originate from an electrochemical cell, which consists of two metal electrodes immersed into an electrolyte with electrode reactions occurring at the electrode-solution surfaces.

Electrochemical actuators are used in the micro region. Electrochemical actuators have the big benefit of a liquid to gas phase transformation, which gives the huge change in volume and/or pressure that can be obtained. They have been investigated to be used for regulating the pressure in the eye [36].

According to Cameron and Freund [27], an electrochemical phase transformation actuator with great theoretically efficiency, strain and stress has been developed. It is based on the electrolysis of water to oxygen and hydrogen. The intention is to find new actuation methods based on electrolysis of liquids and gases. This method enables the making of smaller, more efficient and lighter machines from micro-region and larger.

2.3.2 Phase change

When certain phase change materials experience changes between phases such as solid, liquid and gas, they force dimensional changes to the system. These dimensional changes are expansion or contraction. Phase change actuators are built to utilize the forces exerted by the materials, and they generally demonstrate full reversibility. These devices can create a phase change effect over a wide range of speeds and pressures through electrical, thermal or ultrasonic input; depending on which material is used.

Phase change actuators can exert great forces, but the forces are generally across very short distances. Higher voltage is required if the actuator is to manage to maintain a given force for longer distances. These devices are sensitive to ambient temperature, but they give highly efficient actuation because of the extremely low current consumption. Phase change materials are fairly new, and they attract interest within many different applications.

2.3.3 Pyrotechnical

Explosive and pyrotechnic devices transform a small input of mechanical or electrical energy into a higher level of mechanical or thermal energy that is applied to perform practical work on a one-time basis. Therefore, they are not used for vibration control. These devices store chemical energy until it is released by mechanical or electrical input. The original gas generators used in airbags consist of a pyrotechnical charge only, while the new generation of gas generators by Autoliv is a hybrid of a canister of compressed argon gas and a pyrotechnical charge.

2.3.4 Shape Memory

In 1932, the Swedish physicist Arne Olander discovered that an alloy of gold and cadmium could be plastically deformed when cooled and then be heated to return to the original dimensional configuration. Shape memory actuators are constructed to use these property changes in the material when the temperature reaches certain transition levels. This is known as the shape memory effect. This transformation involves changes in strength of the material, deformability and Young's Modulus, as well as the ability of the material to return to a previously conditioned physical shape. Shape memory belongs to a group of materials that are called smart materials, to be able to use shape memory as a phase change actuator through heating, the material must first be educated.

There are shape memory alloys and shape memory polymers, but the only ones of interest to us are shape memory alloys. Polymers require a built-in squeezing mechanism to be used for purposes other than on one-time basis. Shape memory alloys have some benefits such as considerable temperature-dependent expansion/contraction, relatively linear control, very high stress (often over 200 MPa), arbitrary shapes and simple actuation, and have achieved a million cycles in laboratory tests, but life time is still fairly uncertain. Disadvantages are that special alloy materials are needed: high temperature annealing, low efficiency (energy conversion efficiency is approximately 3 %) and long-term thermal constants.

Shape memory alloys have strain of 5-8 % depending on the number of cycles [28]. The main disadvantage with shape memory alloys is the slow speed of response. If they could be made smaller they would be faster, since heating and cooling is faster with small devices. Actuators based on this technology can only be used in low frequency and low precision applications. They are not yet suitable for active vibration control [30].

Shape memory alloys have been tested to improve passive-hydraulic engine mounts, through SMA-wires inside rubber bellows [37]. Changes in current will affect the SMA that changes its temperature and phase. The upper chamber compliance can be changed by 50 percent, and it is shown that this is an effective parameter for use in an adaptive mount, where it is sufficient with a simple on/off control. The dynamic stiffness is reduced by 30 percent for low frequencies and 40 percent for high frequencies.

2.3.5 Thermomechanical

Thermomechanical devices are built to utilize the physical dimensional changes (expansion or contraction) as they undergo temperature changes without changing their phase. Thermal changes are the result of the conduction of heat energy into a material. These changes may occur over a wide range of speeds.

Since the material need is sensitive to changes in ambient temperature, insulation may be required. To improve the reverse transformation, the actuator would require some passive or active cooling system. There are different methods to induce temperature changes into the system: resistive heating at low voltages, thermally, radioactively, or ultrasonically.

The thermomechanical actuator can be more useful in the micro region. The heat dissipation is directly related to the volume to be cooled. Therefore, thermal cycling occurs faster in micro devices than in macro devices. A common micro actuator using the thermomechanical principle is a bimetallic cantilever.

Chapter 3

SENSOR TYPES

The intention with this chapter is to briefly discuss suitable sensors for measuring vibrations. According to both [5] and [30], the piezoelectric accelerometer is the most widely used sensor for vibrational measuring, that is the reason why our focus is within piezoelectric accelerometers. Another popular sensor is the piezoelectric force sensor, but neither piezoelectric acceleration nor piezoelectric force sensor can measure D.C. components, or very low frequency vibrations. According to Colla [30], non-piezoelectric devices are generally based on inductive, capacitive or optical technologies. In Appendix B, other types such as optical sensors are presented. According to Sensor technology information exchange (Sentix) [56], the definition of a sensor is:

"*A device or system that responds to a physical or chemical quality to produce an output that is a measure of that quality*".

The active vibration control system consists of three parts, actuators, control units and sensors. The most relevant quantities to be measured are position, velocity, acceleration, strain and force, see [30]. There exist a number of different commercial sensors with varying price and performance. In general, sensors used in active vibration control systems exist in three forms, they are point acting sensors, arrays of point sensors or continuously distributed sensors. Distributed sensors measure over an area and the motion are integrated over the segment in the structure.

According to Lindahl and Sandqvist [5], sensor can be seen as three main parts, a sensing element, a transducer and a device for internal signal processing, see figure 3.1. The sensing element is directly affected by the input quantity and its task is to transform the physical input to a dimension, which is possible to be transformed into an electrical signal by the transducer. It is the sensing element that determines the nature, selectivity and sensitivity of the sensor. The internal signal processing consists of electrical equipment, which transforms the electrical signal to a useful output signal. (For example, the sensing element can be a diaphragm, which is deformed in proportion to the surrounding pressure. The transducer can be a tensiometer that converts the deformation to a change in resistance and the internal signal processing can be an amplifier.)

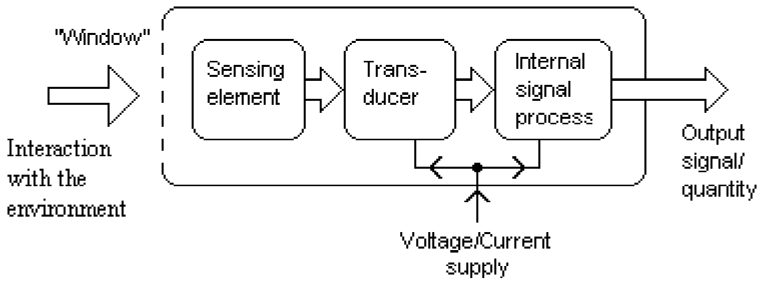

Figure 3.1 Block diagram of a sensor

The choice of sensors is dependent on the application for which they are going to be used. The selection of sensors is influenced by the price, performance, and the requirement for the systems, such as need of precision, reliability, etc. In Chapter 4, the relationship between actuators and sensors is discussed, and their affect on the closed loop etc. Bandwidth, sensitivity, and price are three important properties of a sensor.

3.1 Piezoelectric sensors

Piezoelectric sensors are widely used as force and accelerometer sensors. When a piezoelectric material is subject to a force it deforms elastically and generates electrical charges, this is the direct piezoelectric effect. The change in charge that is detected on the surface of the material originates from rotation of the crystals [13]. To detect the change in charge two conductive coatings are applied to the material. It exhibits good linearity and when the force changes direction the charge changes polarity. There exist several different designs of piezoelectric sensors based on utilization of the different piezoelectric effects, which are forces that affect the material to produce a parallel electrical field (length effect) or perpendicular electrical field (side effect), and shear forces of the plates producing a perpendicular electrical field (shear effect) [49], see figure 3.2.

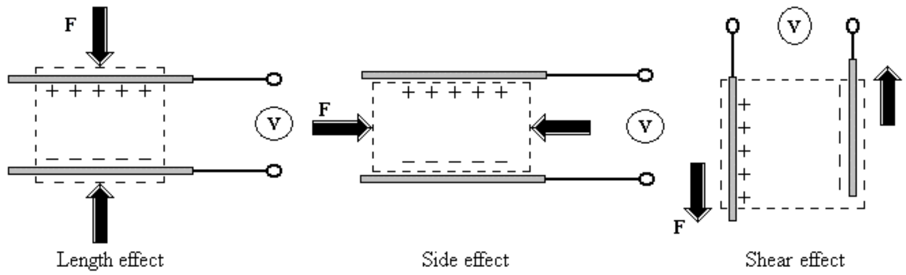

Figure 3.2 Three different effects on piezoelectric materials

Force and pressure sensors are in general utilizing the length effect and side effect, respectively. In Appendix B different designs of piezoelectric accelerometer is presented.

Piezoelectric sensor materials originate from two broad classes, ceramics and polymers which both have been used for active vibration control. Quartz, lead zirconate titanate (PZT) and crystalline are examples of piezoelectric ceramics that have been used widely. Piezoelectric ceramics are used extensively for a wide range of frequencies both as actuators and sensors. The piezoelectric polymers are used mostly as sensors, because they require high voltages and they have a limited control authority (amount of force, moment, strain or displacement, etc.). The best-known piezoelectric polymer is the polyvinylidene fluoride (PVF_2 or PVDF) [13]. Research is going on with porous polymers.

Magnetostrictive and electrostrictive are two materials that remind of piezoelectric materials. These materials and other smart materials can often be used instead of piezoelectric materials in many sensor applications. Piezoelectric is still a step ahead in research and there exist many commercial products based on piezoelectric materials. Piezoelectric sensors are being used in the automotive industry as knock sensors, for distance measurements, acceleration sensors in airbags, flow sensors and liquid level measurements [49].

3.1.1 Piezoelectric accelerometers

There exist several designs of piezoelectric accelerometers, as mentioned earlier there exist designs that harness the different piezoelectric effects, see Appendix B. Some advantages with piezoelectric accelerometers compared to other sensor types. Are that they have no moving parts, are robust, easy to fit, low cost and they have a wide frequency response [48]. There are other advantages that often are mentioned such as high temperature range, high sensitivity, long service life and small spring travel. A disadvantage is that they do not have static measurement.

The piezoelectric material can be suspended between a rigid post and a seismic mass. A force on the piezoelectric element is generated when the accelerometer is subjected to vibration. The size of this force is according to Newton's second law the product of acceleration and the constant seismic mass. A charge output proportional to the applied force is generated due to piezoelectric effect. The sensing element in an accelerometer often consists of a quartz wafer that serves both as sensor and spring. The quartz wafer is used in both compression and shear devices [33]. It produces a charge proportional to the strain. To measure the charge are voltmeters with high input impedance used. A charge amplifier converts the charge to an output voltage, which can then be measured with standard instrumentation. The charge is actually not amplified, it is collected in a capacitor with well-known capacitance and high insulated impedance. The charge amplifier consists often of an OP amplifier together with the capacitor [5]. An enclosed integrated circuit piezoelectric (ICP) is the same as a charge amplifier, which is built into the sensor housing and powered by a constant current [31]. Thus, the piezoelectric sensor output can be either charge or low voltage signal. A benefit with ICP is that it can easy be transferred over long distances without need of special cables.

Accelerometers can be attached to the structure in many ways; some of them are by screwing, by a magnet, silicon, cement, epoxy, or by stud mounting [31, 48]. The choice of mounting technique affects the attachment between the accelerometer and the structure with different strength. It is important to find the best suitable attachment solution for the specific case to minimize influence in the frequency response.

Chapter 4

ACTUATOR AND SENSOR SELECTION

The purpose of this chapter is to briefly discuss how the choice of actuators and sensors will influence the ability to model, update, and control the system. The process of sensor and actuator selection is as important as designing the controller. The process is usually ad hoc, because there is no cohesive approach to designing and selecting actuators and sensors [10, 43].

According to Uhlbrich, Wang and Bormann [29], the realization of an efficient control is distinctly influenced by the choice of actuators and sensors, their positioning within the whole system, and the control concepts.

First, is a discussion about selection of an actuator and comparison of existing technologies. This is followed by a brief discussion about how actuator and sensor parameters are related and their influence on controllability and observability. Finally, the placement of actuator and sensors is briefly discussed.

4.1 Actuator Selection

Historically, passive techniques, such as rubber and hydraulic mechanisms have been used to reduce vibrations. As actuator technology and control design has matured, different types of actuators could be used for active vibration control. To make the right choice of actuator, knowledge is needed of the attributes and technical options for an actuator. According to Crawley, Campbell, and Hall [10], five important parameters can be used to describe the attributes and function of an actuator. They are:

- Type
- Location
- Impedance
- Relative size
- Bandwidth

The actuator can be either linear or angular. It can act in different directions (along different axis) and act at a local point or distributed over an area.

The location of the actuator is somewhat limited: it has a great influence on the behavior of the system, because it affects the controllability and observability of the system. The numbers of actuators to be used and where they should be placed are important factors in design. Regarding the number of actuators, it is advantageous to have as few as possible because the stability of the system is reduced and adaptation-time will increase. This of course depends on the control algorithm. Furthermore, if there are more actuators than necessary for a problem the result can be that the actuators will try to cancel out each other, which gives a strong deteriorating damping. This is the case if the control algorithm has not considered knowledge of interactions between actuators.

The impedance of an actuator defines whether the actuator commands force or displacement. Actuators with low impedance command force, high impedance command displacement and those with intermediate impedance command both force and displacement.

The relative size is defined as the size of the actuator that is needed to produce a specified force, torque, strain or displacement. Relative size is hard to change because it depends on the actuator physics.

The bandwidth describes the operating frequency. Usually, the bandwidth is limited by the dynamics of the actuator or by the power amplifier or a combination of both.

4.1.1 Comparison between Actuator Technologies and Principles

This section attempts to make a simple comparison of technologies described in Chapter 2. Most of them have been tested in laboratories, but not all have been applied in commercial products. For use in linear motion control, there are a number of actuator alternatives. This section gives guidelines to find a proper actuator for a specific application with certain requirements, such as amplitude, bandwidth, force, response time, and size.

It is difficult to make a comparison of all actuator technologies, because the performance of different systems varies greatly, and they can be either voltage or current driven. A direct comparison of the transfer function between input and output can only be carried out with all actuators if the mechanical output is referred to the input electrical power. Apart from that, the development of actuators based on different technologies has reached different stages. A comparison could be made where the actual limit for the technology is dependent on some physical restriction, such as size, weight. Electromagnetic, hydraulic, piezoelectric and pneumatic actuators have been compared through relations between force/displacement and force/response time, see [6] and [42]. Electromagnetic actuators can create high forces, large displacements and fast responses, and with large bandwidth. Piezoelectric actuators can create high forces at very short response time, but the displacements are very small. Electrostrictive ceramics have the same order of strain, force and response time as piezoelectric ceramics. Hydraulic and pneumatic actuators can exert very high forces and at the same time large displacements, but they are often slower than other solutions, and the bandwidth is often small. Figure 4.1 shows two perspicuous graphs for comparison of well-known actuator technologies. Most existing principles of the electromagnetic, electrostrictive, hydraulic, pneumatic and piezoelectric technologies suitable for active engine mounts applications are within the respective areas in Figure 4.1.

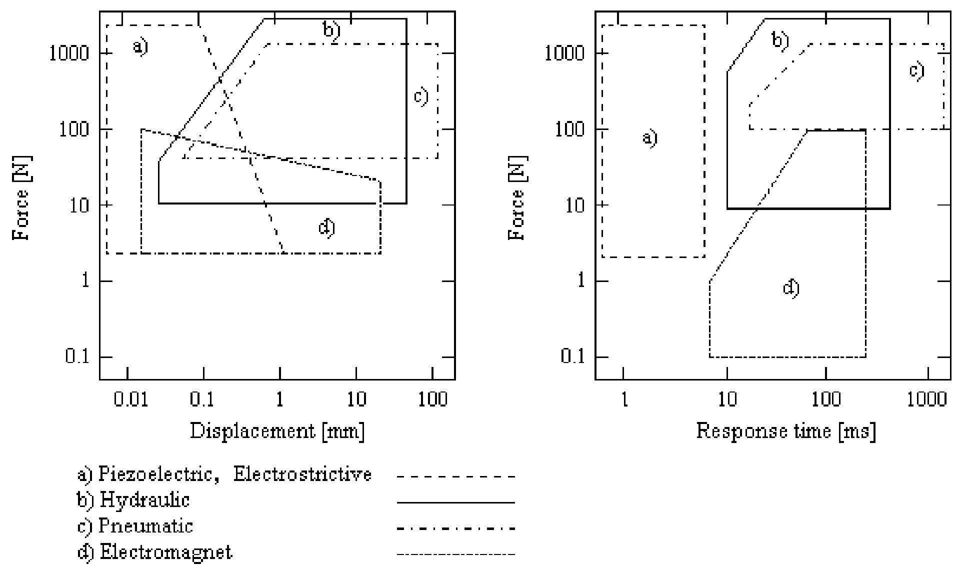

Figure 4.1 Comparison between some actuators concerning force/displacement and force/response time (Earlier presented in [6] and [42])

Even though the main disadvantage with magnetostrictive actuators is their small displacement, the strain in some solutions has been about two times larger than the strain of a stacked piezoelectric actuator. Magnetostrictive can create high forces at very short response time similar to electrostrictive and piezoelectric actuators. However, the amount of force can often be converted into displacement through utilization and modification of hydraulic or mechanic mechanisms. Similar to piezoelectric materials, electrostrictive and magnetostrictive materials are used in high precision applications. They consist of ferromagnetic materials, which experience an elastic strain when exposed to an electric or magnetic field, respectively. Magnetostrictive materials can undergo lower input voltages then most piezoelectric and electrostrictive materials. In addition, magnetostrictive materials have advantages against piezoelectric, for example low weight, no change in characteristics related to age and they can operate at higher temperatures than piezoelectric and electrostrictive actuators. A disadvantage with magnetostrictive materials is that they are not easily embedded in control structures. Magnetostrictive materials are similar to shape memory alloys but the difference is that they react on a magnetic field instead of changes in temperature. A big advantage with magnetostrictive materials is that they are much faster in transforming electrical energy to mechanical motion in comparison to traditional memory metals (shape memory alloys).

Uhlbrich, Wang and Bormann [29] have compared electromagnetic, hydraulic and piezoelectric technologies. Depending on the desired characteristics, they have created a measurement of how well the three different technologies handle each characteristic. For example, both electromagnetic and piezoelectric actuators have higher regulating frequencies and simpler transfer characteristics than the hydraulic ones. Their transfer characteristic depends on fluid dynamics. On the other hand, hydraulic actuators can be made compact. Hydraulic actuators are hard to beat in applications where strong forces are needed. Furthermore, these actuators can establish counterforce without any energy consumption in contrast, for example, to electromagnetic actuators. In Table 4.1, electromagnetic, hydraulic and piezoelectric actuator technologies are compared for each characteristic. This table is carried out through studying [29].

Table 4.1 Three widely used actuator technologies with a simple comparison of each characteristic

↓ Characteristic Technology →	Electromagnetic	Hydraulic	Piezoelectric
Bandwidth	Very wide	Average	Very wide
Displacement	Large	Very large	Very small
Effectiveness	Very high	Low	Very high
Stiffness	Average	Good	Excellent
Realizable force/weight (Active element without periphery)	Average	Excellent	Average
Realizable force/size	Average	Excellent	Weak
Realizable force/total weight	Excellent	Weak	Good
Possibility of stimulation of vibrations due to non-linearities	Good	Weak	Average

Electromagnetic and piezoelectric actuators are highly efficient, generally over 90 and 95 percent, respectively. Piezoelectric is especially well suited when dealing with small amplitude vibrations or in high precision constructions within μm-range. Very low material damping and small realizable regulating distances are two crucial disadvantages of piezoelectric devices. The third technology, hydraulic, can typically have high radial stiffness. It can apply regulating movements without friction and ensure high safety against tilting effects. In general, servo-valve is used to control the pressure in a two-chamber device. Unfortunately, the regulating pressure depends greatly on all fluid mechanical losses of the hydraulics and the dynamic of the servo valve. The maximum material stress in the membranes gives the design limits.

According to Brennan, Garcia-Bonito, Elliot, David and Pinnington [46], the principles of the magnetostrictive, piezoelectric and electromagnetic actuator technologies have been experimentally tested for active vibration control. Before the choice of an actuator can be made, it is explained that the requirements of the actuator must be specified in terms of force, displacement, bandwidth and power. They compare actuator principles through study of the input-output transfer functions for current and voltage driven principles, respectively. Another common method of comparing different actuator technologies is to compare the energy density per unit weight or volume. The energy of density for electromagnetic has been calculated to be 4 J/cm^3 to compare with 0.1, 0.2, 5-10 and 5 J/cm^3 for electrostatic, piezoelectric, SMA and thermomechanical, see Appendix C. This method is often used to compare miniaturized principles. The force is dependent on how strong the magnetic field is in magnetic-based technologies, because the possible magnetic field is dependent on the specific volume or weight.

In electromagnetic, electrostatic and piezoelectric actuators the generated forces are directly in proportion to the voltage level. The main advantages of these technologies are their rapid actuation potential and low power consumption. Piezoelectric micro actuators have been capable of cycle rates in thousands of cycles per second range. Electrostatic actuators can often operate as fast as the electromagnetic actuators. Both of them are usually only limited by their mechanical design and their driving electronics. Electrostatic actuators usually require high operating voltages.

Shape memory alloys can be considered when a large strain is preferred, because compared to piezoelectric they have a larger strain. The main problem with shape memory alloys is the slow response, so fast response can not be a requirement when choosing this type of actuator. Neither can thermally driven actuators be used, because they generally exhibit slower cycle rates than other methods. Actuators that are based on the technologies phase change, thermomechanical and shape memory alloy require heat as a primary driving mechanism. These thermally driven actuators lose heat to the environment and are, typically, low in efficiency.

After we have studied and compared different actuator technologies to be used in an active engine mount our conclusion was that electromagnetic and piezoelectric technologies were the most promising alternatives. These two technologies are already introduced as the selected technologies in sub-chapter 2.2. For a specific application of active vibration isolation one of them is usually better than the other. Piezoelectric actuators are preferred when low power consumption is desired. For example, piezoelectric actuators can be designed to replace almost any solenoid to use less power, but the result is always bulkier and often heavier.

4.2 Relationship between Actuators and Sensors Parameters

The relationship between actuator and sensor parameters, together with the interaction with the closed loop system, determines their effectiveness for control and force changes in actuator and sensor design. The choice of actuators and sensors influence each other and the controllability and observability of the system.

An actuator and sensor pair can influence the characteristics of the transfer function and the ability to control vibrations. For some less successful choices there can be pole-zero cancellation of system modes in the transfer function from the input to output, which gives uncontrollable and unobservable results.

If an actuator and a sensor are the same type, the pair is called dual. If an actuator acts and a sensor measures at the same point they are said to be collocated. If a pair is both dual and collocated they will create a transfer function with residues of the same sign, this creates alternating pole-zero pattern [10]. There are actuator and sensor pairs that are not dual, but still have alternating pole-zero pattern. These are called pseudo-dual pairs. Alternating pole-zero pattern is discussed further is sub-chapter 4.3.

Both the actuator and sensor impedances can have large effects on the structural transfer function characteristics. Actuator and sensor impedances can be low, intermediate, or high. Sometimes the best transfer function characteristics are obtained when the impedances between actuators and sensors are matched, because good pole-zero spacing is obtained. When the impedances are different pole-zero cancellation can occur, leaving the system in an unobservable and/or uncontrollable state. Therefore, it can be worthwhile to examine the choices of actuator and sensor impedances. Usually, the actuator impedance is set to match an application and the next choice is the impedance of the sensor. The best choice is to match the selected actuator impedance in order to obtain the largest pole-zero spacing.

To form an opinion of which size of actuator is necessary can generally not be done before reaching halfway in the design process. Usually, a number of studies are performed after the structure has been designed. In simplistic cases the relative size can be analysed after the dominated mode: this rule of thumb is derived from an example of one degree-of-freedom [10]. Before deciding the relative size of the actuator, consideration must be given to how much noise that can be present in a sensor in order to close the loop, see [10].

Actuator and sensor bandwidth depends on the dynamics of the device, or roll-off within the controller. A normal method is to select the frequency to, at least, mode-controlled.

4.3 Estimate the Effectiveness for Control

With the great number of available options of sensor and actuator designs, the most obvious question is whether it is possible to estimate the best choices for attaining the closed loop objectives. This will be discussed briefly in this sub-chapter.

The choice of a proper actuator and sensor is based upon achieving the best performance and stability robustness in the closed loop control design. To verify if the chosen array of actuators and sensors for the control design is capable, there exist different methods to form an opinion of the capability of the system. Open loop controllability and observability and the closed loop stability are two methods respectively discussed in the sections 4.3.1 and 4.3.2.

4.3.1 Open Loop Controllability and Observability

The effectiveness of an actuator and a sensor regarding the controllability and observability of the system is investigated, together with a static test. This method is creates an idea of how observable and controllable the actuator and sensor pair are in achieving a good control of the closed loop system. The classical test for observability and controllability is to examine the rank of the observability and controllability matrices, but these tests do not give relative information on the observability and controllability of different pair of actuator and sensor. There also exists a test that is based on the size of the modal residue, but it is out of scope of this thesis.

According to Crawley, Campbell, and Hall [10], the best approach to test the static effectiveness of an actuator or sensor is by using the observability and controllability gramians. By combining the gramians, the Hankel Singular Values for each mode can be found and the effectiveness of actuator and sensor pair can be analyzed. According to Campbell [60], the gramians are really a measure of the observability and controllability of the dynamic modes. They do not really take into account the static portion. One way to compare different actuator pairs would be, given a set of sensors, to calculate a different set of gramians for each actuator, and then find the Hankel singular values (HSV). The overall level of these HSV's will give some measure of static effectiveness. Another approach is to do the same thing, but to calculate the DC value (i.e. when xdot = 0 in a regular state-space model), which is another measure of the static effectiveness.

For an internally balanced system the observability gramian O_x and controllability gramian S_x are equal and diagonal, according to Glad and Ljung [3].

$$S_x = O_x = \sum \quad = diag\{\sigma_1, \sigma_2, ..., \sigma_n\} \tag{4.1}$$

where σ_i are called the Hankel Singular Values. Large Hankel Values correspond to highly controllable and observable states, while small Hankel Singular Values correspond to almost uncontrollable and unobservable states [10]. Observability and controllability gramians are often used to simplify system models derived from physically built models. This makes it easier to do an analysis, and control design and realization. Non-observable and controllable modes can be removed without affecting the input-output signal relationship [3].

4.3.2 Closed Loop Stability

The stability of the controller is largely affected by choice of actuator and sensor such as type and location. To ensure good closed loop stability there must be stability robustness of the loop to modal parameter errors and variations, along with the ability to measure and control the system modes. To get the most effective transfer function the following four characteristics should be fulfilled [10]:

- (Nearly) alternating poles and zeros.
- No non-minimum phase zeros in the bandwidth.
- Good pole-zero spacing.
- Good roll-off.

Alternating pole-zero pairs gives good robustness in the closed loop [10]. According to Campbell [60], the main issue with alternating poles and zeros are that they give a bounded phase system, which is a lot easier for the controller to cope with. When a zero pair is missing it is still possible to control in that region, but there will be a gain drop and loss of performance somewhere. It is similar to the non-minimum phase zero, but in this case it is impossible to control in this region. In the frequency region where control is desired, it is preferred to have alternating poles and zeros, and perhaps a missing zero pair near the end of one of these regions, so that the gain drop and loss of performance falls in a less important region, such as after roll-off.

We do not want non-minimum phase zeros in the bandwidth, because for systems with zeros in the right half plane the phase curve will have a larger negative phase shift than for a minimum phase system with the same amplitude curve [3].

Good pole-zero spacing is similar to having modes that are highly controllable and observable [10]. This allows the controller to be more easily designed.

For high frequencies it is important to have high roll-off if the model is not good, thanks to the requirement for sensitivity function and the complementary sensitivity function.

4.4 Placement

There is no easy way of dealing with this problem, and object in this sub-chapter is to just present the problem and give a brief discussion. In consideration placement of actuators and sensors there are alternatives such as direction, location and number of units. Placement of an actuator or sensor influences only the direction and location.

It was briefly discussed in sub-chapter 4.2, however, an actuator and sensor collocated pair usually creates an alternating pole-zero pattern that is quite beneficial to closed loop control. The placement of actuators and sensors influences the controllability, observability and the closed loop performance. There have been many studies on optimization of placement and how it influenced the control design, see [10, 43, 44].

Chapter 5

STUDY OF ELECTROMAGNETIC AND PIEZOELECTRIC ACTUATORS

Piezoelectric and electromagnetic actuators are two popular choices for AVNC and are promising technologies for use in active engine mounts. In this chapter we will discuss them and their characteristics. To predict the performance of the actuators and to investigate design parameters, models are crated using Matlab/Simulink. A good model of the actuator is also advantageous when designing its control algorithm.

5.1 Electromagnetic Actuators

In this section the principle of a voice coil actuator is discussed and how different parameters affect its behaviour. To enable that discussion, a model of a voice coil is developed. It will be shown that the derived model, with minor modifications, represents a wide range of electromagnetic actuators. Guidelines are given at the end of the chapter on how to design an electromagnetic actuator for specific needs.

5.1.1 Model of a Typical Voice Coil

The voice coil consists of two basic parts: a magnet and a coil. The magnet creates a magnetic flux that passes through the coil. When current passes through the coil a force will be generated that is proportional to the current, the length of the wire in the flux field and the magnetic flux passing through the coil. The arrangement of a typical voice coil is shown in Figure 5.1.

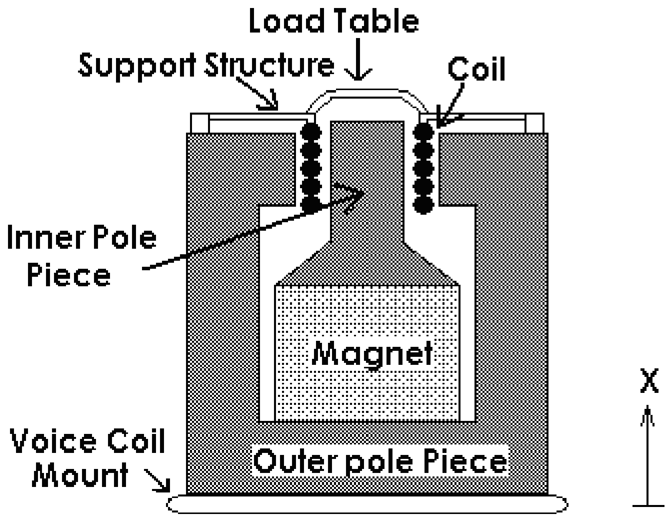

Figure 5.1 Typical voice coil actuator

The intense magnetic field that is required by the voice coil is created by a permanent magnet or an electromagnet, as shown in Figure 5.1. Soft iron is used to transmit the magnetic flux to the air gap between the face of the north-polarized inner pole piece, which is round, and the hole in the south-polarized outer pole piece where the armature coil is placed. This creates a radial flux field in the air gap between the outer and inner pole piece. The air gap between the pole pieces is minimized to reduce the reluctance of the magnetic circuit and thus maximizing the intensity of the fixed magnetic field.

The force that is exerted on the coil is given by:

$$F_X = 2\pi RN(\mu_0 H_0 i + \frac{\mu_0 N i^2}{2g}) \tag{5.1}$$

Where:

F_x: axial force
R: radius
N: number of turns
μ_0: permeability of free space
H_0: magnetic field intensity
i: current
g: gap between the outer and inner pole pieces

The first term in equation 5.1 is due to the interaction between the coil (current) and the magnetic field, H_0. The last term is due to the interaction between the wires in the coil itself. H_0 is usually made much larger than $Ni^2/2g$ to achieve linear operation by reducing the importance of the last term. Thus, F_X becomes simply $2\pi RN\mu_0 H_0 i$. The force that is exerted on the coil is then simply proportional to the current, i.e. $F = k_I \cdot i$.

In Figure 5.2 the mechanical components of the shaker are shown and in Figure 5.3 the electrical components. The two are cross-coupled because the back electromotive force e_{back} in Figure 5.3 depends on the movements of the coil according to equation 5.5. In the model the shaker is divided in four different masses: body (M_B), coil (M_C), table (M_T) and Load (M_L). The connections between them are modelled with a spring and a damper system.

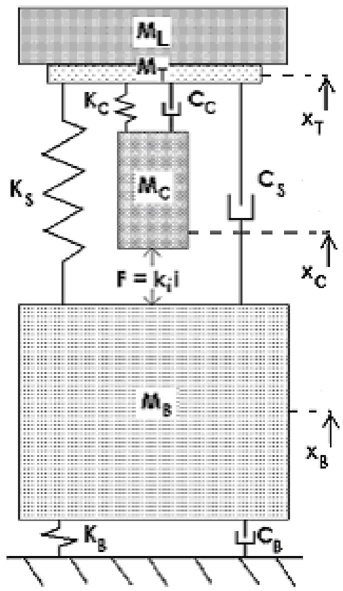

Figure 5.2 Mechanical diagram of a voice coil diagram of a voice coil

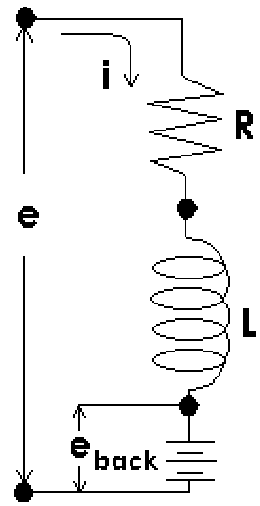

Figure 5.3 Electrical circuit

K_B and C_B are stiffness and damping of the voice coil mounts. The values for K_C and C_C depend on coil design. In early designs, the coil was wound around a stiff thin walled tube. Modern armatures use epoxy bonding techniques to fasten a rigid epoxy-stabilized coil to a light magnesium table structure [12]. The load table is assumed rigid and is held in place by the support structure, which has stiffness, K_S and damping C_S. The armature is only allowed to move axially and it needs to be restrained from all other motions. To accomplish this and accurately centre the armature in the gap the support structure must be soft in the axial direction.

Applying Newton's Second Law of motion, three equations of motion will be derived, one for each mass. The first equation is for the coil, the second for the table and the load, and the last one is for the body.

$$M_C x_C + C_C \dot{x}_C - C_C \dot{x}_T + K_C x_C - k_C x_T - k_1 i = 0 \tag{5.2}$$

$$(M_T + M_L)\ddot{x}_T - C_C \dot{x}_C + (C_C + C_S)\dot{x}_T - C_S \dot{x}_B - K_S x_C + (K_S + K_C) x_T - K_S x_B = 0 \tag{5.3}$$

$$M_B \ddot{x}_C - C_S \dot{x}_T + (C_B + C_S)\dot{x}_B - K_S x_T + (K_B + K_S) x_B + k_1 i = 0 \tag{5.4}$$

The voice coil's electrical model includes the resistance and inductance of the coil. The resistance, R, defines the minimum impedance exhibited at the voice coil input. The resistance increases by about 40% per 100°C for copper, and slightly with frequency. The coil inductance, L, is large because the coil is attracted strongly to the iron of the pole pieces. This cause the complex electrical impedance to be equal to $R + j\omega L$, which hence increase with frequency.

When the coil begins to move in response to the generated force, a voltage is induced in the coil caused by its motion in a magnetic field. This voltage is called back electromotive force or "back emf" and is proportional to speed, magnetic field strength and current. It reduces the voltage across the coil, thus lowering the current and the rate of acceleration. It is modelled as:

$$e_{back} = k_2(\dot{x}_C - \dot{x}_B) \tag{5.5}$$

According to Kirchhoff's second law for an electric circuit, the equation for the electrical circuit in Figure 5.3 is:

$$(\dot{x}_C - \dot{x}_B)k_2 + L \cdot \frac{di}{dt} + Ri = e \tag{5.6}$$

Equation 5.2 to 5.6 now yields:

$$\begin{bmatrix} M_C & 0 & 0 & 0 \\ 0 & M_T+M_L & 0 & 0 \\ 0 & 0 & M_B & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{Bmatrix} \ddot{x}_C \\ \ddot{x}_T \\ \ddot{x}_B \\ 0 \end{Bmatrix} + \begin{bmatrix} C_C & -C_C & 0 & 0 \\ -C_C & C_C+C_S & -C_S & 0 \\ 0 & -C_S & C_B+C_S & 0 \\ k_2 & 0 & -k_2 & L \end{bmatrix} \begin{Bmatrix} \dot{x}_C \\ \dot{x}_T \\ \dot{x}_B \\ di/dt \end{Bmatrix} + \begin{bmatrix} K_C & -K_C & 0 & -k_1 \\ -K_C & K_S+K_C & -K_S & 0 \\ 0 & -K_S & K_B+K_S & k_1 \\ 0 & 0 & 0 & R \end{bmatrix} \begin{Bmatrix} x_C \\ x_T \\ x_B \\ i \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \\ 0 \\ e \end{Bmatrix} \tag{5.7}$$

K_B and C_B are the stiffness and damping of the voice coil mount and are not part of the actuator itself, therefore can they be set to infinity. The mass M_B is then directly connected to the ground, thus x_b will be zero, has no impact on the system and can therefore be removed from the model. The equation (5.7) is then simplified to:

$$\begin{bmatrix} M_C & 0 & 0 \\ 0 & M_T+M_L & 0 \\ 0 & 0 & 0 \end{bmatrix} \begin{Bmatrix} \ddot{x}_C \\ \ddot{x}_T \\ 0 \end{Bmatrix} + \begin{bmatrix} C_C & -C_C & 0 \\ -C_C & C_C+C_S & 0 \\ k_2 & 0 & L \end{bmatrix} \begin{Bmatrix} \dot{x}_C \\ \dot{x}_T \\ di/dt \end{Bmatrix} + \begin{bmatrix} K_C & -K_C & -k_1 \\ -K_C & K_S+K_C & 0 \\ 0 & 0 & R \end{bmatrix} \begin{Bmatrix} x_C \\ x_T \\ i \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \\ e \end{Bmatrix} \tag{5.8}$$

5.1.2 Parameter Identification

To enable the use of the model obtained in the previous section, to predict the behaviour of voice coil actuators, two methods have been developed for identification of the parameters in Table 5.1. The first is called the Heuristic method and is based on knowledge obtained by experiment. The second is based on the least square method which is a more automatic method.

Table 5.1 Parts of a Voice Coil with moving magnet

Parts and property for voice coil with moving coil	**Name**
Load, mass	M_L
Table, mass	M_T
Table, stiffness	K_T
Table, damping	C_T
Coil, mass	M_C
Coil, stiffness	K_C
Coil, damping	C_C

A problem that has been noted is that some manufacturers only provide the magnitude of the frequency response and let the phase be unknown. In this case we can assume that the system is linear and the poles and zeroes are placed in the left half plane. This makes it possible to re-create the phase from the magnitude. This can be done with the use of the Bode relation [3]. However when the input data is experimentally obtained we can assume that it is not well conditioned. A better solution is then to use a data-fitting method on a parameterized model to obtain the phase [53]. This will be the case in both method one and method two.

5.1.2.1 Method 1: The Heuristic Method

Unlike algorithms, the heuristic method does not guarantee optimal solutions and has no theoretical guarantee. This method is based on experimental knowledge and educated guesses. Therefore, it is suitably described by an example, as in example 1, section 5.1.4.1. The results can be summarized: The damping C_B and stiffness K_B control gain and frequency for the first peak. The second peak depends similarly on the values of C_S and K_S. Values C_C and K_C respectively, affect the last peak. And k1 affects the gain for all frequencies. Figure 5.4 shows a typical frequency response for a voice coil.

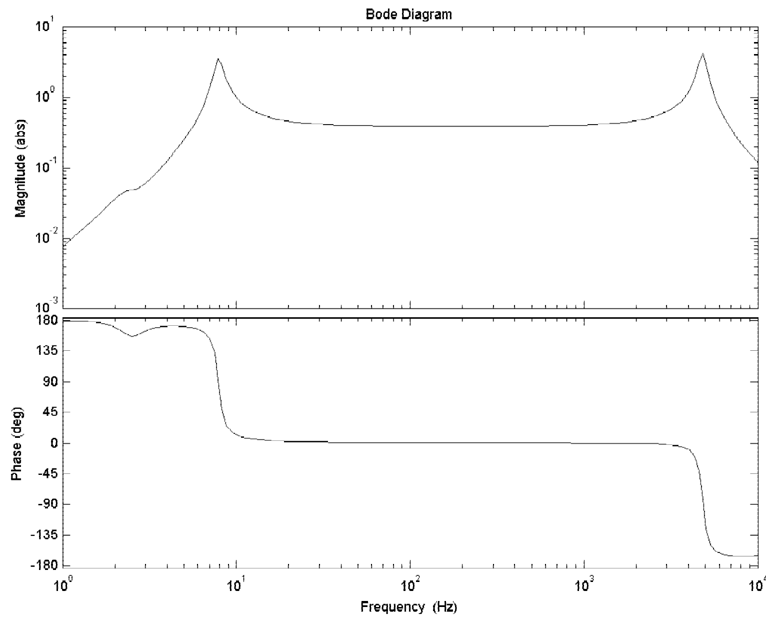

Figure 5.4 A typical frequency response for a Voice Coil

Damping C_S, C_C and C_B can be calculated with the 3dB-method. For a single degree of freedom system SDOF the damping ratio is:

$$\zeta = \frac{c}{c_0} = \frac{c}{2m\omega_0} = \frac{c}{2m\sqrt{\frac{k}{m}}} = \frac{c}{2\sqrt{mk}} \tag{5.9}$$

Where:
- c : damping coefficient
- c_0 : critical damping coefficient
- ω_0 : resonant frequency
- m : oscillating mass
- k : stiffness coefficient

The 3dB-method gives the damping ratio for a SDOF system:

$$\zeta = \frac{\Delta\omega_{3dB}}{2 \cdot \omega_0} \tag{5.10}$$

Equations (5.9) and (5.10) now give the damping coefficient:

$$c = \frac{\Delta\omega_{3dB}}{\omega_0} \cdot \sqrt{m \cdot k} \tag{5.11}$$

The stiffness K_S, K_C and K_B can be calculated from data given in Figure 5.4, with the regular method, $K = m \cdot \omega_n^2$.

This method of calculating the parameters has been shown to give good results. The drawback is that it does not guarantee an optimal solution. It could be profitable to use the Least Square method described in section 5.1.2.1 as a complement to this method.

5.1.2.2 Method 2: The Least Square Method

The second method is the least square method, and can be used by itself or as a complement to the heuristic method. It is based on the least square method and uses a frequency response, which can be found in product information given by manufactures.

To solve the curve-fitting problem a solution based on the Matlab function lsqcurvfit is used. This function is based on the interior-reflective Newton method described by Coleman, T.F. and Y. Li in SIAM Journal on Optimization [45]. The function that will be minimized is:

$$\min_x \frac{1}{2}\left\|F(x, freq) - mag(freq)\right\|_2^2 = \frac{1}{2}\sum_i \left(\left|F(x, freq_i)\right|, - mag_i(freq_i)\right)^2$$

where mag_i is the magnitude at the frequency given by $freq_i$. The resulting "best fit" coefficients is here called x for the parametric model. F is the transfer function for the system of equations (5.8).

The method is implemented in Matlab. The least square algorithm is implemented in the m-file parest.m. It needs the user to implement the transfer function into the m-file. Then the function inputs are start values for the parameters, frequency and amplitude data. Outputs from the function are the estimated values for the parameters. For an example of this method we direct the reader to example two in subsection 5.1.4.2.

A problem that has been noted with this way of calculating parameters is that the least square method can find a local minimum that gives a good magnitude correspondence but incorrect phase. The problem is minimized if the initial values are chosen with care. The second problem that may arise is that the masses can be estimated to be abnormal. In that case it is better to have them fixed.

Besides the least square algorithm implemented in the m-file parest.m, an additional tool for converting pictures to transfer functions is found in the m-file tfpic2tfvalues.m. It takes a picture of a frequency response and converts that to magnitude and frequency data in vector form that is used in parest.m.

5.1.3 Validity of the Model

The voice coil actuator model (5.8) can be used for both regular voice-coil actuators and voice-coil, reaction-mass actuators (RMA) with either moving-magnet or moving-coil design. The difference is that the parameters symbolize different parts of the actuator. The differences can be seen in Table 5.2.

Table 5.2 Parameters for different electromagnetic actuators

Part, property:	Moving coil:	Moving magnet:	RMA, moving coil:	RMA, Moving magnet:
Reaction mass	-	-	M_L	M_L
Load, mass	M_L	M_L	-	-
Table, mass	M_T	M_T	M_T	M_T
Table, stiffness	K_T	K_T	K_T	K_T
Table, damping	C_T	C_T	C_T	C_T
Coil, mass	M_C	-	M_C	-
Coil, stiffness	K_C	-	K_C	-
Coil, damping	C_C	-	C_C	-
Magnet, mass	-	M_C	-	M_C
Magnet, stiffness	-	K_C	-	K_C
Magnet, damping	-	C_C	-	C_C

Some assumptions have been made when deriving the model. The following phenomenon have been ignored: temperature dependency, nonlinearity because the magnetic field that affects the coil depends on the coil displacement, the resistance in the coil is temperature and frequency dependent, and friction losses.

The nonlinearity because of the magnetic field that affects the coil is assumed small. When it is possible to construct the pole pieces higher than the coil, the results will be that the coil is always in the area where the radial flux is constant. This is also discussed in section 5.1.1.

Temperature dependency tests have been made on the SA10 actuator at CSA Engineering. According to Eric Anderson at CSA Engineering, the conclusion is that there is no significant change in actuator output gain for current input over the -40ºC to +60ºC temperature range. The magnetic properties of the system did not change significantly over the range. But CSA also recommended that a more comprehensive test may be necessary in the future. The test done by CSA Engineering was carried out in an ambient environment of 24ºC. They measured at four different amplifier inputs: 0.1, 0.2, 0.5 and 1.0 Vrms at 200 Hz, and found that the output was nearly linear with the drive amplitude. The use of a current amplifier rather than a voltage amplifier means the change in resistance over this temperature range does not influence the current applied for a given drive input. This is because it is not affected by changes in impedance with temperate. Assuming that the coil resistance changes with temperature in the same way as the conductor it is approximately 0.4%/ºC [5]. Data for conductor materials are given in Table 5.4. The high temperature extreme presents the largest voltage requirements because of ambient temperature increases and temperature rise due to self heating. The results of the test are presented in Table 5.3.

Table 5.3 Results of temperate measurements for the SA10 actuator.

	Room T (24 C)	-40 C	+60 C
Measured response	40.5 N/A	42.17 N/A	42.12 N/A
Difference from RT	0%	+4%	+4%
Resistance re RT and implied voltage increase for same current	0%	-25.6%	+14.4%

Table 5.4 Data for conductor materials [5]

Attribute	Unit	*Copper*	*Aluminium*
Density	g/cm^3	8.9	2.7
Resistivity	$\Omega mm^2/km^2$	17.2	28.3
Temperature coefficient	per ºC	0.0039	0.0040
Melting point	ºC	1083	660
Tensile strength	N/mm^2	200-250	70-100
Permitted tensile stress	N/mm^2	50	30

5.1.4 Examples of parameter identification

In this chapter the model of a typical voice coil has been developed. In this section two examples of model parameter identification will be shown. The first example is a voice coil with a moving magnet, SA–100 from Data Physics. The second example is a voice coil reaction mass actuator, SA10 from CSA Engineering. Both the heuristic and the more automated least square method will be exemplified.

5.1.4.1 Example A: Voice Coil Reaction Mass Actuator

To identify parameters for a typical voice coil with moving coil, we contacted Henrik Isaksson at Saven Hitech, agent for Data Physics in Sweden. He contributed with some estimated values for the voice coil actuator S-100 [54] as follows:

$M_D = 10$ kg
$M_T + M_C = 1.4$ kg
$M_B = 80$ kg
$K_S = 22$ kN/m
$K_C = 10$ GN/m
$R = 4.3$ ohm
$L = 0.31$ mH
$k1 = 44$

The remaining values were calculated from Figure 5.5 which is extracted from the article by George Fox Lang and Dave Snyderin Sound & Vibration, October 2001 [12]. The upper curve shows gain of the transfer function from current to table acceleration and the lower curve shows the same for voltage.

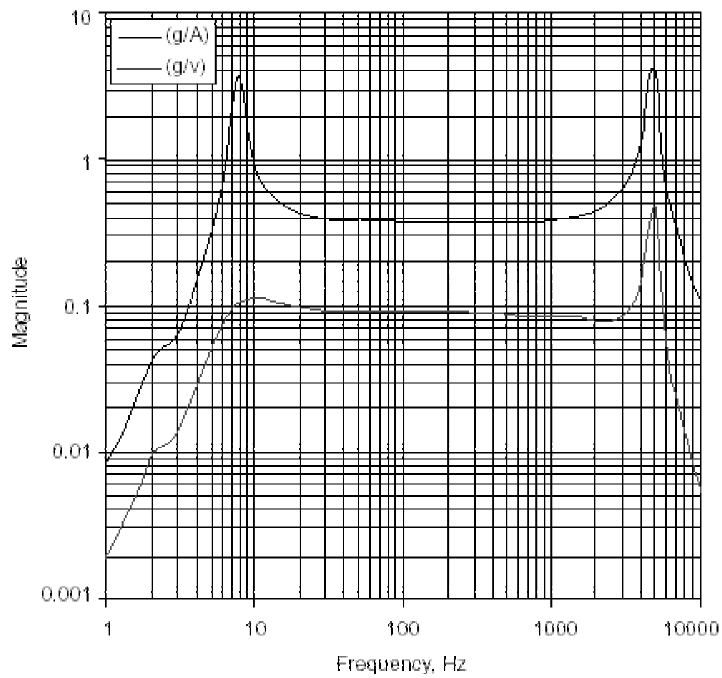

Figure 5.5 Frequency response for model of S-100 in Sound & vibration [12]. Upper curve is gain from current to table acceleration, and lower curve is gain from voltage to table acceleration.

Damping C_S, C_C and C_B was calculated with the 3dB-method according to (5.11).

$$c = \frac{\Delta\omega_{3dB}}{\omega_0} \cdot \sqrt{mk}$$

The remaining stiffness K_B was calculated from data given in the Figure 5.5 with the regular method, $K = m\omega_n^2$. The values were calculated to be:

C_S = 340 Ns/m
C_C = 273 000 Ns/m
C_B = 167 Ns/m
K_B = 19 739 Ns/m

In Figure 5.6, the frequency response can be seen for transfer function 5.7. The dotted line is from the article in Sound & Vibration [12]. The amplitude is ten times what is expected. The conclusion is that k_l is ten times higher than that used in the article. And the damping for the second and last peak seems too high. Figure 5.7 shows the frequency response for k_l, C_C, C_B ten times smaller: k_l = 4.4, C_C = 273 00, C_B = 16 which seems more accurate.

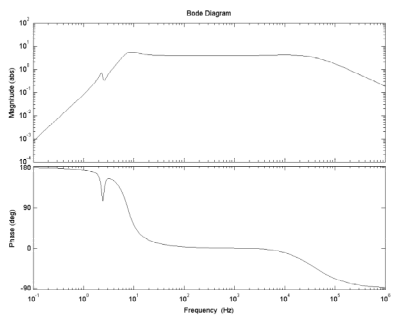

Figure 5.6 Bode diagram, voice coil, given values.

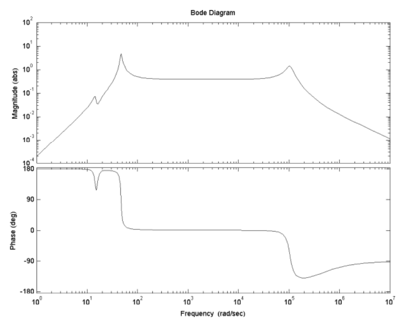

Figure 5.7 Bode diagram, voice coil with smaller values.

The two different methods were tried, to obtain the right values of the parameters. The first, heuristic method, is based on experimentally identifying the effect the parameters on the model. The second used to calculate the parameters was the least square method.

As expected, the damping C_B and stiffness K_B control gain and frequency for the first peak. The second peak depends similarly on the values of C_S and K_S. Values C_C and K_C respectively, affect the last peak.

After some adjustments of the values the following results were obtained:

Table 5.5 Estimated vales for a voice coil

M_D = 10 kg $M_T + M_C$ = 1.3 kg M_B = 78.8 kg k1 = 4.4	K_S = 24 kN/m K_C = 0.44 GN/m K_B = 20 kN/m	C_S = 43 Ns/m C_C = 1350 Ns/m C_B = 500 Ns/m

The resulting frequency response is compared to those in the article by George Fox Lang and Dave Snyder in Sound & Vibration, in Figure 5.5. The conformity with the frequency response in their model is high. The results can be seen in Figure 5.8.

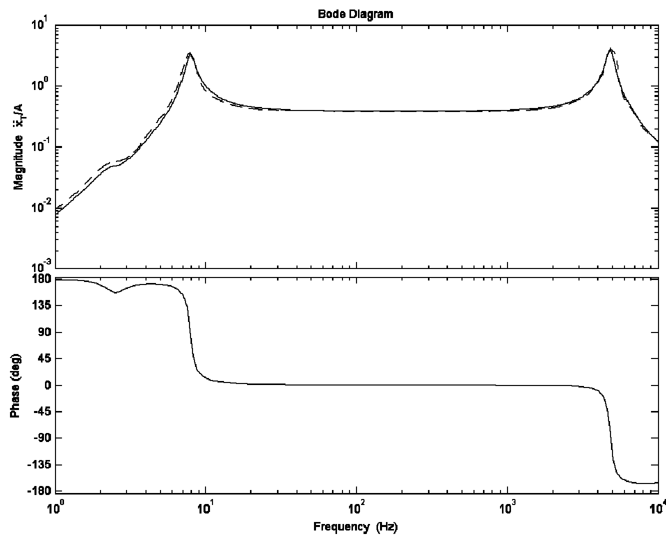

Figure 5.8 Bode diagram for S-100 voice coil, relating table acceleration to applied current as a function of frequency. Solid line is voice coil model with values in Table 5.5. Dotted line is values from Sound & Vibration [12].

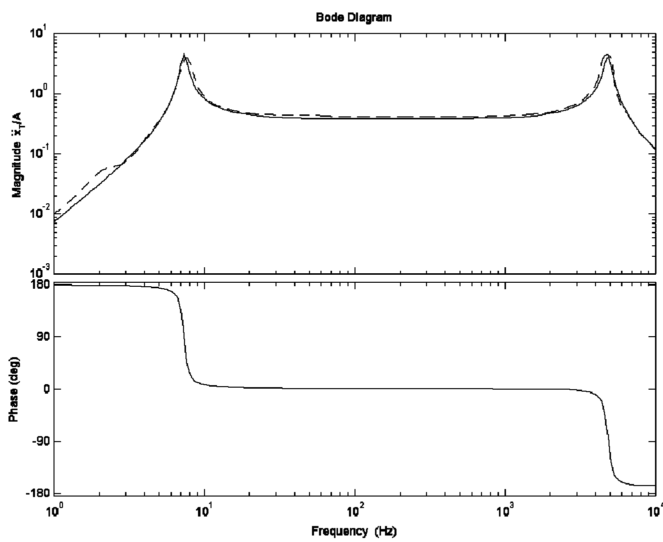

Figure 5.9 Bode diagram for S-100 voice coil, relating table acceleration to applied current as a function of frequency. Solid line is voice coil model without mount, with values in Table 5.5. Dotted line is values from Sound & Vibration [12].

The second method based on the least square method can be used by itself or as a complement to the method discussed above. The method was used for the transfer function representing system one (5.7) and two (5.8). The same initial values in Table 5.6 were used in both cases:

Table 5.6 Initial values for parameter estimation

$M_D = 0.1$ $M_T + M_C = 1$ $M_B = 10$ $k1 = 10$	$K_S = 10^4$ $K_C = 10^7$ $K_B = 10^4$	$C_S = 10$ $C_C = 10^3$ $C_B = 10^3$

The results can be seen in the Figure 5.10 below and the agreement with the Figure 5.5 is good.

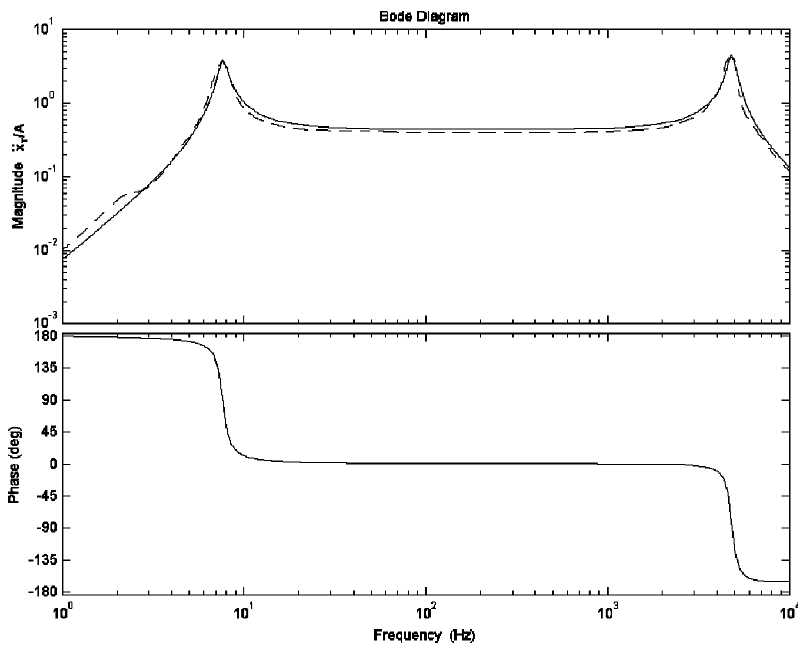

Figure 5.10 Bode diagram for S-100 voice coil, relating table acceleration to applied current as a function of frequency. Solid line is least square parameter estimation of acceleration. Dotted line is values from Sound & Vibration [12].

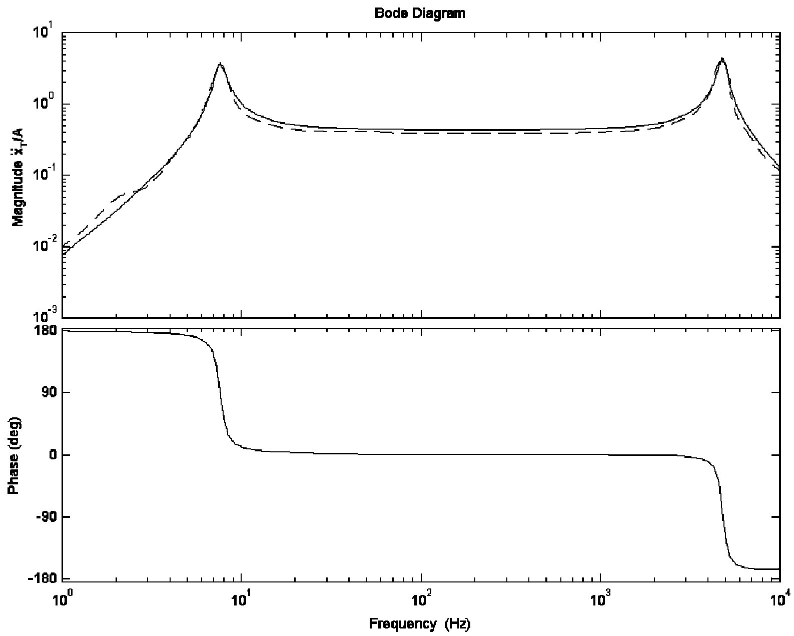

Figure 5.11 Bode diagram for S-100 voice coil, relating table acceleration to applied current as a function of frequency. Solid line is least square parameter estimation. Dotted line is values from Sound & Vibration [12].

It can be noted that the first peak in Figure 5.10 is missing. The reason is that the least square method does not weight that small change high. In Figure 5.11 it is also missing as expected, because it is originally from the body moment and in this model the body is rigidly connected to the ground. But the results for both the heuristic and the least square method are satisfying.

5.1.4.2 Example B: Voice Coil Reaction Mass Actuator

In this example the least square method will used to identify the model parameters for a reaction mass actuator from CSA Engineering, SA10. The specifications for SA10 are taken from a data sheet published by CSA Engineering [62]. The Bode diagram is shown in Figure 5.10.

Table 5.7: Specifications for SA10

Rated Force Output:	44.48 N	Resistance (User-specified):	2 Ohm
Bandwidth:	20-100 Hz	Total Mass:	2.49 kg
Motor Constant (Typical Values):	22.24 N/A	Diameter:	93 mm
Resonant Frequency:	30-200 Hz	Height:	92 mm

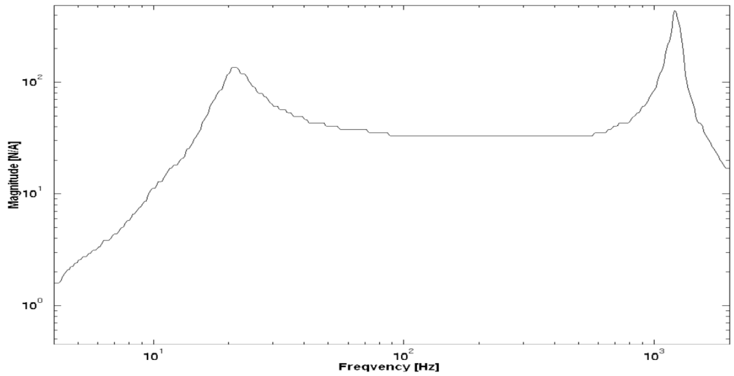

Figure 5.12 Bode diagram for SA10 [62].

The method used was to guess the initial values and then use the least square method discussed in sub-chapter 5.1.2.1. The initial values that were used are presented in Table 5.8. The results are shown in Figure 5.13 and in Table 5.9.

Table 5.8 Initial values for SA10

$M_C = 0.1$ $M_T + M_D = 1.6$ $k1 = 90.0$	$K_S = 6.0{\cdot}10^3$ $K_C = 2.0{\cdot}10^7$	$C_S = 20.0$ $C_C = 200.0$

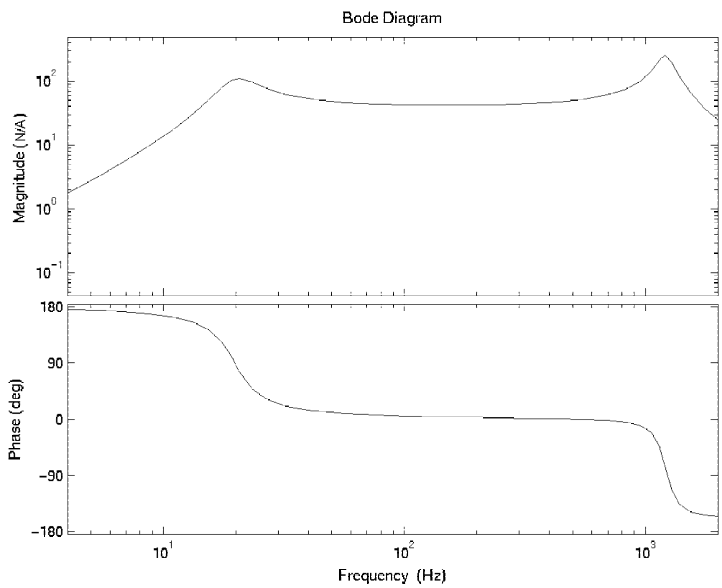

Figure 5.11 Bode diagram for SA, parameters estimated with least square method.

Table 5.8 Results for SA10 with least square method

$M_C = 0.28$ $M_T + M_D = 1.88$ $k1 = 88.70$	$K_S = 3.33 \cdot 10^4$ $K_C = 1.43 \cdot 10^7$	$C_S = 101.93$ $C_C = 311.29$

The results correlate well with the measured response that can be seen in Figure 5.12. What can be noted is that the total mass given in the specification, Table 5.7, is 2.49 kg and the total mass of the modelled parts is estimated to 2.16 kg. We can assume that the actuator housing weight is approximately 0.4 kg and that gives a total estimated mass of 2.56 kg, which is close to the given value. If the results had been unusual, the masses could have been fixed and the least square method could have been used again. This is not necessary this time.

5.1.5 Specification-dependent Design

To be able to use the voice coil actuator capacity to its maximum, consideration must be given to the wanted frequency response for the specific application. This enables the system designer to use a smaller actuator then otherwise would be needed. This can save space, weight and money.

For instance, the actuator resonance frequencies can be placed close to where the largest forces are needed. Thus enable the use of a smaller actuator and less power. Another desired effect can be that the actuator should give the same response in the entire operating range. That can be accomplished by placing the resonance frequencies outside the actuator's operating range.

Voice coil response characteristics with constant current or voltage amplifier coupling can be seen in Figure 5.4. When choosing between controlling the actuator with current or voltage, both the wanted frequency response and the accessibility must be considered.

The following steps can be used as guidelines for voice coil actuator design from frequency-domain specifications. The steps are given for the moving coil, the moving magnet, and the reaction mass actuator cases:

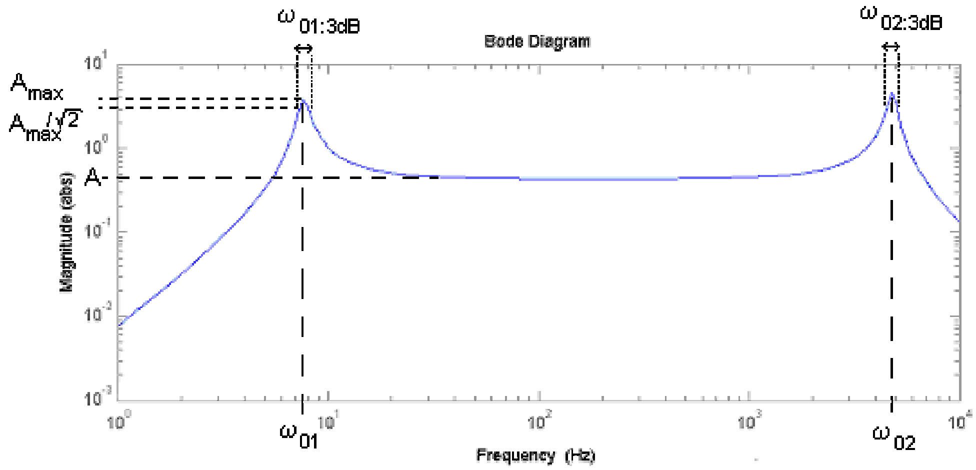

Figure 5.14 Bode diagram, specification-dependent design

Step 1: Decide the weight of the masses:

Moving coil: Coil M_C, table M_T and load M_D.
Moving magnet: Magnet M_C, table M_T and load M_D.
RMA (moving coil): Coil M_C, table M_T and reaction mass M_D.
RMA (moving magnet): Magnet M_C, table M_T and reaction mass M_D.

Step 2: Place the resonance frequencies:

First resonance: $\omega_{01} = \sqrt{\dfrac{K_S}{M_C + M_T + M_D}}$

Second resonance: $\omega_{02} = \sqrt{\dfrac{K_C}{M_C}}$

Step 3: Chose the damping coefficients:
First resonance:

$$C_S = \frac{\Delta\omega_{01:3dB}}{\omega_{01}} \cdot \sqrt{K_S(M_C + M_T + M_D)}$$

Second resonance:

$$C_C = \frac{\Delta\omega_{02:3dB}}{\omega_{02}} \cdot \sqrt{K_C M_C}$$

Step 4: Adjust amplitude A:

$$\mathrm{k}_{1:\mathrm{wanted}} = \mathrm{k}_{1:\mathrm{now}} \frac{\mathrm{A}_{\mathrm{now}}}{\mathrm{A}_{\mathrm{wanted}}}$$

Step 5: Are the parameters reasonable?
No, go back to step 1. and adjust the values.

When choosing the parameters, the values must be considered so they are physically possible. For example, as discussed in section 5.3.1, the force from the coil which is described by the constant k_1 depends on the size of the coil. This makes k_1 coupled to the mass of the coil M_C. And the generated heat depends on the resistance in the wiring, which depends on its area, and which will also affect the possible weight of the coil, as discussed in subsection 5.3.3.

5.2. Piezoelectric Actuators

Among the different actuator technologies that are available today piezoelectric actuator devices offer a number of benefits for use in active vibration control. Their high stiffness results in isotopic high actuator performance. Piezoelectric actuators give fast response, are small in size and weight, and are easily controlled. [7]

5.2.1 Piezoelectric Model

This section will describe the behaviour of the piezoelectric material. The notations used are the IEEE standard on piezoelectricity:

Table 5.9 The IEEE standard notations for piezoelectricity

D : Electric Displacement, Coulumb/m^2
E : Electric field, V/m
ε : Dielectric constant of the material, Farad/m
ε^T : Dielectric constant under constant stress
S : Strain
T : Stress, N/m^2
s : Compliance of the material, m^2/N
s^E : Compliance under constant electrical field, m^2/N

When the material is unstressed the electric displacement, D, is simply related to the electric field, E, in the one-dimensional case by:

$$D = \varepsilon E \tag{5.12}$$

Strain and stress are similarly related in a zero electric field:

$$S = sT \tag{5.13}$$

For a piezoelectric material, the electrical and mechanical equations are coupled. In the pseudo-static case the equations become [7]:

$$S = s^E T + dE \tag{5.14}$$
$$D = dT + \varepsilon^T E \tag{5.15}$$

Equations (5.14) and (5.15) are transformed for easier understanding of the physical phenomenon they describe. The quantities that will be used are displacement u [m], force F [N], applied voltage V [V], charge Q [C], height h [m] and area A [m^2]. Using the following variable transformations:

$$S = \frac{u}{h} \qquad T = -\frac{F}{A} \qquad E = \frac{V}{h} \qquad D = \frac{Q}{A}$$

the new equations will then describe actuator displacement as a function of force:

$$u = -\frac{hs^E}{A} F + dV \tag{5.16}$$

and the force as a function of displacement:

$$F = \frac{A}{hs^E}(-u + dV) \tag{5.17}$$

In the unloaded actuator case, the displacement as a function of applied voltage is shown in Figure 5.15. One relevant feature for practical applications is the hysteresis behaviour of the piezoelectric material. This originates in the movement of ferroelectric domain walls in the piezoelectric material. [7] This creates a practical problem for using applied voltage for controlling displacement. In general, additional sensors are needed to monitor the actual displacement.

The problem with hysteresis can be reduced by controlling the transferred charge instead of the voltage. In Figure 5.15, displacement is shown as a function of voltage (dashed line) and displacement as a function of charge (solid line). The hysteresis in the later case is almost suppressed.

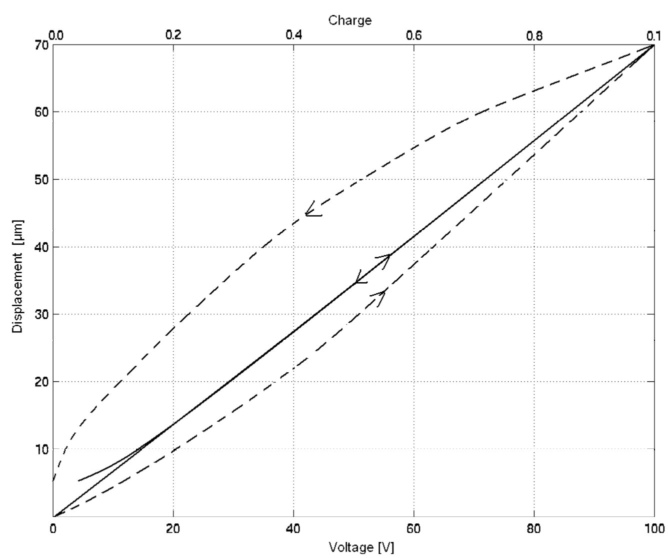

Figure 5.15 Example of displacement. The dashed line is displacement as a function of voltage and the solid line is displacement as a function of charge for a free actuator

When the actuator is subjected to an external force, there are two basic situations that must be considered. The first case is when the external force is proportional to the displacement, like a spring. The other case is when the actuator is subjected to a constant preload.

If no internal forces are considered and the actuator is assumed to be in a quasi-static case, the counterforce will increase with displacement as, $k\Delta u$. That will decrease the maximum actuator displacement. The final position will be reduced by $k\Delta u/K$, where K is the stiffness of the actuator.

The second case is with a constant load F on the actuator. The "displacement" of the actuator will initially be "reduced" by F/K, where g is the gravitational constant. The displacement due to an applied charge or voltage will then be approximately the same as in the free case. The two restraint cases and the free case are shown in Figure 5.16.

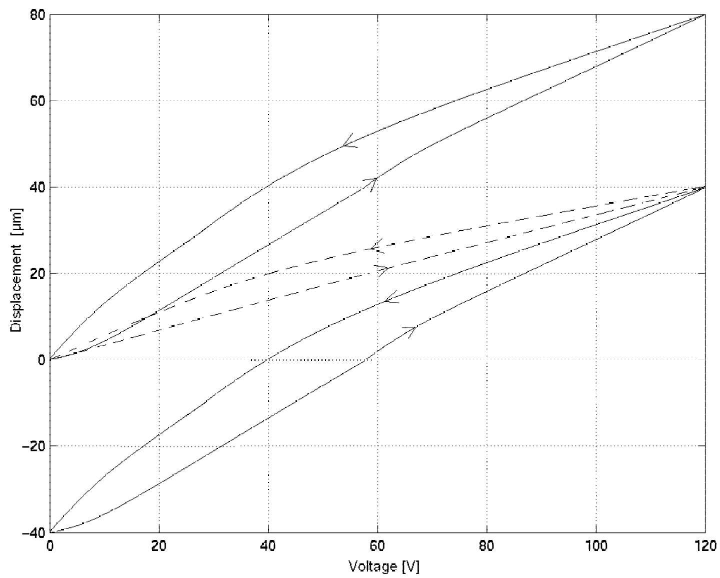

Figure 5.16 Upper curve shows free displacement, middle curve shows displacement with proportional counter force, and the lower curve shows displacement with constant load.

Another important property is the blocking force at "maximum" applied voltage. With free displacement it can be used to decide if the actuator can fulfil the requirements. The blocking force is the force that the actuator gives when it is restrained so no strain can be developed, but a force will be developed against the object. To illustrate the applicability of an actuator, a plot over displacement and force can be drawn. A line is drawn from the maximum blocking force (zero displacement) to the free displacement (zero force). An example can be seen in Figure 5.17. The relation between displacement and blocking force can also easily be seen in equations 5.16 and 5.17.

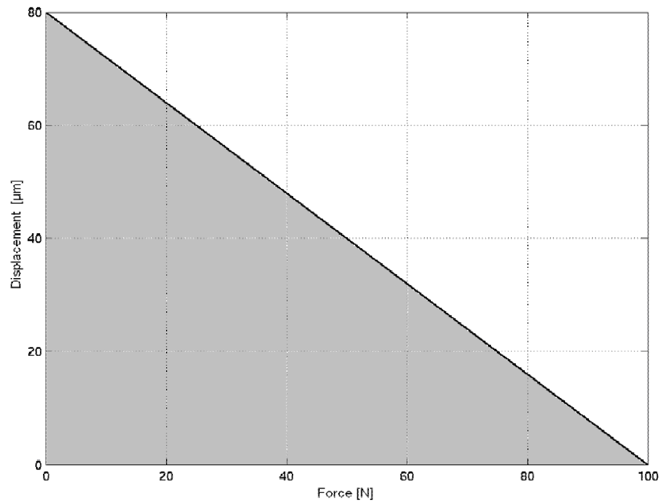

Figure 5.17 Force versus displacement for "maximum" voltage. The grey area is where the actuator can be operated

A simple mechanical model of a piezoelectric material can be shown in Figure 5.18. The mass m_{eff} is effective mass, about 1/3 of the mass of the ceramic stack. K_T is the actuator stiffness. The force F is the actuators internal force.

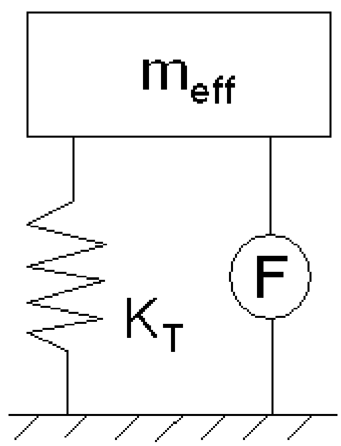

Figure 5.18 Simple mechanical model of piezoelectric material

The resonant frequency for the spring and mass system is a function of its stiffness and effective mass. The resonant frequency given in the technical data tables refers to unloaded actuators, with one end rigidly attached. The resonance frequency will be estimated to:

$$\omega_0 = \sqrt{\frac{k_T}{m_{eff}}} \tag{5.18}$$

One desirable feature of a piezo actuator is fast response. The key to obtain fast response is rapid drive voltage change, which results in a rapid position change. This is necessary in active vibration-cancellation systems.

A piezoelement can reach its nominal displacement in approximately 1/3 of the resonant frequency period, although with significant overshoot that can be seen in Figure 5.19.

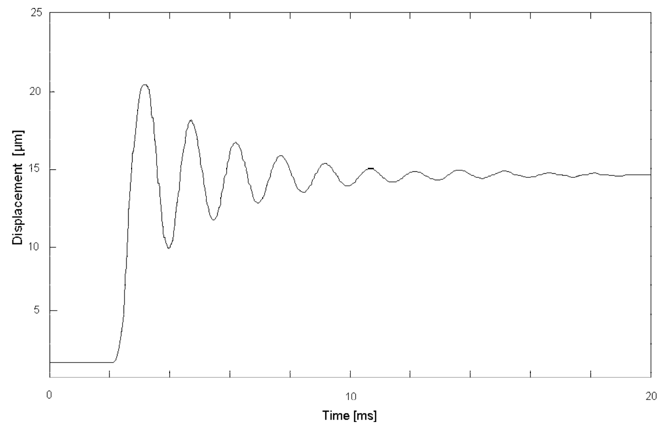

Figure 5.19 Example of a step response for a piezoelement.

5.2.2 Piezo Stack Actuator

In this section a one degree of freedom electromechanical model of a piezoelectric actuator will be developed. The linear, constitutive equations (5.14) and (5.15) will be used. The resulting model can be directly coupled to a dynamic model of an amplifier and the surrounding structure.

The equations (5.14) and (5.15) are in the axial direction (third direction):

$$S_3 = s_3^E T_3 + d_{33} E_3 \tag{5.19}$$

$$D_3 = d_{33} T_3 + \varepsilon_{33}^3 E_3 \tag{5.20}$$

First, the equations for one layer are described. D_3 is the electrical displacement and can be written as, $D_3 = \frac{1}{l \cdot w} q_l$. [2] E_3 is the electrical field in the piezoelement and can be written as, $E_3 = \frac{V_a}{h}$. By use of equation 5.20 the charge q_l that enters each layer can be obtained as:

$$\frac{1}{l \cdot w} q_l = \varepsilon_{33} \frac{V_a}{h} + d_{33} T_3$$

The equation can be rewritten to:

$$q_l = \varepsilon_{33} \frac{l \cdot w}{h} V_a + d_{33} lw T_3 = C_l V_a + d_{33} lw T_3 \tag{5.21}$$

By use of equation (5.21) we can state the equations for a stack with n layers. The charge entering all layers will be $q=nq_l$ and the total capacitance $C = nCl \ = \ \varepsilon_{33} \frac{nlw}{h}$. Equation (5.21) for n layers becomes:

$$q_l = CV_a + d_{33} nlw T_3 \tag{5.22}$$

Solving equation (2.22) for voltage:

$$V_a = \frac{1}{C} q - \frac{d_{33} nlw}{C} T_3 = \underbrace{\frac{1}{C} q}_{I} - \underbrace{\frac{d_{33}}{\varepsilon_{33}} hT_3}_{II} \tag{5.23}$$

The first component, I, is the direct capacitive effect and the second part, II, is the contribution from the mechanical stress. Replacing the electric field with its voltage relation in equation 5.19 the following equations are obtained:

$$S_3 = s_{33} T_3 + d_{33} E_3 = s_{33} T_3 + d_{33} \frac{V_a}{h} = s_{33} T_3 + \frac{d_{33}}{h} \left(\frac{1}{C} q - \frac{d_{33}}{\varepsilon_{33}} hT_3 \right) = \frac{d_{33}}{nlw\varepsilon_{33}} \cdot q + (1-k^2) s_{33} T_3 \tag{5.24}$$

Introducing the electromechanical coupling coefficient:

$$k^2 = \frac{d_{33}^2}{s_{33} \cdot \varepsilon_{33}} \tag{5.25}$$

Equation 5.24 could be rewritten to:

$$T_3 = \frac{1}{s_{33} \left(1 - k^2\right)} \left(S_3 - \frac{1}{nlw} \frac{d_{33}}{\varepsilon_{33}} q \right) \tag{5.26}$$

The strain in the actuator will cause displacement according to:

$$x = nhS_3 \tag{5.27}$$

The force from the stress in the actuator becomes:

$$F_a = lwT_3 \tag{5.28}$$

The relation between current i_a and charge q are:

$$q = \int i\, dt \tag{5.29}$$

From equations (5.26) to (5.27) the connection between current, displacement and force are conducted:

$$F_a = \frac{1}{lws_{33}(1-k^2)}\left(\frac{1}{nh}x - \frac{1}{nlw}\frac{d_{33}}{\varepsilon_{33}}\int i\, dt\right) \tag{5.30}$$

The equations 5.30 and 5.23 are implemented in Matlab as shown in Figure 5.18:

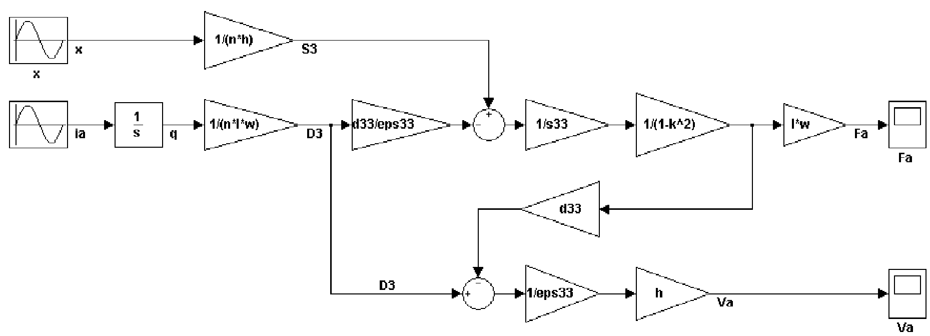

Figure 5.20 Electromechanical model of a piezo stack actuator

5.2.3 Amplified Piezo Actuators

The increase in displacement gained with a mechanical amplifier reduces the actuator's stiffness and maximum operating frequency. They consist of a piezoelectric element and either a mechanical or hydraulic displacement amplifier. A more detailed discussion is out of scope for this thesis work.

5.2.4 Simulation of Piezoelectric Stack Actuator Model

For validation we have constructed a simple model of an actively controlled structure. It is assumed that the actuator is bonded rigidly to the mass M. K is the parallel spring and C is the parallel damper. The mechanical model is in Figure 5.19. The structure is implemented in Simulink as showed in Figure 5.19.

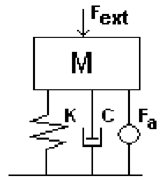

Figure 5.21 Mechanical diagram of the controlled structure

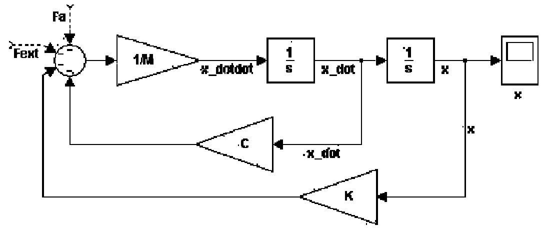

Figure 5.22 Simulink block diagram of the controlled structure

To minimize the vibration in the mounting point, the force $F_a + K \cdot x + C \cdot \dot{x}$ must be small. In this example this is obtained using proportional feedback coupling with the constant K_{reg}. The simulink block diagram of the complete system is shown in Figure 5.23.

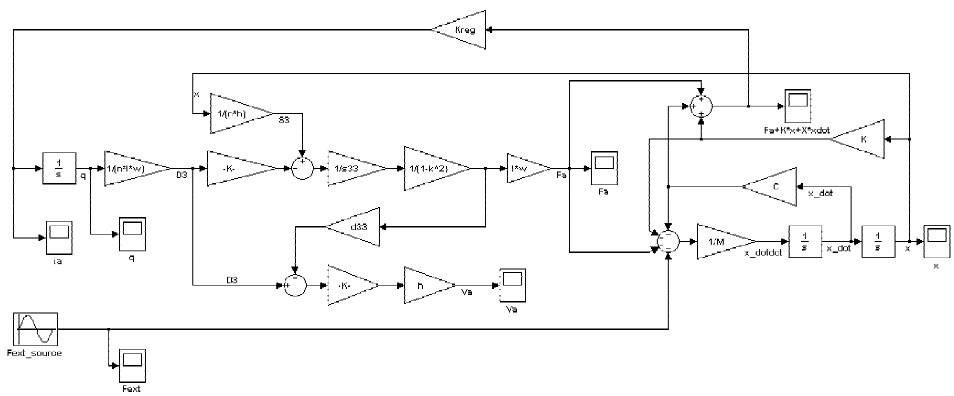

Figure 5.23 Simulink block diagram of the complete system.

The system was run with an external sinusoidal disturbance force F_{ext} with amplitude of 20 N and frequency 100 Hz. The results are presented in Figures 5.23, 5.24; they are plotted with the regulator on and off. The results shown that the displacement is higher for the case with actuator and regulator which is as expected because it will decrease the force against the ground as can seen in Figure 5.24.

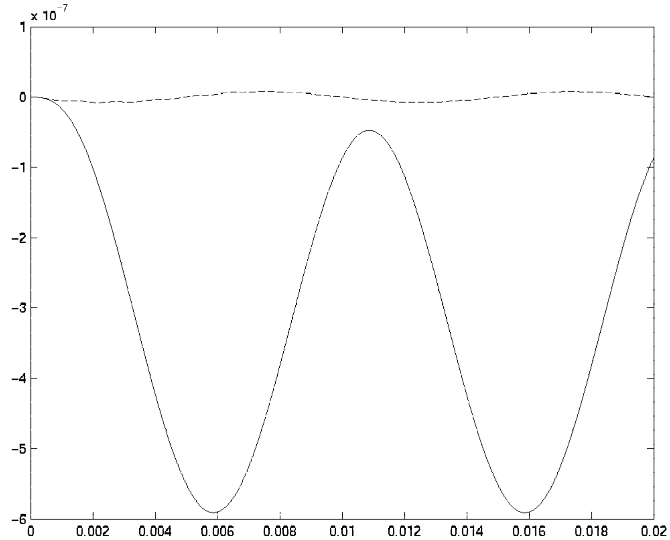

Figure 5.23 The solid line is the displacement x with actuator and regulator, the dotted line is without actuator.

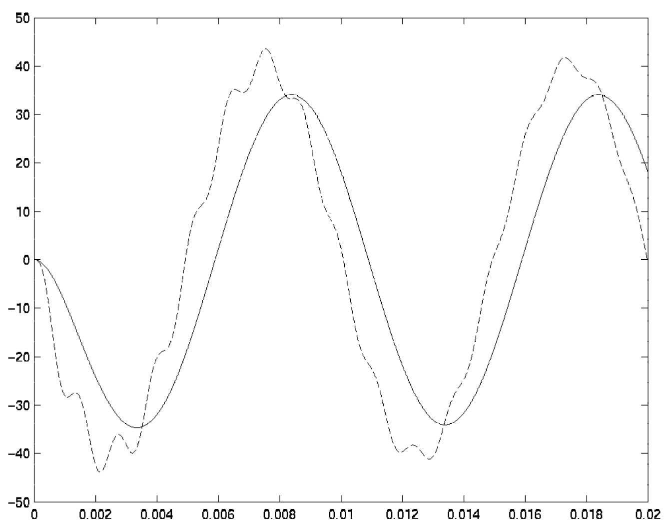

Figure 5.24 The solid line is the force against the ground $F_a + Kx + C\dot{x}$ with actuator and regulator, the dotted line is without actuator

The simulation results are as expected but the physical measurements must be carried out to completely verify the model, but that is beyond the scope of this thesis.

5.2.5 Validity of the Model

Some assumptions have been made when deriving the model of a piezoelectric stack actuator: The piezoelectric element has been seen as pseudo-static. Temperature dependency and changes over life time have been ignored.

Temperature has two different effects on piezoelectric actuators: linear thermal expansion and temperature dependency of the piezoelectric effect.

Thermal stability of piezoelectric ceramics is better than other materials like steel or aluminium. The change is usually specified by the manufacturer and is given as relative change in length ΔL/L per unit change in temperature. The following values could be used as guidelines: $11 \cdot 10^{-6}$m/°K for high voltage piezoelectric ceramics and $-3.5 \cdot 10^{-6}$m/°K for low voltage ceramics. For example, for the piezoelectric stack actuator DPA60 from Cederat, it is $0.20 \cdot 10^{-6}$m/°K and for their amplified actuator APA120ML it is $1.17 \cdot 10^{-6}$m/°K. The thermal expansion values change with temperature and are normally given for room temperature. [8]

The piezoelectric effect is based on electric field and thus works down to zero degrees Kelvin. But temperature changes cause a voltage to appear across the electrodes. This is due to the pyroelectric properties of piezoelectric ceramic. Temperature also affects other properties of piezoelectric ceramics such as elastic, dielectric and piezoelectric coupling. There is no general trend and each dependence must be measured separately. [9] The piezoelectric effect varies with temperature for several reasons, but around room temperature it is very stable. For example, at cryogenic temperatures (ultra-low temperatures) it is approximately 20 to 30 % of its room temperature value. Figure 5.25 shows the temperature dependency for the piezo effect (piezo gain) given by Physik Instrumente. [8]

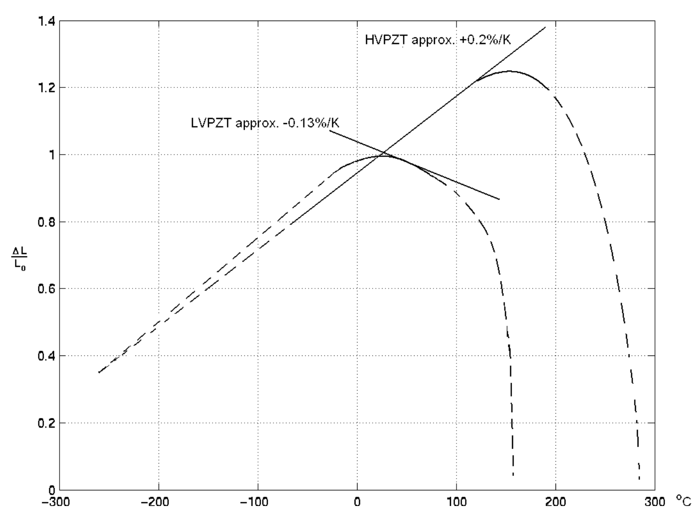

Figure 5.25 Temperature dependency for the piezoelectric effect, given by Physik Instrumente. [51]

To create the piezoelectric effect in a piezoelectric ceramic it is heated during the polarization and an electric field is applied to allow alignment of the dipoles. Conversely, a poled piezoelectric material will depolarize when heated above the maximum allowed operating temperature. It can be somewhere around 100°C. But there are piezoelectric actuators that can operate in higher temperatures.

The life time is pretty difficult to estimate but tests at Physik Instrumente have shown that piezoelectric elements can perform billions of cycles under suitable conditions. And at Piezo Systems they have had a piezoelectric fan running constantly since 1982. But no conclusive tests have been done.

But some things can be said. Generally, as with capacitors, the lifetime of a Piezo is a function of the applied voltage and the average voltage should be kept as low as possible. But many other factors affect the life time, such as temperature, humidity, voltage, acceleration, load, operating frequency and insulation materials.

Chapter 6

MODELLING OF AN ACTIVE HYDRAULIC ENGINE MOUNT

It exist different principles of active engine mounts. The one that will be presented and modelled in this chapter are based on the commonly used passive hydraulic engine mount. Use of the present passive hydraulic engine mount extended with an active part would benefit from the well-known passive characteristics as well as the active. The type of active hydraulic mount that is modelled in this chapter is a hydraulic mount with a voice coil actuator as the active element. This seems to be the most common principle today for an active hydraulic engine mount. Therefore, it is most interesting to construct a model for this type of mount. During this thesis work we have not found any published papers on construction of an active hydraulic engine mount model that works both for low and high frequencies. The models that we have found are only valid up to around 40 Hz. To reduce the time to design the mount and its application, it is desirable to construct a model to predict the behaviour of the system before it is physically assembled. A good model is also a key element when designing the control algorithm for the system.

First a regular passive hydraulic engine mount will be modelled and the results will be compared to measured data. The object is not to obtain the optimum parameters for the passive hydraulic engine mount model. The purpose is to show that the modelling technique is satisfactory and can serve to model the active hydraulic engine mount. It is important that the model has a high known correspondence to reality, since the amount of data that can be used to verify the active hydraulic mount model is limited.

After the passive hydraulic mount is modelled an actuator is included in the mount to provide mechanical energy. The main structure will be similar to the passive hydraulic engine mounts. The difference is that that the decoupler is replaced with a diaphragm that is attached to a voice coil actuator. That will be shown in section 6.1.4 Complete Active Engine Mount.

6.1 Passive part

The general principles of a hydraulic engine mount are shown in Figure 6.1. It consists of two parts. The rubber part supports the engine weight and the hydraulic part gives the mount its main dynamic behaviour.

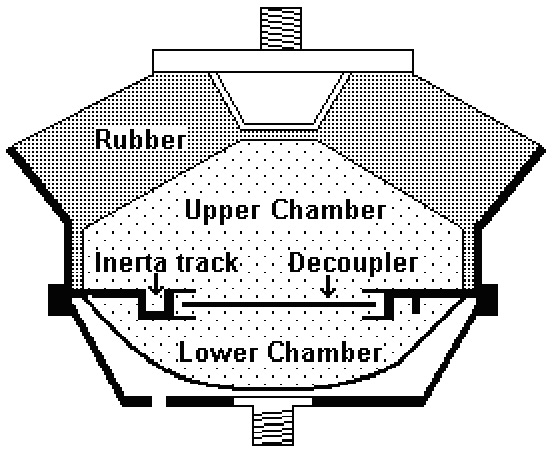

Figure 6.1 Passive hydraulic engine mount

The hydraulic part consists of two fluid-filled chambers. The fluid can be a mixture of water and anti-freeze liquid such as glycol. The bottom of the lower chamber is of thin rubber; the chamber acting as a reservoir when fluid is forced into it. The two chambers are connected through an inertia track (damping channel) and a decoupler. The decoupler is a rubber or fabric diaphragm which can move freely in the passage connecting the two chambers.

Hydraulic mounts with only an inertial track can be tuned to give high damping at low frequencies. But this kind of mount gives higher dynamic stiffness at all frequencies for lower amplitudes compared to an elastomeric mount. To solve that problem a decoupler is used to make the hydraulic mount amplitude-dependent. The decouple works like a floating piston. At low amplitudes it allows the fluid to pass between the chambers and the mount behaves like an elastomeric mount. This gives low dynamic stiffness and provides good vibration isolation. Higher amplitudes will force the decoupler to saturate, which forces the fluid through the inertial track and thus increases the dynamic stiffness. To obtain a model of a passive hydraulic engine mount, the different parts of the mount will be modelled. The affect that the surrounding structure has on the mount is parameterized by X, which is the displacement of the mount. It is also the input to the system and the output will be the force F_T transmitted to the frame.

The rubber around the upper chamber (chamber one) is modelled with its stiffness k_r and damping property b_r. The rubber will add volumetric compliance C1 to the upper chamber. The upper rubber structure also works as a piston on the upper chamber with an effective pumping area A_p. The lower chamber has volumetric compliance C_2, which comes from the rubber below. The flow that passes through the decoupler is called Q_d and the flow through the inertial track is called Q_i.

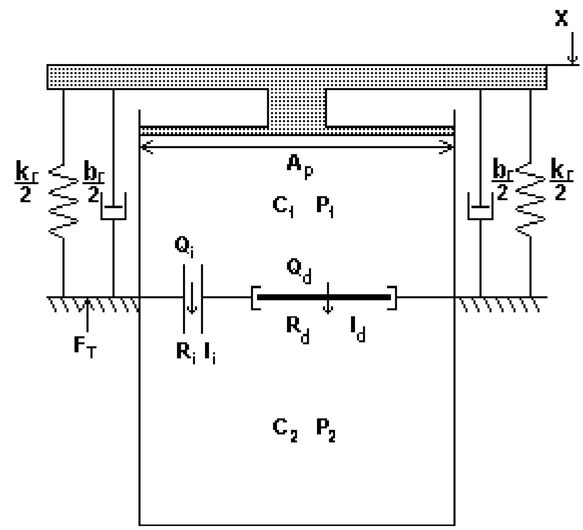

Figure 6.3 Schematic of a hydraulic engine mount

The continuity equations for the internal dynamics of the system can now be derived:

$$C_1 \dot{P}_1 = A_p \dot{X} - Q_i - Q_d \tag{6.1}$$

$$C_2 \dot{P}_2 = Q_i + Q_d \tag{6.2}$$

6.1.1 Inertia Track

The inertia track is a long column of fluid between the two chambers which has a damping effect. It is assigned the resistance $\widetilde{R}_i$ and the effective inertia I_i. The momentum equations become:

$$P_1 - P_2 = I_i \dot{Q}_i + \widetilde{R}_i Q_i$$

This equation describes the linear behaviour of the inertia track. From experimental results the resistance unlike the inertia does not exhibit constant behaviour [57]. Since the resistance is dependent on the Reynolds number will it change because of the oscillatory flow. According to A.A. Geisberger [8], the resistance is dependent on the amplitude and frequency of the flow and can be modelled as: $\widetilde{R}_i = R_i + R_i^{'} |Q_i|$. Where

R_i represents the laminar resistance term and $R_i^{'}$ represents the resistance in the turbulent region. The non-linear momentum equation becomes:

$$P_1 - P_2 = I_i \dot{Q}_i + (R_i + R_i^{'}|Q_i|)Q_i \qquad (6.3)$$

6.1.2 Decoupler

Most of the amplitude-dependent behaviour originates from the decoupler. When the flow is oscillating across the decoupler orifice and the decoupler plate does not have contact with the cage it is considered to be free. In that case the decoupler system behaves like a simple orifice. The decoupler is assigned inertia and a resistance value, similar to the inertia track. The non-linear equation for the free decoupler now becomes:

$$P_1 - P_2 = I_d \dot{Q}_d + (R_d + R_d^{'}|Q_d|)Q_d \qquad (6.4)$$

When the decoupler contacts the cage it will stop the flow. To model this flow stop a large resistance is added to the equation (6.4) and the equation become:

$$P_1 - P_2 = I_d \dot{Q}_d + (R_d + R_d^{'}|Q_d| + R_{stop})Q_d \qquad (6.5)$$

The position of the decoupler plate is called X_d and is related to the decoupler volume V_d by $V_d = X_d A_d$ where A_d is the decoupler area. The decoupler volume can be integrated from the flow of fluid through the decoupler. In Figure 6.4 the corresponding position of the decoupler can be seen. The expression for X_d becomes:

$$X_d = \frac{1}{A_d} V_d = \frac{1}{A_d} \int_0^T Q_d \, dt \qquad (6.6)$$

Figure 6.4 The positions and sign for the decoupler

When the decoupler contacts the cage the resistance R_{stop} should be high to reduce the flow to zero in the decoupler, and when it is free it should be low so it doesn't have any affect on the flow. The resistance R_{stop} should also be low when the decoupler contact the cage but the flow change direction. This because the decoupler then can move freely in the wanted direction even if it still contact the cage. The desired values of R_{stop} resistance can be seen in Figure 6.5.

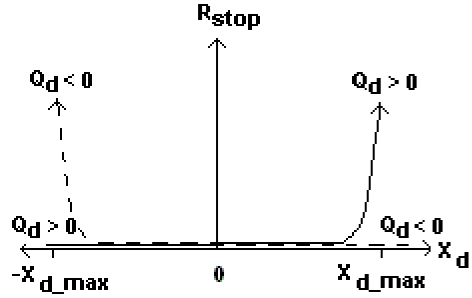

Figure 6.5 The nonlinear switching function of the resistance R_{stop}

To describe the function in Figure 6.4 an exponential function is used with an arctangent function to get the switching behaviour of the exponential function [8]. Three constants R_0, X_0 and Q_0 are also used to get the resistance R_{stop} in the right domain. The function R_{stop} will be $R_{stop} = R_0 e^{(X_d / X_0)\arctan(Q_d / Q_0)}$. The momentum equation for the decoupler becomes:

$$P_1 - P_2 = I_d \dot{Q}_d + (R_d + R_d' \left|Q_d\right| + R_0 e^{(X_d / X_0)\arctan(Q_d / Q_0)}) Q_d \tag{6.7}$$

6.1.3 Transmitted Force

When the decoupler comes into contact with the cage, the transmitted force will be as in equation 6.8. That is, because the force can work on the total area A_p it can be seen as the decoupler area A_d becomes almost zero and the flow Q_d become negligible.

$$F_T = k_r X + b_r \dot{X} + A_p P_1 \tag{6.8}$$

The force from the rubber is modelled as $k_r X$ and $b_r \dot{X}$ like a normal spring and damper system. The force from the pressure in the upper chamber is $A_p P_1$. The lower chamber's bottom rubber is soft and acts as a reservoir; therefore it only contributes with a negligibly small force.

To capture the force from the decoupler, a similar method as in the decoupler resistance case is used. To model the changes in force from the decoupler, the decoupler area will be modelled as dependent on the decoupler position and the pressure differential. The total area which the upper chamber pressure affects will be $A_p - A_{d\_fnc}$, and this describes the behaviour discussed. When the decoupler plate comes into contact with the cage the decoupler is modelled as zero. Though, when the pressure differential reverses direction the decoupler will appear open even if the inertia still holds the decoupler at rest. The difference from the conditions used for decoupler resistance is that the area is directly related to pressure differential and not flow. The desired behaviour for the switching function can be seen in Figure 6.6.

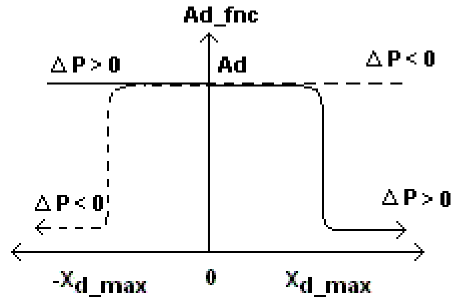

Figure 6.6 The decoupler area function A_{d_fnc}

According to A. A. Geisberger [57], the function can be modelled as:

$$A_{d\_fnc} = \frac{1}{\pi} A_d \left(\frac{\pi}{2} - \arctan\left(\frac{(2/\pi) X_d \arctan((P_1 - P_2)/P_0 - X_{d\_\max})}{X_1} \right) \right) \quad (6.9)$$

Where P_0 and X_1 is constants and X_{d_max} is the maximum displacement in the decoupler as can be seen in Figure 6.4.

The transmitted force now becomes:

$$F_T = k_r X + b_r \dot{X} + (A_p - A_{d\_fnc})(P_1 - P_2) + A_p P_2 + A_d (R_d + R_d^{'} |Q_d|) Q_d \quad (6.10)$$

6.1.4 Complete Passive Mount Model

The total model of the hydraulic mount can now be summarized from equation (6.1) to (6.10):

The continuity equations for the mount:

$$C_1 \dot{P}_1 = A_p \dot{X} - Q_i - Q_d$$
$$C_2 \dot{P}_2 = Q_i + Q_d$$

The momentum equations for the mount:

$$P_1 - P_2 = I_i \dot{Q}_i + (R_i + R_i^{'} |Q_i|) Q_i$$
$$P_1 - P_2 = I_d \dot{Q}_d + (R_d + R_d^{'} |Q_d| + R_0 e^{(X_d / X_0) \arctan(Q_d / Q_0)}) Q_d$$

Where X_d is:

$$X_d = \frac{1}{A_d} \int_0^T Q_d \, dt$$

The transmitted force:

$$F_T = k_r X + b_r \dot{X} + (A_p - A_{d\_fnc})(P_1 - P_2) + A_p P_2 + A_d (R_d + R_d^{'} |Q_d|) Q_d$$

$$A_{d\_fnc} = \frac{1}{\pi} A_d \left(\frac{\pi}{2} - \arctan\left(\frac{(2/\pi) X_d \arctan((P_1 - P_2)/P_0 - X_{d\_\max})}{X_1} \right) \right)$$

To enable simulations the equations are implemented in Matlab. The system can then be implemented as a numeric differential problem in Matlab.

6.1.5 Validity of the Passive Engine Mount Model

Some assumptions have been made constructing the model. The effects that have not been taken into account are as follows:

- Rubber stiffness k_r depend on excitation amplitude, driving frequency and preload force.
- Damping b_r depend on excitation amplitude, driving frequency and preload force.
- Effective pumping A_p depends on the preload force.
- Upper chamber compliance C_1 depends on driving frequency, volume amplitude and preload force.
- Some hydraulic engine mounts permit high resistance flow through the decoupler even when the decoupler contacts the cage.
- Lower chamber compliance C_2 depends on the preload force.

This is according to A. A. Geisberger [4]. One solution is to measure the different parameters' dependency and take that into account. One parameter that is heavily influenced by frequency is the stiffness of the rubber. The rubber stiffness increases, roughly, with 1 N/mm per frequency step (Hz); depending of course on the chosen rubber material. This dependency is relatively linear and can easily be modelled. It could be modelled as: $k_r = k_{r\_static} + k_{r\_dynamic} \omega_X$ where k_{r_static} is the rubber stiffness for very low frequencies and $k_{r\_dynamic}\omega_X$ is the change in stiffness by frequency and there ω_X is the frequency of excitation.

The other effects are not in order of size to have a significant affect on the model. This conclusion is taken from studying the results that the derived model gives without consideration of the effects and studying of measurements done [4].

6.1.5.1 Superimposed inputs

One interesting question about the validity of the model is the reaction to superimposed input. As far as the author knows, no measurements of superimposed inputs have been done successfully for a hydraulic mount. In an attempt to verify the correctness of the model, measurements were performed at Volvo Cars. The results are inconclusive, most likely because of equipment limitations.

However, measurements were also done on a conventional rubber grommet to verify the function of the measurement equipment. The result is seen in Figure 6.8. The dynamic stiffness for the base frequency 10 Hz with 1 mm amplitude was 661 N/mm. The 0.1 mm superimposed force dynamic stiffness can be viewed in Figure 6.8.

The conclusion that can be made from the measurements on the rubber is that the dynamic stiffness is lower for superimposed input then it would be without the base frequency. This is probably a result of the higher base amplitude making the rubber soften. To really understand the behaviour of the rubber in the case of superimposed input, a more detailed investigation is needed. It is likely that the rubber part of the hydraulic mount behaves in a similar way as the rubber grommet. This makes it likely that the dynamic stiffness for the hydraulic engine mount is lower in the superimposed case similar to the case of a rubber grommet.

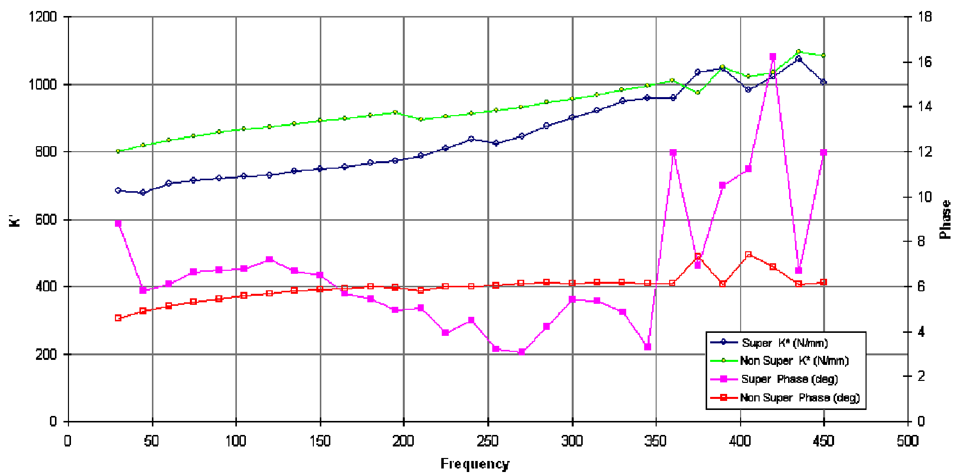

Figure 6.8 Dynamic stiffness for conventional rubber grommet for superimposed input and non superimposed input. Base frequency 15 Hz with 1 mm amplitude and 0.1 mm superimposed.

6.1.6 Experimentally Validation of the Passive Engine Mount Model

To verify the model of the passive hydraulic mount, measurements were done on the hydraulic engine mount. The parameters needed for the model were identified by studying their influence on the frequency response. In Figures 6.9 to 6.19, the different parameters' affects on the model can be studied. The model was linearized to make these plots by letting the flow Q_d be zero at low frequencies, as the amplitude normally is high (1 mm) which makes the decoupler to bottom. And Q_i becoming zero at high frequencies because the amplitudes are normally small (0.1 mm) and all flow goes through the decoupler. R_i' and R_d' is zero in the examples.

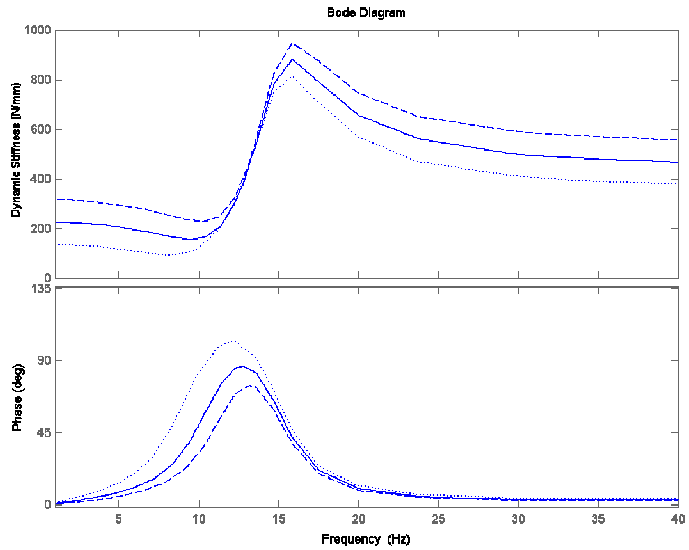

Figure 6.9 Model response for change in the rubber stiffness k_r.
· · · · 40 % increased values of the parameter
- - - 40 % decreased values of the parameter

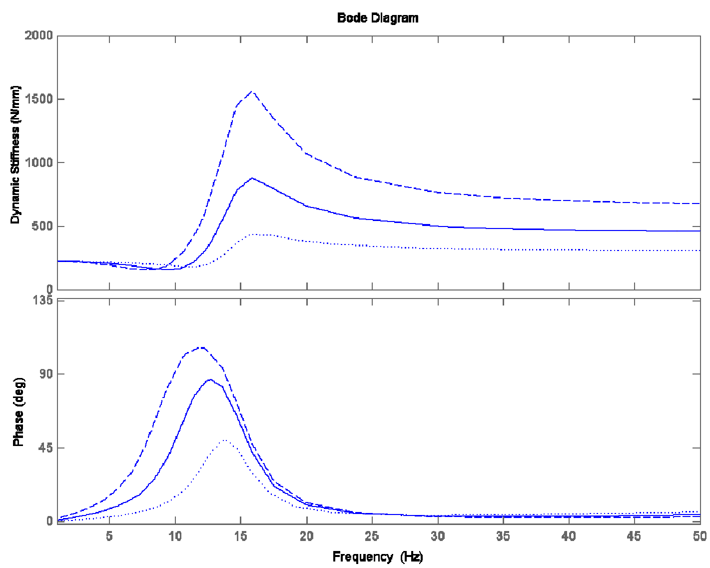

Figure 6.10 Model response for change in the affective pumping area A_p.
· · · · 40 % increased values of the parameter
- - - 40 % decreased values of the parameter

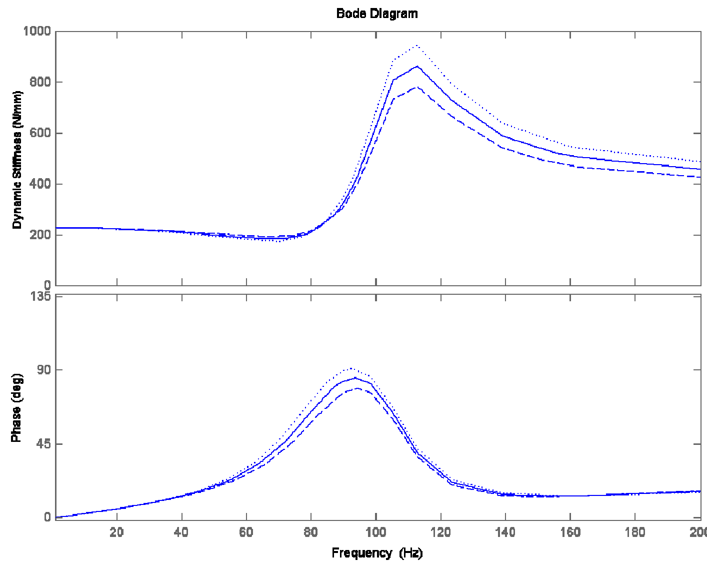

Figure 6.11 Model response for change in the decoupler area A_d.
· · · · 40 % increased values of the parameter
- - - 40 % decreased values of the parameter

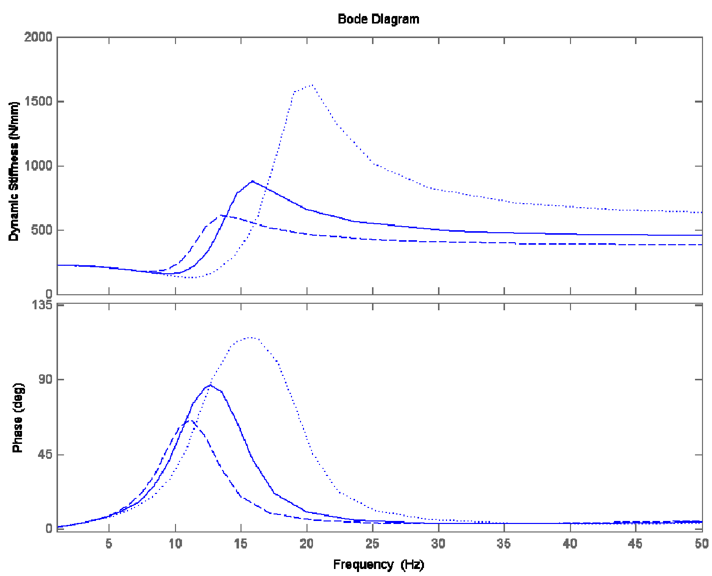

Figure 6.12 Model response for change in the upper compliance parameter C_1
· · · · 40 % increased values of the parameter
- - - 40 % decreased values of the parameter

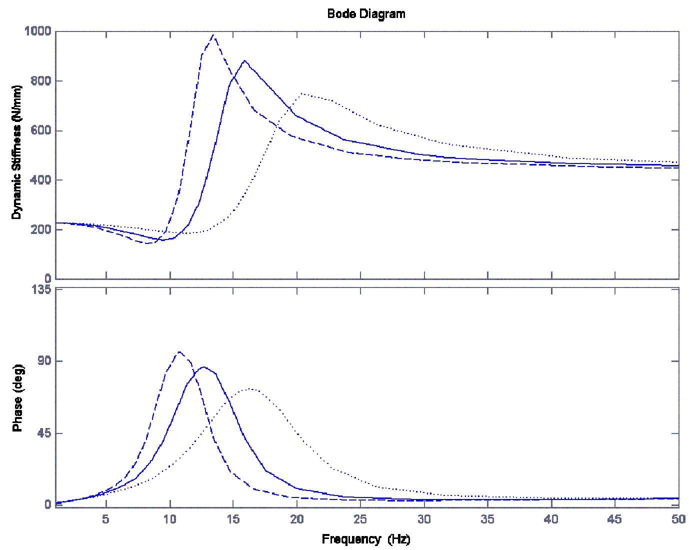

Figure 6.13 Model response for change in the inertia for inertia track I_i.
· · · · 40 % increased values of the parameter
- - - 40 % decreased values of the parameter

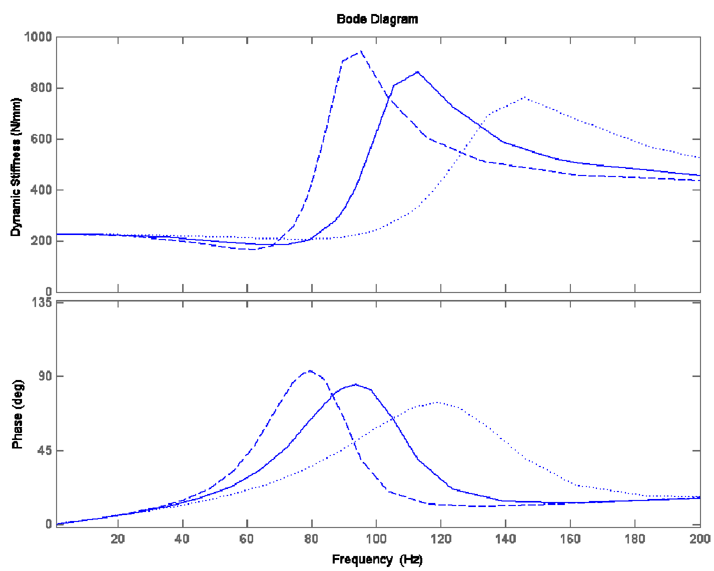

Figure 6.14 Model response for change in the inertia for decoupler track I_d.
· · · · 40 % increased values of the parameter
- - - 40 % decreased values of the parameter

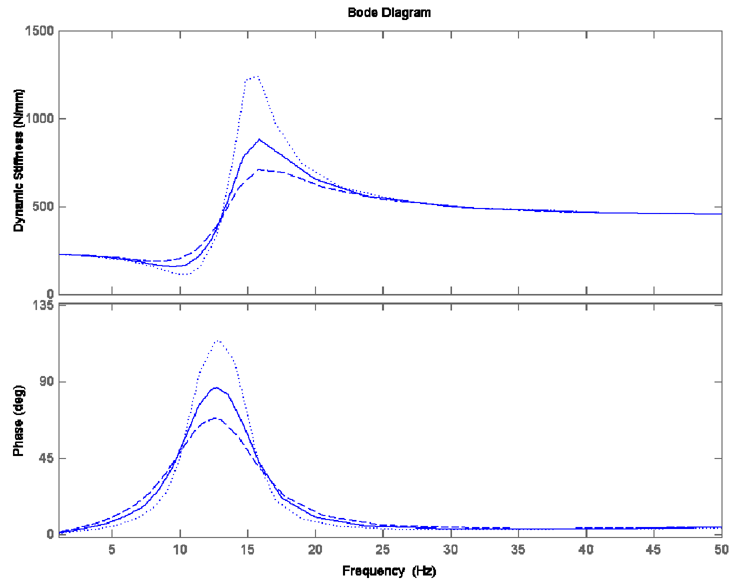

Figure 6.15 Model response for change in the resistance for inertia track R_i.
· · · · 40 % increased values of the parameter
- - - 40 % decreased values of the parameter

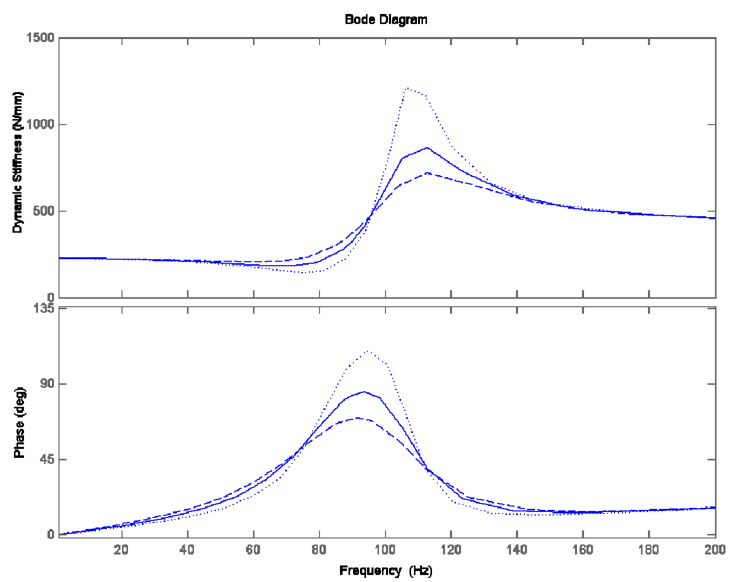

Figure 6.16 Model response for change in the resistance for decoupler track $R_{d.}$
· · · · 40 % increased values of the parameter
- - - 40 % decreased values of the parameter

The resulting model and the measured results for 1 mm amplitude of X can be seen in Figure 6.17. This result must be seen as satisfactory when the objective was not to optimize the parameters for the model but to study the model behaviour. For higher frequencies the results is shown in Figure 6.18. As can be viewed are the results inconclusive after 350 Hz. This is probably an effect of the measurement equipment limitations.

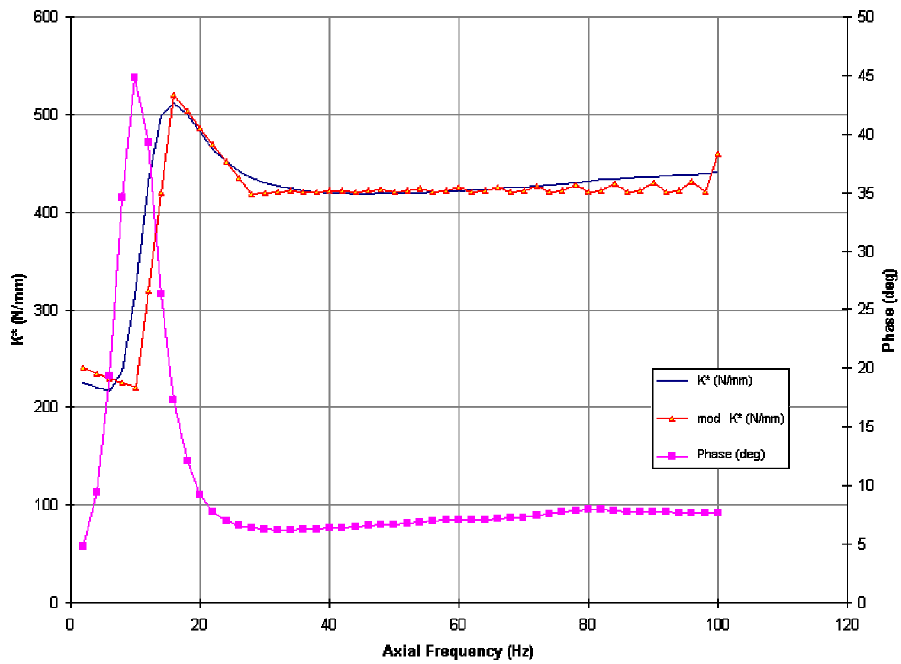

Figure 6.17 Measured and calculated response for 1 mm amplitude for a hydraulic engine mount.

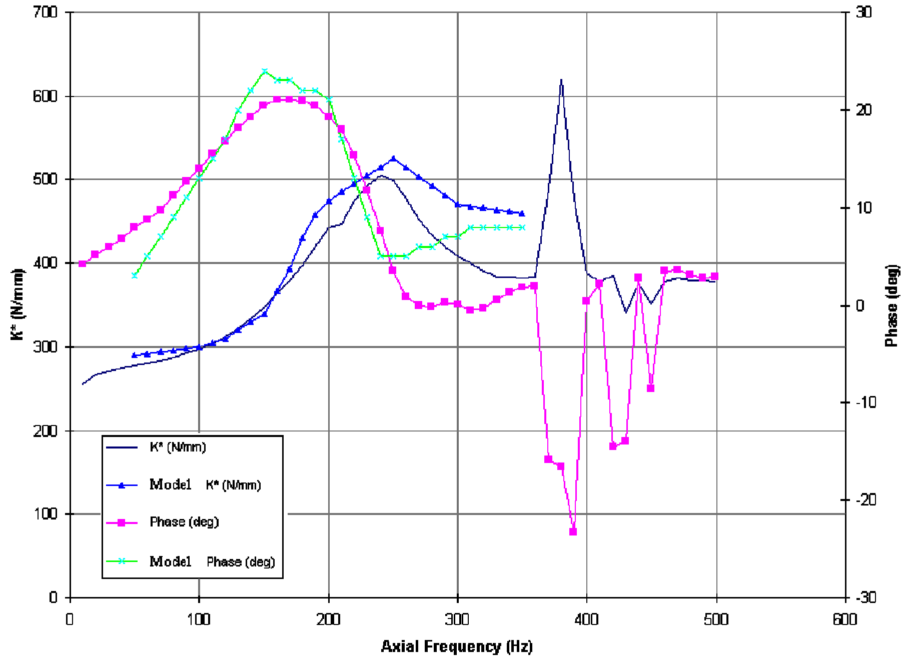

Figure 6.18 Measured and calculated response for 0.1 mm amplitude for a hydraulic engine mount.

6.2 Complete active engine mount

Different techniques exist to go from a passive engine mount to an active engine mount. However, there are several different ways to provide controlled mechanical energy to the engine mount system. The secondary vibrations produced by the active engine mount are used to cancel the primary vibrations. The main principle is to replace the decoupler in the passive hydraulic engine mount model with a diaphragm connected to a voice coil actuator; that will be used to provide mechanical energy to the engine mount system.

Figure 6.20 Schematic of a decoupler

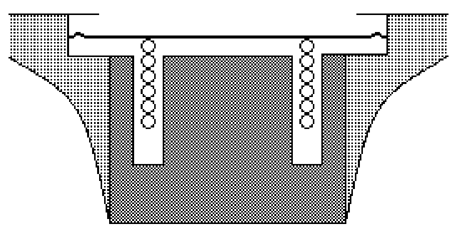

Figure 6.21 Schematic of a diaphragm

Figure 6.20 shows a schematic of a decoupler, and Figure 6.21 a schematic of a diaphragm. The coil of the voice-coil actuator is mounted on the diaphragm and the dark shaded part symbolizes the magnet. The characteristics and function of a voice coil are described in Chapter 5. The difference between the decoupler and the diaphragm is that the diaphragm does not have the same dynamic stiffness amplitude-dependency. This is because it is never totally free when it is attached to the side. Instead, the dynamic stiffness can now be controlled by controlling the diaphragm position with the voice coil actuator. In theory this will make it possible to have a dynamic stiffness that is zero for higher frequencies, which will give total vibration isolation at these frequencies. This will be derived if the actuator forces the mount to follow the vibration displacements from the engine, which will isolate the vibration from the engine to the frame. It could also be used to make the engine mount stiffer at certain frequencies by providing an anti-phase force to the mount.

The diaphragm is modelled with a stiffness k_{dia} and damping properties b_{dia}. This makes the diaphragm model linear differing from the non-linear decoupler model. This because the diaphragm is not free floating and we assume it never contacts the top or bottom of its cage as the decoupler does. The force from the voice coil is called F_a. The dynamics of the force F_a can be derived from the voice-coil actuator model in Chapter 5. Basically, it is proportional to the current provided to the actuator. The voice coil could also be controlled with voltage, which would give a different dynamic behaviour. A_{dia} is the area of the diaphragm and the force from the pressure in the upper chamber is called F_{P_1}. The mechanical representation of the diaphragm is seen in Figure 6.22.

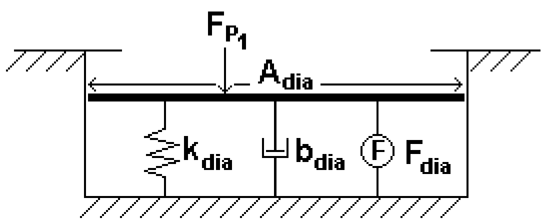

Figure 6.22 Mechanical representation of a diaphragm

Because pressure is force per area a fluid momentum equation similar to the one for the decoupler (6.7) can be derived. The pressure from the lower chamber P_2 which the decoupler was affected by is replaced with the force from the stiffness and damping of the diaphragm, together with the force from the actuator. The equation becomes:

$$P_1 - (k_{dia}X_{dia} + b_{dia}\dot{X}_{dia} + F_{dia}) / A_{dia} = I_{dia}\dot{Q}_{diad} + (R_{dia} + R_{dia}^{'}|Q_{dia}|)Q_{dia} \tag{6.11}$$

X_{dia} is the position of the diaphragm, I_{dia} is the effective inertia for the liquid diaphragm column and the coil. The resistance is assumed to be dependent on the amplitude and frequency of the flow in the same way as in the passive hydraulic mount case. The frequency and amplitude dependency is modelled with R_{dia} that represents the laminar resistance term, and $R_{dia}^{'}$ that represents the resistance in the turbulent region.

The force is calculated in a similar way as in the passive case. The force equation (6.10) can be simplified because the diaphragm does not flow freely in its cage. The transmitted force then will be the sum of the force from the upper chamber pressure P_1 on the chamber floor and the force from the rubber k_rX and $b_r\dot{X}$: like a normal spring damper system. The differences here, as mentioned, are that the diaphragm is not free and the pressure affects the total floor area (effective pumping area) Ap. The effect of the pressure in the lower chamber P_2 can be negligible because the rubber below is soft and it acts as a reservoir. The resulting force becomes:

$$F_T = k_r X + b_r \dot{X} + A_p P_1 \tag{6.12}$$

The continuity equations will also be different in the active case. The flow into the diaphragm chamber only affects the upper chamber and the continuity equations become:

$$C_1 \dot{P}_1 = A_p \dot{X} - Q_i - Q_{dia} \tag{6.13}$$

$$C_2 \dot{P}_2 = Q_i \tag{6.14}$$

The momentum equation for the inertia track will be the same as in the passive hydraulic engine mount case when no changes to the inertia track have been done:

$$P_1 - P_2 = I_i \dot{Q}_i + (R_i + R_i^{'} |Q_i|) Q_i \tag{6.15}$$

The final model of the active hydraulic engine mount can now be summarized from equation (6.11) to (6.15):

The continuity equations for the mount:

$$C_1 \dot{P}_1 = A_p \dot{X} - Q_i - Q_{dia}$$
$$C_2 \dot{P}_2 = Q_i$$

The momentum equations for the mount:

$$P_1 - P_2 = I_i \dot{Q}_i + (R_i + R_i^{'} |Q_i|) Q_i$$
$$P_1 - (k_{dia} X_{dia} + b_{dia} \dot{X}_{dia} + F_{dia}) / A_{dia} = I_{dia} \dot{Q}_{dia} + (R_{dia} + R_{dia}^{'} |Q_{dia}|) Q_{dia}$$

The transmitted force:

$$F_T = k_r X + b_r \dot{X} + A_p P_1$$

In the same way as the decoupler position X_d was calculated (6.6) the diaphragm position X_{dia} is calculated:

$$X_{dia} = \frac{1}{A_{dia}} \int_0^T Q_{dia} dt \tag{6.16}$$

6.2.1 Validity of the complete active engine mount model

The assumptions made in constructing the model of the active hydraulic engine mount are similar to the ones for the passive hydraulic engine mount model. The affects that have not been taken into account are the same as for the passive hydraulic mount model are as follows:

- The diaphragm effective area is probably not constant when it bends.
- The stiffness k_{dia} and damping b_{dia} of the diaphragm probably depend on the excitation amplitude and driving frequency.

It would be useful to be able to make more accurate, conclusive measurements on a specific diaphragm. This is, however, beyond the scope of this thesis.

6.2.2 Validation of complete active engine mount

The model has been verified with experimental data provide by a manufacturer. The broken line in the frequency response seen in Figure 6.23 is for their active engine mount in passive mode. The solid line is the calculated model frequency response. The agreement is very good, as can be seen.

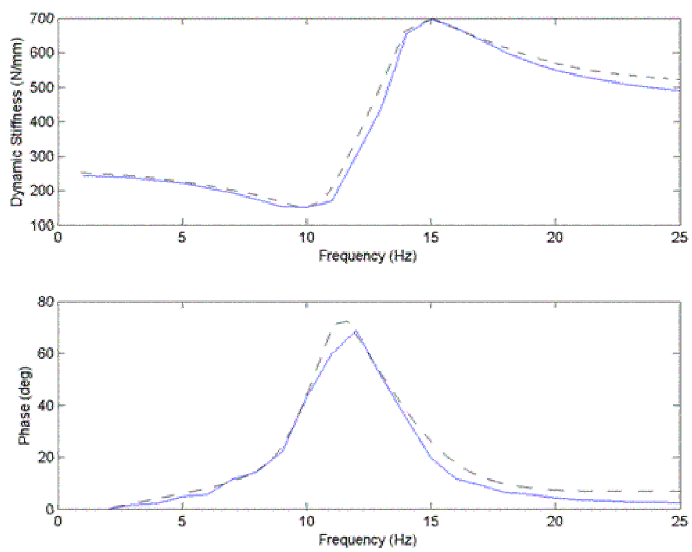

Figure 6.23 Frequency response for an active engine mount. Dotted line is the measured frequency response and the solid line is the model frequency response.

For higher frequencies, no data was found to verify the model. If a decoupler were to be used, similar resonance will be present at 200 Hz. This is because the resonance phenomenon of the column of fluid in the diaphragm chamber is similar to that in the decoupler passage. The model's frequency response for higher frequencies can be seen in Figure 6.24. The results are as expected with a resonance peak at 200Hz. This is because the liquid column, diaphragm and the coil of the actuator are resonating.

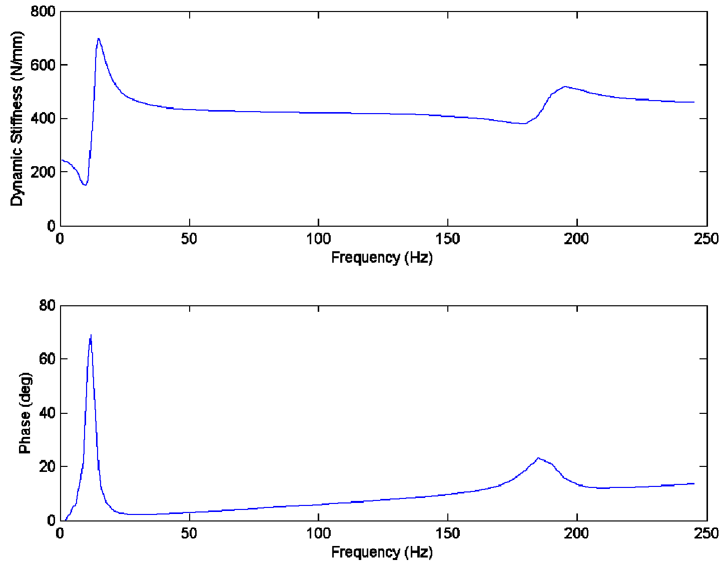

Figure 6.24 High frequency response for the model

6.2.3 Linearization of the Complete Active Engine Mount Model

The non-linearity in the active engine mount model origin in the non-linear behaviour of the resistance in the momentum equations for the inertia track and the diaphragm. If we assume that the non-linearity in the fluid column resistance for the inertia track $(R_i + R_i^{'}|Q_i|)$ and diaphragm $(R_{dia} + R_{dia}^{'}|Q_{dia}|)$ are small can they be replaced by $\widetilde{R}_i$ and $\widetilde{R}_{dia}$. The loss is that the non-linear behaviour of the resistances is not taken into account. Where this non-linearity phenomenon originates from is described in section 6.1.1, where it is added to the model. Have this affect the model has to be investigated further but the results seen in figure 6.25 are similar to the results with the non-linear model shown in figure 6.24. This indicates that the non-linearity is quite small. But a more detailed investigation is out of scope of this work. The equation system for the active engine mount becomes:

The continuity equations for the mount:

$$C_1\dot{P}_1 = A_p\dot{X} - Q_i - Q_{dia}$$
$$C_2\dot{P}_2 = Q_i$$

The momentum equations for the mount:

$$P_1 - P_2 = I_i\dot{Q}_i + \widetilde{R}_iQ_i$$

$$P_1 - (k_{dia} X_{dia} + b_{dia} \dot{X}_{dia} + F_{dia}) / A_{dia} = I_{dia} \dot{Q}_{dia} + \widetilde{R}_{dia} Q_{dia}$$

$$X_{dia} = \frac{1}{A_{dia}} \int_0^T Q_{dia} dt$$

The transmitted force:

$$F_T = k_r X + b_r \dot{X} + A_p P_1$$

The equations now give the systems transfer function G_1, relating the transmitted force F_T to the displacement X, dynamic stiffness. Its also gives the transfer function G_2 relating the force crated by the actuator on the diaphragm F_{dia} to the transmitted force F_T. In this case a voice coil actuator is used and the force F_{dia} exerted on the diaphragm is simply proportional to the current, i.e $F_{dia} = k_1 \cdot i$.

$$F_T = G_1 X + G_2 F_{dia} = G_1 X + G_2 k_1 i \tag{6.17}$$

$$G_1 = k_r + b_s s + UZA_p^2 s$$

$$G_2 = \frac{U}{A_{dia}}$$

$$U = \frac{1}{1 + ZC_1 s + WZC_2 s} \qquad Z = \frac{k_{dia}}{A_{dia}^2 s} + \frac{b_{dia}}{A_{dia}^2} + I_d s + \widetilde{R}_{dia} \qquad W = \frac{1}{1 + I_1 C_2 s^2 + \widetilde{R}_i C_2 s}$$

The dynamic stiffness for the linear active engine mount model in passive mode, $F_{dia} = 0$, is seen in Figure 6.25.

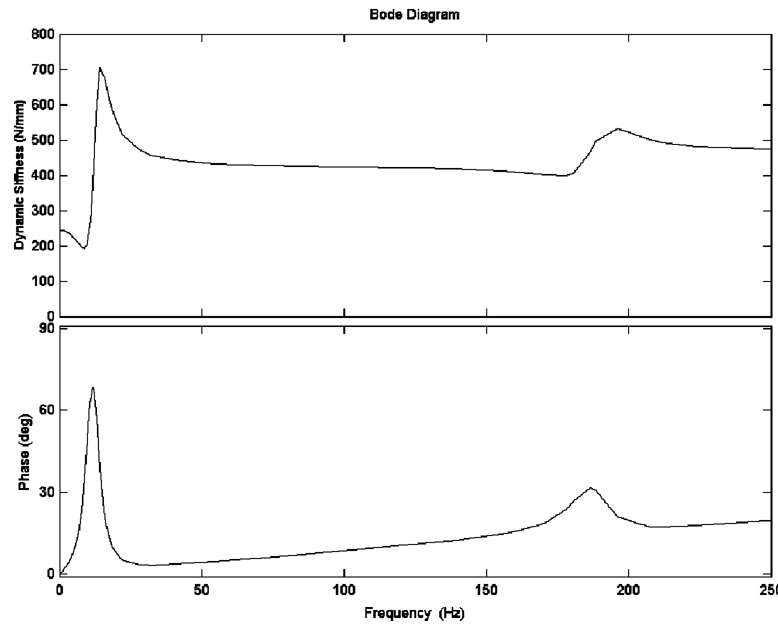

Figure 6.25 Bode diagram for linear model of active engine mount, dynamic stiffness as a function of frequency.

Chapter 7

CONCLUSIONS AND RECOMMENDATIONS

The main objective of this thesis work was to create a complete parameterized model of an active hydraulic engine mount. This was done with success and the model show great potential to predict the behaviour of the active engine mount over its complete frequency range. A benchmark of the now existing and upcoming technologies and principles was carried out to find suitable actuators for active vibration control. The actuator technology is placed in three groups depending on their suitability for active vibration control. The groups are: *promising actuators*, *selected actuators* and *other actuators*. The actuator technologies selected are the piezoelectric and the electromagnetic, because of their good performance and that their wide use. These two are discussed more in Chapter 4, where models of them are also developed. The electromagnetic actuator principle of voice coil with moving coil was chosen for use in the active hydraulic mount model.

The model of the active hydraulic engine mount can be used to predict the behavior of the mount before it is physically constructed. It is also important for designers of active engine mounts and control algorithms to understand how the design parameters affect the characteristics of the mount, which is explained in this thesis work. This can be done because the model technique enables studies of internal behavior of liquid columns and components of the mount. This gives an extra contribution to the work. A model can also shorten the period of designing an active hydraulic mount system and financial savings can be made.

Two ways to estimate the parameters for the voice coil actuator were obtained. A method was achieved to go from specification in the frequency domain to specification of the voice coil actuator. This could be used both for design and to understand the possibilities of the different voice coil actuators.

To summarize, a benchmark over actuators for active vibration control was also accomplished and an overview of the sensors used mostly for that purpose. Models for the two chosen actuators, a common passive hydraulic engine mount and an active hydraulic engine mount were created with good results.

7.1 Conclusions

Several conclusions were made in the different parts of the work:

- Several actuators are suitable for use in active vibration control as discussed in Chapter 2. A number of actuators also exist that are promising but need further development.

- When choosing sensors for active vibration control, there are several alternatives. The most widely used is the piezoelectric accelerometer, as referred to in Chapter 3.

- Two methods have been developed that can be used to estimate the parameters for the voice coil model. It is also shown that they give good results independently of whether phase information is available or not.

- The models created for the voice coil actuator with moving magnet could also be used for other types of voice coil actuators, with small changes, as described in section 5.1.3. The actuators that could be described with this model are: voice coil with moving magnet, voice coil with moving coil, and reaction mass actuators with moving coil or moving magnet.

- It is possible to go from a specification in the frequency domain to design parameters of a voice coil actuator. This can be done by use of the five step method that has been developed and is shown in section 5.1.4.

- A model of an active voice coil engine mount is developed and the model shows great potential in predicting the behavior of the mount between 0Hz and about 300Hz. Good agreement with experimental data has been achieved.

7.2 Recommendations

Recommendations for future work are made as a consequence of the experience from this thesis work:

- The actuator field is developing and information on it needs to be continuously updated.

- The model of the active voice coil engine mount could be enhanced by studies of the effects that have not been taken into account, as mentioned in section 6.2.1.

- Measurements need to be done to verify the high frequency characteristics of the active voice coil engine mount in sub-chapter 6.2.

- Study of the effect that superimposed input has on both passive and active hydraulic mounts would be interesting; it has not been done with any success as far as the authors is aware. A short introduction and measurements are given in section 6.1.5.1.

- Development of an algorithm for parameter identification for both the passive and the active hydraulic engine mount. This may be done in a similar way as for the voice coil actuator in section 5.1.1.2 and would be of great use for practical use with the models.

- The development of good hydraulic displacement expanders in recent years makes it interesting to study the possibilities of piezoelectric engine mounts. These have the potential advantages of smaller dimensions, less heat dissipations, faster response time and power consumption only at expansion [62].

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APPENDIX

Appendix A
ACTUATOR TECHNOLOGIES AND PRINCIPLES

This appendix presents further information about the technologies and principles presents in Chapter 2, together with diamagnetism and electrohydrodynamic technologies for use in active vibration control. Benefits and drawbacks are presented along with typical materials.

A.1 Diamagnetism

Definition
"Diamagnetism is the ability to reflect an external magnetic field" [35].

An external magnetic field affects the movement of electrons in a material so that it works against the external magnetic field. The subject's magnetization is directed in the opposite direction to the external magnetic field. Diamagnetism can be found in all materials, but it is hard to see because of much stronger paramagnetic or ferromagnetic qualities. The relative magnetic susceptibility is low in most diamagnetic materials except in superconductors. Superconductor materials are associated with diamagnetism because of their qualities. Bismuth, graphite and silicon are some non-superconducting materials that also possessing diamagnetic ability, but at very low levels [35].

The Meissner Effect is one well-known result of the diamagnetic effect, often explained as a magnet floating steadily above a superconductor. This technology has been promising for devices in micro and nano regions, because of problems creating frictionless and self-levitating bearings. In general, these devices do not force maintenance. This technology is sensitive to ambient temperatures outside the proper region [1]. Inside a superconductor the magnetic field becomes zero below its critical temperature. Superconducting transition temperatures have been studied for different materials. For example for $MgB_2H_{0.03}$ and MgB_2 materials, the result is that the former is higher.

Diamagnetic actuator benefits have high efficiency and fast response time, but the forces are generally weak per unit mass. Another drawback is that diamagnetic materials are highly susceptible to impurities.

Recently, diamagnetism has not been used for vibration control except in some unique cases in the micro and nano region. The forces that a diamagnetic device can distribute are generally weak per unit mass, and diamagnetic materials are highly susceptible to impurities.

A.2 Electrochemical

Definition
An electrochemical cell is a device that converts chemical energy into electrical energy or vice versa when a chemical reaction occurs in the cell.

This technology is based on the electrolysis-field, with an electrochemical cell. An electrochemical cell consists of two metal electrodes immersed in an electrolyte with electrode reactions occurring at the electrode-solution surfaces. The principle of an electrochemical actuator is to build up a gas pressure by electrolysis of an aqueous solution [36]. There have been studies to develop new materials for actuation and pumping to improve the use of scalable electrochemical phase transformations. Electrochemical actuators have been developed with strains of 136% and non-optimized work cycle efficiencies near 50% [37].

There is not much information about electrochemical actuators and those that exist, are rarely used, but there are some actuators in the micro region. There are some practical problems concerning the micro fabrication and operation of the device [36].

A.2.1 Principles

Electrochemical micro-actuator
Electrochemical micro-actuators have been used in medical research. The electrochemical cell contains two electrodes, platinum and copper in a 1molar (1M) $CuSO_4 \times 5H_2O$ solution. The platinum electrode is used as the anode and they should have a relatively large active area because the oxygen gas has to react at the platinum electrode to form water, when the actuator is in the pressure reduction state. The copper electrode is used as the cathode and it should be thick enough to ensure the required lifetime. This particular micro-actuator was developed to adjust the pressure in the eye [36].

A.3 Electrohydrodynamic

Definition
"Electrohydrodynamic motion arises when the particle of a polar fluid are subjected to a strong electric field" [35].

Electrohydrodynamic motion generates a fluid pressure and creates a flow or fluid circulation. Electric force induces flow in dielectric fluids, which alternates in 3-phase wave (positive, negative, neutral). They are often simple in design and are produced to provide direct conversion of electricity to flow of fluid. These devices do not require much maintenance, because of few moving parts. Drawbacks concerning electrohydrodynamic actuators are low power density and low current, but they require high operating voltage. The maximum efficiency is medium and they have medium speed in comparison to other technologies [35]. Electrohydrodynamic actuator operation is greatly influenced by the electric properties of the fluid. They can produce a high volume of flow in comparison to piezoelectric or thermally driven pumps. They can also be used as a driver for pumps to move other types of fluid inappropriate for electrohydrodynamic flow.

For instance, this technology is used as motive power in submarines, to avoid the noise of rotating propellers, by pumping huge quantities of seawater through special tubes [35]. Another application is in the micro region, where an ethanol pump has been developed with grids charged by etched silicon [35].

A.4 Electromagnetic

Definition
"Electromagnetism arises from electric current moving through a conducting material" [35].

Electromagnetism is one of the four known forces in nature. Electricity and magnetism were long seen as separate phenomena. It was not until the 19th century that they were treated as different types of the same electromagnetic field. The motion of a charge carrier, like an electron, will cause an electromagnetic field as a property of space. In contrast to a stationary charge, that will only produce an electric field in the surrounding space, if the charge is moving a magnetic field is also produced. If the magnetic field is changing an electric field can be produced. The lines of magnetic flux are between and around a pair of opposite magnetic poles. Equation (A.1) is Lorentz Force Law for contribution of electric and magnetic forces.

$$F = q\vec{E} + q\vec{v} \times \vec{B} \tag{A.1}$$

An electromagnetic actuator has many benefits, such as extreme positioning accuracy that is independent of load or velocity. The scope of this thesis covers linear electromagnetic actuators.

Electromagnetic actuators have many advantages, such as they can generate attractive and repulsive forces, and in some principles is the force in proportion to input current. They have usually quick responses, high efficiency, wide bandwidth, and large displacements. The mayor reasons that have done this technology so successfully are because it is well-used and often cheaper than other alternatives.

Some drawbacks with electromagnetic actuators are that they have an upper temperature limit and they are difficult to miniaturize. There are two ways to create the magnetic field, by use of a permanent magnet or an electromagnet. When an electromagnet is used instead of a permanent magnet, the difference is increasing dimensions and a continuous power loss. Disadvantage of permanent magnets is the upper temperature limit of 200 degrees and its cost. Usually costs are only initial investment, which are compensated very quickly by the advantage of lower losses. The wire manages temperatures up to approximately 180 degrees depending on which material is used in the conductor according to Compter [55]. According to Sjöstrand [59] engine mounts will be exposed to ambient temperatures of 90 ºC continuous and 110 ºC short duration. (Engine overheating in driving up a steep hill.)

This technology was selected to be more investigated, because it is commonly used and has good characteristics that match with our requirements for use in an active engine mount. We are especially interested in the voice coil principles.

A.4.1 Principles

Voice Coil – Moving Magnet
Unlike a solenoid, a linear actuator produces a constant force regardless of actuator position. Some advantages that are often stated with a moving magnet are higher forces, no frictional wear, by design, and the structure can utilize heat sinking more effectively. A voice coil with moving coil or moving magnet can be chosen if the actuator fulfills certain requirements. These requirements are:

In an actuator built on the voice coil principal: with a single coil, force is proportional to current and the force is generated in one direction. Furthermore, it is friction-free (non-contacting action) and the force is bi-directional and hysteresis-free.

If it is important that the moving mass is low, select a moving coil and if better heat transfer is a desirable effect, chose a moving magnet instead.

Voice Coil – Moving Coil "Loudspeaker"
The loudspeaker is the most commonly used linear motor, thanks to good characteristics such as high forces and good displacement. It is well-tested and often cheaper than other alternatives. This principle produces the fastest actuators in electro-mechanics, according to Compter [55]. Voice coils are capable of moving an inertial load at extremely high acceleration. Voice coils are used, for example, in audio speakers and computer disk drives.

Suppliers often call a more robust voice coil a shaker, but that is the only difference between them. The voice coil consists of a coil of wire, suspended in a fixed radian magnetic field. The force provided by the actuator is proportional to the current flowing through the coil, with the assumption that there are no interactions between the coil wires.

The performance of the voice coil is limited by displacement, moving mass, the total mass of the voice coil, thermal power of the coil and stress safety factor of the armature [12]. The coil resistance increases with temperature and slightly with frequency and coil inductance depends on frequency.

Reaction Mass Actuator
Reaction mass actuator consists in essential terms of a mass suspended on a spring that is driven by an electromagnetic circuit [14]. The suspended mass is constituted either by the magnets and supporting structures or the coil itself.

The Slide Motor
This principle is used in some CD players for moving the laser and the lens actuator in the radial direction of the disc [13]. The moving part is the copper coil. In a CD player two parallel shafts are used to guarantee guidance.

Linear Motor, Long Stroke
This concept is based on moving magnets and a stationary coil. This principle achieves positional accuracies of 100 nanometers and better, therefore this principle is used in exposure machines for IC production [13].

Short Stroke Linear Motor
Draw magnets are the cheapest alternative for short stroke linear motors, as described further in [13]. They are typically used in doorbells. Draw magnets always have non-linear behavior between the force and the current. These actuators are not suitable for servo applications [13].

Micro-actuator
Micro-actuators are becoming a growing field of interest. Their applications range from low-force actuators, such as optical mirrors or magnetic printing systems, to large-force actuators, such as motor relays and valves. As there is a demand for actuators with large displacement and large force, in particular for applications like microrelayes, a magnetic actuation principle has been chosen. Different approaches for the magnetic actuation of micro-relays have been evaluated, such as magneto-thermal and electromagnetic with magnet or without magnet.

Electromechanical
In some way electromechanical as technology can include many other technologies, but when using electromechanical we think of the use in a gearbox or other structure that are used to transmit motion. This technology is one of several different types commonly used in power-by-wire actuation [34]. In comparison to other solutions, they can be more efficient, smaller, lighter, stiffer and more complex, because of the absence of an internal hydraulic system.

A.5 Electrorheological and Magnetorheological

Definition
"If fluids are exposed to an electric field there are going to be some changes in their rheology, i.e. viscosity or flow rate" [35].

With special structures, a device can be successfully built so that changes in rheology generate or control fluid motion. According to [35], in an electrorheological fluid subjected to an electric field, the particles can react in the millisecond range. The particles are lining up and causing the fluid to be thicker or even non-flowing. These devices are usually not used in micro regions. When this technology is used in an actuator it results in medium efficiency and includes few or no moving parts. Electrorheological actuators can be found in semi-active hydraulic systems.

The drawbacks with electrorheological fluids are some problems that in the worst case can lead to device failures. Water-based fluids can develop thermal runaway problems if they are not adequately cooled: thermal stresses can dry out the working fluid, and low temperatures can cause problems, especially around the fluid's freezing point [35].

The viscosity characterizes an electrorheological fluid with mechanical tension as a parameter. It will be affected by an electrical potential, where the change in viscosity is proportional to electrical field strength. Electrorheological fluid is a collection name for several different substances, but the most common construction is a polar polymer suspended or pulverized in a fluid with the same characteristics as oil.

One variant of an electrorheological fluid is magnetorheological fluid. It functions in a similar way to the electrorheological fluids, but the difference is that it reacts on a magnetic field instead of an electric. There are some commercial products that use this type of fluid. The flow rate is controlled by the strength of a magnetic field.

This technology is not presently suitable for active vibration control, because it is not sufficiently tested, generates weak forces and is sensitive to ambient temperatures. But one idea is to use this technology in cooperation with another technology, like a hybrid actuator, to use the benefits of both. According to laboratory results this technology has good potential to accomplish great things in this area if the quality of the fluids can be improved.

A.6 Electrostatic

Definition
"Electrostatic charge arises from a build-up or deficit of free electrons in a material, which can exert an attractive force on oppositely charged objects, or a repulsive force on similarly charged objects" [35].

When two objects have different electrical charges and are located near each other, an electrostatic field exists between them. There also exist electrostatic fields around any single electrically charged object with consideration to its surroundings. Metallic objects block electrostatic fields, unlike magnetic fields, which can pass through most metals. According to [21], the attractive force between two conductive plates with unlike charges (Coulombic force) is:

$$F_N = \frac{1}{2} \cdot \varepsilon \cdot A \cdot \left(\frac{U}{d} \right)^2 \tag{A.2}$$

The strength of the field depends on the gap size as well as surface roughness. Electrostatic actuators are affected very little by ambient temperatures and they are highly efficient in actuation, because of their extremely low current consumption.

This technology is better than electromagnetic in smaller dimensions [55]. Electrostatic actuators are suitable for active vibration control, but there are not as many commercial products as electromagnetic ones for purposes other than micro and nano regions.

A.6.1 Principles

The parallel plate capacitor
In micro-electromechanical systems, or MEMS technology, this principle is used commonly and is very basic: consisting of two parallel plates (capacitor). The lower plate is fixed, while the upper plate can move. To simplify the expressions we first assume that the electrical field is uniform between the plates of the capacitor, and zero outside.

A.7 Electrostrictive

Definition
Electrostrictive materials are deformed when they are exposed to an electric field.

Strain sensitivity is affected by ambient temperature. Commercial actuators exist in small regions: typically micro regions [21]. The usual principles are crystal stack design and polymers.

Electrostrictive actuators can be suitable for active vibration control, but they can in some instances be too costly [22]. They produce small displacements, which can limit the possibilities. The strain is sensitive to ambient temperature.

A.8 Hybrid – combination of two or more Technologies

Definition
Hybrid actuators are actuators that are built of two or more different technologies in cooperation.

Sometimes it is possible to merge two or more technologies together to utilize advantages of each technology. It may be interesting to use piezoelectric, electrostrictive and magnetostrictive materials together with other technologies to enlarge the strain. For instance hydraulic has been successfully used together with piezoelectric.

A.8.1 Principles

Piezohydraulic
According to Anderson, Linder and Regelbrugge [38], piezoelectric and closed hydraulic systems have been produced with large displacement and high force capacity.

Electrostatically-driven Phase Change Actuator
Electrostatically driven phase change actuators can produce large displacement and, at the same time, high forces [16].

A.9 Hydraulic

Definition
A definition by example: a rod is forced to move because the pressure is higher on one side of a plate attached to it than on the other side.

In constructing a hydraulic actuator is it important to try to minimize the power loss of transmission of the fluid, through reducing the mass flow of the fluid and reducing the frictional drags.

Semi-active hydraulic devices are filled with fluid, typically ethanediol: often divided in two or more different chambers that are connected through channels. These channels have an adjustable area of opening controlled by servo-valves. The damping characteristic of semi-active hydraulic actuators with a flow through the adjustable channels changes because the friction in the channels depends on the adjustable opening area.

Hydraulic actuation is the technology most often used in the majority of the earth moving equipment. It is used, for instance, in brakes (sometimes together with pneumatic), steering and implements for excavators implements.

Hydraulic actuators exist that are successfully used in active vibration control for frequencies below 100 Hz [15]. They generate high forces and large displacement, but they can be slow in response time and usually have low bandwidth.

A.9.1 Principles

Electrohydrostatic
Electrohydrostatic actuators are used in power-by-wire actuation, together with several other types of actuators [34]. This principle is based on a reversible, electrically driven pump motor to directly pump self-contained hydraulic fluid to a piston. This drives the ram in the same fashion as a standard hydraulic actuator. They tend to be more complex, larger, heavier, less efficient and smoother than other devices used in similar applications.

Hydraulic actuator for vibration control
In [15] a fist-sized hydraulic actuator is presented, which can apply forces of several kN with displacements in the region of millimeters and with a frequency range of up to approximately 100 Hz. This actuator was developed to meet the requirement to be used for small displacement and to avoid the disadvantages of hydraulic cylinders that have some problems with body friction, leakage, extended length and relatively low stiffness due to large oil volume. To accomplish this they use elastic membranes that allow small axial movements and also act as a radial guide. The two annular membranes connect the lower and upper body and give flexibility to the system together with high radial stiffness. The oil pressures in the two chambers are to be controlled by servo-valves.

To obtain some specific requirements, such as maximum force, pressure, displacements and outer diameter the parameters that should be changed are membrane thickness and inner diameter this will also affect maximum stress in the membranes.

A.10 Magnetostrictive

Definition
"Magnetostrictive materials exhibit very small but strong shape changes when subjected to magnetic fields" [35].

James Prescott Joule first discovered magnetostrictive materials in the 1840s. He noticed that changes in magnetism affected the length of iron, this phenomenon he named the Joule Effect (magnetostriction). The phenomenon magneto-mechanical effect (Villari Effect) refers to the change in magnetic energy when a magnetostrictive material is stretched or compressed.

These materials give new possibilities in the development of components with high density, rapid reaction time and extremely good precision. Typically, magnetostrictive rod is placed inside a coil to be activated by an external magnetic field, changing the shape of the core [35].

Typical magnetostrictive materials include combinations of rare earth elements with iron such as TbFe (Terfenol) and TbDyFe (Terfenol-D). Terfenol-D has been the most widely used magnetostrictive material. Iron, nickel, cobalt and ferrite are examples of other magnetostrictive materials.

Under a magnetic field these materials offer less then 0.15% strain. Advantages of this technology are that these types of actuators can be used in high frequency and high precision applications and have a long life. But the actuators are quite complicated, both in mechanical and electrical construction. That is because a magnetic coil is required for control of the driving magnetic field. One other major drawback is the need for a large bias field. That implies the use of heavy permanent magnets. [30] The conclusion is that this type of material can be used, but the drawback is the need for a large magnetic field that can be complicated to create.

A.11 Phase change

Definition
"Phase Change systems use the dimensional changes, like expansion and contraction, which occur in materials as they undergo changes between phases, such as solid, liquid and gas" [35].

Device can be built which harness the forces exerted by the phase changes, and they generally demonstrate full reversibility. Depending on the material, a phase change may be induced electrically, thermally, or ultrasonically, and may happen over a wide range of speeds and pressures.

Commercial thermal phase change actuators exist in micro-region and in MEMS techniques.

A.12 Piezoelectric

Definition
Piezoelectric ceramics generates an electric charge when mechanically deformed. Conversely, when an external electric field is applied to piezoelectric materials they mechanically deform.

Even though piezoelectricity can be found in several types of natural materials, most modern devices use polycrystalline ceramics such as lead-zirconate-titanate (PZT). PZT is the most commonly used piezoelectric ceramic since its discovery for more than 40 years ago.

There are ranges of piezoelectric actuators: high-voltage and low-voltage piezos. Piezoelectric devices show good resolution and are used commonly as both actuators and sensors.

Basic piezoelectric modes are thickness expansion, thickness shear and face shear. According to Compter [55], one successful application for piezoelectric actuators has been to drive a system of an electron microscope of FEI, where magnetic fields are not allowed.

Hydraulic mechanisms can be used to enlarge the displacement. For example, a piezoelectric multilayered stack actuator acting on a diaphragm that forces a fluid (non compressible medium). The diameter of the rod is smaller than the diameter of the diaphragm. A force is thus been converted into displacement.

We can not discount piezoelectric as an interesting technology. Most commercial applications today are based on electromagnetic or piezoelectric principles. They can exert high output forces, very quick responses, often linear to charge and they are highly efficient. The drawbacks with piezoelectric actuators are that they are hard to attach, expensive, hysteresis and have unknown lifetime [55].

A.12.1 Principles

Amplified Piezoelectric Actuators (APAs)
Amplified piezoelectric actuators consist of stacks that are pre-stressed inside a steel elliptic frame, which produces a natural amplification ratio of the displacement (between 2 and 5 according to the two axes ratio) [5]. This principle will obtain deformation between 0.3 and 3 percent with only a slight decrease of the force capability. Other piezoelectric principles have strokes about 0.1 percent of their total length.

Bimorph
Two independent flat piezoelectric elements, stacked on top of each other. By driving one element to expand while contracting the other one, the actuator is forced to bend, creating an out-of-pale motion.
Disk bender
Disk benders consist of two piezoelectric disks separated by a central electrode and two electrodized external surfaces. The whole sandwich is poled in one direction and can be operated in different ways, depending on how the voltage is applied.

They appear to represent an ideal solution for many applications with demanding space conditions. A major drawback is that disk benders are usually quite expensive because the fabrication method requires the cutting of two thin piezoelectric disks from a ceramic cylinder.

A new low-cost disk bender was recently developed and shows very little difference from standard disk benders.

Electrodistortive
Work on the inverse piezoelectric effect, the PZT (lead-zirconate-titanate) material is used. Applying voltage causes the material to expand.

Moonie
The Moonie consists of a piezoelectric disk with electrodes and two metal caps glued on the surface. By shrinking the disk the caps are forced to bend.

Multilayered Stacked
Actuators of this kind show quite large displacements, in the range of 10-300 µm, with large blocking forces, up to several kN, and quite low driving voltages. The mayor drawback is that these actuators are complicated to fabricate and therefore expensive.

Rainbow
The rainbow is a piezoelectric disk such as lead-lanthanum-ziconate-titanate (PLZT) with one of the two faces of the disk reduced.

A.13 Pneumatic

Definition
Differences in pressure of air or gas in these devices create force.

Pneumatic can establish a counterforce without any energy consumption. Pneumatic actuators are similar to hydraulic actuators, but they use another medium. There are some problems with friction like the ones with hydraulic actuators. Pneumatic actuators are very common in level-regulation systems.

A.14 Pyrotechnical

Definition
Explosive and pyrotechnic devices transforms a small input of mechanical or electrical energy into a higher level of mechanical or thermal energy that is applied to perform practical work on a one-time basis.

Some companies, such as Hirschmann, supply ignition units for pyrotechnical actuators in restraint and safety systems in the car. This is accomplished by releasing the stored energy in an explosive or pyrotechnic mixture through a precisely controlled reaction.

Actuator devices transform pyrotechnical-generated energy into motion to perform work against an external load.

There is no information about this technology used for active vibration control, and that is understandable because its use is on a one-time basis.

A.15 Shape Memory

Definition
Shape Memory effect arises in some unique metal alloys that change form with temperature, but "remembers" the original shape and when it is cooled it reverts to its original shape (cycle as figure below) [35].

The term shape memory refers to the ability of certain materials to cancel its deformation and recovers its predefined or "imprinted" shape. The SM effect is based on a solid-solid phase transition of the shape memory alloy that takes place within a specific temperature interval [9].

These materials are best suited for heating through the phase change. Creating phase change through heating is only possible after "education" of the material. And the difficulties of cooling must be solved. These types of actuators can only be used in low-frequency and low-precision applications [30]. We have only found Shape Memory actuator solutions in the micro region. The conclusion is that SMA-actuators are not suitable for active vibration control [30].

A.15.1 Principles

SMA - Shape Memory Alloy
Nickel Titanium is a widely used alloy. This principle is used more commonly than SMP. Memory metals are another name for Shape memory alloys. The function of memory metals is similar to piezoelectric elements, but the difference is that they react to changes in temperature. This characteristic implies that after the metal has been deformed it can recover to its original form if it warms up. The process that changes the form is a thermo-elastic martensitic transformation. The most common memory metals are Zn-Al, Cu-Al-Ni and Ni-Ti alloys. Memory metals have big advantages, as they are small and not sensitive to environmental influences apart from temperature. A control signal can be put direct into the material without transformation to or from digital form.

There exist so-called Giant magnetostrictive materials (GMM), which are in competition with piezoelectric ceramics, but they are commonly used in specific applications such as low voltage actuators, large force actuators, high power low frequency sensors and space cryogenic positioning.

SMP – Shape Memory Polymer
Shape memory polymers are not based on the same physical principles as shape memory alloys, even though they have similar names. They are thermo-elastic polymers, which undergo a glass transition; their viscosity becomes low at high temperatures (rubber

state) and increases as the temperature decreases. Consequently, when the temperature is sufficiently low, the viscosity becomes high enough to make the polymer stiff. This condition is called the glassy state.

This principle has been utilized as an actuator by squeezing the polymer into a very small volume at high temperatures, and then lowering the temperature while maintaining the pressure bringing the polymer to the glassy state. The polymer remains compact when the pressure is removed. When the actuator is exposed to high temperatures the polymer strives to return to its initial value and the polymer restores its original shape. In [11] can the amount of exploitable motion of this actuation technique can be very high up to 4000% since the polymer can be squeezed into a very small volume. As long as there is no built-in squeezing system, this device can only be used once.

A.16 Thermomechanical

Definition
Thermo mechanical systems use the physical expansion or contraction that occurs in materials as they undergo temperature changes within their phase (solid, liquid or gas) [35].

Thermomechanical materials change their dimensions with temperature. To avoid undesirable changes in temperature in the surroundings, the materials may be forced to be isolated. In micro devices the characteristics of usefulness and speed of a thermomechanical actuator change radically. To re-establish the previous condition, the heat must be removed in some way.

A.16.1 Principles

Bimetallic cantilever
A bimetallic cantilever is a micro actuator using gold on silicon with a beam length of 500 micrometer, producing deflections of up to 100 micrometers while using about 200 mW of power. A 200 micrometer long thermally activated cantilever beam made only of silicon, silicon oxides and phosphorous doped silicon - standard elements of CMOS-type electronic circuits - produced a displacement of 4 micrometers and operated at a frequency of over 1 kHz [35].

Appendix B
SENSORS - TYPES AND DESIGNS

B.1 Accelerometers

There are two main types of accelerometers, which measure either translation accelerations or angular accelerations. Most of the translation accelerometers belong to the category of seismic instruments, which implies that the acceleration is not measured in proportion to a reference point [4]. Accelerometers can be either mechanical or electromechanical devices. Further, the electromechanical accelerometers can be classified as variable resistance accelerometer, variable inductance accelerometer, piezoelectric accelerometer, piezoelectric transistor and servo accelerometer [4].

There are several kinds of accelerometer that are based on rotation. In one kind the damping fluid functions as a seismic mass, and during rotation acceleration of the fluid is relative to the chamber creating a pressure on two symmetrically placed vanes. This pressure is a measure of the rotation acceleration.

B.1.1 Design alternatives - piezoelectric accelerometers

There are a few different types of piezoelectric accelerometer, which harness one of three effects: length, side and shear. The following sections B.1.1.1, B.1.1.2 and B.1.1.3 present examples of different designs of piezoelectric accelerometers, which utilize each one of the three effects.

B.1.1.1 Compression-based design

A piezoelectric sensor of the compression type consists of a seismic mass, piezoelectric crystals and a pre-stressed spring washer, see figure B.1. This type of design harnesses the length effect of piezoelectric materials, shown in sub-chapter 3.1. The pre-stressed spring washer presses the movable mass towards the piezoelectric crystals. To improve the sensitivity two piezoelectric crystals are often mechanically connected in series.

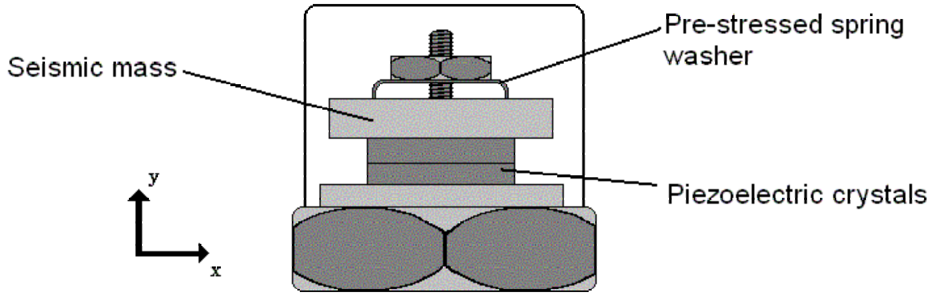

Figure B.1 Piezoelectric sensor based on compression (side view).

When a piezoelectric sensor of the compression type is subjected to a sine curve vibration along the direction of the symmetry axis (y), the voltage per frequency output will be as shown in Figure B.2. The figure shows the bandwidth of the piezoelectric compression accelerometer between the frequencies f_1 and f_2. In the figure, two cut-off frequencies appear. The upper cut-off frequency is determined by the seismic system and the lower cut-off frequency depends on which system that is used. It originates either from leakage resistance by voltage amplification or from the time constant of the charge amplification. The frequency f_0 is the natural frequency of the seismic system, and because the internal spring rate is really high the natural frequency in general is going to be 10 kHz or higher.

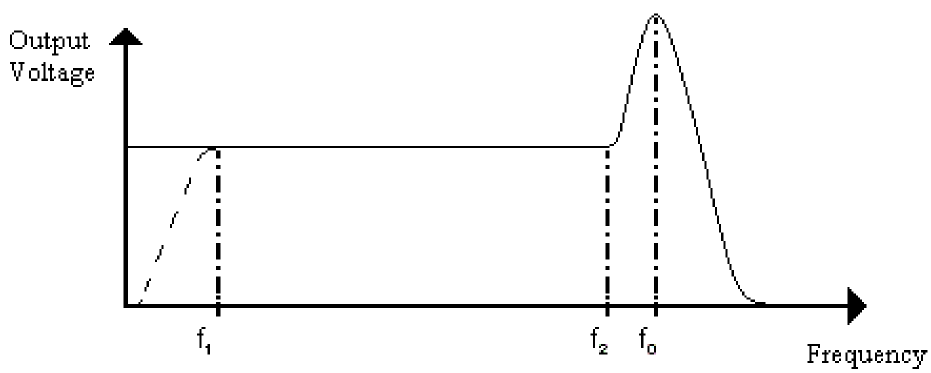

Figure B.2 Output voltage of a piezoelectric compression accelerometer with sine curve vibration.

The force that will affect the crystals becomes a linear function of the acceleration of the moving part relative to the fixed part, since the crystals deform elastically. The output voltage becomes proportional to the oscillatory movements of the movable mass if the frequencies are low compared to the natural frequency, see equation B.1.

$$l \approx -\frac{1}{\omega_0^2}\frac{d^2 s}{dt^2} \tag{B.1}$$

Here l is sensed by its position, which is proportional to the acceleration of the test object. The sensor is sensitive to accelerations at low frequencies if the damping ratio is $\zeta \approx 0.7$, see [5, 33]. This is not the case with this piezoelectric sensor since it does not have any damping material [5]. Manufactures typically give the bandwidth (operating range) for the sensor instead of the damping ratio.

B.1.1.2 Shear-based design

When compression-based piezoelectric sensors are mounted on a frame that is subject to bending stress they can be compared to force sensors, because the bending stress is mistaken for acceleration by the crystals. This is one reason why the shear-based piezoelectric accelerometer was developed. In contrast to compression types, it can sense the acceleration component, which is perpendicular to the symmetry axis. It senses acceleration in the symmetry axis direction. This design harnesses the shear effect of piezoelectric materials. Figure B.3 shows that the crystals are mounted free from the bottom of the housing, this makes the sensor insensitive to bending stress of the frame. The crystals are held in place by two half-moon seismic masses and a clamping ring (pre-load ring). During acceleration the crystals are subject to shear stress forces, and charge is created by the transversal component of the shear stress. In new developed accelerometers is the majority design based on shear effects

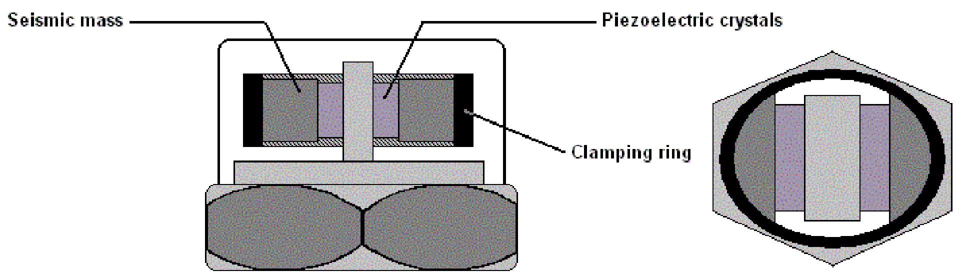

Figure B.3 Side view and top view of a piezoelectric sensor based on shear effects.

B.1.1.3 Bending-based design

Piezoelectric accelerometer designs based on bending utilize beam-shaped sensing crystals, which are fixed in one end, with the other end having a seismic mass that create strain on the crystals when the mass is accelerated. Bending-based design harnesses the side effects of piezoelectric materials. The upper and lower parts of the carrier beam, respectively, are subjected alternately to compression and expansion effects, which is the same as the piezoelectric side effect. This design offers a low profile, light weight, and a competitive price. Generally, bending-beam designs are well suited for low frequency and low acceleration applications. They are sensitive to very high acceleration, high amplitude and high frequencies. Figure B.4 gives a basic layout sketch of a piezoelectric sensor based on bending design.

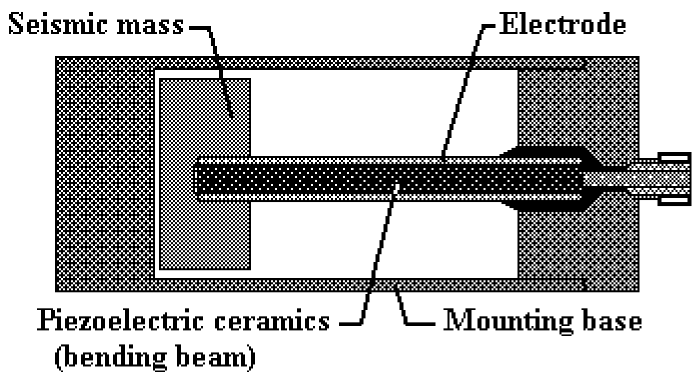

Figure B.4 Piezoelectric sensor based on bending (side view).

Comparison between the piezoelectric accelerometer designs compression, shear and bender has been carried out through study [31] and are stated in table B.1.

Table B.1 Comparison between different piezoelectric accelerometers based on different designs

	Compression	**Shear**	**Bender**
Benefits	+ high sensitivity-to-mass ratio + robustness + technological advantages	+ low temperature transient sensitivity + low base strain sensitivity	+ best sensitivity-to-mass ratio
Drawbacks	- high temperature transient sensitivity - high base strain sensitivity	- lower sensitivity-to-mass ratio	- fragile - relatively high temperature transient sensitivity

B.1.2 Other accelerometers

The purpose of this section is to present other types of accelerometer than piezoelectric, even though piezoelectrics are used more commonly than others.

B.1.2.1 Servo accelerometer

Servo accelerometers consist of an electromechanical servomechanism, which holds a mass fixed in a certain position when the device is accelerated. Consequently, the force that is consumed is a measure of the acceleration. Some servo accelerometers can measure up to 100 g [4].

B.1.2.2 Variable inductance accelerometers

Actually, these sensors measure velocity, but can be modified to be sensitive for acceleration [5]. The measuring instrument is supported by weak springs. Therefore, the natural frequency of the system is low, at about 10 Hz. The system is an electrodynamic type, having a limiting coil (copper ring) that moves in the field of a permanent magnet, thereby inducing damping eddy currents. The damping ratio ζ becomes 0.5-0.7. Deflections of the moving part will be in proportion to the measured velocity. The output voltage is proportional to the velocity. Variable inductance accelerometers are characterized by comparatively low natural frequency. The measured capacity can be as high as 50 g with high output voltage, which means that the requirement for amplification falls away [4]. Other information [5] specifies the upper limit of acceleration and amplitude of oscillation to be approximately ± 10g and ± 1mm. They exist in different sizes.

B.1.2.2.1 Geophone

The geophone is nowadays only used in investigation of vibrations associated with blasting operations. In these applications it is generally of no importance that the sensor responds to velocity, because it is often compared to standard specifications and empirical values, and these values can also be based on velocity.

B.1.2.3 Variable resistance accelerometers

A variable resistance accelerometer is constructed on the principle that the electrical resistance of the conductor is a function of its dimensions. When these dimensions mechanically change at the same time as a constant electrical current flows through the conductor the voltage will vary over it. The change in voltage is in proportion to the mechanical change in dimension. They are often used to measure sluggish processes, for instance to measure the acceleration of vehicles such as aeroplanes [5].

B.2 Microphones

Microphones can be used as sensors that can enable a system to be optimized by noise. There are sensors that react to sonic waves. They consist of a diaphragm that interacts with the waves. When the diaphragm moves in a condenser microphone the electrical potential changes due to energy changing in an electrical field when one plate of the parallel plate capacitor is moving. The fixed plate is loaded by voltage supply.

B.3 Optical Sensors

Optical sensors are used to measure velocity, amplitude or position in relation to a reference point. One kind of optical sensor is based on fibre optics. Optical sensors can be used to measure rotation vibration.

B.3.1 Photoelectric cells

B.3.1.1 Distance measuring

Position is often measured with a sensor that consists of photoelectric cells, a source of light, and a movable glass plate furnished with alternating black and transparent fields according to a certain system. When the glass plate moves light will be allowed to pass through it to the photoelectric cells that have a transparent field between them and the source of light. The photoelectric cells have two possible conditions, which give the signal 0 or 1. By combining several photoelectric cells binary numbers are generated, which correspond to a certain distance to the reference point, see figure B.5.

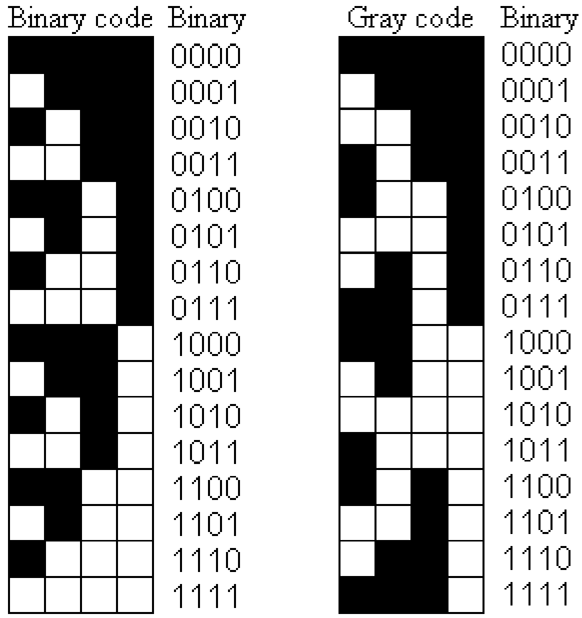

Figure B.5 Binary and Gray code that corresponds to $2^4 = 16$ possible distances (four photoelectric cells are used).

To get the correct binary number go from left to right. One problem with this technology is how to code the different states, if regular binary numbers is used the result is that a big change in the binary number shaping will affect the output only a minor amount. This can give rise to disproportionately large faults, and because of this Grey codes are commonly used when the plate changes sectors.

B.3.1.2 Amplitude measuring

For optical measuring of vibrational amplitudes a sensor is often used consisting of two plates with thin lines engraved. One plate is fixed and the other one is attached to the body whose movement is going to be measured. The two plates are moving relative to each other and at the same time the amount of light that will pass through the plates is changing: in the ideal case as a sine curve. These changes can be read off by a photoelectric cell, which generates an output signal. A bigger output can be measured through calculating the number of crests.

B.3.2 Optical fibre

Thanks to inner reflections, optical fibres have the ability to enclosure a light beam inside the fibre. There is a distance and time difference between a straight fibre and a curved fibre when a light beam is travelling through it, because the amount of reflections is not the same. Therefore, the light beam travels much faster in a straight fibre than in a curved one, and thus time displacement can be measured as a phase displacement compared to the straight fibre. Unfortunately the phase displacement is not in proportion to the deflection, so the fibre sensor must therefore be calibrated along the length of the fibre.

B.3.3 Laser

Only recently have laser sensors been used to measure direct displacement and velocity on the surface: such sensors utilize the Doppler Effect. The lasers rely on not-contacting the vibrating surface and they measure the phase difference between an internal reference and the measurement beam that has been reflected to the surface. Polytec has developed lasers that measure on rotating surfaces at speeds of 30 m/s and at frequencies of 30 MHz.

Advantages with lasers based on Doppler Effect are insensitivity to ambient light, long measuring ranges, measurement on any target surfaces, high resolution and repeatability. They use high measuring frequency and permits measurement on hot, miniature or soft surfaces, and even under water.

There are also laser sensors that measure distance through optical triangulation. Helium-Neon is a common low-power laser that is eye-safe class 2, which can operate at distances of hundreds of meters.

B.4 Force Sensors

When piezoelectric force sensors are subjected to a stress, they produce a charge that is proportional to the force. In force sensors, quartz is commonly used, because it has good mechanical properties, good resistance to high temperature and the resistively is very high. Furthermore, the piezoelectric effect becomes free from hysteresis, has extraordinary linearity and is insignificantly dependent on temperature. They are well suited for high precision in the nanometer range and have been used in ultrasonic applications.

The piezoelectric polymer polyvinylidene fluoride (PVDF) has been used in force sensors. PVDF is a flexible piezoelectric material, which produces a current proportional to extension. This material is a polymer (semi-crystalline) with high polarity, which can be attached to almost any surface. The characteristics of the material are dependent on the ambient temperature. Pyroelectric application with PVDF films has been used in thermometers. Piezorubber (PZR) is a flexible composite material (polymer) that consists of lead titanate particles embedded in a neoprene elastomeric matrix. The U.S. Navy has tested PZR in hydrophones. Neither PVDF nor PZR is piezoelectric naturally, but they have been polarized in the manufacturing process by exposure to a strong electric field during cooling down from a high temperature.

Appendix C
MAX ENERGY DENSITY

Max energy density comparison between different technologies is a good measure when the objective is to compare micro actuators. In literature there exist many different values of maximum energy of density at a specific volume or weight, depending on the estimated conditions. The estimated conditions are obtained through a study of relevant material and estimation. According to [61], the max energy of density is 0.025 J/cm^3 for electromagnetic to compare with 0.015, >5 and >0.1 J/cm^3 for electrostatic, Shape memory alloy and piezoelectric ceramic. Other literature has mentioned that the max energy of density of SMA is 10 J/cm^3.

Max Energy Density formulas	**Estimated conditions**	**Approx.** (J/cm^3)
Magnetic		
$\frac{B^2}{2 \cdot \mu_0}$		~ 4
B = magnetic field	B = 0.1 T	
μ_0 = magnetic permeability		
Electrostatic		
$\frac{\varepsilon_0 \cdot E^2}{2}$		~ 0.1
E = electric field	E = 5 V /μm	
ε_0 = dielectric permittivity		
Piezoelectric		
$\frac{Y \cdot (d_{33} \cdot E)^2}{2}$		~ 0.2
E = electric field	E = 30 V/μm	
Y = young's modulus	Y = 100 GPa	

d_{33} = piezoelectric constant d33 = $2 \cdot 10^{-12}$ C/N

SMA

According to literature 5 - 10

Thermal

$$\frac{Y \cdot (\alpha \cdot \Delta T)^2}{2}$$ ~ 5

α = coefficient of expansion α = $3 \cdot 10^{-6}$ /ºC

ΔT = temperature rise ΔT = 100 ºC

Y = young's modulus Y = 100 GPa

According to [35], high work output density the different technologies are compared:

Technology	**Work output density**
Diamagnetic	High
Electrohydrodynamic	Low
Electromagnetic	High
Electrorheological	Medium
Electrostatic	Low
Magnetostrictive	Very high
Phase change	High
Piezoelectric	High
Shape Memory	Very high
Thermomechanical	Medium

Appendix D
TECHNOLOGY – PRINCIPLE – SUPPLIER

The purpose with this appendix is to more easily understand the possibilities and restrictions of certain actuator requirements for existing commercial actuators of electromagnetic and piezoelectric technologies. First, electromagnetic technology is introduced and the possibilities are stated. This is following by detailed information about different electromagnetic actuators from different suppliers.

Finally, piezoelectric technology and detailed information about commercial piezoelectric actuators from different suppliers are presented.

Technology	*Electromagnetic*
References	http://ndeaa.jpl.nasa.gov/nasa-nde/lommas/eap/actuators-comp.pdf
Strengths/ Weaknesses	• Very fast operating speeds • Extreme positioning accuracy • Scale ability • High degree of efficiency • Have an upper temperature limit (≈180ºC) • Performance is primarily limited by the properties of the material used in constructing the actuator • Difficult to build small electromagnetic coils • The most devices require perpendicularity between the current conductor and the moving element • Simply on/off control but relatively limited functions-usually only lift and not lift states available See Appendix A and Chapter 2 for more detailed information.
Maximum Displacements (%)	50[1]
Maximum Pressure (MPa)	0.1[1]
Frequency Range (Hz)	0-70000
Sensitivity	Extremely high
Response Time (ms)	<< 1 Theoretical [35]
Relative Speed (full cycle)	Fast [35]
Maximum Efficiency (%)	> 90[1]
Need of Voltage (V) / Current (A)	Dependent on relative size and principle
Size (Height/Length/Width)	From micro/nano region and bigger
Weight	Depend on the force that is required. Permanent magnet instead of electromagnet increases the weight.
Price	Most of them have low costs.
Comments	Magnetic actuators are perhaps among the oldest types of actuators. Magnetic actuation methods offer the possibility of generating repulsive forces in addition to attractive forces. Voice coils can be built with a max specific work of 30 J/kg and up to 70 kHz (Ref.).
Accuracy	Extremely high

1 (Voice coil) These values are based on an array of 0.01 m thick voice coils, 50% conductor, 50% permanent magnet, 1 T magnetic field, 2 ohm-cm resistivity, and 40,000 W/m2 power dissipation (http://ndeaa.jpl.nasa.gov/nasa-nde/lommas/eap/actuators-comp.pdf)

Technology	*Electromagnetic*				
Principle	Voice coil – moving coil (Shaker)				
Supplier	Data Physics Corporation				
Products Names	**DP-V002**	**DP-V004**	**DP-V009**	**DP-V011**	**DP-V016**
References	http://www.dataphysics.com/shakers.html				
Strengths/ Weaknesses	More robust than ordinary voice coils. The created force is in proportion to the current. The actuator is limited by, for example, displacement, moving mass, thermal power of the coil and stress safety factor of the armature.				
Max Displacements (mm)	3	5	10	15 (25.4)[1]	13
Maximum Force (N)	9[2]	18[2]	40	50	70
Frequency Range (Hz)	2-12000	2-11000	DC-7000	DC-7000	DC-7000
Resonance Frequencies (Hz)	12000	11000	5500	5500	6500
Stiffness (N/µm)	0.004	0.0044	0.0065	0.005	0.008
Damping					
Sensitivity					
Response Time (ms)					
Maximum Efficiency (%)					
Need of Voltage (V) / Current (A)	2.7 A	3.8 A	5.5 A	5.5 A	5.5 A
Size (H/L/W) (mm)					
Weight (g)	1900	4530	18000	30000	10000
Price					
Comments	The coil resistance increases with the temperature and slightly with the frequency. The impedance depends on the frequency.				
Other characteristics					
Nominal Impedance (Ω)	4	4	4	4	4
Max Velocity (m/s)					
Moving Mass (g)	15	28	150	200 (260)[1]	230

1 Optional displacement
2 Peak force for sine instead of random

Technology	***Electromagnetic***			
Principle	Reaction Mass Actuator (RMA) Counterforce vibration control			
Supplier	CSA Engineering, Inc.			
Products Names	**VIS6**	**VRS1**	**SUITE**	**SA-10**
References	http://www.csaengineering.com/hex/vis6.html http://www.csaengineering.com/indus/SAshort.pdf			
Strengths/ Weaknesses				
Max Displacements (mm)				
Maximum Force (N)	0.61	4.47	38.4	44.48
Frequency Range (Hz)	40-500			20-1000
Resonance Frequencies (Hz)	42	38.3	44	21.5
Stiffness (N/µm)	663	2409	33630	32442
Damping		0.03		0.115
Sensitivity				
Response Time (ms)				
Maximum Efficiency (%)				
Need of Voltage (V) / Current (A)				
Size (H/L/W) (mm)	35.0[1]	25.4[1]	43.2[1]	88.9[1]
Weight (g)	88	107	1200	3640
Price				
Comments	CSA have also two smaller actuators than SA-10, they are SA-1 and SA-5.			
Moving Element	Coil	Magnet	Magnet	Magnet

1 The actuator have diameter instead of length and width.

Technology	*Electromagnetic*	
Principle	Voice coil – Moving Coil (Shaker)	
Supplier	Derritron	
Products Names	**VP 5**	**VP50**
Strengths/ Weaknesses	• Low static support	
Max Displacements (mm)	10.1[1]	17.8[1]
Maximum Force (N)	105	2230
Frequency Range (Hz)	DC-8900	DC-4500
Resonance Frequencies (Hz)	8500-8900	3800-3900
Stiffness (N/µm)		
Damping		
Sensitivity		
Response Time (ms)		
Maximum Efficiency (%)		
Need of Voltage (V) / Current (A)	30 A rms, max	80 A rms, max
Size (H/L/W) (mm)	Height: 260.9, diameter: 228.6	Height: 500.1, diameter: 388.3
Weight (g)	50 000	234 000
Price		
Comments	The standard VP 5 will support a static load of 1.8 kg. For heavier loads, up to 23 kg an additional load support system may be fitted. Designed for testing of small components, subassemblies and modal testing of large complex structures. The VP 5 can be configured to perform "back-to-back" calibration of piezoelectric accelerometers. Rated ouput sine force is 222 N. Air cooled by suction type remote centrifugal blower.	
Moving Mass (g)	480	2500

1 Peak to peak, continous

Technology	***Electromagnetic***		
Principle	Voice Coil – Moving Magnet		
Supplier	H2W Technologies, Inc.		
Products Names	**NCM08-15-025-2 LB**	**NCM02-05-005-4X**	
References	http://www.h2wtech.com/noncommdcactu.htm		
Strengths/ Weaknesses	It is a compact linear driven actuator that can be controlled like a servo motor.		Smaller stroke than moving magnet. The low moving mass allows high acceleration of light payloads.
Max Displacements (mm)	19.05	0.15	50.8
Maximum Force (N)	33.36	6.72	1333
Frequency Range (Hz)			
Resonance Frequencies (Hz)			
Stiffness (N/µm)			
Damping			
Sensitivity			5 µm repeatability
Response Time (ms)			
Maximum Efficiency (%)			
Need of Voltage (V) / Current (A)			
Size (H/L/W) (mm)	Diameter: 38.1		
Weight (g)			
Price			
Comments	NCM08-15-025-2 LB has a max acceleration of 20g. The voice coils have very low electrical and mechanical time constants.		
Accuracy (µm)	8.3	8.3	8.3

Technology	*Piezoelectric*
Strengths/ Weaknesses	• High stiffness results in isotropic high actuator performance • Easily controlled • Provide fast response • Small dimensions and weight • Simply driven by voltage • Maximum strain under electrical field can approach 0.2% • They can cover a wide range of frequencies • High precession • They can hardly be used at very low frequency and at dc. • Very large force • Very small displacement See Appendix A and Chapter 2 for more detailed information.
Maximum Displacements (%)	Ceramics 0.2, Single Crystal 1.7, Polymer 0.1
Maximum Pressure (MPa)	Ceramics 110, Single Crystal 131, Polymer 4.8
Frequency Range (Hz)	> 100 kHz
Sensitivity	Infinity
Response Time (ms)	<< 1
Relative Speed (full cycle)	Fast [35]
Maximum Efficiency (%)	> 90
Need of Voltage (V) / Current (A)	From 100 V to 1000 V for full extension. Very low current use (mA).
Size (Height/Length/Width)	Small
Weight	Low
Price	Depend on the design, from low to very high
Comments	The best known and most used piezoceramic, Lead Zirconate Titanate (PZT), can deliver up to 0.1% strain.
Accuracy	High in closed loop design

Technology	***Piezoelectric***			
Principle	Multilayered Stacked			
Supplier	Adaptronics, Inc			
Products Names	**DPA40**	**DPA60**	**DPA80**	**PPA40M**
References	http://www.adaptronics.com/products/piezoelectric _actuators/direct_piezo_actuators/index.html			
Strengths/ Weaknesses	• High power efficiency • Low drive voltage			
Max Displacements (mm)	0.040	0.060	0.080	0.032
Maximum Force (N)	3500	3500	3500	800
Frequency Range (Hz)				
Resonance Frequencies (Hz)	9000	7000	5000	2000
Stiffness (N/µm)	87.5	58.3	43.8	25
Damping				
Sensitivity				
Response Time (ms)	0.08	0.10	0.14	0.04
Maximum Efficiency (%)				
Need of Voltage (V) / Current (A)	-20 to 150 V	-20 to 150 V	-20 to 150 V	-20 to 150 V
Size (H/L/W) (mm)	52.5/25.0/ 25.0	82.5/25.0/ 25.0	112.5/25.0/ 25.0	49.0/10.0/ 8.0
Weight (g)	155.0	230.0	310.0	25.0
Price				
Comments	DPA40, DPA60 and DPA80 have diameter 25.0 mm. PPA40M exist with less displacement.			

Technology	***Piezoelectric***			
Principle	Moonie			
Supplier	Adaptronics, Inc.			
Products Names	**APA120M**	**APA230ML**	**APA400ML**	**APA500L**
References	http://www.adaptronics.com/products/piezoelectric _actuators/direct_piezo_actuators/index.html			
Strengths/ Weaknesses	• Very large displacements			
Max Displacements (mm)	0.130	0.236	0.400	0.500
Maximum Force (N)	1400	1350	38	570
Frequency Range (Hz)				
Resonance Frequencies (Hz)	6200	2700	2350	1700
Stiffness (N/µm)	10.8	5.70	0.10	1.10
Damping				
Sensitivity				
Response Time (ms)	0.11	0.26	0.30	0.41
Maximum Efficiency (%)				
Need of Voltage (V) / Current (A)	-20 to 150 V	-20 to 150 V	-20 to 150 V	-20 to 150 V
Size (H/L/W) (mm)	45.0/78.9/ 22.5	85.0/78.9/ 22.5	14.3/55.1/ 11.5	55.0/145.0/ 12.5
Weight (g)	160.0	275.0	19.0	200.0
Price				
Comments	Response time and resonance frequency for a free actuator. APA129ML and APA400M exist with less displacement and higher force. The moonie consists of a piezoelectric disk with electrodes and two metal caps glued on the surface. By shrinking the disk the caps are forced to bend.			

Technology	***Piezoelectric***		
Principle	Bimorph Strip-benders	Bimorph Disk-benders without centerbore	Bimorph Strip-benders with centerbore
Supplier	Piexomechanik GmbH		
Products Names	**BM 300/70/1.5mm**		
References	http://www.piezomechanik.com/		
Strengths/ Weaknesses	Bimorph have large displacements compared to piezoelectric plate and multilayered stacked actuator (normally 5-10 % of total length for strip-benders), quite small resonance frequency, low stiffness, and low blocking force.		
Max Displacements (mm)	± 1500	± 0.70	± 0.280
Maximum Force (N)	0.3	3	20
Frequency Range (Hz)			
Resonance Frequencies (Hz)	80	6500	3000
Stiffness (N/µm)			
Damping			
Sensitivity			
Response Time (ms)			
Maximum Efficiency (%)			
Need of Voltage (V) / Current (A)	± 150 V	± 100 V	± 200 V
Size (H/L/W) (mm)	0.8/70/10	0.6/35/35	1.3/100/25
Weight (g)			
Price			
Comments	They exist with less displacement. Bimorph Strip-benders exist also with blocking force.		

Technology	*Piezoelectric*	
Principle	Dynamic Spring	Multilayered Spring-like
Supplier	EDO	Adaptronics, Inc
Products Names	**E400P-4**	**PPA80L**
References	http://www.edoceramic.com/ECPImages/PDFs%20for%20download/CatProd.pdf	
Strengths/ Weaknesses		
Max Displacements (mm)	0.050	0.080
Maximum Force (N)	4400	3500
Frequency Range (Hz)		
Resonance Frequencies (Hz)	18000	5000
Stiffness (N/µm)	140	43.8
Damping		
Sensitivity		
Response Time (ms)	0.30	0.14
Maximum Efficiency (%)		
Need of Voltage (V) / Current (A)		-20 to 150 V
Size (H/L/W) (mm)	25/60/12	107.0/22.5/18.0
Weight (g)	0.113	147.0
Price		
Comments	The maximum force for E400P-4 is preload. It exists with less displacement and less force. PPA80L exist with less displacement.	