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# ACTIVE NOISE AND VIBRATION CONTROL LITERATURE SURVEY: Sensors and Actuators

*T.S. Koko – U.O. Akpan – L. Guertin – A. Berry – P. Masson*

*Martec Limited*

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*T.S. Koko – U.O. Akpan – L. Guertin – A.Barry – P. Masson*

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## ABSTRACT

This study arose from the need to control noise in naval vessels to reduce their detectability and hence vulnerability to enemy attack. In this report a detailed review of sensor and actuator technologies that could be used for active noise and vibration control in ship structures has been provided. The review focused on a wide range of sensor and actuator materials, such as piezoelectric and electrostrictive materials; magnetostrictive materials; shape memory alloys (SMAs); optical fibers; electrorheological and magnetoeheological fluids, microphones; loudspeakers; electrodynamic actuators; and hydraulic and pneumatic actuators. The diesel engine noise problem was the focus of the study, and consideration was given to the noise transmission paths, namely, the engine mounting system; the exhaust stacks and piping systems; the drive shafts and mechanical couplings; and the air borne radiated noise. Sensor and actuator technologies that are suitable for active control of noise through these paths (or similar structural systems) were reviewed. Based on factors, such as cost, frequency of the disturbance, the operating (marine) environment, experience in other applications, ease of implementation, and the expected performance, detailed recommendations on sensors and actuators for the paths were provided. For the engine mounting system and the drive shafts and mechanical couplings, sensors and actuators, such as accelerometers, force transducers, hydraulic actuators, piezoelectric materials, and electrodynamic actuators, were recommended. On the other hand, acoustic sensors and actuators, such as microphones and loudspeakers, were recommended for the exhaust stacks and piping systems, and the air borne noise. It was also recommended that the active control strategies be combined with passive treatments when ever possible, to increase the robustness of the control system and to provide a fail-safe design. A systematic and pragmatic program, based on combined experimental and numerical investigations, was suggested in order to implement the sensor and actuator technologies recommendations made in the study.

## RÉSUMÉ

L'étude en cause a résulté du besoin de limiter le bruit à bord des navires de guerre pour réduire leur détectabilité donc leur vulnérabilité à l'attaque ennemie. Le présent rapport contient une étude approfondie des technologies des capteurs et des actionneurs qui pourraient être utilisées pour la limitation active du bruit et des vibrations dans les structures de navire. L'étude s'est concentrée sur une vaste gamme de matériaux de capteur et d'actionneur, tels que les matériaux piézoélectriques et électrostrictifs, les matériaux magnétostrictifs, les alliages à mémoire de forme (SMA), les fibres optiques, les fluides électrorhéologiques et magnétorhéologiques, ainsi que sur les microphones, les haut-parleurs, les actionneurs électrodynamiques et les actionneurs hydrauliques et pneumatiques. L'étude portait en premier lieu sur le problème du bruit des moteurs diesel, et l'on a pris en considération les trajets de propagation du bruit, notamment le système de suspension du moteur; les cheminées et tuyauteries d'échappement, les arbres d'entraînement et les accouplements mécaniques, de même que le bruit rayonné dans l'air. On a étudié les technologies des capteurs et des actionneurs convenant à la limitation active du bruit se propageant par ces trajets (ou des systèmes structuraux semblables). Compte tenu de facteurs

tels que le coût, la fréquence des perturbations, l'environnement d'utilisation (maritime), l'expérience d'autres applications, la facilité de mise en œuvre et les performances anticipées, des recommandations détaillées relatives aux capteurs et aux actionneurs sont données. Pour le système de suspension du moteur, les arbres d'entraînement et les accouplements mécaniques, des capteurs et des actionneurs, tels que des accéléromètres, des transducteurs de force, des actionneurs hydrauliques, des matériaux piézoélectriques et des actionneurs électrodynamiques sont recommandés. D'autre part, des capteurs et actionneurs acoustiques, tels que des microphones et des haut-parleurs sont recommandés pour les cheminées et tuyauteries d'échappement et le bruit rayonné dans l'air. Il est également recommandé que les stratégies de limitation active soient combinées, dans la mesure du possible, à des traitements passifs en vue d'augmenter la robustesse du système de limitation et de fournir une conception à sécurité intégrée. On suggère un programme systématique et pragmatique basé sur des expériences et des calculs, afin de mettre en œuvre les recommandations relatives aux technologies des capteurs et des actionneurs faites dans l'étude.

DREA CR 1999-110

# Active Noise and Vibration Control Literature Survey: Sensors and Actuators

T.S. Koko, U.O. Akpan, L. Guertin, A. Berry, and P. Masson

## EXECUTIVE SUMMARY


### Background

Reducing or controlling the underwater acoustic signatures of naval ships and submarines has historically meant the use of "passive" techniques, many of which rely on the energy dissipative properties of polymeric (particularly lossy elastomeric) materials. Components such as engine mountings, flexible pipework couplings and joints, vibration isolators and interior acoustic absorbers can be used to prevent acoustic energy from being coupled into the structure of the surface ship or submarine. While these methods have proven to be effective in general, there are usually low frequency limitations which cannot be overcome by passive means. Active noise control methods show promise either as replacements for, or in combination with passive techniques. The purpose of this work was to conduct a literature review of sensors and actuators that would be appropriate for use in the active control of noise and vibration in ship structures. The study was conducted as part of a Technology Investment Fund sponsored project entitled "Warship Underwater Signature Reduction by Active Means".

### Principal Results

The review focused on a wide range of sensor and actuator materials or devices. Particular attention was given to noise due to the diesel engine, and consideration was given to the following noise transmission paths: the engine mounting system, the exhaust stacks and piping systems; the drive shaft and mechanical couplings; and air borne noise. Specific recommendations on sensor and actuator technologies for the various paths were made based on factors such as cost, frequency of disturbance, operating (marine) environment, ease of implementation, and expected performance. In general, non-acoustic sensors and actuators (such as piezoelectric materials or electrodynamic shakers) were recommended for non-acoustic paths such as engine mounts and drive shaft, and acoustic sensors and actuators were recommended for acoustic paths such as cooling and piping systems.

### Significance of Results

This report, along with a companion review on controller technologies, provide a useful starting point for research and development in the area of active noise control in ship structures.

### Future Work

Future work in this area will include proof of concept studies to demonstrate the potential benefits of active noise reduction technologies in naval ships, using a laboratory environment and/or CFAV Quest as demonstration platforms.

## TABLE OF CONTENTS

TABLE OF CONTENTS - Continued

TABLE OF CONTENTS - Continued

## LIST OF TABLES

## LIST OF FIGURES

## LIST OF FIGURES - Continued

# 1. INTRODUCTION

## 1.1 Background

The success of surface warships and submarines in combat may depend significantly on how difficult it is for them to be detected by enemy vessels. One of the ways by which enemy ships can detect the vessels is through sound or noise that is radiated from the vessels. The dominant sources of noise radiation into water from ships are the ship engine and machinery, the propeller cavitation noise, the noise radiation from propeller blades, and the hydrodynamic pressure fluctuations induced by turbulent water flow along the ship hull. At speeds below propeller cavitation inception, a ship's acoustic signature is generally dominated by structurally transmitted noise from on-board machinery.

In order to reduce the detectability and hence the vulnerability of surface ships and submarines to enemy attack, it is necessary to reduce their acoustic signatures. Reducing or controlling ship noise has traditionally been implemented by passive means, such as the use of vibration isolation mounts, flexible pipe-work and interior acoustic absorbing materials. However, these passive noise control techniques are effective mostly for attenuating high frequency noise and are generally ineffective for controlling low frequency noise. On the other hand, active noise control methods have proven to be effective in controlling low frequency and tonal noise. Consequently, active control methods may be used as replacements for, or in combination with, passive techniques, for controlling or reducing ship noise.

Active noise control (ANC) involves the purposeful reduction/elimination of noise, either by modification of the dynamic properties of the system, or by active noise cancellation through linear superposition of a secondary noise field of equal but opposite strength. An active noise control system will typically consist of all or some of the following ingredients:

(i) Sensors, which provide information on the primary noise source;
(ii) Actuators, which provide means of modifying the system characteristics or providing secondary noise cancellation excitation; and
(iii) Controllers that determine the manner in which the secondary excitations are applied to the system to reduce the noise.

This study arose from a need by the Defense Research Establishment Atlantic (DREA) to investigate the applicability of active noise control methodologies for reducing or controlling the underwater acoustic signatures of Canadian naval ships and submarines. To do proper justice to the subject, it is necessary to have a proper understanding of all three ingredients of ANC listed above. This study attempts to provide this needed understanding on the subject.

## 1.2 Objectives and Scope

The objective of this study is to conduct a comprehensive literature review on the subject of active control of ship engine noise, with emphasis on actuators and sensors technologies. A review of controller technologies is provided in a companion study (Akpan et al, 1999).

To put the study in proper perspective, a brief overview of the fundamental concepts of ship noise control is first presented in Chapter 2. Then a review of possible materials that can serve as sensors and actuators for noise and vibration control are provided in Chapter 3. This detailed overview covers sensors and actuators such as piezoelectric and electrostrictive materials; magnetostrictive materials; shape memory alloys (SMAs); optical fibres; electrorheological and magnetorheological fluids; microphones, LVDT; accelerometers; loudspeakers; and hydraulic and pneumatic actuators are discussed.

Chapter 4 provides a review of active noise control studies with consideration given to the various acoustic paths through which noise can be transmitted from ship structures. The paths considered include the mounting system; exhaust stacks and piping systems, drive shafts and mechanical couplings; and air borne radiated noise. The chapter ends with a list of vendors and prices of sensors and actuators. Chapter 4 also contains a review of passive, active and hybrid noise and vibration control methodologies. In Chapter 5, recommendations on suitable sensors and actuators for controlling noise arising from the possible paths are provided. Chapter 6 provides a summary of the study and recommendations on the way ahead.

# 2. FUNDAMENTAL CONCEPTS ON SHIP NOISE CONTROL

## 2.1 Noise Sources and Transmission Paths in Ship Structures

There are many sources of noise in a ship structure. These include propulsion systems, exhaust stacks, and various types of on board equipment. In the present study the engine system has been identified as the principal noise source of interest. A typical ship engine along with its mounting system is schematically depicted in Figure 2.1. The figure shows the various vibro-acoustic paths through which the engine vibration is transmitted to the ship structure and eventually radiated into the surrounding mediums. The ship structure transmits noise after it has been excited by mechanical motion (engine vibration). The engine vibration is due primarily to unbalanced rotating or oscillating parts, bearing noise, gear meshing and combustion. In general the spectrum of the excitation is broadband but exhibits spikes at frequencies corresponding to the shaft rotational speed and its harmonics. The various vibro-acoustic paths transmit noise in different ways. For example:

- The noise from the exhaust stack and the fuel intake and the cooling systems can be viewed as duct and piping noise. In this mechanism the pressure wave in the duct is excited and transmitted as noise;
- The mounting systems, which consist of the engine cradle, isolation mount, raft and foundation are mechanical connections between the ship hull and the machine. Vibration is transmitted from the engine motion to the ship hull through these connections. The induced hull vibration is transmitted to the surrounding medium and is radiated as acoustic noise. This noise transmission mechanism is referred to as structural acoustic radiation; and
- The engine vibration leads to airborne radiation within the enclosure, which may induce an acoustic load on the ship hull. This resulting excitation is radiated to the surrounding medium as acoustic noise.

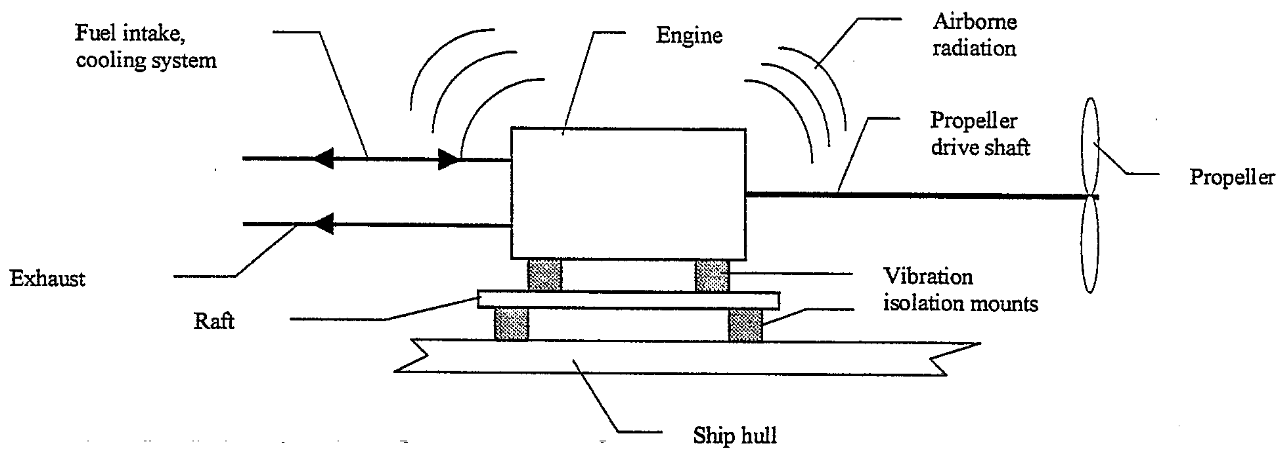



Figure 2.1. Schematics of a Ship Diesel Engine Mounting

### 2.1.1 Objective of Ship Noise Control

There are many reasons why ship noise is undesirable. These include:

- Significant levels of radiated noise could constitute a potential threat because of the higher susceptibility to detection by enemy naval vessels such as submarines and surface ships;
- High levels of ship noise may cause discomfort to crew; and
- High levels of noise and vibration in the vicinity of the engine mounts could cause localized acoustic fatigue.

The objective of ship noise control is the minimisation of the acoustic radiation from the ducting and piping systems, the ship hull and appendages to the surrounding water.

## 2.2 Prediction and Assessment of Noise Levels

Once the noise sources and transmission paths have been identified, it is essential to have capabilities to accurately assess the noise and vibration levels either experimentally or computationally. Typically, acoustic sensors can be installed to measure the noise levels at different locations. The deployment and maintenance of such sensors throughout the service life of the ship structure has several practical difficulties and challenges. Some of these include disconnection after some time of operation, resistance to water and environmental conditions (for submerged locations) and the difficulty of taking far field measurements.

Because of the challenges associated with full reliance on experimental measured acoustic levels, it is very important to have capabilities to accurately predict the acoustic fields pertaining to different ship noises and transmission paths. Several proprietary and commercial computer programs are available for vibro-acoustic modelling. Examples in the connection include AUTOSEA, SYSNOISE and AVAST, (Brennan, 1998).

In particular, a general-purpose code called AVAST (acronym for Acoustic Vibration and Strength Analysis Program) has been developed over the years by MARTEC Limited under the sponsorship of the Defence Research Establishment Atlantic (DREA). AVAST is a vibro-acoustic modelling/analysis program that utilizes a coupled boundary element and finite element modelling strategy. The finite element method is used to model the structure and the boundary element method is used for the acoustic medium (fluid). AVAST capabilities have been specifically targeted at ship structure modelling. As a result, we believe it is an appropriate and logical starting point for further development of noise modelling and predictive capabilities that are needed to support the design of adequate control strategies.

In general, there are two distinct methods that are used for the reduction of acoustic noise and radiation. These are passive and active control methods. The methods are briefly described in the next sections.

### 2.2.1 The Concept of Passive Noise Control

Passive noise control is essentially the process of reducing unwanted noise by utilising the absorption property of materials. In this approach, sound absorbent materials are mounted on or around the primary source of noise or along the acoustic paths between the source and the receivers of noise. Traditional methods for reducing acoustic noise and vibration have employed passive techniques and these techniques have been shown to be very effective at high frequencies (Harris, 1991; Barenik and Ver, 1992). At low frequencies, however, passive control techniques are not effective because the long acoustic wavelength of the noise requires large volumes of the passive absorbers (Fuller and von Flotow, 1995). Furthermore, at low frequencies it is difficult to stop the transmission of noise from one space to another (Elliot and Nelson, 1993). Attempts to overcome the limitation of passive control led to the development of active noise control by Lueg in 1933.

### 2.2.2 The Concept of Active Noise Control

Active noise control can be viewed from two perspectives:

- System dynamical properties modification; or
- Active cancellation.

In system dynamic properties modification the active control system changes the physical characteristics of the overall acoustical system such as the input impedance presented to the external disturbance; the impedance of the modes or the nature of the boundary conditions. The changes in the dynamical characteristics, in turn, reduces the response of the system to the external excitation. This reduction may be due to a variety of mechanisms such as reduction of transmissibility across discrete connections or other identifiable transmission paths. This approach includes the emerging technology of Active Structural Acoustic Control (ASAC) which uses smart structures.

In active noise cancellation the system actuators inject sound which by linear superposition is additive to the field. It operates on the principle of superposing waveforms, by generating a cancelling waveform whose amplitude and envelope match those of the unwanted noise, but whose phase is shifted by 180° (Leitch and Tokhi, 1989). The source of the unwanted noise is referred to as the primary source. The source of the cancelling noise is referred to as the secondary source and is usually driven by a controller. The cancelling noise is sometimes referred to as anti-noise or secondary noise. The concept of active noise cancellation is shown in Figure 2.2. Elliot and Nelson, 1992, have shown that the process of generating a stable destructive noise can only be achieved at low frequencies, that is, the effectiveness of active noise cancellation is restricted to low frequencies. Since passive noise cancellation techniques are effective at high frequencies, then active and passive noise control methods can be combined together to minimise both high and low frequency noise. Therefore, the best solution to a noise control problem is a combination of passive and active control.

A great amount of literature has been published on the origin and development of active noise control. Several studies, (Warnaka, 1982; Stevens and Ahuja 1989; Elliot and Nelson 1990, 1993; Fuller and von Flotow, 1995), have chronicled the progression and historical development of active noise control from its inception with Lueg's idea in 1933. These studies demonstrated that practical implementation of active noise control has been driven primarily by the development of modern electronics. Furthermore, Elliot and Nelson, 1993, emphasised that an understanding of both the acoustic principles of the physical system and the control system is essential for successful application of active noise control technology.

The present study is concerned with the review of sensor and actuator technologies that have found practical applications particularly in the marine environment. Emphasis is placed on active control methodologies that have been found to be effective in noise and vibration control. For clarity of presentation, the main features of an active control system are illustrated in Figure 2.3. The basic components are the physical system (this encompasses the plant, the sensors and the actuators) and the electronic control system (Nelson and Elliot, 1992). The main features are:

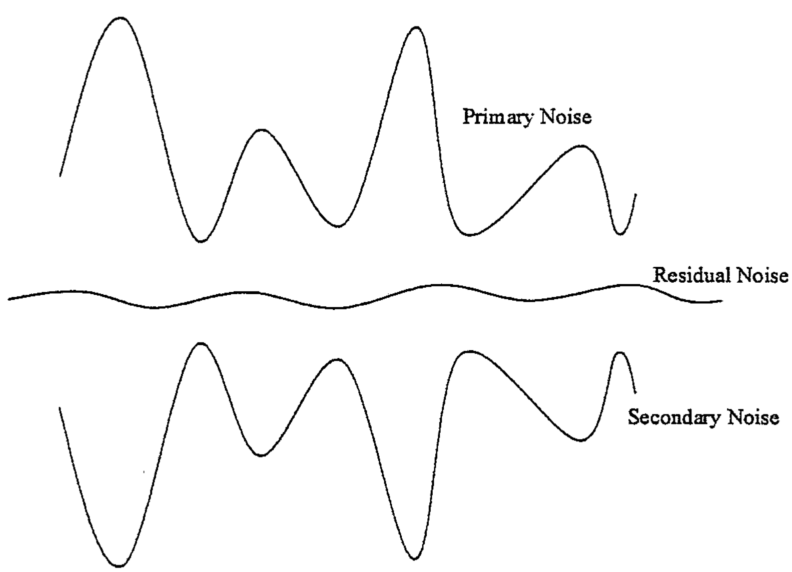



Figure 2.2: The Concept of Active Noise Cancellation

1. The primary source of noise/disturbance and the system to be controlled. This is usually referred to as the plant.

2. The input and error sensors: The input sensors are the electro-acoustic (microphones) or electro-mechanical (accelerometers, tachometers) devices that measure the disturbance from the primary source and communicate it to the controller. They are often referred to as reference sensors. The error sensor monitors the performance of the active controller.

3. The Actuators: These are the electro-acoustic or electro-mechanical devices that generate the secondary noise or anti-noise in order to reduce or cancel the primary noise. In some cases, the actuators modify the dynamic properties of the system in order to reduce their noise radiation efficiencies. Examples of actuators include speakers, piezoelectric material and vibration shakers. The actuators, plant and the sensors are collectively referred to as the physical system.

4. The Active Controller: This is the signal processor (usually a digital electronic system) that gives command to the actuators. The controller bases its output on sensor signals (primary noise sensor/error sensor) and usually on some knowledge of how the plant responds to the actuator.

The performance of an active controller depends on several factors. They include causality condition, acoustic feedback, and the physical arrangement of the actuators and sensors (Nelson and Elliot, 1992).

Causality condition relates the acoustic delay to the electrical delay. In a practical active control system, there is an acoustical delay between the primary noise and the secondary noise (see Figure 2.3). This delay depends on the speed of propagation of the primary noise in the medium. Also, there is an electrical delay between the primary noise sensor (commonly referred to reference sensor) and the actuator. The electrical delay depends on the time it takes the controller to process the input signal. An active control system is said to be causal when the controller produces the anti-noise at a downstream location at the same time that the primary noise arrives. Causality condition requires that, for effective control, the acoustic delay must be longer than the electrical delay, that is, the maximum response time of the sensor-controller-actuator system should be less than the sound propagation time.

When the actuator generates the anti-noise, it travels downstream to generate a zone of quietness. The anti-noise also travels upstream towards the primary source of noise. The secondary noise that moves upstream corrupts the primary noise that is measured by the reference input sensor. The secondary noise that propagates upstream is referred to as secondary effect or acoustic feedback. Acoustic feedback must be taken into consideration in the design of active controllers because it has the potential to undermine the effectiveness of the controller.

The physical arrangements of the control sources (actuators) and the sensors play an important role in determining the effectiveness of the control system. Moving the location of the control sources and sensors can affect the controllability, observability and the stability of the control system.

Active Noise Control Methodologies (ANC) can be classified into two main categories:

- Feedforward Control (FFC) and;
- Feedback Control (FBC).

A summary of the description of the two methodologies is given in the companion report (Akpan, et al. 1999). The controllers that have been used in active noise control methodologies (FFC and FBC) have evolved over the years from analogue to digital designs. Elliot and Nelson, 1992, have noted that progress in active noise control has been tied to advancement in digital electronics. Therefore a review of digital control technologies is undertaken in the next chapter.

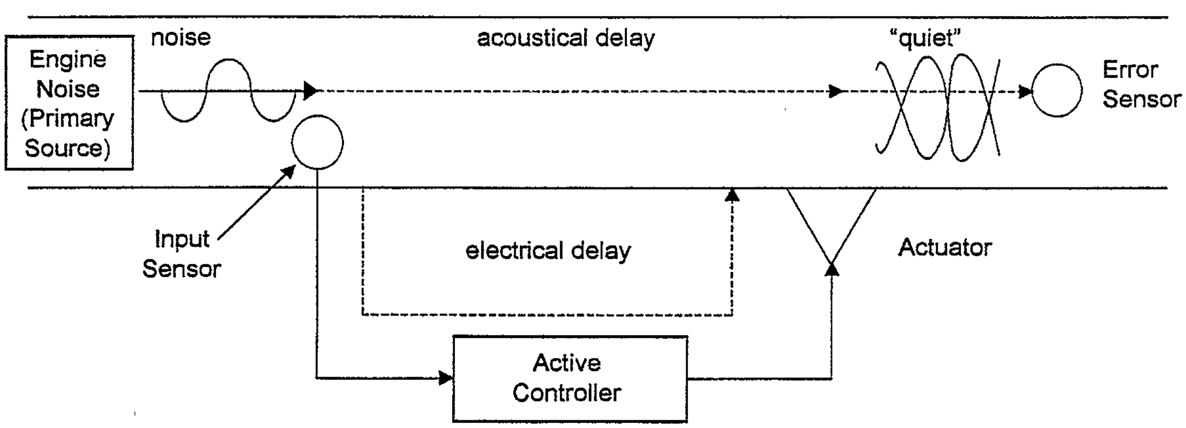



Figure 2.3: The Main Features of an Active Noise Control System

# 3. OVERVIEW OF SENSORS AND ACTUATORS FOR ACTIVE NOISE AND VIBRATION CONTROL (ANVC)

## 3.1 Piezoelectric Materials

Piezoelectric materials are the most mature and developed of all high-speed sensor and actuator technologies. Piezoelectricity was discovered by the Curie brothers in 1880. When a piezoelectric material is subjected to a mechanical stimulus, an electrical charge or voltage is induced in the material. This is called the direct piezoelectric effect, which enables the material to be used as a sensor. On the other hand, when the piezoelectric material is subjected to an electrical charge or voltage, a mechanical force or strain is induced in the material. This is called the converse piezoelectric effect, which enables the material to be used as an actuator.

The induced strain is directly proportional to the applied electric field and the linear piezoelectric constitutive equations are given by

$$\begin{aligned} T &= cS - eE \\ D &= e^T S + \varepsilon E \end{aligned} \tag{3.1}$$

where $T$, $E$, $S$ and $D$ are the vectors of stress, electric field, strain and electric displacement (charge per unit area), $c$, $e$ and $\varepsilon$ are the matrices of the elastic stiffness coefficients, piezoelectric stress constants and dielectric coefficients, respectively. The first and second of equations (3.1) describe the direct and converse effects, respectively. Piezoelectric materials are also called soft ceramics because they are characterized by high dissipation factors (dielectric losses). As a result, they have high hysteresis in the displacement versus voltage curves (Simonich, 1996).

The three most important piezoelectric materials are lithium niobate (Li $NbO_3$), polyvinylidene (PVDF or $PVF_2$) and lead zirconate titanate (PZT) (Rao and Sonar, 1994). $LiNbO_3$ is a crystal with a high electromechanical coupling and very low acoustical attenuation. Piezoelectricity is obtained from a strip of PVDF by stretching it under a high voltage. PVDF, originally discovered in 1969, is known for its flexibility, lightweight durability, and relatively low acoustic impedance. PZT is by far the most commonly used piezoelectric material. This is a ferroelectric ceramic material with direct and converse piezoelectric properties. A wide variety of PZT formulations have been developed, with PZT-5 being one of the most widely used formulations for actuator applications (Giurgiutiu and Rogers, 1996; Bowen et al., 1996; Sherrit et al, 1992, 1995, and 1997; Thunder, 1997). PZT can be used as sensors or actuators. For actuators, the device usually consists of a stack of many layers of the PZT, alternatively connected to the positive and negative terminals of a high voltage source as shown in Figure 3.1. The stacks are either bonded together using a structural adhesive, or assembled in the green state and subjected to hot isostatic pressure (HIP) (Giurgiutiu et al. (1996)). In the former method of construction, the stiffness of the actuator device is limited by the lower modulus of the adhesive (typically 4-5 GPa compared to 70-90 GPa for the PZT Ceramic). The second method results in much stiffer products, but processing limitations restrict their applicability to small size stacks only.

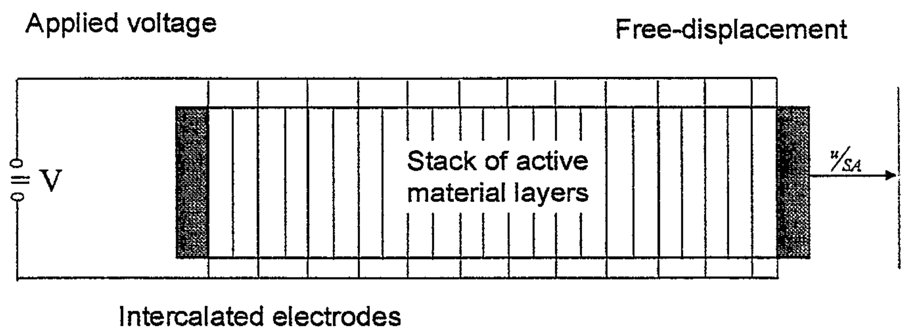



Figure 3.1: Induce Strain Actuator Using PZT or PMN Stack
(Guirgiutiu et al. 1996)

The mechanical, dielectric and electro-mechanical coupling properties of some piezoelectric materials are shown in Table 3.1. As the table shows, the strains for these materials are typically less than 0.1%. The energy capabilities of some commercially available piezoelectric ceramic materials have also been discussed by Giurgiutiu et al. (1996). It is also widely known that piezoelectric materials are applicable over a wide range of frequencies and have very quick response times, which make them suitable for high precision measurements. Because of these advantages, piezoelectric material have been used as sensors and actuators in several structural systems for shape, vibration, and noise control. Excellent reviews of some of these applications have been presented by several researchers including Rao and Sonar (1994), Housner et al. (1997) and Tani et al. (1998).

Table 3.1: Properties of Selected Piezoelectric Materials

| Property | PZT-4 | PZT-5H | BM532 Piezoceramic | Motorola 3203 HD PZT | PZT-G1195N |
|---|---|---|---|---|---|
| Elastic Properties: | | | | | |
| $E_{11}$ (GPa) | 81.3 | 170.0 | 71.4 | 1.77 | 63 |
| $E_{22}$ (GPa) | 81.3 | 170.0 | 71.4 | 1.77 | 63 |
| $E_{33}$ (GPa) | 64.5 | 158.0 | 5.0 | 1.77 | 63 |
| $G_{12}$ (GPa) | 30.6 | 23.0 | 27.5 | 0.681 | 24.2 |
| $G_{23}$ (GPa) | 25.6 | 23.0 | 27.5 | 0.681 | 24.2 |
| $G_{13}$ (GPa) | 25.6 | 23.0 | 27.5 | 0.681 | 24.2 |
| $\nu_{12}$ | 0.33 | 0.3 | 0.3 | 0.3 | 0.3 |
| $\nu_{13}$ | | 0.3 | 0.3 | 0.3 | 0.3 |
| $\nu_{23}$ | | 0.3 | 0.3 | 0.3 | 0.3 |
| Piezoelectric Coefficients ($10^{12}$ m/V): | | | | | |
| $d_{31}$ | -122 | | -200 | -295 | -254 |
| $d_{32}$ | -122 | | -200 | -295 | -254 |
| $d_{33}$ | 285 | | 580 | 569 | 374 |
| $d_{24}$ | | | 0 | 560 | 584 |
| Piezoelectric Stress Constants (C/m$^2$): | | | | | |
| $e_{31}$ | | -6.5 | | | |
| $e_{32}$ | | -6.5 | | | |
| $e_{33}$ | | 23.3 | | | |
| $e_{23}$ | | 17.0 | | | |
| Electric Permittivity: | | | | | |
| $\varepsilon_{11}/\varepsilon_0$ | 1480 | 1695 | 3250 | 2418 | 1729 |
| $\varepsilon_{22}/\varepsilon_0$ | 1480 | 1695 | 3250 | 2418 | 1729 |
| $\varepsilon_{33}/\varepsilon_0$ | 1300 | 1695 | 3250 | 3333 | 1695 |
| Mass Density $\rho$ (kg/m$^3$) | 7600 | 7500 | 7350 | 7600 | |
| $\varepsilon_0 = 8.85 \times 10^{-12}$ farad/m, electric permittivity of air) | | | | | |

A good number of studies, covering theoretical and experimental investigations, have been focused on the application of piezoelectric materials for vibration control of flexible structures (eg. Crawley and de Luis (1987), Varadan et al. (1996), Rao and Sunar (1994), Tzou and Ye (1994); Koko et al. (1997) and Aglietti et al. (1997)). Studies on the use of piezoelectric materials for static shape control have also been reported by Koconis et al. (1994a,b) and Wang et al. (1997) and Varadarajan et al. (1998).

Of direct relevance to the noise control problem that is of interest in this study is the work by Sumali and Cudeny (1994) who developed an actuator from a stack of layers of piezoelectric material in an actively controlled engine mount that was designed to reduce structural vibrations. They developed a simple impedance model in which the velocity at the structure interface was expressed as a linear function of the engine force and actuator input voltage. The effects of several parameters, such as stiffness, dielectric constant and size of the piezoelectric stacks, on the actuator authority were evaluated. The active model is shown in Figure 3.2.

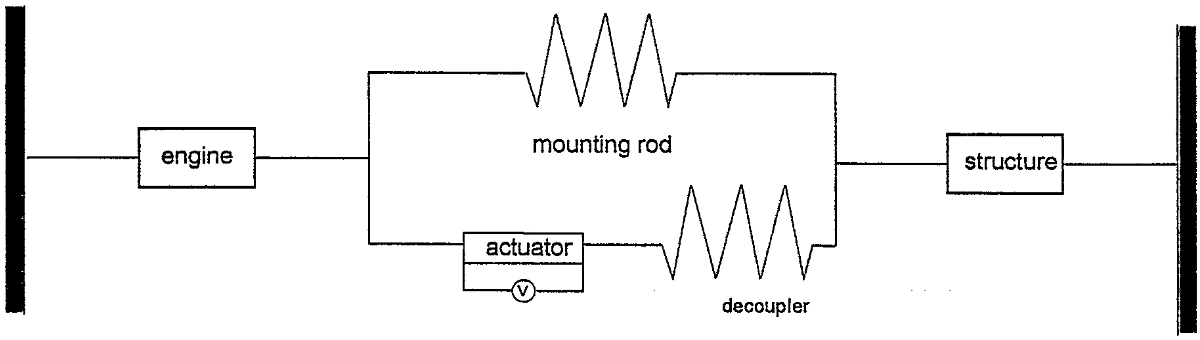



Figure 3.2: Active Mount Model (Sumali et al. 1994)

Most of the theoretical and experimental studies on the use of piezoelectric materials for active noise control have been directed at aircraft cabin noise control. For instance, Grewal et al. (1998) have investigated the use of piezoceramic elements to reduce cabin noise in the de Havilland Dash-8 series 100/200 aircraft. Their study showed that by judicious actuator and sensor design considerations systems using bonded piezoelectric actuators and vibration sensors alone are capable of simultaneously providing significant noise reduction as well as vibration suppression.

Sutliff et al. (1997) have investigated the use of PZT actuators for active noise control of low speed fan rotor-stator modes. The aim of their study was to assess the feasibility of using wall mounted secondary acoustic sources and sensors within a duct of a high bypass turbo fan aircraft engine for active noise cancellation of fan tones. PZT elements were used to excite a plate radiator that was designed so that the resonance frequency of its first mode corresponded to the design frequency. In their study active noise control was demonstrated for a single mode (tonal noise) and the target spinning mode.

Simonich (1996) also investigated the application of various types of piezoelectric actuators for active noise control of gas turbine engines. This was also a feasibility study aimed at investigating suitable actuator technologies for the problem. In his study Rainbow piezoceramic actuators (Heartling (1994)) were identified as the most favourable actuators for the application. This was based not only on their large strain capabilities (up to 500%) but also on preliminary cyclic testing and long-term endurance capability. Other researchers have also discussed the advantages and applications of the Rainbow actuators (Tani et al. (1998), Prasad et al. (1998)).

## 3.2 Electrostrictive Materials

Electrostrictive materials are similar to piezoelectric materials. When a mechanical force or strain is applied to the material an electric charge or voltage is induced; and conversely, when an electric field is applied across an electrostrictive material a mechanical strain is induced. Hence, electrostrictive material can be used as sensors or actuators, just like piezoelectric materials. The most commonly used electrostrictive material is the Lead Magnesium Niobate (PMN) ceramic material.

There are several differences between electrostrictive and piezoelectric materials. In electrostrictive materials, the induced mechanical strain is proportional to the square of the electric field, whereas it is proportional to the electric field in piezoelectric materials. Thus electrostrictive materials always produce positive displacements regardless of the polarity. As a result, they are always in compression when doing work and avoid typical weakness of ceramics in tension (Prasad et al. (1998)). Electrostrictive materials exhibit microsecond recovery time upon withdrawal of the electric field, compared to milliseconds for piezoelectric materials. Electrostrictive materials have lower dissipation factors (and low displacements and hysteresis) compared to piezoelectric materials, and are regarded as hard ceramics (Simonich (1996)). Figure 3.3 shows the strain-electric field behaviours of typical electrostrictive and piezoelectric materials. Although electrostrictive ceramics have lower strain levels than piezoceramics, strains that are comparable to PZT have been induced in PMN by poling the material at high d.c. fields (3-4kV/mm) (Prasad et al. (1998)).

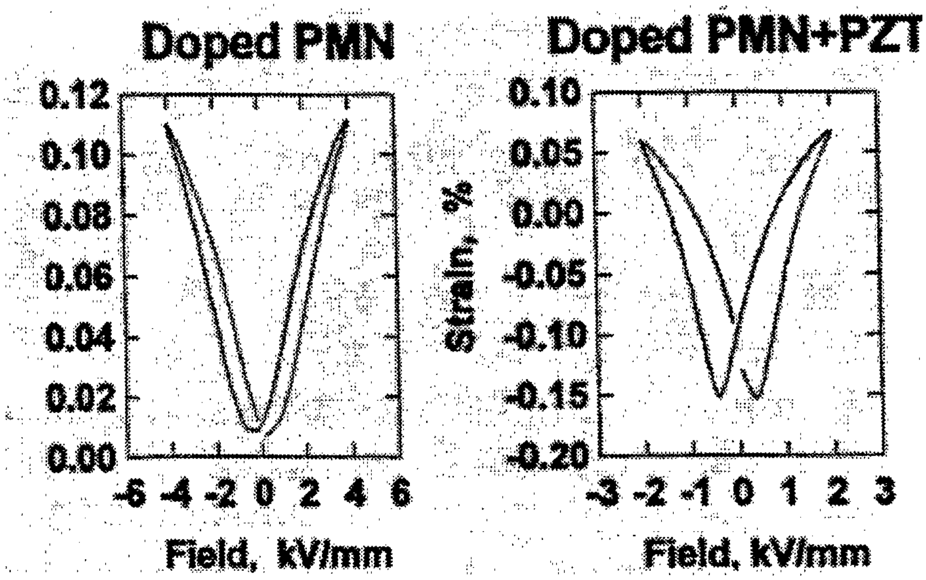



Figure 3.3: Hystersis Behaviours of Typical Piezoelectric and Electrostrive Materials (Prasad, 1998)

## 3.3 Magnetostrictive Materials

Magnetostrictive materials are those materials which when subjected to a magnetic field undergo an induced mechanical strain. On the other hand, when a mechanical stress (or strain) is applied to the material it undergoes domain changes, which yield a magnetic field. These materials can thus be used as sensors and actuators due to the direct and converse effects. The most common magnetostrictive material is TERFELOL (consisting of TERbium, FE (Iron) and dysprosium, that was developed by NOL (the Naval Ordinance Laboratory). The most commonly used formulation is TERFENOL-D. Magnetostrictive materials can exhibit strains of up to 0.2% at reasonably low magnetic field strength (Tani et al. (1998)).

Figure 3.4, originally presented by Giugiutu et al. (1996), shows the construction of a terfenol actuator. The actuator consists of a TERFENOL-D rod inside an electric coil which is enclosed in an annular armature. When the coil is activated, the TERFENOL rod expands and produces a displacement as shown in the figure. The TERFENOL-D bar, coil and armature are assembled between two steel washers and put inside a protective wrapping to form the basic magnetoactive induced strain actuator unit (Guignitu et al., (1996). The main advantage of terfenol is its high force capability at relatively low cost (Hansen and Snyder, (1997)). It also has the advantage of small size and light weight which makes it suitable for situations where no reactive mass is required, such as in stiffened structures in aircraft and submarine hulls.

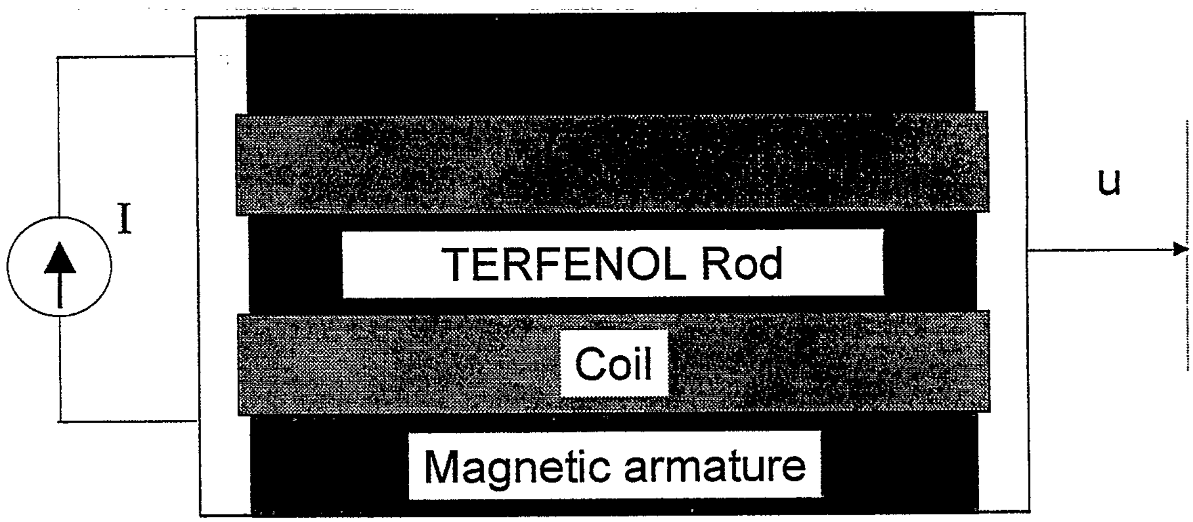



Figure 3.4: Induced Strain Actuator Using TERFENOL Magnetostrictive Rod
(Guirguitiu et al. 1996)

The disadvantages of terfenol include its brittleness and low tensile strength (100 MPa) compared to compressive strength of 780 MPa. Its low displacement capability is also a major disadvantage especially in the low frequency range (less than 100 Hz). In addition, it also exhibits large hysteresis resulting in a highly nonlinear behaviour that is difficult to model in practical applications (Hansen and Snyder (1977), Tani et al. (1998)).

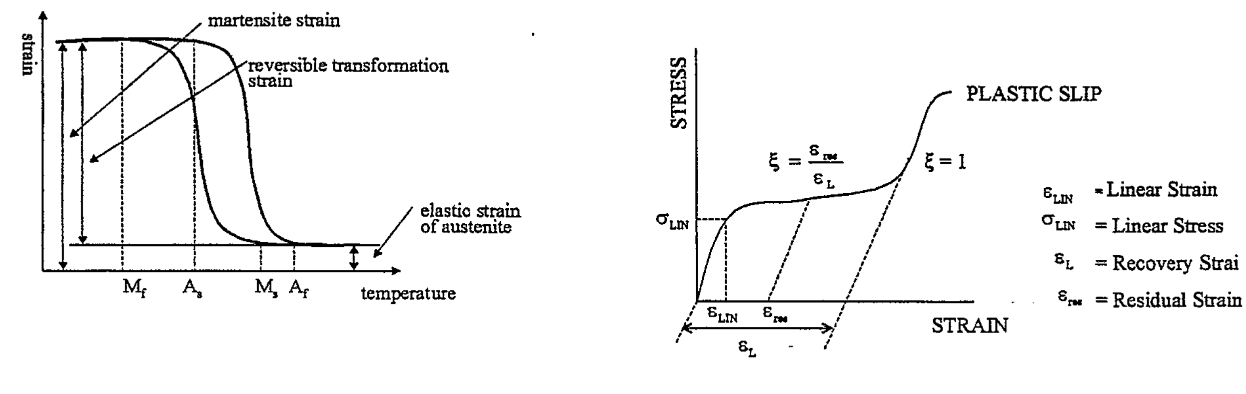
A review of studies on modelling the nonlinear behaviour of TERFENOL-D has been presented by Tani et al. (1998). These authors have also provided a review of recent applications of magnetostrictive materials in smart structures, Ackermann et al. (1996) developed a transduction model for magnetostrictive actuators through an impedance analysis of the electro-magneto-mechanical coupling of the actuator device. This model provided a tool for in-depth investigation of the frequency-dependent behaviour of the magnetostrictive actuator, such as energy conversion, output stroke and force. The feasibility of using embedded magnetostrictive mini actuators (MMA) for vibration suppression has been investigated by Anjanappa and Bi (1994). Several other applications of magnetostrictive materials for vibration suppression have also been presented by Tani et al. (1998).

### 3.4 Shape Memory Alloys (SMAs)

Shape memory alloys (SMAs) are materials that undergo shape changes due to phase transformations associated with the application of a thermal field. When a SMA material is plastically deformed in its martensitic (low temperature) condition, and the stress is removed, it regains its original (memory) shape by phase transformation to its austenite (high temperature) condition, when heated. SMAs are considered as functional materials because of their ability to sense temperature and stress loading to produce large recovery deformations with force generation. TiNi (nitinol), which is an alloy comprising approximately 50% nickel and 50% titanium, is the most commonly used SMA material. Other SMA material including FeMnSi, CuZnAl and CuAlNi alloys have also been investigated (Tani et al. 1998, Housner et al. 1997).

Typically, plastic strains of 6-8% can be completely recovered by heating nitonol beyond its transition temperature (of 45°C-55°C). According to Liang and 90, restraining the material from regaining its memory shape can yield stresses of up to 500 MPa for 8% plastic strain and a temperature of 180°C. By transformation from the martensite to austenite phase, the elastic modulus of nitinol increases three-fold from 25 GPa to 75 GPa and its yield stress increases eight-fold from 80 MPa to 600 MPa (Hansen and Scott, 1992).

SMAs can be used for sensing or actuation, although, they are largely used for actuation due to their large force generation capabilities. They have very low voltage requirements for operation and are very suited for low frequency applications. However, their use is limited by their slow response time which makes them suitable for low precision applications only. Also, they exhibit complex constitutive behaviour with large hysterises, which makes it difficult to understand their behaviour in active structural systems. To provide a better understanding of the behaviour of SMAs several researchers have focused on the development of constitutive models for SMAs. Some of the most prominent and commonly used ones are those by Tanaka (1986), Liang and Rogers (1990) and Boyd and Lagoudas (1996). These models are derived from phenomenological considerations of the thermo-mechanical behaviour of the SMAs. Figure 3.5 shows typical strain-temperature and stress-strain curves for SMA material (Liang and Rogers, 1990; Guenin and Gaudez, 1996). Tani et al. 1998 have highlighted several other SMA consititutive models in the review paper.




(a) (b)

Figure 3.5: Typical Curves for SMAs, (a) Strain Temperature Behaviour at Constant Stress, (b) Stress Strain Relationship

Because of the numerous advantages they offer, several investigations on the application of SMAs have been carried out within the present decade. Overviews of these applications have been provided by Housner et al., 1997 and Tani et al., 1998 in their review papers. As described by these authors, applications of SMAs have been in six main areas, namely:

(i) Fabrication of SMA hybrid composites, where SMA elements have been integrated (either embedded or bonded) into various composite materials in order to achieve a specific behaviour;

(ii) Active frequency tuning, where SMA wires are used to induce stress in the target structure or SMA elements are used to induce changes in the structural stiffness to tune the frequencies and amplitude of occillations in structures;

(iii) Active shape and deflection control, where SMA wires are embedded in composite structures to induce significant changes in the shape of host structure;

(iv) Active damage control, in which the SMA recovery force is used to reduce damage in host structure;

(v) Modelling of SMA elements within composite materials, focusing on several approaches for modelling the behaviour of SMA elements incorporated with a host structure; and

(vi) Structural response control, which has received the most attention on the application of SMAs. The applications in this area include seismic resistance design and vibration control.

## 3.5 Optical Fibers

For many applications, ideal sensors would have such attributes as low weight, small size, low power, environmental ruggedness, immunity to electromagnetic interference, good performance specifications and low cost. As technology has advanced, the need for these sensors has become increasingly acute in such areas as aerospace, defense, manufacturing, medicine and construction. The emergence of fiber optic technology, which was largely driven by the telecommunication industry in the 1970s and 1980s, in combination with low-cost optoelectronic components, has enabled fiber optic sensor technology to realize its potential for many applications (Udd, 1991; Tobin, 1991; Troy, 1997; Ansari, 1998).

A wide variety of fiber-optic sensors are now being developed to measure strain, temperature, electric/magnetic fields, pressure and other measurable quantities. Many physical principles are involved in these measurements, ranging from the Pockel, Kerr and Raman effects to the photoelastic effect (Yeh, 1990). These sensors use intensity, phase, frequency or polarization modulation (Peetermans, 1993). In addition, multiplexing is largely used for many-sensor systems. Fiber-optic sensors can also be divided in discrete sensors and distributed sensors to perform spatial integration or differentiation (Turner, 1990). The emphasis is herein put on strain sensors, in accordance with the scope of this review.

Three types of fiber-optic strain sensors will be reviewed in this section: extrinsic inter-ferometric sensors, Bragg gratings and sensors based on the photo-elastic effect.

### 3.5.1 Extrinsic Inter-ferometric Sensors

The most widely used phase modulating fiber-optic sensors are the inter-ferometric sensors. Two fibers and directional couplers are generally used for these sensors. One of the fibers acts as a reference arm, not affected by the strain, while the other fiber acts as the sensing arm measuring the strain field. By combining the signals from both arms, an interference pattern is obtained from the optical path length difference. This interference pattern is used to evaluate the strain affecting the sensing arm (by fringe counting, for example) (Jackson, 1994). These sensors have a high sensitivity and can simultaneously measure strain and temperature. Figure 3.6(a) shows some fiber-optic interferometers.

Another inter-ferometer now being used in industrial applications is the Fabry-Perot interferometer, where a sensing cavity is used to measure the strain (Belleville, 1993; Morin, 1996; Liu, 1997). This sensor uses a white-light source, a single multi-mode fiber and provides absolute measurements. This extrinsic inter-ferometer sensor is shown in Figure 3.6(b).

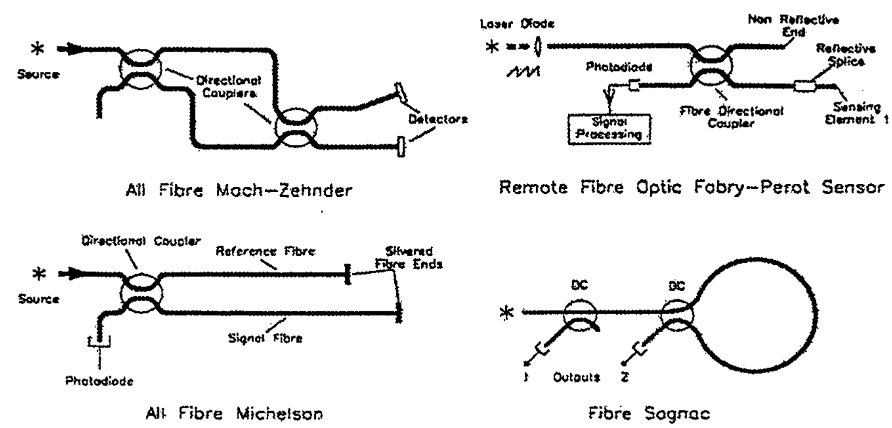



Figure 3.6 (a): Fiber-optic Interforemeters (Jackson, 1994)

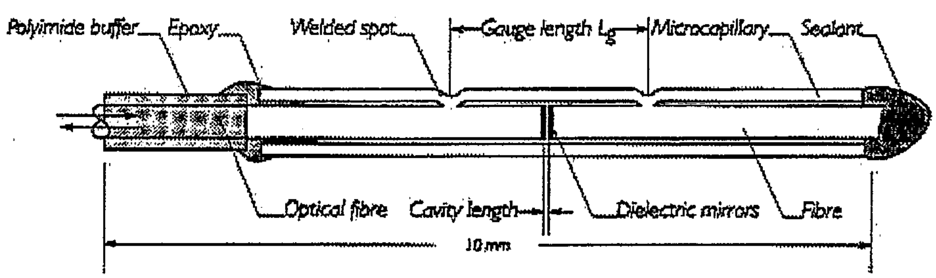



Figure 3.6 (b): Fabry-Perot Sensors Used for Ice Impact Monitoring (Morin, 1996)

### 3.5.2 Bragg Gratings

Bragg grating reflectors can be written on an optical fiber using a holographic system or a phase mask to generate a periodic intensity profile (Meltz, 1989; Hill, 1997; Friebele, 1998). The fiber is illuminated from the side and a permanent grating is induced by the photorefractive effect. Dependent parameters such as the central reflective wavelength, the reflectivity and the bandwidth of the grating can be selected. These sensors can be used as point or quasi-distributed sensors. The reflected signal from these sensors consist of frequency components directly related to the number of lines per millimeter of each grating reflector, and thus, to the strain experienced by the sensor.

Fiber-optic sensors based on Bragg gratings are used to measure strain and temperature, either simultaneously or individually (Du, 1999). The Bragg gratings are traditionally interrogated using a tuneable Fabry-Perot or a Mach-Zender interferometer. Recently, long-period gratings have been used to interrogate Bragg sensing gratings (Fallon, 1998). Bragg gratings have been used to measure vibrations either directly (Jones, 1998) or through the development of novel accelerometers (Todd, 1998). A typical fiber Bradd grating (FBG) system is illustrated in Figure 3.7.

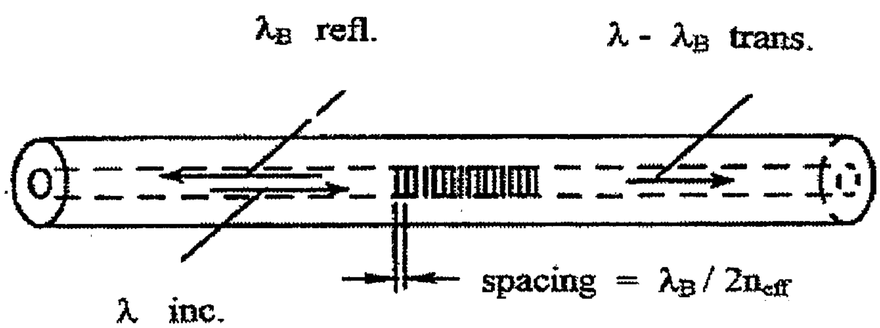



Figure 3.7: Bragg Grating on an Optical Fiber (Hill,1997)

### 3.5.3 Sensors Based on the Photo-elastic Effect

The photo-elastic effect is an expression of the phase variation of the light passing through a material (fiber) that is undergoing a strain variation. This phase variation $\Delta\phi$ is given by the following general relation:

$$\Delta\phi = \beta\Delta L + L\Delta\beta$$

where $L$ is the optical path length and $\beta$ is the propagation constant of the material. The first term on the right side of the equation represents the variation of the length produced by the strain while the second describes two effects: the photo-elastic effect and the modal dispersion caused by the variation of the diameter of the fiber.

These sensors are classified in modal inter-ferometric sensors and polarimetric sensors. The first class exploits a phase difference while the second class exploits the rotation of the plane of polarization in a two-mode fiber, all caused by the strain through the photo-elastic effect.

The modal inter-ferometric sensors are made of a two-mode elliptical or circular core fiber and can simultaneously measure, if desired, the strain and the temperature. In these sensors, the linearly polarized light is launched in a two-mode fiber and the phase difference between the two modes is used to produce an interference pattern at the output splice. The phase difference is proportional to the strain experienced by the fiber (Layton, 1979; Shankaranarayanan, 1987; Sirkis, 1991). These sensors suffer from the fragile splicings needed at both ends of the sensing region. As it integrates the strain effect over its length, this type of sensor can act as a spatial filter if the characteristics of the sensing fiber are adjusted so that the propagation constant is given a spatial weighting (Murphy, 1990; Murphy, 1992; Cox, 1991; Vengsarkar, 1991).

The polarimetric sensors use a highly birefringent fiber with a linearly polarized light source. A polarizer is used to measure the rotation of the plane of polarization after passing through the birefringent fiber subjected to strain (Spillman, 1989; Asundi, 1998). Figure 3.8 shows a typical polarimetric sensor.

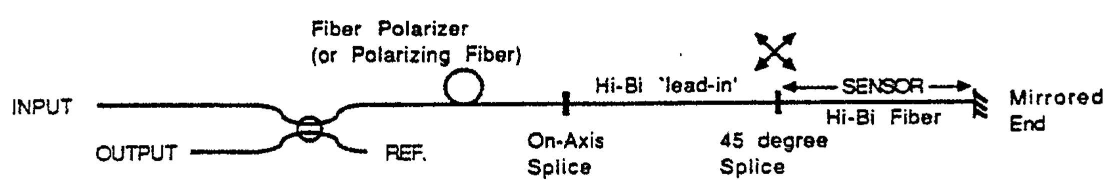



Figure 3.8: A Polarimetric Sensor (Turner, 1990)

## 3.6 Electro-rheological Fluids

Electro-rheological fluids (ER) are a class of controllable fluids that respond to an applied electric field with a dramatic change in rheological behaviour. The essential characteristic of ER fluids is their ability to reversibly change from free-flowing linear viscous liquids to semi-solids having controllable yield strength in milliseconds when exposed to an electric field (Housner et al. 1997). The ER fluids provide very simple, quiet and rapid response interfaces between electronic controls and mechanical systems. They are very suitable for vibration control, because of the ease with which their damping and stiffness properties can be varied with the application of an electric field.

ER materials consist of a base fluid (usually a low viscosity liquid) mixed with non-conductive particles, typically in the range of 1-10 μm diameter. These particles become polarized on the application of an electric field, leading to solidification of the material mixture. Typical yield stresses in shear for ER materials are about 5-10 kPa. The most common type of ER material is the class of dielectric oils doped with semi-conductor particle suspensions, such as alumino-silicate in paraffin oil. The material exhibits nonlinear behaviour, which is still not completely understood by the research community. This lack of understanding has hindered efforts in developing optimal applications of ER materials. However, many devices that can potentially use ER materials are continually being developed. Some of these include shock absorbers, engine mounts, robotic devices and valves with no moving parts (Housner, 1997, Hansen and Synder, 1997).

## 3.7 Magnetorheological Fluids (MR)

Magnetorheological fluids (MR) are similar to ER materials in that they are also controllable fluids. These materials respond to an applied magnetic field with a change in the rheological behaviour. MR fluids, which are less known than ER materials, are typically non-colloidal suspensions of micron-sized paramagnetic particles. The key differences between MR and ER fluids are summarized in Table 3.2. In general, MR fluids have maximum yield stresses that are 20 to 50 times higher than those of ER fluids; and may be operated directly from low-voltage power supplies compared to ER fluids which require high voltage (2-5 kV) power supplies. Furthermore, MR fluids are less sensitive to contaminants and temperature variations, than ER fluids. MR fluids also have lower ratios of $\eta_p / \tau_y^2$ than ER materials, where $\eta_p$ is the plastic viscosity and $\tau_y$ the maximum yield stress. This ratio is an important parameter in the design of controllable fluid device design, in which minimization of the ratio is always a desired objective (Housner, et al., 1997). These factors make MR fluids the controllable choice for recent practical applications.

Several MR fluid devices developed by Lord Corporation in North Carolina under the Rheonetic trademark have recently been reviewed by Housner et al., 1997. These devices include:

- An MR fluid rotary (damper) brake that is used as a controllable resistance element for programmable aerobic exercise equipment;
- An MR fluid linear damper – a small monotube damper designed for use in a semi-active suspension system in highway vehicle seats; and
- An MR seismic damper – a full scale 200 kN (20t) MR fluid damper, which was designed to demonstrate that the technology of semi-active MR fluid dampers will be scalable to full-sized engineering structures.

Table 3.2: Comparison of Properties of Typical MR and ER Fluid (Housner, et al. 1997)

| Property | MR Fluids | ER Fluids |
|---|---|---|
| Max yield stress $\tau_{y(field)}$ | 50-100 kPa | 2-5 kPa |
| Maximum field | ~250 kA/m | ~4 kV/mm |
| Plastic viscosity $\eta_p$ | 0.2-1.0 Pa-s | 0.2-1.0 Pa-s |
| Operable temperature range | -50°C-150°C | +10°C-90°C |
| Stability | Not affected by most impurities | Cannot tolerate impurities |
| Response time | Ms | Ms |
| Density | 3-4 g/cm$^3$ | 1-2 g/cm$^3$ |
| $\eta_p / \tau_{y(field)}^2$ | $10^{-10}$-$10^{-11}$ s/Pa | ms |
| Power supply (typical) | 2-25 V<br>1-2 A<br>(2-50 W) | 1-2 g/cm$^3$<br>$10^{-7}$-$10^{-8}$ s/Pa<br>2,000-5,000 V<br>1-10 mA<br>(2-50 W) |

## 3.8 Microphones

Microphones are usually the preferred type of acoustic sensors in active noise control applications. Relatively inexpensive microphones can be used in most active noise control systems, because accurate *absolute* sound pressure measurements are generally not necessary. Also, the frequency response flatness of the microphones is not critical in digital active control systems since this frequency response is compensated in the identification of the control path. The most common types of microphones are omni-directional, directional and probe microphones. These are discussed below.

### 3.8.1 Types of Microphones

#### 3.8.1.1 Omni-directional Microphones

In applications involving omni-directional microphones (eg. in the active control of enclosed sound fields), the two most common types are the prepolarised condenser (or electret) microphone, and the piezoelectric microphone.

*Condenser microphone*

A condenser microphone consists of a diaphragm which forms one electrode of the condenser, and a polarized backing plate, parallel to the diaphragm and separated by a narrow air gap. A prescibed constant electric charge is applied to the backing plate; therefore, variations of the air gap thickness in response to external acoustic pressure loads on the diaphragm will result in variations of the condenser capacitance and condenser voltage. Provided the diaphragm displacements are kept small relative to the static spacing, the output voltage is proportional to the diaphragm displacement, and external sound pressure magnitude. Details of the construction and use of electret microphones can be found in (Fredericksen, 1979). It is possible to purchase very inexpensive electret microphones for about $3 US; however, there may be significant deviations between the sensitivity or frequency response of individual electrets selected in a large batch, and manual selection and matching is sometimes necessary. The electret microphone is sensitive to dust and moisture on its diaphragm, but it can operate at higher temperatures than the piezoelectric microphone, and is less sensitive to mechanical vibration. Typical condenser microphone sensitivities range between –25 and –60 dB re 1 $Vpa^{-1}$.

*Piezoelectric microphone*

In the case of a piezoelectric microphone, sound incident upon the diaphragm will stress a small piezoelectric crystal which in turn induces a bound charge across its capacitance. The variable charge is converted into an output voltage. The theory of the piezoelectric microphone (Chap. 14 in Hansen, 1997) shows that the output voltage is a linear function of the incident acoustic pressure. The piezoelectric microphone is less sensitive to dust and moisture than the electret, but can be damaged by high temperature environment, and is more sensitive to mechanical vibration. The sensitivity of piezoelectric microphones is usually less than the electret. However, the self-noise floor of piezoelectric microphones is usually less than the electret. Self-noise floor is an important parameter, which dictates the maximum attenuation achievable by an active control system. Consequently, piezoelectric microphones can be less expensive than electrets microphones for a given self-noise floor and are often more adequate for use in active noise control systems.

### 3.8.1.2 Directional Microphones

In some active noise control situations, it is desirable that the microphones be sensitive only to sound propagating in a given direction. This is especially true for a *reference* microphone used to monitor the sound emanating from a disturbance source in a duct. In this case, the reference microphone should be sensitive only to the disturbance source, and not to the control sources placed downstream from this reference microphone. However, in practice, there are alternatives to the use of directional reference microphones, such as the use of directional control sources, or the use of IIR control filters which inherently account for the feedback effect of the control sources on the reference microphone.

A relatively standard technique to realize a directional microphone is shown in Figure.3.9. It consists of a long, narrow tube with a small axial slit and with a standard electret or piezoelectric microphone mounted at one end. This tube microphone will have a higher sensitivity to sound propagating in the direction of the tube, towards the microphone (the device should thus point towards the sound source). The principle of operation lies in the relative phase velocity of the external sound wave, and the internal sound wave transmitted through the axial slit and propagating in the tube. These two waves arrive in phase at the microphone when the tube is oriented towards the source, but there is a phase shift when the tube is oriented at some non-zero angle with the direction of the external sound wave. The device thus has a maximum sensitivity to the sound propagating in the direction of the tube.

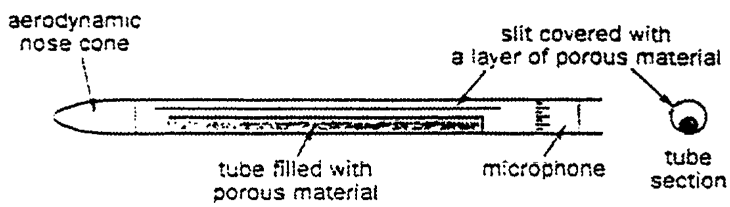



Figure 3.9: Directional Microphone With Internal Absorption Tube (Hansen, 1997)

Further discussions on the construction details, proper selection of porous materials in the tube, and sensitivity with respect to frequency and tube length can be found in (Davy, 1993; Munjal, 1989; Chapter 14 in Hansen, 1997). An alternative to the tube microphone for achieving directional sensing is the use of one-dimensional microphone arrays, with post-processing techniques similar to those used in directional acoustic antennas. Microphone arrays are however more expensive than tube microphones, and they have a very sharp sensitivity function in the frequency domain. The use of tube microphones is therefore generally recommended, as this device also filters turbulence noise, as discussed further below.

### 3.8.1.3 Turbulence Filtering Sensors (Probe Microphones)

Whenever turbulent flow is present in the acoustic medium, (e.g. a turbulent flow in a duct conveying a gas or a fluid) turbulent random pressure fluctuations are generated in the flow, which add to the disturbance pressure field, such as the harmonic sound field due to rotating machinery. The turbulent pressure fluctuations contribute very little to the noise radiated from the end of the duct. Thus, in an active control system, it is essential that their contribution be removed from both the *reference* and *error* sensors located in the flow. If this is not done, the performance of the active control system in reducing the disturbance noise will be rapidly impaired.

The most common way of reducing the influence of turbulent noise is to use a probe tube microphone, very similar to the directional tube microphone discussed previously. A typical probe tube is depicted in Figure 3.10. It consists of a long, narrow tube with a standard microphone mounted at the end. The walls of the tube are either porous, or contain holes or an axial slit. The probe tube microphone must be oriented with the microphone facing the flow. The principle of operation is based on the fact that both external turbulent pressure fluctuations and external sound waves will be transmitted inside the tube through the slit or openings, in the form of an internal sound wave. The internal sound wave component due to the external sound wave will travel at the same speed as the external sound wave, resulting in reinforcement at the microphone, whereas the internal sound wave due to the external turbulent pressure flucuations will cancel out because of their random phase distribution. Probe tube microphones are convenient as reference sensors in active control systems in ducts since they act as both directional sensors and turbulence filtering sensors. Details on the principle of operation can be found in (Shepherd, 1989).

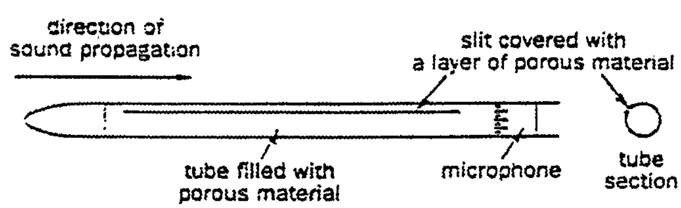



Figure 3.10: Probe Tube Microphone in a Duct (Hansen, 1997)

The sensitivity of the probe tube microphone depends on a number of parameters – slit flow resistance, flow speed, probe tube orientation, probe tube diameter, reflective duct termination – which have been investigated by (Neise, 1975; Munjal, 1989) and summarized in (Chapter 14 in Hansen, 1997). Alternatives to the probe tube microphone involve one-dimensional microphone arrays with an appropriate post-processing in order to extract the acoustic wave component travelling at the sound speed and statistically average out the turbulent pressure fluctuations; this technique turns out to be costly and not necessarily more effective than the probe tube.

### 3.8.2 Operating Considerations for Microphone Sensors

For acoustical ANC (AANC), the sound pressure measurement at the error sensor is obviously of an important concern, since the efficiency of all the control processes will depend on the quality of the measurement at the controlling point and of its representativeness of the sound field to be reduced. It is useful to distinguish two aspects of this problem: the location of the error sensor in the sound field and the hardware that can be used.

*Error microphone location*

- *ANC in 3-D space:* When the control algorithm minimizes the pressure at a given error microphone, it creates a '*zone of quiet*' around this error microphone (for example, a 1*0 dB zone of quiet* corresponds to the zone where the pressure is reduced by at least 10 dB around the error microphone) (A. David, S.J. Elliott, 1994). The location and arrangement of the error sensors represent an important problem. Various studies have been done and are still under progress in order to determine the optimum location of the error sensors (S.J. Elliott, P. Joseph, A.J. Bullmore and P.A. Nelson , 1988).

- *ANC in duct:* For ANC in duct, multi-channel acoustical ANC systems are necessary and *M* error sensors have to be used to control *M* modes, for high order propagation cases. The error sensors should not be located at the nodal lines (observability condition) (S. Douglas and J. Olkin, 1993). For a rectangular duct, the location of the error sensors is relatively simple because the nodal lines are fixed along the duct axis. However, in circular ducts, the location of the nodal lines changes along the duct axis since the modes usually spin as a function of the frequency, of the temperature and of flow speed (C.L. Morfey, 1964), (A. Bihhadi and Y. Gervais, 1994). Those variations of the nodal lines may explain why ANC of high order modes in circular or irregular ducts appears to be difficult (S. Laugesen, 1996). Instead of using the modal approach (that is, the shape of the modes to be controlled) to determine the error sensors location, an alternative strategy has recently been proposed by A. L'Espérance, 1999 - the *error sensor plane concept.* The objective of this concept is to create a *quiet cross-section* in the duct so that, based on the Huygen's principle, the noise from the primary source cannot propagate over this cross-section. Using this strategy, multi-channel ANC in a circular duct is possible (A. L'Espérance, 1997).

*Error microphone hardware*

Depending on the application, the microphone hardware can be a significant problem, mainly due to the environmental conditions. Flow speed as well as corrosive and hot gas are current problems that have to be considered. However, it is generally possible to use a relevant microphone installation to handle those problems.

The following issues should be noted in the choice of the microphone sensor probes:

(i) The absolute value of the sound field pressure at the error sensors is not required, a relative level is sufficient;
(ii) When considering a tonal ANC application, a flat spectral response of the microphone is not necessary;
(iii) It is not necessary to have a direct measurement of the sound field to reduce, but a representative signal of this sound field; and
(iv) In the case of multi-channel active noise control, the response of each microphone (or error sensor probes) has to be similar (or the difference should be known to include a compensation process of the input in the control algorithm).

- Tonal AANC: For a tonal AANC application, the coefficient of the control filter will be used by the controller to minimize the spectral component that has the most important energy. Then, unless the level of the pure tone to reduce is more important than the level of other spectral component, the control of the pure tone(s) will be efficient even if the response of the microphone is not flat. Moreover, when the energy of other parts of the spectrum becomes important compared to the energy of the frequency to reduce (for example, low frequency noise due to airflow), the error signal can be filtered with a high pass or a pass band filter in order to fix the controller on the specific frequency to control.

- Wide band: For a wide band controller, it is necessary to use a microphone and an acquisition process that have a flat response in order to allow the controller to reduce the noise at the spectral component of the signal where the sound field is the most important. Note, however, that even in this case, if the response of the microphones and acquisition process is not flat, a compensation process can be implemented in the controller.

- Multi-channel Control: If the relative sensitivity of an error microphone is not important for a single AANC system, in the case of a multi-channel AANC, the sensitivity of each error sensor has to be similar (or compensated in the control algorithm) to ensure a good performance of the control system. Because the response of the sensor probe may change with the temperature, static pressure or other factors, special procedures may be used (in the control algorithm process) in order to calibrate the probes at their operating locations (A. L'Espérance, 1998).

*Mechanical set-up for a reliable installation*

From the above discussion, the response of the microphones itself is generally not an important concern. Economic standard electric microphones can thus be used. The mechanical set-up that should be used in order to have a reliable installation is, however, of an important concern. Flow speed as well as corrosive and hot gas are current problems that have to be faced (S.D. Goodma, 1990).There are only a few details in the literature on how to install and/or protect microphones for an AANC in problematic environmental conditions. The following gives different set-ups and systems found to meet the requirements of different environmental conditions.

- Bruel & Kjear probe Microphone Type 4182. For measurement in stack with high temperature, corrosive gas etc. Useful when a flat response is needed.
- Microphone probe for a hot corrosive environment (from Soft dB Inc).This microphone probe allows the taking of measurements in a hot corrosive environment. This stainless steel probe can be fixed to the wall of the duct where the sound has to be reduced using the fixation flange. This probe has two parts fixed together with Teflon insulation material. The microphone itself is located at the end of the outside part (see Figure 3.11). The inner part, open at its extremity, allows the sound wave to propagate in the tube and then be measured by the microphone. The signal measured is proportional to the sound at the open extremity of the probe. This type of probe, which is inexpensive and reliable for industrial environments, is adapted for tonal control, but not appropriate for wide band control since the frequency response is not flat.

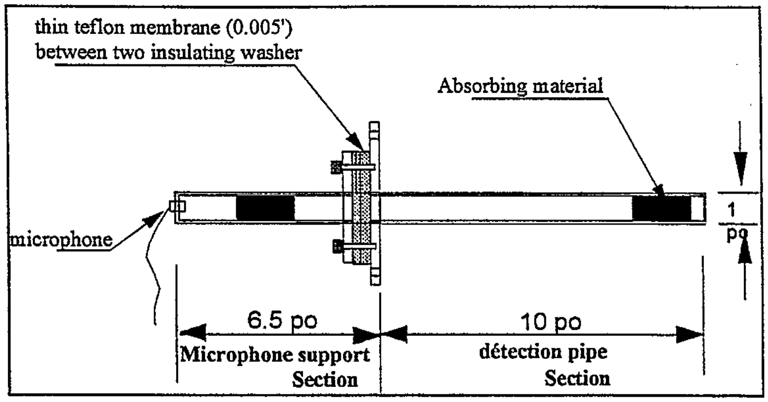



Figure 3.11: Sound Pressure and Particle Velocity Sensing

A number of alternatives to sensing and minimizing squared sound pressure have been suggested for active noise control applications. These include the active control of duct noise (Wang, 1997; Zander, 1993), and more importantly the active control of enclosed sound fields, for which sensing strategies based on acoustic potential energy density minimization, and total energy density minimization have been suggested (Park, 1997; Sommerfeldt, 1995; Cazzolato, 1998). Such quantities can be measured by combinations of (2 to 6) microphones. In this case, finite differences between individual microphones are applied to obtain approximate measurements of the pressure gradient in several directions. Precise measurements of the pressure gradient requires an excellent phase matching of the individual microphones, which can result in more expensive microphones. Associated adaptation algorithms for the minimization of energy-based quantities have been derived (Sommerfeldt, 1994).

## 3.9 Displacement and Velocity Transducers

Although the dynamic range of displacement and velocity transducers is usually much less than that for accelerometers, displacement and velocity transducers are often more practical for very low frequencies (0-10Hz), where vibration amplitudes can be of the order of a millimetre or more for heavy structures, where corresponding accelerations are small. Also, in some active control systems, displacement or velocity rather than acceleration can be the preferred quantities to minimize, and attempts to derive these from acceleration measurements can lead to errenous noise measurements at low frequency. The displacement and velocity transducers are described below.

### 3.9.1 Displacement Transducers

There are three types of displacement transducers, namely proximity probes, linear variable differential transformers (LVDT) and linear variable inductance transformers (LVIT). These are described below:

*Proximity probes:*

Proximity probes are the most common type of displacement transducers. There are two main types of proximity probes, the capacitance probe and the eddy current probe. The capacitance probe measures the change of the capacitance between the vibrating surface and the stationary probe, in response to vibration displacements of the surface. The eddy current probe measures amplitude changes in a high frequency (>500kHz) voltage applied to a stationary coil or fine wire; these changes are due to eddy currents which form in the magnetic field of the probe. These eddy currents are in turn proportional to the size of the gap between the vibrating surface and the probe. The main advantage of proximity probes is that they allow non-contact measurement of vibration displacements. They are therefore well suited to vibration displacement measurements on rotating structures, such as rotating shafts. Usually, eddy current probes are more common and easier to use than capacitance probes. The dynamic range of proximity probes, however, is very small – typically 100:1. The resolution varies from 0.02mm to 0.4mm. The limited dynamic range restricts its practical application to frequencies below 200Hz.

*Linear variable differential transformer (LVDT):*

This type of transducer consists of a single primary and two secondary coils wound around a cylindrical bobbin (see Figure 3.12). A moveable nickel iron core is positioned inside the windings and it is the movement of this core which is measured. To operate the transducer, it is necessary to drive the primary coil with a sine wave, at a frequency between 1 and 15kHz; the amplitude and phase of the output from the secondary coil are linearly related to the position of the core. The dynamic range of an LVDT is typically 100 :1, with a resolution ranging from 0.01mm to 1mm. The frequency range is typically d.c to 100Hz. The total length of the sensor varies from 30-50mm for short stroke transducers to about 300mm for long stroke transducers.

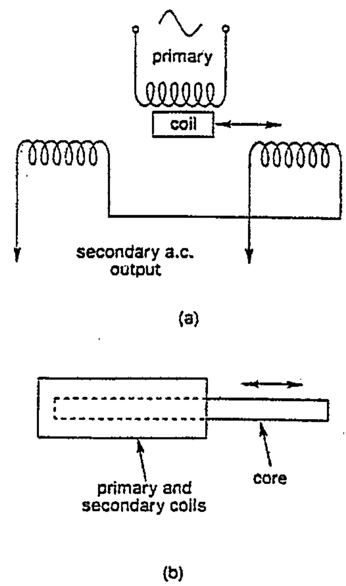



Figure 3.12: (a) Schematic of an LVDT transducer; (b) LVDT transducer (Hansen, 1997)

*Linear variable inductance transformer (LVIT)*

This type of transducer is based on the measurement of inductance changes in a cylindrical coil. The coil is excited at about 100kHz and the inductance change is caused by the introduction of a highly conductive, non-ferrous coaxial rod sliding along the coil axis. It is the movement of this coaxial rod which is measured. The inductance change is measured most effectively using a resonant LC oscillator. This type of transducer is particularly suited for measuring relative displacements in vehicle suspension systems. Transducer sizes vary from diameters of a few millimeters to tens of millimeters, and lengths depend on the displacement amplitudes to be measured.

### 3.9.2 Velocity Transducers

One type of velocity transducer is the non-contacting magnetic type consisting of a cylindrical permanent magnet on which is wound an insulated coil. A voltage is produced by the varying reluctance between the transducer and the vibrating surface. This type of transducer is generally unsuitable for absolute measurements, but is very useful for relative velocity measurement such as needed for active suspension systems. Another device suitable for active suspensions consists of a magnet fixed to the receiving structure (the structure to be isolated) and a coil fixed to the suspension. Relative motion between the two produces a voltage proportional to the relative velocity. This type of device derives from the most common type of absolute velocity transducer, which consists of a moving coil that surrounds a permanent magnet. Inductive electromagnetic field proportional to the relative velocity of the coil with respect to the magnet is set up when the transducer is vibrated. This transducer is designed to have a very low natural frequency, such that the magnet remains virtually fixed, and the absolute velocity of the coil is then measured. The frequency range of operation is 10Hz – 1kHz; the low resonance frequency of the transducer makes it relatively heavy. Velocity transducers cover a dynamic range between 1 and 100$mms^{-1}$. Low impedance, inexpensive voltage amplifiers are suitable. Temperatures during operation or storage should not exceed 120°C.

## 3.10 Accelerometers

Accelerometers are the most employed technology for vibration measurements; they provide a direct measurement of the acceleration - usually in the transverse direction - of a vibrating object. The acceleration is a quantity well correlated to the sound field radiated by the vibrating object. Therefore, accelerometers can be a convenient alternative to microphones as error sensors in active structural acoustic control systems. Accelerometers usually have a much larger dynamic range than displacement or velocity sensors. A potential drawback of accelerometers, in low-frequency active noise control systems, is their low sensitivity in low frequency (typically 0-10Hz).

### 3.10.1 Types of Accelerometers

There are two main, commercially available types of accelerometers: piezoelectric and piezoresistive accelerometers. These are discussed below.

#### 3.10.1.1 Piezoelectric Accelerometers

A piezoelectric accelerometer consists of a small seismic mass attached to a piezoelectric crystal. When the accelerometer is attached to a vibrating body, the inertia force due to the acceleration of the mass produces a mechanical stress in the piezoelectric crystal which is converted into an electric charge on the electrodes of the crystal. Provided the piezoelectric crystal works in its linear regime, the electric charge is proportional to the acceleration of the seismic mass. The mass may be mounted to produce either compressive/ tensile stress or alternatively, shear stress in the crystal. The latter arrangement allows a smaller and lighter accelerometer for the same sensitivity. Various piezoelectric accelerometer types are illustrated in Figure 3.13. A piezoelectric accelerometer should be used below the resonance of the seismic mass – piezoelectric crystal system. Practical details on the operation of piezoelectric accelerometers can be found on B&K's Web site at the address http : // www.bk.dk / pdfs / proddata / english / bp019620.pdf, as well as in (Chap. 15 in Hansen, 1997)

Since piezoelectric accelerometers essentially behave as electric charge generators, they must generally be used with relatively expensive high impedance charge amplifiers. The cost of such amplifiers can represent a significant amount of the total cost of an active control system when a large number of accelerometers are used.

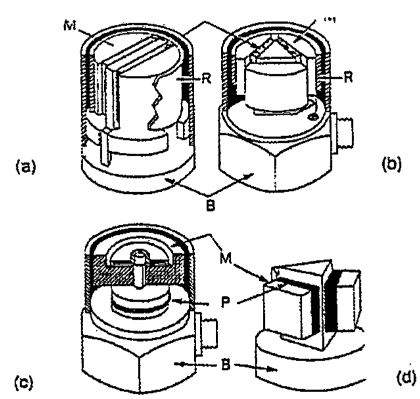



Figure 3.13: Piezoelectric Accelerometers Configurations (Hansen, 1997):
(a) planar shear type; (b) delta shear type; (c) compression type;
(d) schematic of delta shear type. (Bruel and Kjaer.)
M = seismic mass, P = piezoelectric element, R = clamping ring, B = base.

### 3.10.1.2 Piezoresistive accelerometers

Piezoresistive accelerometers rely on the measurement of resistance change in a piezoresistive element subjected to stress. The piezoresistive element is generally mounted on a beam as shown in Figure 3.14. Piezoresistive accelerometers are less sensitive than piezoelectric accelerometers by an order of magnitude for the same size and frequency response. They also require a stable, external d.c. power supply to excite the piezoresistive elements. However, piezoresistive accelerometers have a better sensitivity at low frequency, and require less expensive, low impedance voltage amplifiers.

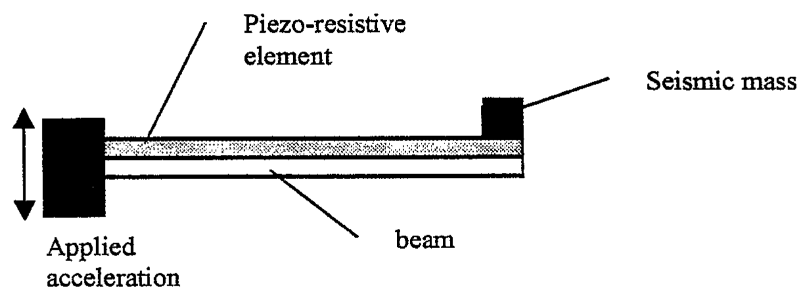



Figure 3.14: Beam-type Piezoresistive Accelerometer

There is another type of piezoelectric accelerometer with an operation principle similar to the piezoresistive accelerometer described above; in this case, the piezoresistive element is replaced by a piezoelectric polymer film (PVDF), and the electric charge across the electrodes of the PVDF is collected as the sensor output. Such a PVDF accelerometer has a sensitivity and frequency response similar to the piezoresistive accelerometer, and is less expensive than the piezoelectric accelerometer. PVDF film accelerometers seem to be a promising technology due to their high performance to cost ratio, and they have been used to make *rotational* acceleration transducers which are now commercially available.

### 3.10.2 Considerations in Use of Accelerometers

When selecting an accelerometer for a particular application, a compromise has to be made between weight, sensitivity and frequency response. Small accelerometers can measure higher frequencies, and are less likely to affect the dynamics of the structure by mass loading it; however, they have a lower sensitivity, which puts a limit on the smallest acceleration they can measure. Accelerometers range in weight from miniature 0.65g for high level vibration amplitudes up to

18kHz on light weight structures, to 500g for low level vibration amplitudes on heavy structures up to 1kHz. Depending on their size and sensitivity, accelerometers can measure acceleration amplitudes as low as $10^{-4}ms^{-2}$, and as high as $10^{6}ms^{-2}$. Because of the three-dimensional sensitivity of piezoelectric crystals, piezoelectric accelerometers are sensitive to vibrations at right angle to their main axis. This is called the transverse sensitivity, which should be less than 5% of the axial sensitivity.

Particular care should be taken when mounting accelerometers on the structure. There are numerous methods of attaching an accelerometer to a vibrating surface. These are described for example on B&K's Web site at the address http : // www.bk.dk / pdfs/ proddata / english / bp019620.pdf, and the associated upper frequencies for which measurements are valid are also discussed at this address. Other considerations on the operating environment involve temperature effects – reversible sensitivity changes of up to 12% occur at 200°C -, sensitivity to electric field, and handling.

Recent developments in micromachining and solid-state sensors have made it possible to design miniaturized accelerometer arrays and integrate them in mechanical structures (Subramanian, 1997; Corsaro, 1997). Such transducers are particularly needed in active sound and vibration control and smart-materials applications. Commercial areas of interest include advanced underwater, aerospace or robotic-sensing applications.

## 3.11 Loudspeakers

### 3.11.1 Description

The electrodynamic loudspeaker is by far the most commonly employed actuator technology for active noise control applications. When selecting a loudspeaker for an active noise control system, the important parameter is the cone volume velocity required to cancel the primary sound field. For an electrodynamic loudspeaker, the electrical power $W$ consumed by the voice coil is related to the cone volume velocity $v$ by (Chap. 14 in Hansen, 1996)

$$W = \frac{v^2}{2S^2}\left[\frac{|Z_m|^2 R}{(BL)^2} + \Re\{Z_m\}\right],$$

where $S$ is the cone area, $Z_m$ is the total mechanical impedance of the loudspeaker, $R$ is the resistance of the voice coil, $B$ is the magnetic flux density, $L$ is the conductor length and $\Re$ designates the real part. This equation shows interesting features which are relevant for active noise control applications.

- The required electrical power to the speaker is proportional to the square of the cone volume velocity; and
- When the speaker is backed by a small enclosure, and in low frequency, the mechanical impedance of the loudspeaker $Z_m$ is usually dominated by the stiffness of the backing enclosure; the required electrical power will then be approximately proportional to the square of this enclosure stiffness (which in turn is inversely proportional to the volume of the enclosure). Such a backing enclosure is usually necessary to enhance the response of the loudspeaker in low frequency, and can be used to enhance the efficiency of the loudspeaker in a limited frequency range, as discussed further on.

### 3.11.2 Electrical Power and Cone Excursion Requirements

Electrodynamic loudspeakers exhibit a non-linear behaviour when they are driven close to maximum power or maximum membrane deflection. This non-linear behaviour can significantly degrade the performance of active control systems based on linear filtering techniques. It is thus important that loudspeakers in active control should be driven at a fraction of the maximum power or maximum deflection specifications, especially in situations where single-frequency or harmonic noise has to be attenuated. For random noise, the peak cone velocity requirements for active control are likely to be four or five times the estimated r.m.s. velocity requirements (Shepherd, 1986).

The above considerations are also important in terms of the speaker life. Provided low power and low cone excursion requirements are satisfied, the life of loudspeakers used in clean industrial environments has been found to be approximately three years (chap. 14 in Hansen, 1997).

In some active control applications using loudspeakers, one may be faced with excessive electrical power requirements to achieve adequate noise attenuation; a few potential solutions to this problem are examined in what follows:

- In some cases, it will be found that multiple loudspeakers (instead of one) can be used to obtain the required volume velocity; if the loudpeakers are small compared to the acoustic wavelength, then two closely spaced loudspeakers driven with a voltage $V$ provide the same volume velocity as a single loudspeaker driven with $2V$. For example, in a duct system it is thus preferable to use a number of small loudspeakers placed in the duct walls at the same axial location, rather than a single large loudspeaker.
- The equation relating the electrical power requirement to the cone volume velocity shows that the required electrical power is minimized at the frequency corresponding to the mechanical resonance of the loudspeaker ($Z_m = 0$) (Shepherd, 1986; Ford, 1984). Thus, in active control of single-frequency noise, it is desirable to design the loudspeaker so that its mechanical resonance lies at – or close to – the frequency of interest. This resonance frequency can be adjusted to suit a particular application either by adding mass to the cone (to reduce the frequency), or by adding a backing enclosure to the speaker (to increase the frequency).

- Finally, there are a number of alternatives to electrodynamic loudspeakers in active control. A first alternative involves replacing the loudspeaker coil and magnet by a low-inertia, high-speed d.c. servo motor similar to those used in computer tape drives. Because of its low inertia and ability to rotate in either direction, the motor is capable to responding to audio signals up to about 130 Hz. Peak to peak deflections achievable are larger than with conventional electrodynamic loudspeakers (Leventhall, 1988). Speakers based on this technology, and capable of producing 130 dB at 1 m, between 30 Hz and 130 Hz, are now available (ServoDrive, see section 3.14). A second alternative involves an electrodynamic inertial actuator (similar to a coil and magnet) which is directly attached to a vibrating structure; in this case, the loudpeaker cone is replaced by the vibrating structure itself, which can be a more efficient acoustic source because of its large dimensions. Such actuators are also commercially available (Stanford Acoustics, see section 3.14). A third alternative involves a carefully designed vibrating plate driven by a piezoceramic actuator; the radiator in this case is the vibrating plate, the geometry and material characteristics of which can be tailored for a maximum radiation at a frequency – or in a frequency range – of interest. Such a device has been used to replace loudspeakers in a prototype active control system for the attenuation of humming noise generated by large electrical power transformers (McLoughlin, 1994).

### 3.11.3 Operating Environment

*Domestic or commercial application (low and medium power speaker)*

Depending on the Acoustic Active Noise Control(AANC) applications, the choice of the loud speaker may or may not be easy. For small systems, (small duct, low noise, domestic ventilation system), AANC can be achieved with small commercial medium quality speakers (radio type speaker) (A. Boudreau et al. 1998). To reduce the noise in the ventilating duct of a building where the noise levels are generally below 90-100 dBlin, standard loudspeakers can be used.

*Industrial environment (high power speaker)*

Use of conventional loudspeakers as active control sources in industrial environments requires considerable precautions, even supposing that the speakers will operate to only a fraction of the maximum deflection capability. Operation in high humidity, high temperature and corrosive environments requires special care: water-resistant adhesives must be used for the cone-coil connections in high humidity and/or high temperature applications; the paper cone must be sprayed with a protective coating or replaced by some inert plastic depending on the nature of the environment. In cases of extreme heat, the loudspeaker cone must be protected with a heat shield (L'Esperance, 1997) which in some cases may require the addition of cooling air flowing past the shield. Alternatively, the loudspeaker can be removed from extreme conditions by placing it at the end of a short tube. An example of loudspeaker implementation in an industrial active control system is shown in Figure.3.15.

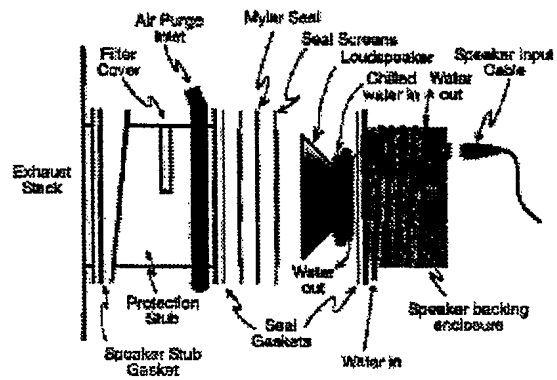



Figure 3.15: Industrial Loudspeaker Enclosure Design for a Hot, Dirty Environment (Hansen, 1998)

In industrial applications, the actuator often appears to be the critical part and limiting factor of an AANC application. In fact, when noise problems occur, the noise levels are generally high. Also, when ANC is considered, it is often for low frequency problems, that is lower than 300 Hz, classical passive systems usually being efficient for medium and high frequencies. For those situations, low frequency and high powered actuators are required for AANC. It is a common situation, for example, where the needed acoustical power is *Lw =110 dB(A)* at 100 Hz, which corresponds to *Lw=130 dBlin,* a sound power that no standard actuator can easily generate. For instance, in classical technology, one of the most powerful models is the *JBL 2023H professional series,* with 250 Watt, 101 dBlin/1w/1m, which allows a total sound power of about 125 dBlin.

Finally, one may note that at low frequency, the displacement of the speakers' membrane can become important (about 1/2'' to generate 125 dBlin at 100 Hz with a 10'' speaker). For lower frequency, the stroke is increased. These high displacements create significant mechanical fatigue on the speaker suspension and will affect the durability of the speakers.

Several companies that produce and develop loud speakers exist. Examples of some of the important companies are JBL (Harman International), BOSE, AUDAX, Electro-Voice. Some companies are specialized in low-frequency transducers (SunFire, http://www.sunfire.com/, STechnologies, www.supremetechnologies.com/). For example, the SunFire Subwoofer can generate about 100 dBlin between 18-100Hz with 2,5'' stroke using 2400W. Very few loudspeakers are however designed especially for ANC, that is for intensive and extreme environmental conditions.

*High temperature loudspeakers*

To our knowledge, Harman Applied Technologies (JBL) and Motran (Motran Industries Inc. is a designer, developer and manufacturer of inertial actuators, linear actuators, and acoustic transducers) are two of the few companies that have developed speakers for high temperature operation (see annex and http://www.harman.com/), but those speakers are limited in output power and durability. Special mechanical set-ups can also be used to protect the speakers from the medium where the control has to take place. Speaker protection systems (Speaker System, US patent 4949386) or special set-ups minimizing the contact of the hot/corrosive gas with the speaker membrane are the general methods reported in the literature (C.H.Hansen,1996) As an example, the ANC silencer proposed by Noise Control Technologies (NCT) uses speakers fixed near the exit of the exhaust, so that the speaker is not directly in contact with the exhaust gas.

Soft dB used a Teflon membrane and a perforated metal sheet to protect the membrane of the speaker from corrosive gas. Those protection systems however reduce the efficiency of the speakers, since they introduce more resistance for the sound wave generated by the speakers (see Figure 3.16).

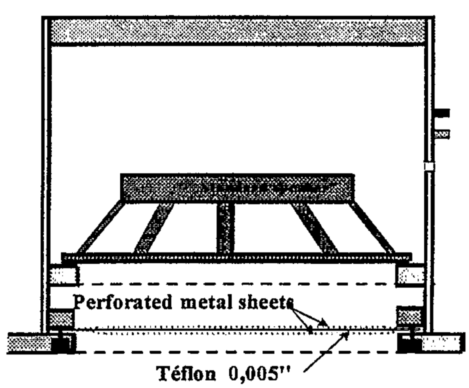



Figure 3.16: Protective System for Loudspeaker Membrane

## 3.12 Electromagnetic Actuators

For vibration control purposes, electromagnetic actuators can be classified into electrodynamic shakers and electrical motors. The latter can be used for low frequency vibration control, provided that a linear configuration is used or that appropriate mechanical coupling is employed. Both of these electromagnetic actuators will be briefly reviewed in this section.

### 3.12.1 Electrodynamic Shakers and Proof-Mass Actuators

Electrodynamic shakers are generally defined as devices having a central inertial core surrounded by a winding. The core is generally a permanent magnet coupled with an inertial mass. This type of inertial actuator applies a point force to a structure by reacting against the inertial mass. As in a loudspeaker, a time-varying voltage is applied to the coil in order to move the inertial mass and to force the movement of the structure onto which the shaker is attached. Compact systems are available (Stanford Acoustics) (see Figure 3.17).

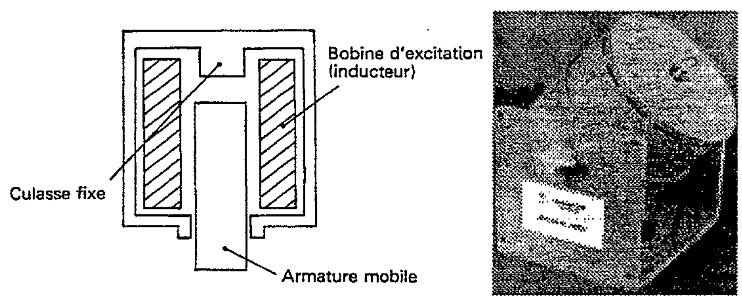



Figure 3.17: Physical Principle of the Electrodynamic Shaker

Other inertial type actuators are available which use, for example, the piezoelectric effect, instead of a coil, to move the inertial mass (see Figure 3.18, PCB Piezotronics). There are several reasons for considering piezoelectric inertial actuators in active control applications:

- Inherent passive vibration absorber effect which greatly reduce the power requirements of an active system;
- Permit attachment to curved surfaces; and
- Actuator resonant frequency can be tuned to a desired frequency.

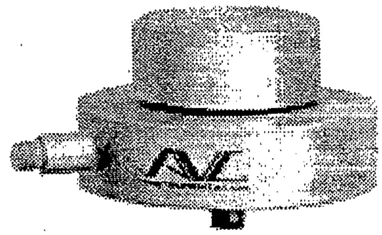


Figure 3.18: The Piezoelectric Inertial Actuator (PCB Piezotronics)

*Proof-mass actuators*

Proof-mass actuators (also called inertial actuators) are very similar in their operation to electrodynamic shakers. They usually consist of a mass, which is moved by an alternating electromagnetic field. The motion of the proof mass results in a reaction force on the electromagnetic stator which is attached to the structure to be controlled. This device can generate relatively large forces and displacements and can be an interesting alternative to costly electrodynamic shakers. Very inexpensive proof mass actuators are commercially available from several companies (eg. Aura, Stanford Acoustics) at a price varying between 40 and 100$, including power supply, with excellent linearity characteristics and a large bandwidth. Such devices have been tested by the GAUS, and by the active control group at the University of Adelaide, as potential control actuators on very stiff structures such as electrical power transformers, with very promising results. Another advantage of proof mass actuators is that their resonant frequency can be easily tuned for optimal efficiency at a given frequency.

### 3.12.2 Electrical Motors

Most industrial motor applications use AC induction motors. The reasons for this include high robustness, reliability and low price. However, the use of induction motors also has its disadvantages, these lie mostly in its difficult controllability, due to its complex mathematical model, its nonlinear behaviour during saturation effect and the electrical parameter oscillation which depends on the physical influence of the temperature (Texas Instruments, 1999). The same disadvantages apply for most electrical motors traditionally used in the industry. The use of electrical motors in active control of vibrations has long been limited due to the same reasons.

The advent of new control strategies has revolutionized the way electrical motors can be used and now allows for the use of motor technologies which were previously difficult to implement in practical applications. Simple motor drives were traditionally designed with relatively inexpensive analog components which suffer from susceptibility to temperature variations and component aging. The advent of digital structures for the control of electrical motors opens the way for far more complex drives allowing real-time identification, modelization and linearization of motor models. The high performance of digital controllers allows them to perform high resolution control and minimize control loop delays. These efficient controls make it possible to reduce torque ripples and harmonics and to improve dynamic behaviour in all speed ranges. The motor design is optimized due to lower vibrations and lower power losses such as harmonic losses in the rotor. Smooth waveforms allow an optimization of power elements and input filters. Overall, these improvements result in a reduction of system cost and better reliability.

Electrical motors can be divided into motors with a permanent magnet rotor (AC and DC motors) and motors with a coiled rotor. Figure 3.19 illustrates a detailed classification of the electrical motors. With the advent of new controllers, the tendency is to classify electrical motors under AC or DC according to the control strategy.

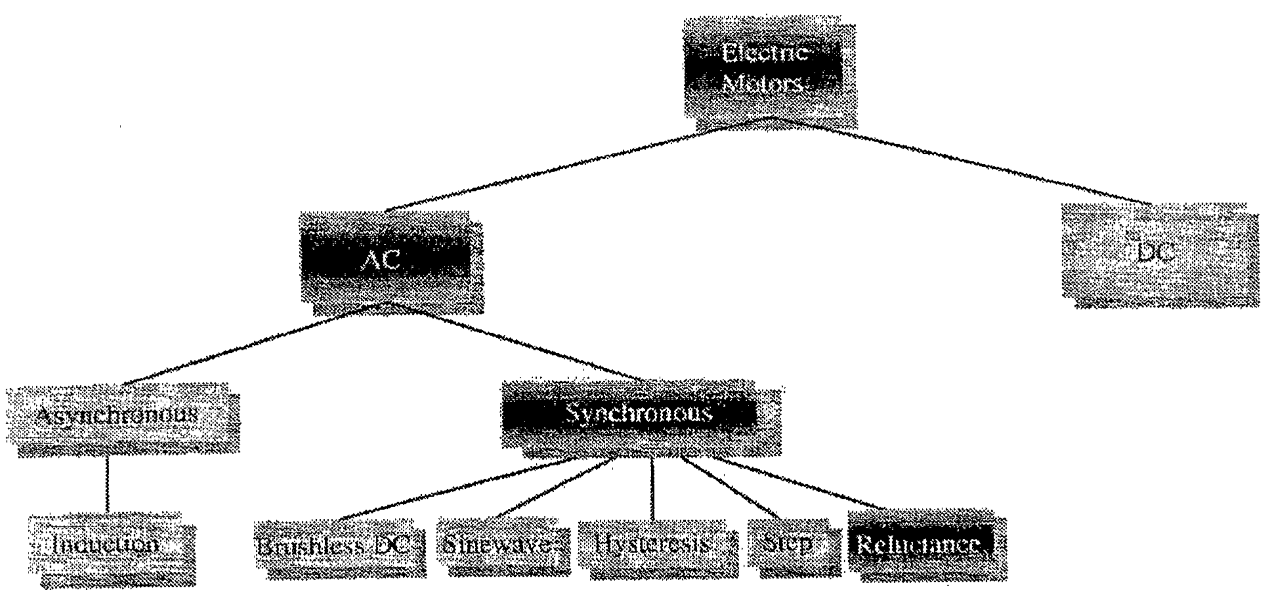



Figure 3.19: Classification of Electric Motors (Texas Instruments, 1999)

*Brushless Motor*

Due to its high reliability and high efficiency in a reduced volume, the *brushless motor* (see Figure 3.20*)* is actually the most interesting motor for application to active vibration control (Lecoufle, 1987; Texas Instruments, 1999). Although the brushless characteristic can be applied to several kinds of motors, the brushless DC motor is conventionally defined as a permanent magnet synchronous motor with a trapezoidal back EMF waveform shape while the brushless AC motor is conventionally defined as a permanent magnet synchronous motor with a sinusoidal back EMF waveform shape. New brushless and coreless motors are now available which are very linear over a wide speed range (Micro Mo Electronics, 1999). By using the appropriate sequence to supply the stator phases, a rotating field on the stator is created and maintained in the brushless motor. The lead between the rotor and the rotating field must be controlled to produce torque. This synchronization implies knowledge of the rotor position. The key to effective torque (no ripples) and speed control of a brushless motor is based on relatively simple torque and back EMF equations, which are similar to those of the DC motor. The brushless motor control consists of generating variable currents in the motor phases. The regulation of the current to a fixed 60° reference can be realized in two modes: *pulse width modulation* (PWM) or *hysteresis* mode. Shaft position sensors (incremental, Hall effect, resolvers) and current sensors are used for the control. Linear permanent magnet motors are also available which, in addition to the linear action, allow better magnetic dissipation in the core as it is distributed in space.

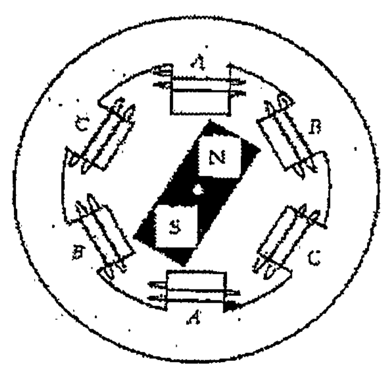



Figure 3.20: The Brushless DC Motor

*Induction Motor*

If volume is not a major concern, a second type of motor to be used in active vibration control is the *induction or AC motor* (Texas Instruments, 1999), see Figure 3.21. As with the brushless motor, the performance of an AC motor is strongly dependent on its control. DSP controllers enable enhanced real time algorithms. There are several ways to control an induction motor in torque, speed or position; they can be categorized in two groups: *scalar* and *vector control.* Scalar control means that variables are controlled only in magnitude and the feedback and command signals are proportional to DC quantities. A scalar control method can only drive the stator frequency using a voltage or a current as a command. The vector control is referring not only to the magnitude but also to the phase of these variables. This method takes into consideration not only successive steady-states but real mathematical equations that describe the motor itself; the control results obtained have a better dynamic for torque variations in a wider speed range. Pulse width modulation techniques are also used for the control of induction motors and indirect current measurement (using a shunt or Hall effect sensor) is used as a feedback information for the controller.

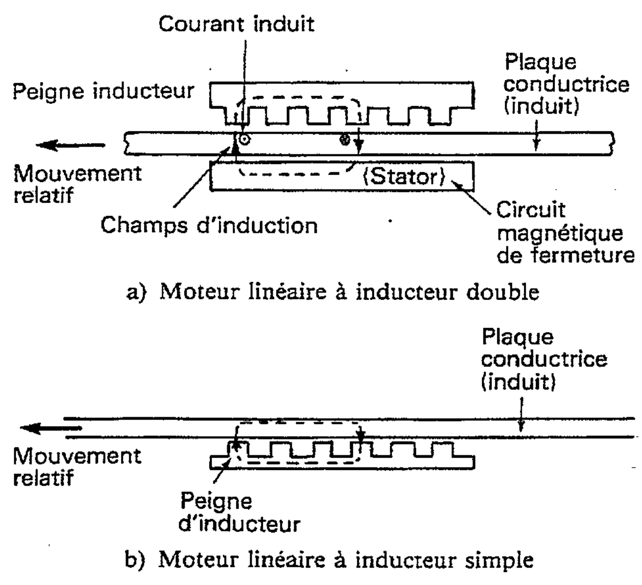



Figure 3.21: Linear Induction Motor

*Switched Reluctance Motor*

The third electrical motor used for active vibration control is the *switched reluctance motor* (Texas Instruments, 1999). This motor is widely used mainly because of its simple mechanical construction and associated low cost and secondarily because of its efficiency, its torque/speed characteristic and its very low requirement for maintenance. This type of motor however requires a more complicated control strategy. The switched reluctance motor is a motor with salient poles on both the stator and the rotor. Only the stator carries windings. One stator phase consists of two series-connected windings on diametrically opposite poles. Torque is produced by the tendency of its moveable part to move to a position where the inductance of the excited winding is maximized. There are several possible ways to control the switched reluctance motor in torque, speed and position. Torque can be controlled by two methods: the *current control method* or the *torque control method*. In the first control method, the magnitude of the current flowing into windings is controlled using a control loop with a current feedback. Both current and position feedback are then needed for controlling the switched reluctance motor. Although the preceding approach has many practical applications, a major disadvantage exists: controlling a constant value of current will result in torque ripple because of the non-linearity of the relationship between torque and current for a switched reluctance motor. A solution to this problem is to profile the current such that torque is the controlled variable. This can only be accomplished using *a priori* information about the motor's torque-current-angle characteristics. The pulse width modulation (PWM) strategy is used in both current and torque control approaches to drive each phase of the switched reluctance motor according to the controller signal. Three PWM strategies exist: the single pulse operation, the chopping voltage strategy and the chopping current strategy. The position information required to generate precise firing commands for the power converter is obtained from incremental sensors or Hall effect sensors. The current is obtained from a current/voltage converter or a current sense resistor connected in series with the phase.

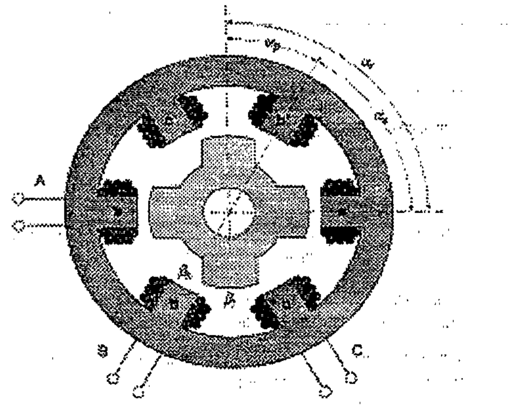



Figure 3.22: Switched Reluctance Motor

## 3.13 Hydraulic and Pneumatic Actuators

Hydraulic and pneumatic actuators are good candidate technologies when low frequency, large force and displacements are required.

*Hydraulic actuators*

Hydraulic actuators consist of a hydraulic cylinder in which a piston is moved by the action of a high-pressure fluid. Alternating motion of the piston is obtained by switching the action of the fluid from one end of the cylinder to the other end. The most common method to alternate the action of the fluid is to use a servo-valve, which consists of a moveable spool connected to a solenoid. The design of the servo-valve generally determines the frequency range of operation of the actuator. The main advantage of hydraulic actuators is their large force and large displacement capability for a relatively small size. The disadvantages include: the need for a hydraulic power supply (which can require space and generate noise), the high cost of servo-valves, the non-linear relation between the servo-valve input voltage and the output force or displacement produced by the actuator; and the limited bandwidth of the actuator (0-150Hz). It is important to locate the servo-valve as closely as possible to the cylinder in order to minimize the efficiency loss in high frequency. Hydraulic actuators have been used in the past mostly in active vehicle suspensions (Crolla, 1988; Stayner, 1988), active control of helicopter cabin vibration (King, 1988, Staple, 1990), active isolation of machinery vibration (Tanaka, 1988), or in the design of active dynamic absorbers for ship structures (Hsueh, 1994; Kakinouchi, 1992).

*Pneumatic actuators*

The principle of operation of pneumatic actuators is very similar to hydraulic actuators, except that the hydraulic fluid is replaced by compressed air. Due to the higher compressibility of air, the bandwidth of pneumatic actuators is reduced (typically 0-10Hz), which restricts the application to non-acoustic problems. Pneumatic actuators may be an attractive option when an existing air supply is already available (such as in rail vehicle active suspensions, see Buzan, 1983; Cho, 1985). In the case of active mounts, the same air supply can be used to drive pneumatic actuators and passive air springs mounted in parallel. Day (1993) used pneumatically-controlled vibration absorbers to reduce the vibrations caused by a six-cylinder diesel generator set having a mass of approximately 1 tonne.

Names and addresses of suppliers of sensors and actuators discussed above are provided in Appendix A.

# 4. APPLICATIONS OF NOISE CONTROL IN SHIP STRUCTURES

## 4.1 Introduction

A typical marine diesel engine mounted on a ship hull is schematically depicted in Figure 4.1. The figure shows the various vibroacoustic paths through which the engine vibration is transmitted to the ship structure, and eventually radiated into sea water. In this figure, the coupling between structural and acoustic energy is classified using the following symbols: AA: acoustic to acoustic coupling, SS: structural to structural coupling, AS: acoustic to structural coupling, SA: structural to acoustic coupling. The relative importance of energy coupling for radiation into sea water is illustrated by a number.

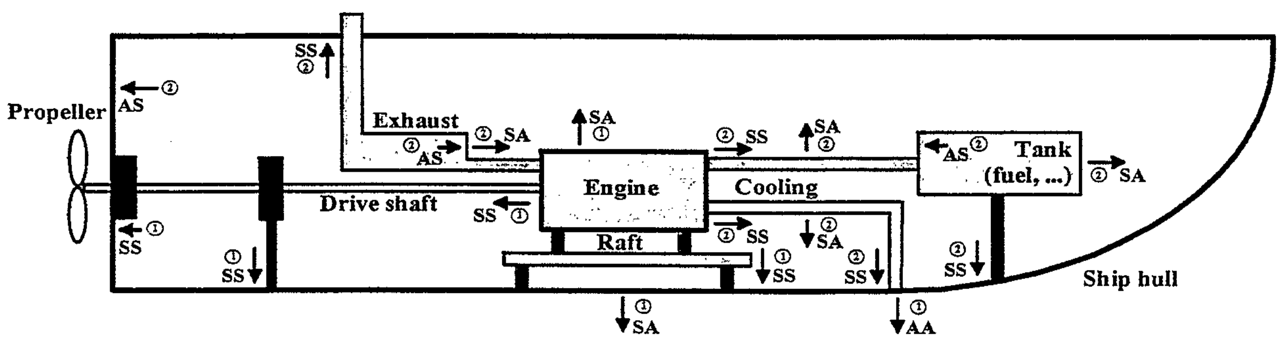



Figure 4.1: Typical marine diesel engine mounted on a ship hull. ( AA: acoustic to acoustic coupling, SS: structural to structural coupling, AS: acoustic to structural coupling, SA: structural to acoustic coupling). The relative importance of energy coupling for radiation into sea water is illustrated by a number, from more important (1) to less important (2).

The following energy transmission paths appear on the figure:

- Mounting system, consisting of the engine cradle, isolation mounts, raft and foundation;
- Exhaust stack;
- Fuel intake and cooling system;
- Drive shaft; and
- Airborne radiation of the engine.

All of these energy transmission paths need to be considered in the implementation of an active control system. In fact, blocking one transmission path may result in an increase of energy transmission through other paths of the system. This is particularly true for periodic disturbances, for which vibration energy transmitted to the ship structure through several paths can destructively interfere; removal of one of the transmission paths can reduce the destructive interference and results in an increase in vibration level.

The proposed methodology addresses a review of active control strategies and actuators/sensors selection for each of the above transmission paths. The above five paths have been grouped in 4 categories, corresponding to generic active control problems:

- **Path 1: Active Vibration Isolation** (mounting system);
- **Path 2: Active Control of Noise in Ducts and Pipes** (exhaust stack; fuel intake and cooling system);
- **Path 3**: **Active Control of Vibration Propagation in Beam-Type Structures** (drive shaft);
- **Path 4**: **Active Control of Enclosed Sound Fields** (airborne radiation of the engine).

These four paths are discussed in Sections 4.2 to 4.5.

## 4.2 Path 1: Active Vibration Isolation (Mounting System)

### 4.2.1 Introduction

Active vibration isolation involves the use of an active system to reduce the transmission of vibration from one body or structure to another. Of interest here will be the transmission of periodic vibration from a ship's engine to the ship hull. Such an active isolation system will be used in practice to complement passive, elastomeric isolation mounts between the engine and supporting structure. An active isolation system is usually much more complex and expensive than its passive counterpart, but has the advantage of offering better low frequency isolation performances, and can be designed for a better static stability of the supported equipment.

In the broad sense, there are two main categories of active vibration isolation systems, corresponding to two distinct situations:

- Isolation of a body, structure, equipment from the vibration of a base structure (such as the isolation of a car body from the road irregularities through an active suspension);
- Isolation of a supporting structure from the vibration of a machinery or vibrating source.

Each of the above categories gives rise to distinct active control approaches. The second category of active systems will be of primary interest here. There is a vast body of literature on the subject of active isolation of a supporting structure from a vibrating source, and the following literature review concentrates on the previous work related to *active vibration isolation for heavy machinery and structures*. Therefore, earlier studies dealing with the first category of systems above, or studies involving active vibration isolation of small dynamic forces will not be covered in detail.

### 4.2.2 Types of Active Isolation Systems

For simplicity and illustration purposes, the case of a one-degree of freedom system will be considered here. Consider a vibrating rigid body modelled as a mass, and connected to a rigid, immovable base through a spring and damper (Figure 4.2(a)). There are several configurations of active systems that can enhance the vibration isolation of the basic passive system shown in Figure 4.2(a). These configurations are listed below and illustrated in Figure 4.2(b) – (g).

- A first class of system involves the control of system damping, and is often referred as a *semi-active* isolation system (Figure 4.2(b). The damping modification is usually achieved by a hydraulic damper with varying orifice sizes. This system is often used for active suspensions in cars. Such a system involves control time constants significantly longer than the disturbance time constants, with the advantage of a simpler and less expensive implementation. However, low frequency performance is much less than for fully active systems described in the following.

- A second class of system involves controlling the vibration of the primary source itself, and is depicted on Figure 4.2(c). Such a system has several disadvantages as compared to an active system acting at the transmission point between the primary source and the receiving structure. The amount of required control force is larger above the natural frequency of the system

(Chapter. 7 in Fuller, 1996), and in practice, globally controlling the vibration of the primary source will necessitate many control actuators (especially if the primary source can no longer be considered as a rigid body, and exhibits modal behaviour).

- Another class of system involves an active control actuator in parallel with the passive system, with the actuator exerting a force on both the base structure or the rigid mass (Figure 4.2(d)). In this parallel configuration, the actuator is not required to withstand the weight of the machine; as compared to the configuration of Figure 4.2(c), the required control force is smaller above the natural frequency of the system (Chapter. 7 in Fuller, 1996). The main disadvantage of this configuration is that at higher frequency (outside the frequency range where the actuator is effective), the actuator itself can become a transmission path. At low frequency, the large displacement/large force requirements for heavy structures preclude the use of piezoelectric and magnetostrictive actuators. Instead, hydraulic, pneumatic or electromagnetic actuators with their associated weight, space and possibly fluid supply problems, must be used. As far as practical application of active control is concerned, the use of an actuator in parallel with a passive isolation stage could have distinct advantages. In a given application, if an actuator can be found that provides a control force of the order of the primary force exciting the machine, then it may be possible to use of much higher mounted natural frequency associated with the passive isolation stage than would be otherwise possible. This in turn has advantages with regards to the stability of the mounted machine.

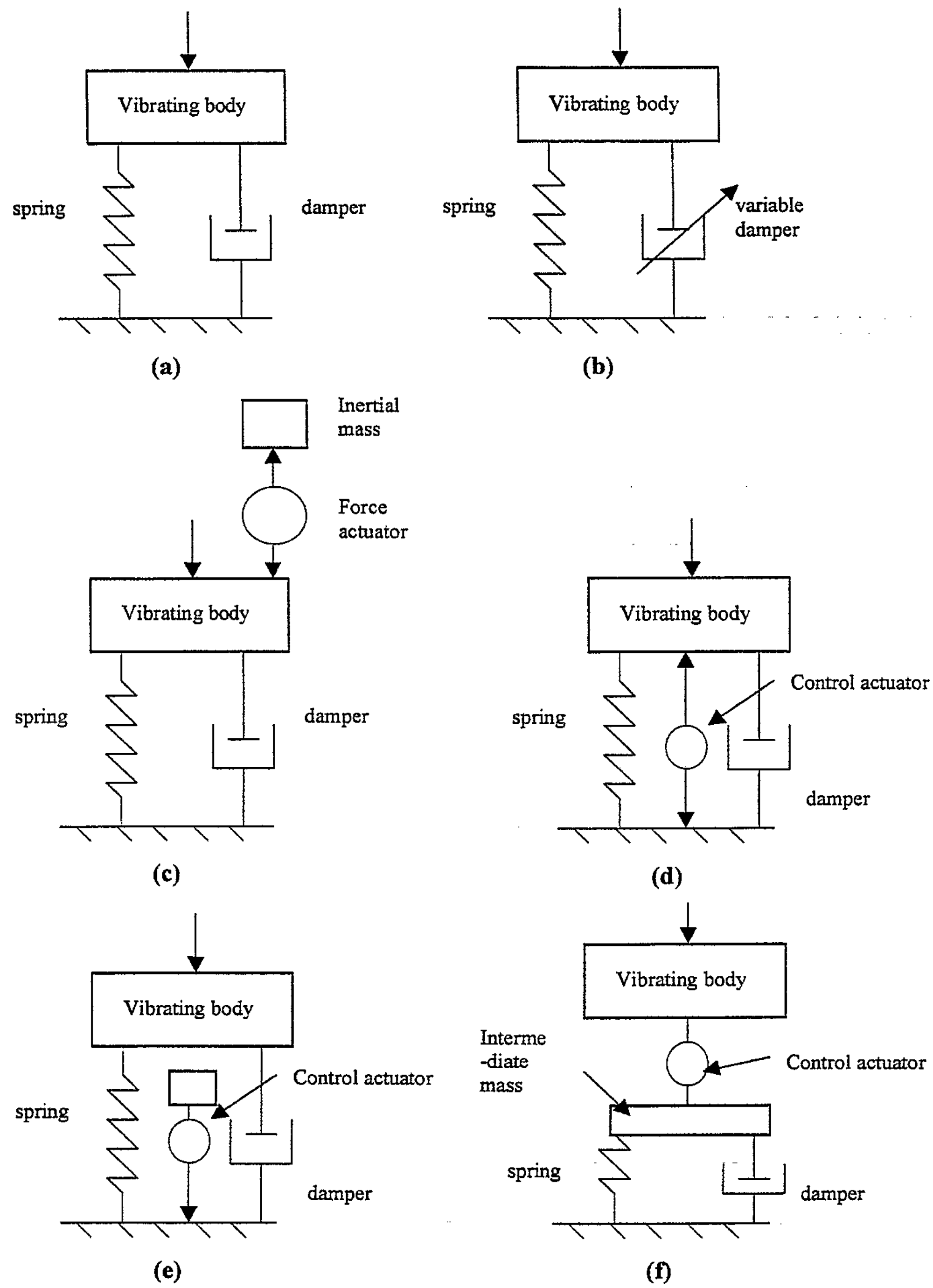
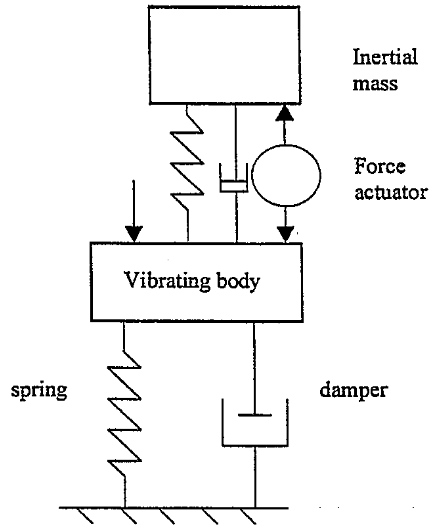



Figure 4.2: Passive and active vibration isolation systems: (a) Passive system; (b) Semi-active system with variable damper; (c) Active system with control force applied to vibrating body; (d) Active system with control force applied to both vibrating body and base structure; (e) Active system with control force applied to base structure; (f) Active system with control force in series with passive mount.




(g)

Figure 4.2(g): Active Dynamic Vibration Absorber

- Figure 4.2(e) shows another configuration of an active control actuator in parallel with the passive system, but with the actuator now acting on the base structure only. A reaction (or inertial) mass is therefore necessary for the actuator in this case, as in the case of Figure 4.2(c) This reaction mass needs to be large for heavy structures and in some cases this may be impractical. As compared to the case of Figure 4.2(d), the required control force will be much larger in the frequency range close to the natural frequency of the system, but will be approximately the same at frequencies larger than the natural frequency of the mount.

- A configuration with the active system in series with the passive mount is shown on Figure 4.2(f). Such a system has several advantages over the parallel configuration. The active system is now isolated from the dynamics of the receiving structure, which simplifies the control in the case of a flexible base structure, and the use of an intermediate mass creates a two-stage isolation system, which offers better isolation performance in higher frequency. However, its main disadvantage is that the control actuator is required to support the full weight of the vibrating body.

- Finally, in cases where it is desired to attenuate the vibration of a machine at a specific frequency, an active vibration absorber can be attached to it as shown in Figure 4.2(g). A *semi-active* version of a dynamic vibration absorber is often encountered, where the control aims at continuously adjusting the natural frequency of the vibration absorber in order to track variations of the primary disturbance frequency.

### 4.2.3 Control Methods for Vibration Isolation

Feedforward control involves sensing an advanced signal related to the disturbance to be cancelled at the error sensors, and feeding this signal into the controller which then generates an appropriate signal to drive the control actuators. On the other hand, feedback control uses the error sensor signals as inputs of the controller; these signals are passed through compensation filters to generate the inputs of the control actuators. Feedforward control has been shown to provide better results than feedback control, provided that a reference signal well correlated with the disturbance is available to the controller. Obtaining such a signal is particularly simple for periodic excitation. Therefore, for periodic vibration generated by rotating or reciprocating machinery, feedforward control is preferable. In cases where the disturbance is broadband, random noise and if a reference signal of this disturbance is available and can be processed through the control filter before it reaches the location of the control actuators, then feedforward control is also possible. In other cases where no advance reference signal is available, feedback control is the only option (this is the case for active vehicle suspensions that minimize the effect of road irregularities on passengers).

### 4.2.4 Effect of Control Force Location

When the physical system has large dimensions (such as a ship engine), some caution must be excercised in locating the control forces with respect to the primary forces transmitted to the base structure. A theoretical analysis has been presented by Jenkins (1989, 1993) to study this effect in the case of an infinite plate which supports pure bending waves and which is excited by two transverse point forces separated by a distance *d*, simulating the primary and control forces. It was shown that a reduction of the total power transmitted into the plate can only be achieved with the control force, if the separation *d* is well within $3\lambda/8$, where $\lambda$ is the bending wavelength in the plate. For example, in the case of a 5mm thick steel plate, a reduction of the base plate vibration is possible at 100Hz only when the control actuator is at a distance less than 260mm from the primary force, and at a distance less than 82mm from the primary force at 1000Hz. For a 10dB attenuation, these distances should be less than 69mm and 15mm respectively. These conditions can be important physical limitations when large actuators are involved.

### 4.2.5 Effect of Rotational and Translational Vibration Transmission Through the Mounts

In practice, mounts do not behave as one-degree-of-freedom systems. Disturbances can be applied on the mounts as either moments or forces in the various directions, and mounts have finite compliances in all of these directions. In situations where vibration energy is transmitted through the mounts via several types of motion, active mounts must be designed to control the various motion components involved. The active mount suggested by Ross (1988) and re-visited by Hansen (1997) for heavy machinery isolation is designed to control forces in both horizontal and vertical directions, as well as moments. This mount uses an intermediate mass in a two-stage isolator, all degrees of freedom of the intermediate structure being controlled by control electrodynamic shakers (see Figure 4.3).

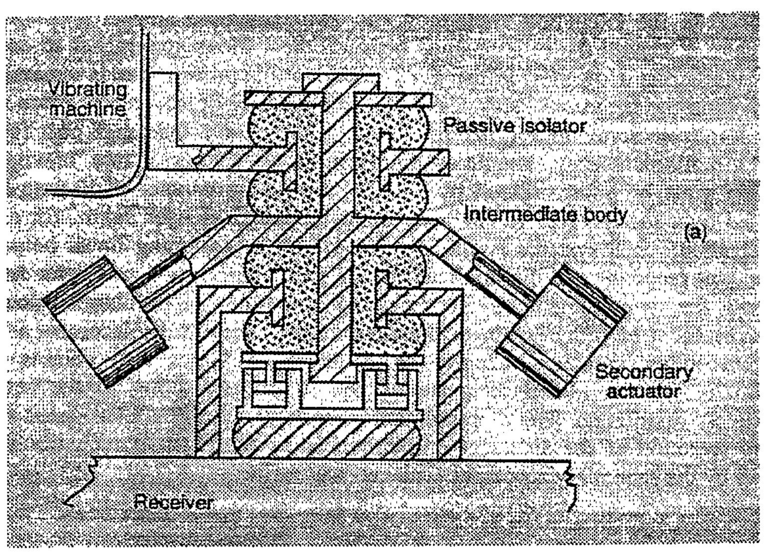



Figure 4.3: The Active Mounting Configuration Suggested by Ross (1988) for the Control of Multi-Degree-of-freedom Transmission

Another approach is suggested by Smith (1983) who used passive isolators which were very compliant to shearing motions, in parallel with electrodynamic actuators at each mounting point, which acted to control only vertical motion. A further approach successfully used by Jenkins (1991), employs a pneumatic isolator which can in principle only transmit forces in the transverse direction. The pneumatic isolator is in turn controlled by only a single translational actuator which is capable of controlling the pressure fluctuations within the isolator. This is illustrated in Figure 4.4.

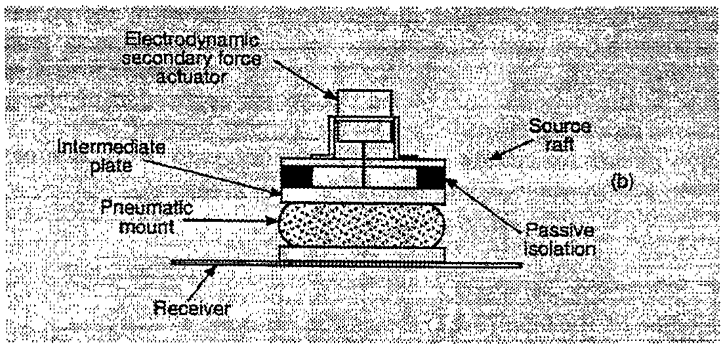



Figure 4.4: Active Mounting Configuration used by Jenkins (1991) to Enable Control Only in the Vertical Direction

Finally, Howard (1999) has recently investigated a multi-degree-of-freedom active isolator, consisting of up to seven electrodynamic actuators arranged in such a way as to control all six dof (three translations and three rotations) at the attachment point.

### 4.2.6 Sensing Strategies in Active Isolation Systems

Sensing strategies are investigated for the case of the active isolation of a base structure from a vibrating source such as a machine. This discussion is mostly restricted to *feedforward* control systems. In general, the receiving structure is not perfectly rigid, and two main sensing strategies can be envisioned – minimizing the forces transmitted to the base structure, and minimizing the vibration accelerations at specific locations on the base structure:

- The minimization of the transmitted forces requires force transducers mounted between the suspension and the receiving structure; the force sensors must measure the force transmitted through both the passive and active components if these are in a parallel configuration.

- The minimization of the transmitted accelerations requires accelerometers attached on the receiving structure at various locations (not necessarily the mount attachment points). Jenkins (1989, 1993) has undertaken a detailed investigation of an active isolation system representative of a machinery isolation problem. In the study the effects of various accelerometer configurations on the receiving structure as error sensors were studied. It was concluded that "overdetermined" configurations (with more error sensors than control actuators) seem preferable for active isolation. The results indicate that a requirement of two error accelerometers per structural wavelength on the receiving structure are an adequate criterion for a *global* control of the receiving structure.

An important question is whether force sensors or acceleration sensors are preferable. Burke (1991) indicates that in single axis applications, minimizing either force or acceleration (at the mount) results in global control of the receiving structure. However, Scribner (1993) and Sievers (1989) argue that force minimization is preferable since it allows vibration isolation *in both directions* (in other words, minimizing the acceleration of the receiving structure may actually increase the vibration of the primary source). Additionally, for a system with multiple mounts between the primary source and the receiving structure, Hansen (1997) recommends that the *total transmitted vibratory power* be used as the objective function to be minimized by the controller. The transmitted vibratory power is obtained by combining the transmitted forces and vibration velocities at the various attachment points on the receiving structure. Work undertaken by Pinnington (1981, 1987), Scheuren (1995), Elliott (1997) and Pavic (1987) indicates that vibratory power is a suitable cost function for minimizing the overall vibration response of a flexible structure. Vibratory power minimization has been investigated experimentally in multi-degree of freedom active isolation by Howard (1998).

### 4.2.7 Review of Active Isolation Systems for Ship Structures or Heavy Structures

*Machinery*

Tanaka (1988) has demonstrated the feasibility of a *feedback* approach to reducing the transmission of machinery vibration to the foundation. An outline sketch of their experimental system is shown in Figure 4.5. The secondary force is applied via hydraulic actuators capable of producing forces of up to 4kN and quartz force transducers were used to sense the force applied to the concrete receiving structure. Details on the compensating feedback filter can be found in

(Tanaka, 1988). The system was successful in reducing the transmitted forces by more than 10dB in the frequency range 2-20Hz.

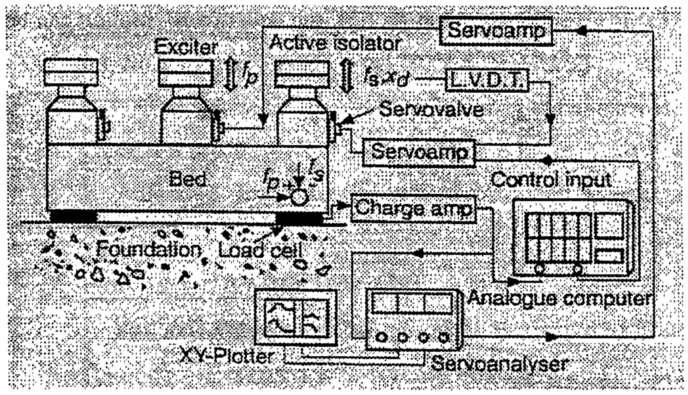



Figure 4.5: The Approach Used by Tanaka (1988) to Reduce the Transmission of Machinery Vibration to a Foundation

Work presented by Watters (1988) used a similar feedback strategy in experiments on active isolation of a diesel engine vibration. Scheuren (1995) has reported experiments on the active isolation of vibration produced by a small electric motor to a base structure, using electrodynamic actuators. To demonstrate the operational robustness of the active system, it was installed to isolate the vibrations of a converter from the foundation connecting it to a ship structure. It could be shown that under normal operating conditions of the ship it was possible to attenuate the mean square vibration level on the target structure by up to 20 dB.

*Helicopters and aircraft*

The application of active techniques to helicopter vibration problems is described by Staple (1990). Vibrations due to aerodynamic imbalance on the rotor are transmitted to the airframe at the blade passing frequency and its harmonics. An active isolation system using four hydraulic actuators in parallel with elastomeric mounts was installed at the airframe-raft interface. The hydraulic actuators used are shown in Figure 4.6; they were capable of producing a force of 8.9KN and displacement of 0.01in at the blade passing frequency (21.7Hz). The actuators were driven in a feedforward control system to minimize the weighted sum of squares of the outputs from 12 accelerometers plus the weighted sum of squared actuator inputs.

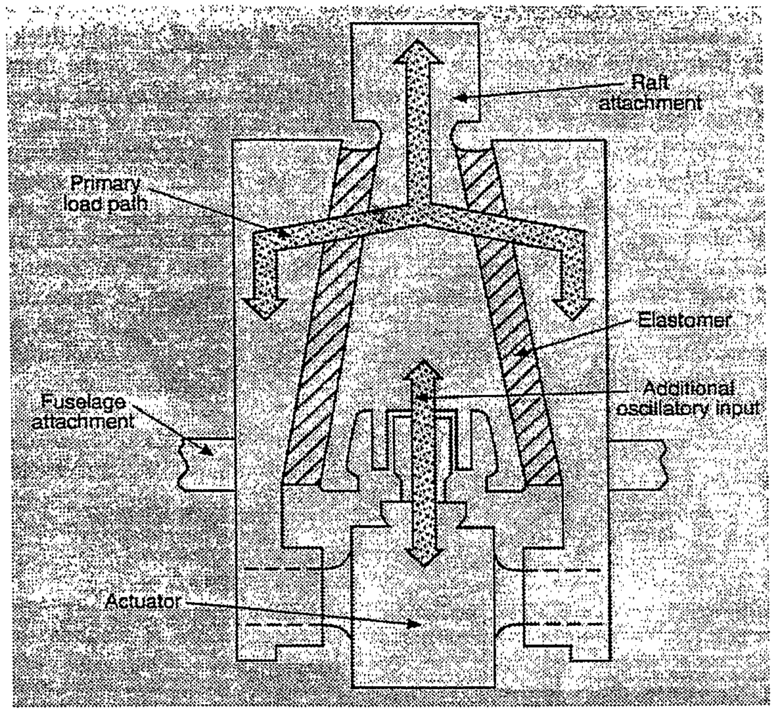



Figure 4.6: The Active Mount Used by Staple (1990) for Vibration Isolation of a Helicopter Airframe

The elastomer provides the passive stiffness between the raft (supporting engine and gearbox) and fuselage. The hydraulic actuator provides an additional force in parallel with the passive stiffness.

Swanson (1993) has described the design of an active engine mounting for the isolation of an aircraft structure to the vibrations produced by mass imbalances in jet engine turbofans. The mount is essentially electromagnetic in its operation.

*Ship Structures*

Day (1993) used pneumatically-controlled vibration absorbers to reduce the vibrations caused by a six-cylinder diesel generator set having a mass of approximately 1 tonne. Reductions in vibration levels near the engine mount of about 10dB were achieved using four absorbers with a total mass of 18kg. The absorbers incorporated air springs whose stiffness can be tuned depending on the contained air pressure within the springs to track the frequency of interest. A nonlinear behaviour of this type of actuator was reported by the authors, and prevented an automatic tracking of the target frequency.

Experimental work was also presented on the use of active or semi-active absorbers directly attached to the ship structure in order to attenuate ship hull or ship superstructure vibrations. Hsueh (1994) proposed to use an active absorber consisting of a 10-tonne mass moved by a hydraulic actuator attached to the hull of a 87,000-tonne oil tanker in order to attenuate propeller and engine induced vibrations of the ship. A new type of Tuned Liquid Damper (TLD) was suggested by Kagawa (1994) to attenuate the vibration of ship structures. This dynamic absorber uses water in a U-shaped tank as the mass element, and air in the tank as a variable stiffness element; the stiffness can be varied by varying the air pressure in the tank. Also, a two-axis active mass damper driven by a hydraulic actuator was developed by Kakinouchi (1992) to reduce the vibration of ship superstructures from external vibrations.

Finally, an inertial, electrodynamic shaker has been investigated as a control actuator in active isolation mounts for ship engine vibration, by Winberg (1998). The boat used in these experiments was a Storcbro Royal Cruiser 33, powered by two Volvo Penta TAMD engines. The main idea was to prevent the engine vibrations from propagating in the ship structure and radiating sound into the cabin. Therefore, microphones, mounted in the cabin, were used as error sensors in the active control system. The active element was an inertial mass damper designed by Metravib in France. The actuator yields a large output force for its volume and utilizes Metravib's patented multipolar technology. The inertial mass damper was mounted between the engine and the hull in parallel with a passive mount, as in Figure 4.2(e).

## 4.3 Path 2: Active Control of Noise in Ducts and Pipes

### 4.3.1 Introduction

The reduction of duct noise is the first known application of active noise control. Active control systems for duct noise are now a mature technology, with several commercial systems available for ventilation systems, chimney stacks or exhausts. All existing commercial systems are based on *feedforward* adaptive control systems. In the case of ducts containing air or a gas, loudspeakers are generally used as control sources, and microphones as error sensors.

Two important classes of systems must be distinguished, depending on the frequency and the cross-sectional dimension of the duct:

- The systems for which only plane wave propagation exists in the duct; such systems will necessitate a single channel control system (one control source and one error sensor); and
- The systems for which higher-order acoustic modes propagate in the duct; such systems will require a multi-channel control system.

The occurrence of higher-order modes in a duct depends on the value of the cut-on frequency. For a rectangular duct, the cut-off frequency is given by $f_c = c_0/2d$ where $d$ is the largest cross-sectional dimension and $c_0$ is the speed of sound in free space. For a circular duct: $f_c = 0.586c_0/d$ where $d$ is the duct diameter. Higher-order modes will propagate at frequencies larger than $f_c$.

There exists a large amount of literature on the active control of duct noise, and the reader is referred to the Chapter 5 in Nelson (1992) and Chapter 7 in Hansen (1996) for a general presentation and access to many references on the subject. The following review gives a summary of important physical aspects involved in active control of duct noise, and concentrates on the active control of exhaust noise and control of pressure fluctuations in liquid filled ducts.

### 4.3.2 Passive Control of Duct Noise

The passive control of duct noise is covered in many textbooks, such as (Bies, 1996). Passive control is usually accomplished using reactive mufflers (which change radiation impedance seen by the source, and consist of expansion chambers and perforated pipes) or dissipative mufflers (which dissipate the sound energy using baffles lined with sound absorbing materials). For large ducts, such as an exhaust stack, passive mufflers are bulky, and can be very expensive. In general, the advantages of active systems are a small size, low pressure drop and better low-frequency performance. In high frequency, passive systems are usually preferable in terms of cost and complexity. Combinations of active and passive measures for low and high frequency control in ducts has been discussed by Munjal (1989a).

One of the practical difficulties associated with active control systems is the acoustic feedback from the control source to the reference microphone in feedforward systems. Also, turbulent pressure fluctuations associated with flow in the duct create contaminating noise at the reference and error microphones. Poor frequency response of the control loudspeakers at low frequency and non-uniform frequency response in high frequency are important difficulties associated with the control sources. Reflections of the sound waves from duct bends, section changes and duct ends create important conditions on the placement of the control source and error sensor in the duct. The transducers used must in some cases withstand contaminated flows containing corrosive or high temperature gases. The possible transmission of vibration by the duct walls can be important for liquid filled ducts. Aspects related to turbulent pressure fluctuations, contaminated flows and actuator requirements have already been discussed in the Sections 3.8 and 3.11. The other aspects are discussed below.

### 4.3.3 Acoustic Feedback

In a feedforward system a reference sensor is necessary to measure an incoming signal correlated with the primary noise. The reference sensor is usually an upstream microphone for random noise control. For periodic noise (eg. engine noise), a non-acoustic independent sensor such as a tachometer can be used to measure the rotational speed of the machine. It is then necessary to convert the tachometer output, usually a pulse train, into a periodic analogue signal containing sinusoids at the fundamental and harmonic frequencies to be controlled.

When a reference microphone is used upstream of the control source, there is generally an acoustic feedback between the control source and reference sensor, which has to be minimized or compensated (otherwise there is a risk of instability of the control system). This can be achieved by using a directional reference sensor and control source (Swinbanks, 1973, Eghtesadi, 1983, LaFontaine 1983, 1985), implemented as sensor arrays or source arrays, or more practically using a probe tube in the case of the reference sensor (see Section 3.8). The acoustic feedback can also be compensated in the control loop by using an IIR (recursive) control filter, which inherently accounts for the acoustic feedback (Eriksson, 1987). Additionally, in the control of random noise the distance between the reference microphone and the control source should be such that the electronic delay associated with A/D and D/A conversions, low-pass filters and control filter is smaller than the acoustic delay between the microphone and the control source.

### 4.3.4 Internal Reflections in the Duct, Control Source Locations, and Error Microphone Locations

For finite ducts, internal acoustic reflections put important conditions on the optimal locations of the control speakers and error microphones in the duct. The following rules to follow are based on (Chapter 7 in Hansen, 1997):

- For a semi-infinite duct with the primary source at the duct end and an infinite impedance at the primary source (i.e. a constant volume velocity primary source), the control source should be placed at an integer multiple of half wavelengths from the primary source. Positions corresponding to odd multiple of quarter wavelengths should be avoided (as infinite volume velocity is required for the control source in this case). If the primary source behaves as a zero impedance (or constant pressure) source (this better represents aerodynamic sources such as in fan noise) then optimal control source locations are odd multiple of quarter wavelengths from the primary source. Control source locations at an integer multiple of half wavelengths from the primary source should be avoided.

- The analysis becomes more complex in the case of a finite duct because of the reflections from both ends of the duct, which create a standing wave. As a general trend, the control source positions should not correspond to nodes of the standing wave in the duct. As far as error microphones are concerned, they should be sufficiently far from the control sources, not to detect the control source near field. For the same reason, error microphones should not be in the close vicinity of duct discontinuities, such as bends or section changes. As for the control sources, the error microphone positions should not correspond to nodes of the standing wave in the duct.

### 4.3.5 Active Control of Higher Order Modes in Ducts

Stell (1990), Zander (1993) have presented theoretical analyses of higher order mode control in ducts. Eriksson (1989), Silcox (1990) have carried out experimental investigations on the control of two propagating modes in a duct with a two-channel control system. When multi-mode propagation exists in a duct, the number of control sources and error microphones required is approximately equal to the number of modes to be controlled (Laugesen, 1993). L'Esperance (1997) has presented a successful implementation of a multi-channel system including 12 control loudspeakers and 12 microphones for the active control of multi-mode sound propagation in a large chimney stack.

### 4.3.6 Applications of Duct Noise Control

*Control of Exhaust Noise*

A feedforward system for controlling the harmonic exhaust noise from a diesel engine was first proposed by Chaplin (1980). It involved a single loudspeaker mounted at the exit of the exhaust (Figure 4.7). The error microphone was located close to the exhaust termination, and the reference sensor was a tachometer directed at the engine cam shaft.

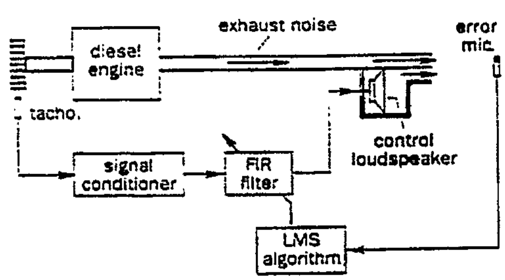



Figure 4.7: Active Control of Exhaust Noise (after Chaplin, 1980)

Kido (1987, 1989) have used a similar arrangement except that the reference signal is taken from a microphone in the exhaust. The physical mechanisms associated with the use of a control source adjacent to the duct exit, is that the control source changes the simple monopole source to a less efficient dipole. The use of a second control source at the duct exit has been investigated by Trinder (1986) to create an even less efficient quadruple-type radiation. A single error sensor outside the duct is required provided $kd \leq 0.5$, where $k$ is the acoustic wave number and $d$ is the duct diameter.

In typical exhaust noise situations, it is impractical to install the sound sources inside the exhaust pipe or in the pipe wall, because it is difficult to find acoustic sources capable of withstanding the hot and dirty environment. It is also difficult for conventional loudspeakers to generate the required high sound levels (typically 170-180dB) needed for control. Thus, for exhaust silencing, it is more practical to locate the control sources outside the exhaust. An example of such an implementation is reported by Foller (1992) and represented in Figure 4.8 in this case, the secondary sound field is introduced in a concentric, larger diameter duct.

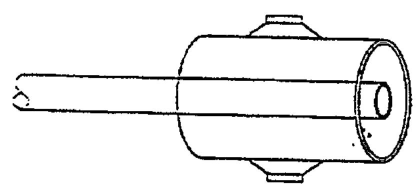


Figure 4.8: Vehicle Exhaust Muffler With Concentric Cylinder to Contain the Secondary Sources (Foller, 1992)

Alternatives to the use of loudspeakers as control sources in the active control of exhaust noise, have also been investigated. Tartarin (1991) has successfully demonstrated the use of an oscillating valve to control the pressure fluctuations inside the duct and reduce the downstream propagation of sound waves. This device is detailed in Section 2.4.

*Control of sound propagation in liquid filled ducts*

The principles of active control of noise propagating in liquid filled ducts are much the same as in air ducts (Culbreth, 1988). The higher speeds of sound in liquids means that plane wave propagation occurs in a larger frequency range than in air ducts. However, considerable care must be exercised to the possible transmission of energy via the flexible duct walls in this case, as a result of the strong coupling between the duct walls and the interior fluid.

As a consequence, a distinct aspect for fluid-filled ducts is the type of control sources and error sensors, which are used. Hydrophones can be used as error sensors in the duct, while commercially available sound sources include sonar sources. Another option is to use purely structural sensors and actuators mounted on the pipe wall. Active control of wave propagation in fluid-filled elastic cylinders has been studied by Brevart (1993) and Fuller (1996) using transverse forces on the pipe as secondary sources and accelerometers as error sensors. This has benefits in realistic implementations such as ease of implementation and nonobtrusive control hardware.

## 4.4 Path 3: Active Control of Vibration Propagation in Beam-Type Structures

### 4.4.1 Introduction

The active control of vibration in one-dimensional systems such as beams, rods, struts, shafts, etc. can be approached from two different perspectives, depending on the description of the structural response. The response can be described in terms of vibration modes, or in terms of waves propagating in the structure. The modal perspective is more appropriate to finite, or short beams, and to *global* reduction of the vibration. The description of the response in terms of structural waves is more appropriate to infinite, or long beams, and to reducing energy flow from one part of the beam to another (control of vibration transmission). The wave description is then more appropriate to the case of the transmission of vibration from a ship engine via the drive shaft, since in this case the source of vibration is known and the objective is to block the vibration transmission along the shaft.

The following sections first address important physical aspects of the active control of structural waves in beams, as revealed by the literature on the subject. A review of previous work on the active control of vibration in rotating machinery or similar problems relevant to the vibration control of a marine drive shaft is then presented.

### 4.4.2 Physical Considerations

The problem of controlling the propagation of structural waves in a beam bears some similarity with the analog problem of controlling sound waves in a duct. However, there are important differences between the two problems. In contrast with acoustic propagation in ducts, there are several types of waves propagating in beams: flexural, longitudinal, and torsional waves. Flexural waves are generally the most important type as they are associated with low frequency, and large amplitude response; also, flexural waves are responsible for most of the acoustic radiation. Flexural (and to some extent longitudinal) waves will be mostly covered in the following discussions. Another difference with the propagation of acoustic waves is that flexural waves in structures are dispersive (their propagation speed depends on the frequency); this distinction has important consequences as far as active control systems must be implemented, as discussed further on. It should be noted however, that torsional waves transmitted to the propeller may also be significant and require attention.

The active control of vibration in beams is covered in Chapter 6 of (Fuller, 1996) and Chapter 10 of (Hansen, 1997). The following presentation is mostly limited to feedforward control systems since it is assumed that for the problem of vibration transmission along a marine drive shaft, an advanced signal correlated to the disturbance, or a measurement of the incoming disturbance wave is possible.

*Simultaneous control of all wave types (flexural, longitudinal, torsional):*

A general theoretical analysis of this problem can be found in (Pan, 1991). Figure 4.9 shows the structure of a general adaptive feedforward controller for the active control of multiple wave types in a beam (after Fuller, 1996). In this system, sensor arrays (such as accelerometer arrays) are used to measure the different types of waves propagating upstream (detection array) or downstream (error array) of the control actuators, and an array of actuators is used to inject and control the various wave types in the beam.

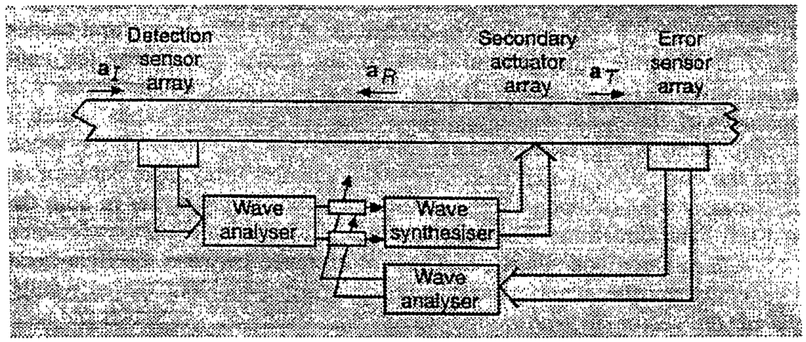



Figure 4.9: Structure of a General Adaptive Feedforward Controller for Active Control of Multiple Wave Types in a Beam (after Fuller, 1996).

Wave analyzers are necessary to extract the independent wave types (assumed uncoupled) from the sensor arrays, and wave synthesisers are necessary to generate the appropriate commands to the individual actuators. This approach has the advantage that independant control filters can be used to control the flexural, longitudinal and torsional waves. However, it necessitates excellent phase matching of the sensors and a detailed knowledge of the structure in which the waves propagate. An experimental laboratory implementation of this approach has been conducted by Clark (1992), on a thin beam, for the control of two flexural wave components and one longitudinal wave using PZT actuators. Another easier option avoids implementing wave analyzers and synthesisers by simply minimizing the sum of squared output of the error sensors to control the different wave types. This approach, however, requires a fully coupled multi-channel control system. Brennan (1992a), Elliott (1994) have tested this approach for the control of two flexural waves and one longitudinal wave in a strut using 3 magnetostrictive actuators. Similarly, (Fuller, 1990) has implemented the same approach for the control of one flexural wave and one longitudinal wave on beam using a pair of collocated PZTs.

*Control of flexural waves:*

As mentioned previously, when only flexural waves are considered, it is important to account for the dispersive nature of the flexural waves in the configuration of the active control system. The dispersive nature of flexural waves implies that a control force applied transversely to the beam generates propagative waves as well as evanescent waves localized close to the point of application of the force. Consider now a flexural wave propagating along a beam, the downstream transmission of which has to be stopped actively. If one transverse control force is applied at some location on the beam, it generates downstream and upstream propagating waves plus downstream and upstream evanescent waves; this actuator can minimize the total, transmitted downstream wave, but generates a reflected wave towards the source and two evanescent components which may be undesirable. In order to minimize the downstream propagating and evanescent waves, two closely spaced actuators will be necessary. A total of four actuators will be necessary to control downstream and upstream, propagating and evanescent components. Therefore, the control of flexural waves in beams will in general require *actuator arrays* (Mace, 1987, McKinnell, 1989; Von Flotow, 1988; Brennan, 1992b; Scheuren, 1985). Combinations of *force* and *moment* actuators can also be used in the actuator array.

The simplest feedforward control system is represented in Figure 4.10. This system uses only one control force and one error accelerometer, together with one reference accelerometer to measure the incoming wave. This system has been studied theoretically by Pan (1993), and tested experimentally by Elliott (1993).

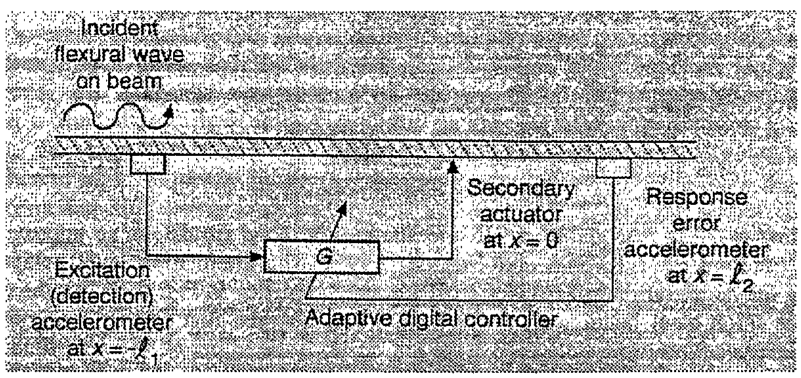



Figure 4.10: Simple Feedforward System for the Control of Flexural Waves in a Beam (after Fuller, 1996)

The previous authors have identified important physical limits of this system. The first limit is associated with the detection of the control actuator evanescent wave by the error sensor. This detection is not desirable since the evanescent wave does not carry energy in the far field of the actuator. This puts a limit on the actuator-error sensor separation, in practice the sensor should be at least 0.7$\lambda$ from the control actuator ($\lambda$ being the flexural wavelength). The second limit is related to the delay between detection and actuation. This delay should be sufficient to allow the active control system to react at the control actuator location before the primary wave has propagated from the detection sensor to the control actuator. This puts a limit on the reference sensor – actuator separation, which depends on the characteristics of the control system. Note that this delay between detection and actuation is considerably reduced for longitudinal waves, which propagate at a much larger velocity than the flexural waves in general.

Similarly to control actuator arrays, error sensor arrays need to be implemented for the control of flexural waves to distinguish between the various propagating waves and evanescent waves at the error sensor locations. This is particularly needed if the error sensors must be located at a short distance from the control actuators. In this case, an array of four accelerometers can discriminate the two propagating waves and the two evanescent waves at the location of the error sensor array, and extract the components which need to be reduced (such as the downstream propagating wave). Other sensing strategies have also been suggested, such as measuring and minimizing the structural intensity due to flexural waves (Schwenk 1994; Swanson, 1997; Audrain, 1998). Structural intensity can be measured in practice using an array of four closely spaced accelerometers.

### 4.4.3 Active Control of Wave Propagation in Machinery Shafts and Related Structures

The above work has concentrated on theoretical analyses or laboratory experiments on relatively small-scale one-dimensional structures. There are a limited number of practical implementations of these principles to large, machinery structures. Brennan (1998) has described an experiment using semi-active or active devices to attenuate the transmission of *longitudinal* vibration on a large tie-rod structure. The tie-rod is similar to that found in marine machinery to maintain the alignment of a machinery raft. A tunable pneumatic vibration absorber was used as the semi-active device, and an electrodynamic shaker or a magnetostrictive actuator were used as active devices. A load cell was used as the error sensor, such that the force applied by the tie-rod to a receiving bulkhead was minimized.

A paper by Kato (1997) describes the suppression of vibration which is generated on rotating machinery with an overhung rotor. In this case, the vibration of the rotor-shaft system is controlled by active bearings. The active bearings consist of a bearing housing supported elastically by rubber springs and controlled actively by electromagnetic actuators. These actuators are controlled by displacement sensors at the pedestal and/or the roller and can apply an electromagnetic force which suppresses any vibration of the roller. Similar electromagnetic active bearings have been used by other Japanese researchers to control the vibration of shat-rotor systems (Okada, 1994; Satoh, 1990).

Palazzolo (1993) has investigated Active Vibration Control (AVC) of rotating machinery utilizing piezoelectric actuators. The paper gives actual test data on an aircraft engine test stand-shaft line, unlike the majority of related papers which are entirely theoretical or provide test results only on small, laboratory rotors. The AVC is shown to significantly suppress vibration through two critical speeds of the shaft line.

## 4.5 Path 4: Active Control of Enclosed Sound Fields (Airborne Noise)

### 4.5.1 Introduction

There exists a vast body of literature on the subject of active control of enclosed sound fields. Only the previous work relevant to the problem of cancelling the sound field radiated by a ship engine in its enclosed space will be reviewed here. More comprehensive presentations of the generic problem can be found in (Chapter 10, 11 in Nelson, 1992) and (Chapter 9 in Hansen, 1997).

Active control of enclosed sound fields has found applications essentially for automobile interior noise (see eg. Guicking, 1991; Bernhard, 1995; Sutton, 1994) and for aircraft interior noise (see eg. Elliott, 1990; Borchers, 1992; Emborg, 1993), leading in some cases to commercial products.

There are two main categories of active control systems related to enclosed sound field minimization:

- The active control of sound transmission through elastic structures into an enclosure; and
- The active control of sound field into rigid enclosures.

Only the second category will be reviewed here. The active control of sound transmission has been investigated by (Bullmore, 1987; Fuller, 1986; Pan, 1990) using essentially modal approaches. The same type of analytical approach based on modes of the acoustic enclosure can be used to investigate the active control of sound field into rigid enclosures. It should be mentioned, however, that finite element approaches have also been used to study the active control of sound field into enclosures of complex geometries (Mollo, 1987; Mollo, 1989; Mollo, 1990; Cunefare, 1991).

Additionally, the objective of the *active* control in an enclosure can be to minimize the sound field *globally*, or *locally*. Only the approaches directed towards global attenuation of the sound field are reviewed here. In this respect, some important physical aspects of this problem are discussed in the following. These physical aspects depend primarily on the modal density of the enclosure.

### 4.5.2 Enclosures With a Low Modal Density

For enclosures with a low modal density (i.e. a small enclosure, or at low frequency), the active control will usually consist of placing a series of control loudspeakers in the enclosure; the loudspeakers are driven to minimize the sound pressure measured by discrete error microphones. In the case of an enclosed acoustic space, the performance metrics for the control should be the acoustic potential energy integrated over the volume of the enclosure,

$$E_p = \frac{1}{4\rho_0 c_0{}^2} \int_V |p(\mathbf{r})|^2 dv$$

where $p(\mathbf{r})$ is the local sound pressure, $\rho_0$ is the density of the acoustic medium and $c_0$ is the sound speed. The active control scheme should aim at reducing the acoustic potential energy as much as possible.

Bullmore (1987), Elliott (1987), Curtis (1990) have shown that active control of sound fields in lightly damped enclosures is most effective at the resonance of the acoustic modes. In these instances, the problem is essentially the control of a single mode. Significant attenuation of the acoustic potential energy is obtained using a single control source and a single error microphone (provided neither the control source nor the error microphone is located on a nodal surface of the acoustic mode). For multi-mode (off-resonance) response of the cavity, the number of control sources and error microphones must be increased. Control sources and error sensors number and placement become critical. However, the potential for attenuation is never as large as at a resonance frequency.

For multi-mode response of the cavity, the control source and error sensor arrangement is crucial in terms of the achievable acoustic potential energy attenuation. The control source and error sensor optimal locations (with respect to maximum achievable potential energy attenuation) are difficult to determine in practice. The corresponding optimization problem is nonlinear and has usually many local minima. Optimization processes, such as multiple regression (Chapter 9 in Hansen, 1997), or genetic algorithms (Baek, 1995) must be used. The number of control sources required is a difficult problem as well, and there are no simple guidelines to determine it. As a general indication, the number and locations of the control sources should be such that the secondary sound field matches as closely as possible the primary sound field in the enclosure.

### 4.5.3 Enclosures with a High Modal Density

As the frequency increases or the enclosure becomes larger, global attenuation of the sound field becomes more difficult to achieve using an active control system. To quantify these limitations, there are some approximate formulas, which are summarized here. These formulas are approximate, but they give useful expected performance of an active control system in a high modal density enclosure.

First, assuming a single primary point source and a single secondary point source in the enclosure, it is possible to derive the ratio of the minimized potential energy (after control) to the original potential energy (before control), (Tohyama, 1987; Elliott, 1989):

$$\frac{E_{p,\min}}{E_{p,0}} = 1 - \left[1 + \frac{\pi}{2} M(\omega)\right]^{-2},$$

where $M(\omega)$ is the modal overlap of the cavity, which quantifies the likely number of resonance frequencies of other modes lying within the 3dB bandwidth of a given modal resonance. For a rigid rectangular enclosure, and for oblique acoustic modes (i.e. three-dimensional modes, such as the (1,1,1) mode),

$$M(\omega) = \frac{\varsigma\omega^3 V}{\pi c_0^{\ 3}},$$

where $\varsigma$ is the damping ratio in the enclosure (assumed identical for all acoustic modes), $\omega$ is the angular frequency of the sound field and $V$ is volume of the enclosure.

If the modal density is low (at low frequency),

$$\frac{E_{p,\min}}{E_{p,0}} \approx \pi M(\omega),$$

which means that the achievable attenuation is dictated by the modal overlap (and hence the modal density and damping of the enclosure).

If the modal density is large (at high frequency),

$$\frac{E_{p,\min}}{E_{p,0}} \approx 1,$$

which means that no attenuation can be obtained after control. Another expression can be derived from asymptotic expression of modal overlap in high frequency (Nelson, 1987),

$$\frac{E_{p,\min}}{E_{p,0}} = 1 - \mathrm{sinc}^2 kd$$

where $k$ is acoustic wavenumber and $d$ is separation between primary and control sources. Thus, as the control source becomes remote from the primary source, such that $kd \geq \pi$, global attenuation of the sound field becomes impossible. This provides an explicit analytical demonstration that global control of enclosed sound fields of high modal density is only possible with closely spaced compact noise sources. In other words, assuming an extended primary source such as a ship engine, the only viable solution in this case is to distribute control loudspeakers around the engine and in the close vicinity of it (within a fraction of the acoustic wavelength).

### 4.5.4 Advanced Sensing Strategies

Recently, alternatives to sensing and minimizing squared sound pressure have been suggested in active control of enclosed spaces. Sensing strategies based on total acoustic energy density minimization instead of sound pressure minimization have been suggested (Park, 1997; Sommerfeldt, 1995; Cazzolato, 1998). The advantage of sensing the total energy density is that the control is less sensitive to the sensor locations and in general a superior attenuation is obtained. The energy density can be measured using combinations of microphones (2 to 6); in this case, finite differences between individual microphones are applied to obtain approximate measurements of the pressure gradient in several directions. Precise measurements of the pressure gradient requires an excellent phase matching of the individual microphones, which can result in more expensive microphones. Associated adaptation algorithms for the minimization of energy-based quantities have been derived (Sommerfeldt, 1994).

### 4.5.5 Applications to Ship Noise

The only application found on active control of enclosed sound fields to ship noise is a work by Dignan (1996) who has implemented such a system to reduce low frequency noise in the aft berthing space of a high speed patrol craft. The system controls both propeller blade harmonic noise, and broad-band noise caused by propeller cavitation and machinery. A local control strategy using an adaptive feedback was used to provide a quiet zone at the head of each bunk. Significant noise reductions were obtained in the octave bands 31, 63 and 125 Hz.

## 4.6 Hybrid Active-Passive Control methods

The combination of active and passive elements has taken different forms in the literature, ranging from tunable Helmholtz resonators and tuned mass dampers to piezoelectric passive damping and active constrained layer damping. Moreover, vibration control systems have to be distinguished from noise control systems. In addition, a new concept referred to as a piezoelectric damper is also discussed.

### 4.6.1 Vibration control

While stability and robustness of active control systems has been studied greatly in research literature, relatively little has been published on the usefulness of adding passive damping to active control systems. Several researchers have shown that the addition of passive damping can increase the closed-loop performance, decrease control power and increase stability and robustness.

It has been noted that passive damping appears to be the most important robustness parameter of a structure (Gueler, 1993). The amount of passive damping needed to maintain the closed-loop stability depends on the pole-zero spacing and uncertainty in the model. Moreover, the unmodeled flexible modes beyond the bandwidth must be gain stabilized. Because the loop gain of a flexible structure reaches a maximum near each resonance with a peak value inversely proportional to damping ratio, it can be shown that undamped flexible modes can never be gain stabilized. The amount of passive damping ratio that is required to ensure gain stabilization of modes beyond the control or model bandwidth depends on a number of things. These include the controller bandwidth, the modal density, sensor/ actuator locations and the controller rolloff.

The closed-loop performance of realistic controllers using output feedback can be greatly improved by the addition of passive damping (Knowles, 1989). The added passive damping also reduces the effects of actuator bandwidth and actuator/sensor noise. Many methods have been proposed to find the optimal active/passive ratio (Gaudreault, 1993). While no general conclusions can be made about the amount of damping needed, it is clear that the amount of passive damping is an important design parameter.

Another hybrid passive-active strategy is to use active elements such as piezoelectric patches in a passive regime, involving analog RL circuitry for example, and to combine it with the action of an active digital control system (Tsai, 1996) see Figure 4.11.

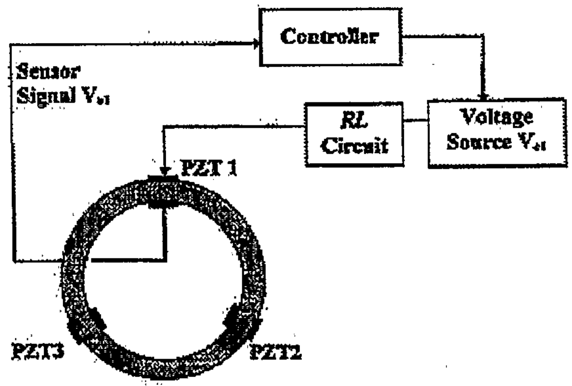



Figure 4.11: Ring with Active-Passive Hybrid Piezelectrical Networks (Tsai, 1996)

Passive control strategies involving appropriate structural modifications can also be combined with active control systems to create hybrid passive-active systems (Sunar, 1993; Gordaninejad, 1994; Hwang, 1994). The issue of optimal passive beam design was studied (Van de Vegte, 1973). The disturbances can also be suppressed using a combination of passive tuning and active control (Von Flotow, 1997). Linear reaction mass actuators (RMAs) have been used in an active control scheme (Garcia, 1995). The tuned-mass dampers were applied to high-rise buildings for vibration attenuation (Nishimura, 1998) see Figure 4.12. Strategies for retrofitting active control into passive vibration isolation systems have also been presented (Margolis, 1998).

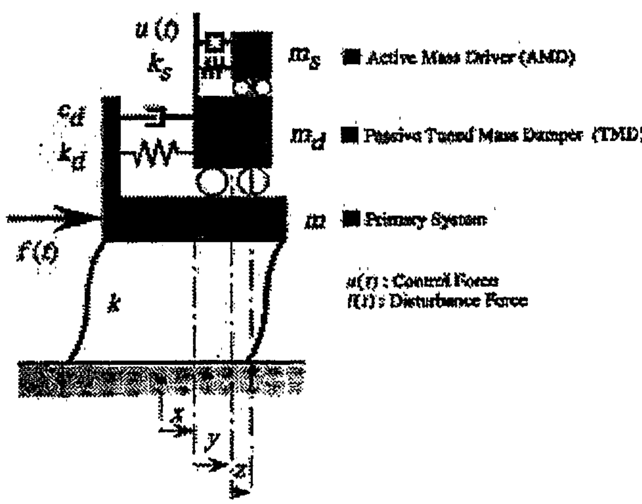



Figure 4.12: Active-Passive Composite tuned Mass Damper (Nishimura, 1998)

Passive and active systems may compete for available space, weight and funds. As a result, a vibration treatment that integrates active and passive damping together has advantages over applying active and passive damping separately. In this respect, Active Constrained Layer Damping (ACLD) has emerged as a promising combined treatment (Baz, 1994; Baz, 1996; Poh, 1996; Lam, 1997; Liao, 1997; Ray, 1997; Shen, 1997a; Shen, 1997b; Tawfeic, 1997). In this approach, a viscoelastic material is attached to a host structure and a PZT element is attached as a constraining layer. A conventional constrained layer damping treatment is thus augmented with efficient active means to control the strain of the constrained layer, in response to the structural vibrations. In a general configuration, (see Figure 4.13) the viscoelastic damping layer is sandwiched between two piezoelectric layers or between one piezoelectric layer and the host structure.

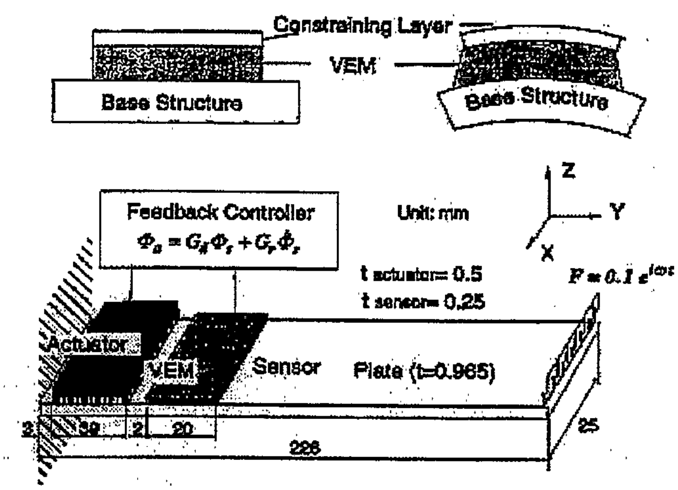



Figure 4.13: A Typical Active Constrained Layer Damping (ACLD) Treatment (Varadan, 1996)

The advantages of ACLD are:

- ACLD is fail-safe since in the event of a power failure or control hardware failure, the system reverts to a traditional constrained layer damping treatment;
- The added natural damping in the system allows higher control gains to be used, thereby increasing performance; the added natural damping across all modes delays the instability of uncontrolled modes so the gains can be increased;
- Constrained layer damping works very well at high frequencies, since higher frequency vibration induces more shear in the viscoelastic material, while active control is more effective at low frequencies; in this way, the passive and active components complement each other.
- The presence of more natural damping makes the control system more robust;

- ACLD is less likely to destabilize unmodeled modes; lightly damped higher modes are easily driven unstable especially when they are poorly modeled or not modeled at all;
- An ACLD treatment is more practical to install than an active and a passive system; both systems are incorporated;
- Since the passive part of the ACLD treatment already absorbs some of the energy, an ACLD treatment will use less power to achieve the same performance as a purely active system.

Many researchers are now working on the numerical modelling of active-passive damping using ACLD. Sensors and actuators embedded into composite host structures are being modelled using finite element techniques (Varadan, 1996; Guertin, 1998).

### 4.6.2 Noise Control

Another family of combined active-passive control techniques is emerging for noise control. Among these, smart foams have been studied for the control of noise transmitted through fuselage panels (Guigou, 1998). The smart foam is made from passively absorbing foam (which is effective at higher frequencies) and the active input from an embedded PVDF element driven by an oscillating electrical input (which is effective at lower frequencies). The use of active membranes to control the sound transmission has been explored (Sadeghipourt, 1992; Van Niekerk, 1995).

Noise control has also been performed using adaptive passive systems such as tunable Helmholtz resonators (Matsuhisa, 1990; Bernhard, 1992; De Bedout, 1997). Another approach involves pulsed flow control using an oscillating flap (Tartarin, 1991; Renault, 1996) see Figure 4.14).

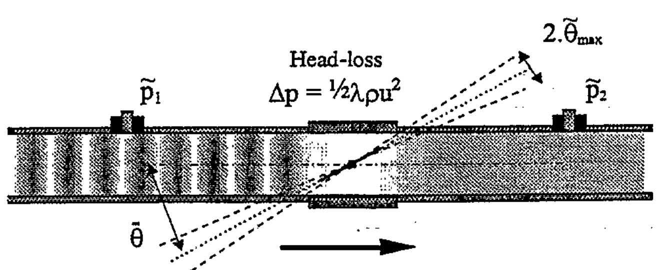



Figure 4.14: Anti-pulsatory System Using an Oscillating Flap (Renault, 1996)

### 4.6.3 Piezoelectric Dampers

Another possibility of increasing the damping in a structural system using piezoelectric materials is referred to as the "piezoelectric damper". In this configuration, the piezoelectric element is mechanically bonded to the structure, and electrically shunted by a simple circuit; the inherent capability of the piezoelectric element to convert mechanical (strain energy) into electrical energy is thus used, and this electrical energy is "absorbed" into the shunt circuit. The addition of a piezoelectric damper thus increases the loss factor of the structure. The principle and underlying equations of such shunted piezoelectric elements has been presented by (Hagood, 1991).

The performance of the device largely depends on the type of electrical shunt and on the stiffness of the shunted piezoelectric as compared to the mass and stiffness of the structure. When using a simple resistive shunt, the shunted piezoelectric has a behavior very similar to a viscoelastic material added to the structure, thereby decreasing the response at the resonances (intrisic loss factors as high as 42% have been reported for the device). When a resonant shunt is used (such as a resistive-inductive shunt), the shunted piezoelectric is very similar in its operation to a conventional dynamic absorber; the shunt parameters can then be tuned to decrease the structural response at a given frequency. Such a device has been commercially marketed by ACX and implemented in various sport goods, such as skis, snow-boards and baseball bats as vibration dampers (http://www.acx.com/home.html).

In the context of marine structures, piezoelectric dampers offer the advantage of being very simple (no need of external electric power supply or controller), and they can provide an interesting alternative to viscoelastic materials (the effective loss factor of the global system not only depends on the loss factor of the components, but also on their strain energy, and the stiffness of piezoelectric materials is several order of magnitude larger than the stiffness of common viscoelastic materials). On the other hand, piezoelectric dampers are effective only on-resonance, or at a targeted disturbance frequency; also, in the case of marine structure, the inherent stiffness of the device has to be large (comparable to the stiffness of the structure) in order to provide efficient damping.

# 5. RECOMMENDATIONS ON SENSORS AND ACTUATORS FOR ANVC OF MARINE STRUCTURES

## 5.1 Introduction

A review of active control methodologies that are suitable for the control of noise in ship structures was presented in Chapter 4. In the review, consideration was given to the sensor and actuator materials that could be used for active control of noise through the various acoustic paths. Based on the review, recommendations are provided below on suitable sensors and actuators to be employed in the development of active noise and vibration attenuation strategies for the four noise paths identified. In making the recommendations, due consideration is given to cost, suitability in the marine environment, experience in other applications, ease of implementation and the potential integration with the controller technology recommended in the accompanying document (Akpan, et al, 1999).

The final choice of the sensors and actuators in an active control system will depend on such factors as the frequency of the disturbance, operating environment, cost, expected performance and magnitude of the vibratory or acoustic disturbance to be controlled along the various vibroacoustic paths (which dictates the (authority) of the control actuators to be used in the system). Figure 5.1 suggests a systematic approach that proceeds through the various steps of the active control system design.

As a general recommendation, the first (and perhaps most important) phase of the design of every active control system is to acquire a thorough understanding of the vibroacoustic behavior of the system on which active control is to be applied. This involves carefully identifying and ranking the various paths along which vibroacoustic energy flows (this may imply addressing questions such as the transmission of moments or in-plane forces through the engine mounting, or the relative contribution of fluid-borne and structure-borne energy along pipes). This early phase is crucial in determining the active control strategy to be implemented. A number of experimental techniques and numerical simulation tools can be used to estimate the relative contribution of the various paths at a given receiving point (eg. in water). Based on some of the contractors previous experience, it is felt that a major transmission path is the engine mounting system.

**Phase 1 – Understanding the vibroacoustics of the system**

- Identification of the transmission paths
- Ranking of the transmission paths
- Active control strategy

**Phase 2 – Selecting the control actuators**

- Frequency of the disturbance
- Magnitude of the disturbance
- Mechanical impedance of the structure
- Control paths transfer function measurements

**Phase 3 – Selecting the error sensors**

- Exact quadratic optimization
- Error sensor configuration for global control

**Phase 4 – Testing the active control system**

Figure 5.1: Suggested Design Steps of an Active Control System

The second phase will determine the *type, number, and locations of the control actuators.* When global control is desirable (i.e. when attenuation of the sound field is desired at all positions in water, for example), these parameters are determined by the requirement that the sound field generated by the control actuators should spatially match the primary sound field. The type of control actuators to be used will be determined primarily by the frequency of the disturbance and the magnitude of the disturbance at the actuator location (to simplify, the control actuators need to generate a secondary field with a magnitude equal to the disturbance at the actuator location). Additional information such as the mechanical impedance of the structure at the actuator location is also necessary, eg. to determine force requirements of an active mount. Once the type, number and locations of the control actuators are known, extensive transfer function measurements need to be taken between individual actuators and field points (vibratory or acoustic), with the primary source turned off. Since this may involve a considerable experimental task, numerical simulations can be of a great help here.

The third phase will address the error sensors. Again, if global control is desirable, the *type, number and locations of the error sensors* is dictated by the requirement that if the control actuators are driven to minimize the signal at the error sensors, then the resulting sound field is globally reduced. The measured transfer functions between individual actuators and field points and the magnitude of the primary disturbance at these field points are used together with classical exact quadratic optimization techniques to calculate the optimal control variables (i.e. the required inputs of the control actuators) that minimize the error signals for a given error sensor arrangement. This procedure can be used to investigate (by simulation) various error sensor strategies and select the ones that offer the best global attenuation.

The previous three phases allow the prediction (from experimental measurements on the system), of the best achievable attenuation for a given control actuators/error sensors configuration. The final phase will be to test the active control with a real controller.

## 5.2 Summary on Sensing and Actuating Technologies

Table 5.1 summarizes the advantages and disadvantages of the potential actuator and sensor technologies to be used for active control of a marine engine, in light of the discussions in Chapter 3. The next sections provide recommendations on the choice of sensor/actuator technologies for the various vibroacoustic paths considered.

Table 5.1: Sensor and Actuator Technologies

| **Type of sensor or actuator** | **Advantages** | **Disadvantages** |
|---|---|---|
| **Piezoelectric Materials**<br><br>eg. Lead Zirconate Titanate (PZT) Polyvinylidene Fluoride(PVDF) | ▪ Used as sensors and actuators<br>▪ Very large frequency range<br>▪ Quick response time<br>▪ Very high resolution and dynamic range<br>▪ Possibility of Integration in the structure for thin PZT actuators and PVDF sensors<br>▪ Possibility of shaping PVDF sensors (spatial filtering) | ▪ Relatively low strain and low displacement capability ( typically, less than 0.1% strain, and 1-100 microns displacement for stack actuators)<br>▪ Actuators require relatively costly voltage amplifiers<br>▪ Low recoverable strain (0.1%)<br>▪ Piezoelectric ceramics are brittle<br>▪ Cannot measure direct current<br>▪ Susceptible to high hysteresis and creep when strained in direction of poling (e.g. stack actuators) |
| **Electrostrictive Materials**<br><br>eg. Lead-Magnesium Niobate (PMN) | ▪ Used as sensor and actuators<br>▪ Lower hysteresis and creep compared to piezoelectric<br>▪ Potentially larger recoverable strain than piezoelectric | ▪ More sensitive to temperature variations than piezoelectrics |
| **Magnetostrictive Materials**<br><br>eg. Terfenol-D | ▪ Higher force and strain capability than piezoceramics (typically 1000 microstrain deformation)<br>▪ Suited for high precision applications<br>▪ Suited for compressive load carrying components<br>▪ Very durable | ▪ Low recoverable strain (0.15%)<br>▪ Only for compression components<br>▪ Nonlinear behaviour |
| **Shape Memory Alloys (SMA)**<br><br>eg. NITINOL | ▪ Large recoverable strain (8%) Used largely for actuation due to large force generation<br>▪ Low voltage requirements | ▪ Suited for low frequency (0-10Hz) and low precision application<br>▪ Slow response time<br>▪ Complex constitutive behavior with large hysteresis |
| **Optical Fibers**<br><br>eg. Bragg grating, Fabry-Perot | ▪ Suited for remote sensing of structures<br>▪ Corrosion resistant<br>▪ Immune to electric interference<br>▪ Small, light, and compatible with advanced composite | ▪ Used for sensing alone<br>▪ Behavior is complicated by thermal strains |
| **Electrorheological Fluids (ER)**<br><br>eg. Alumino-silicate in Paraffin Oil | ▪ Simple and quiet devices<br>▪ Suitable for vibration control<br>▪ Offers significant capability and flexibility for altering structural response<br>▪ Low density | ▪ Low frequency applications<br>▪ Nonlinear behavior<br>▪ Cannot tolerate impurities<br>▪ Fluid and solid phases tend to separate<br>▪ Not suitable for low temperature applications<br>▪ High voltage requirements (2-10kV)<br>▪ Higher $\eta_p / \tau_y^2$ ratio than MR* |

| | | |
|---|---|---|
| **Magnetorheological Fluids (MR)** | ▪ Simple and quiet devices<br>▪ Quick response time<br>▪ Suitable for vibration control<br>▪ Offers significant capability and flexibility for altering structural response<br>▪ low voltage requirements<br>▪ behavior not affected by impurities<br>▪ suitable for wide range of temperatures<br>▪ lower $\eta_p / \tau_y^2$ ratio than ER* | ▪ Nonlinear behavior<br>▪ Higher density than ER |
| **Microphones** | ▪ Low cost<br>▪ Large dynamic range<br>▪ Excellent linearity | ▪ Sensitive to turbulent flow<br>▪ Need to achieve directionality in some active control systems (e.g. ducts)<br>▪ Need protection to dust, moisture, high temperature |
| **Displacement sensors**<br><br>eg. proximity probe, LVDT, LVIT | ▪ Good low frequency sensitivity (0-10Hz)<br>▪ Non-contacting measurement (proximity probe)<br>▪ Well suited to measurement of relative displacement in active mounts | ▪ Low frequency range (typically below 100Hz)<br>▪ Low dynamic range (typically 100 :1)<br>▪ Low resolution |
| **Velocity sensors** (magnetic) | ▪ Non-contacting measurement<br>▪ Well suited to measurement of relative velocity in active mounts | ▪ Low dynamic range (typically 100 :1)<br>▪ Low resolution<br>▪ Heavy |
| **Accelerometers** | ▪ Large dynamic range<br>▪ Excellent linearity | ▪ Low sensitivity in low frequency (0-10Hz)<br>▪ Require relatively expensive charge amplifiers (piezoelectric accelerometers) |
| **Loudspeakers** | ▪ Low cost | ▪ Nonlinear behavior if driven close to max. power<br>▪ Space requirement (backing enclosure)<br>▪ Need protection to dust, moisture, high temperature, corrosive environment |
| **Electrodynamic and electromagnetic actuators** | ▪ Relatively large force/large displacement capability<br>▪ Excellent linearity<br>▪ Extended frequency range | ▪ May need a large reaction mass to transmit large forces<br>▪ Space requirement |
| **Hydraulic and pneumatic actuators** | ▪ Large force/large displacement capability | ▪ Low frequency range (0-10Hz for pneumatic; 0-150Hz for hydraulic)<br>▪ Need for hydraulic or compressed air power supply<br>▪ Nonlinear behavior<br>▪ Space requirement |

## 5.3 Recommendations for Various Ship Noise Paths

### 5.3.1 Path 1: Active Vibration Isolation

In selecting sensors and actuators for active vibration isolation of engine noise, due consideration has to be given to the size and weight of the structure (engine) being isolated. Since the engine is a heavy structure weighing over 6000 kg, it is necessary that that the actuators are capable of delivering very high control forces. In addition, the nature of the noise through this path is non-acoustic and hence non-acoustic sensors and actuators have to be used. Based on these considerations, the following materials are recommended:

(i) Sensors: accelerometers and force transducers; and
(ii) Actuators: hydraulic actuators, electrodynamic actuators.

The recommendations are also summarized in Table 5.2. For increased efficiency, the control systems must be designed to provide control forces in translational and rotational directions, since engine vibrations could take place in all directions. Furthermore, the active control systems should be used in conjunction with passive control systems, to reduce cost as well to provide fail-safe designs.

### 5.3.2 Path 2: Active Control of Noise in Ducts and Pipes

All existing control systems for duct or pipe noise are based on a feedforward algorithm and it is recommended that this algorithm be used for the control of noise associated with a marine diesel engine where a reference signal is accessible (see Akpan et al, 1999). The general sensors and actuators configuration to be will depend on the excitation source and on the cut-on frequency of the duct or pipe. For ducts, generally associated with large cross sectional dimension, higher-order modes are more likely to exist, requiring a large number of sensors and actuators with an appropriate positioning strategy. For pipes, generally associated with small cross sectional dimension, it is expected that only plane wave propagation will exist, thereby limiting the number of elements needed to one sensor and one actuator.

Sensing strategies are proposed to get a representation of the acoustic field inside the duct or pipe either from pressure measurements inside the duct or from indirect vibration measurements. In this context, the following sensing configurations are possible:

- Microphones are to be used to measure the pressure field inside the duct or the pipe if air is flowing into them. Piezoelectric omni-directional microphones are recommended to attain the maximum possible noise reduction. In hot and corrosive gasses, special care has to be taken to protect the microphones. The microphone using Teflon insulation material and designed by SoftdB Inc. is then recommended

- Solutions involving the indirect measurement of the interior acoustic field using vibration measurements with piezoelectric sensors or accelerometers should also be considered.

The following actuating configurations are recommended for controlling duct or pipe noise:

- Loudspeakers are recommended for the application. Proper cone volume velocity has to be used, preferably by using multiple loudspeakers. As with microphones, special care has to be taken to protect the loudspeakers in hot and corrosive gasses. Protective coatings and water-resistant adhesives have to be used. A heat shield (Teflon membrane and perforated metal sheet) and a cooling system may also be required under extreme heat conditions.

- Motor-driven or inertial actuators mechanically connected either to the structure to be controlled or to an intermediate radiating structure may also be used.

The actuators can be used together with passive elements such as foams to improve the performance of the control system at high frequency. Tunable Helmholtz resonators can be used to attenuate slowly varying harmonic components in the disturbance field.

### 5.3.3 Path 3: Active Control of Vibration Propagation in Beam-Type Structures

As mentioned in Section 4.4.2, the general problem of controlling the vibration of a beam-type structure may involve the control of several types of waves: flexural, longitudinal and torsional waves. As flexural waves are associated with low frequency and large amplitudes, the control of the vibration of a marine drive shaft should address this problem. The preferred control algorithm for the vibration control of a propeller shaft driven by a rotating engine is the feedforward algorithm.

The general sensors and actuators configuration to be used in the vibration control of the shaft will depend on the excitation source and on the modal behaviour of the shaft.

*Modal control*

If the shaft is known to vibrate in a given mode, a low number of sensors and actuators can be used to measure the disturbance mode and inject the controlling mode. In this case, the sensors can be located either on the shaft itself or beside it using non-contacting sensors. The signals from sensors mounted on the shaft should be either conditioned using on-shaft electronics rotating with the shaft and transmitted to the control unit using telemetry techniques [Morin, 1996] or transmitted to a stationary unit using slip rings. The use of non-contacting sensors requires some sort of synchronization of the measurement with the rotation of the shaft and possibly the use of few sensors along an annular region in the vicinity of the shaft. The following sensing configurations are possible within this context:

- If the rotating speed is low enough, accelerometers mounted on the shaft can be used to measure the acceleration of the beam locally.
- Other potentially interesting mounted sensors include piezoelectric strain sensors.
- At low frequency and for large amplitudes, a good candidate for non-contacting sensors is the proximity probe to measure the displacement of the rotating shaft.
- Another sensing configuration involves the use of a contacting sensor mechanically linked to a bearing mounted on the shaft. The movement of this bearing is then measured using either accelerometers for high frequency or LVDT (or proximity probe) for low frequency.

As with the sensors, the actuators can be located either on the shaft itself or connected to it by a stationary mechanical link, eg. a bearing mounted on the shaft. The use of actuators mounted on the shaft requires the use of slip rings to transmit the electrical actuation power. The following actuating configurations are possible within this context:

- Potential mounted actuators include curved piezoelectric actuators (PZT) for strain generation.
- Magnetostrictive actuators for low displacement generation at high forces. An appropriate mechanical mounting could induce moments in the shaft.
- Actuators controlling the shaft using a stationary mechanical link can be electrical motors at low frequency (induction motors), electrodynamic shakers for large displacements or magnetostrictve actuator for low displacements.

*Wave transmission control*

If the excitation source can be localized, a control approach based on the wave transmission in the shaft is more appropriate. In this case, sensors and actuators arrays are required to measure the downstream propagating and evanescent waves and to inject the control waves in the structure. In this case, due to the precise measurements required to estimate the wave components, sensors and actuators mounted on the rotating shaft have to be used. Strategies similar to those presented before have to be used to transmit the measurement signals and the actuating power signals. The requirement for a larger number of sensors and actuators in this case also calls for compact elements, especially if they have to be mounted on a small portion of the shaft.

Mounted sensors to be used include:

- Piezoelectric (PVDF) sensors to measure the strain.
- Accelerometers, if the rotation speed permits, for acceleration measurement.

Mounted actuators to be used in this case include:

- Piezoelectric (PZT) actuators to induce strain in the structure.

To improve the robustness and performance of the controller, it is recommended that the actuators be combined with passive control elements such as a viscoelastic layer bonded to the shaft. In the case of a piezoelectric actuator, the viscoelastic layer can be located either beside it or under the PZT (ACLD) to protect the latter from excessive shocks.

### 5.3.4 Path 4: Active Control of Radiated Sound Fields

There are two types of radiated noise to be controlled for ship structures. These are the airborne engine noise into an enclosure, and the noise radiated by the noise into the sea. As stated in the companion document (Akpan et al, 1999), both cases require the use of global control techniques that involve multiple input and multiple output transducers. Control of radiated noise can be achieved either by active noise cancellation (ANC) or active structural acoustic control (ASAC) techniques.

For active cancellation the following sensors and actuators are recommended:

(i) Sensors: combination microphones, accelerometers
(ii) Actuators: loudspeakers

For active structural acoustic control the following sensors and actuators are recommended:

(iii) Sensors: piezoelectric sensors (shaped or not), accelerometers
(iv) Actuators: piezoelectric actuators, magnetostrictive actuators.

Table 5.2: Recommended Sensors and Actuators for Ship Noise Control

| SHIP NOISE PATH | RECOMMENDED SENSORS AND ACTUATORS | |
|---|---|---|
| | SENSORS | ACTUATORS |
| Path 1: Active Vibration Isolation | • Force transducers<br>• Accelerometers | • Hydraulic actuators<br>• Electrodynamic actuators |
| Path 2: Active Control of Noise in Ducts and Pipes | • Microphones<br>• Piezoelectric sensors<br>• Accelerometers | • Loudspeakers<br>• Electric motors |
| Path 3: Active Control of Vibration Propagation in Beam Type Structures | • Piezoelectric sensors<br>• Accelerometers<br>• Electrodynamic shakers<br>• LVDT | • Piezoelectric actuators<br>• Magnetostrictive actuators<br>• Electrodynamic shakers |
| Path 4: Active Control of Airborne Engine Noise<br>(a) Sound Field into Enclosures | • Combination microphones<br>• Accelerometers | • Loudspeakers |
| (b) Radiated Noise into Sea | • Piezoelectric sensors<br>• Accelerometers | • Piezoelectric actuators<br>• Magnetostrictive actuators |

# 6. SUMMARY, CONCLUSIONS AND RECOMMENDATIONS

## 6.1 Summary and Conclusions

This study arose from the need to control noise in naval vessels to reduce their detectability and hence vulnerability to enemy attack. In this report a detailed review of sensor and actuator technologies that could be used for active noise and vibration control in ship structures has been provided. The review focused on a wide range of sensor and actuator materials, such as piezoelectric and electrostrictive materials; magnetostrictive materials; shape memory alloys (SMAs); optical fibers; electrorheological and magnetoeheological fluids, microphones; loudspeakers; electrodynamic actuators; and hydraulic and pneumatic actuators.

The review also addressed the applications of these materials to ship structures and other large or massive structural systems. Particular attention was given to the diesel engine noise problem. In this regard, consideration was given to the noise transmission paths, namely, the engine mounting system; the exhaust stacks and piping systems; the drive shafts and mechanical couplings; and the air borne radiated noise. Sensor and actuator technologies that are suitable for active control of noise through these paths (or similar structural systems) were reviewed and detailed recommendations on sensors and actuators for the paths were provided.

In making the recommendations, due consideration was given to factors such as cost, frequency of the disturbance, operating (marine) environment, experience in other applications, ease of implementation, and the expected performance. In general, the following recommendations (as summarized in Table 5.2) were made:

(a) Non-acoustic sensors and actuators (such as accelerometers, force transducers, hydraulic actuators, piezoelectric materials, and electrodynamic actuators) were recommended for the non-acoustic paths, namely, the engine mounting system and the drive shafts and mechanical couplings; and
(b) Acoustic sensors and actuators (such as microphones and loudspeakers) were recommended for the acoustic paths, namely, the exhaust stacks and piping systems, and the air borne noise.

It was also recommended that the active control strategies be combined with passive treatments whenever possible, to increase the robustness of the control system and to provide a fail-safe design.

## 6.2 Recommendations For Future Work

In order to achieve the stated goal of noise control in ship structures, it is recommended that a combined experimental and numerical program be established by DREA to systematically implement the recommendations outlined in this study. Since the costs associated with this effort are expected to be high a pragmatic approach must be utilized. Such an approach should include the use of relevant on-going or prior DREA work; the use of data from friendly navies; and significant reliance on numerical modeling to reduce the costs associated with experimentation. However, it must be emphasized that experimental investigations must form part of any meaningful program.

Since the engine mount is by far the most important noise transmission path, attention must be focused on this path. A combined numerical and experimental study should be developed to implement the sensor, actuator and controller (see Akpan, et al, 1999) technologies recommended. DREA has been investigating numerical modeling of elastomeric mounts for vibration isolation. This study could be extended to include the effects of actuators and control in an integrated passive-active control methodology. The experimental component will address the implementation details and also serve to validate the numerical models.

As was highlighted in Section 4.1, the exhaust stack, fuel intake and cooling systems are also important noise transmission paths to be considered. In this regard, an investigation of the applicability of off-the-shelf pipe and duct noise control systems to the ship noise problem should be performed. Furthermore, numerical and experimental studies on the applicability of active structural acoustic control (ASAC) methods recommended for control of the drive shaft noise should be carried out, since ASAC methodologies are also suitable for global noise control.

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# SUPPLIERS OF SENSORS AND ACTUATORS

Suppliers of the various types of sensors and actuators are listed in Section A.1 to A.10 and the addresses and contact information on the suppliers are provided in Section A.11.

## A.1 Piezoelectric and Electrostrictive Materials

- Piezo Systems, Inc.
- Polytec PI, Inc.
- Sensor Technology
- Thunder
- PCB Piezotronics, Inc.
- DynaOPtic Motion
- Face International Corporation

## A.2 Shape Memory Alloys

- Shape Memory Alloy Applications, Inc.
- Midé Technology Corporation
- NITI ALLOYS TECHNOLOGIES, LTD.
- TiNi Alloy Company
- DYNALLOY, Inc.
- Special Metals Corporation

## A.3 Optical Fibers

- Canadian Marconi Company
- Electrophotonics Corporation
- F&S Inc.
- FISO technologies
- National Optics Institute
- Ocean Optics Inc.

## A.4 Electro-rheological Fluids

- Lord Corporation (Rheonetic Systems)
- TRW
- Bridgestone

## A.5 Microphones

- Brüel and Kjaer
- Shure Brothers Inc.

- Sennheiser Electronic Corporation
- DPA Microphones
- The Modal Shop Inc.

## A.6 Displacement and Velocity Transducers

- North Atlantic Instruments, Inc.
- Macro Sensors
- G. L. Collins Corporation
- Penny & Giles Controls Ltd
- G.W. Lisk Company, Inc.
- Sentech, Inc.
- Kavlico Corporation
- Lucas Control Systems (Schaevitz)
- Columbia Research Laboratories, Inc.
- Sensotec Inc.
- Trans-Tek Incorporated
- RDP Group
- Automatic Timing and Controls

## A.7 Accelerometers

- PCB Piezotronics, Inc.
- Endevco
- Columbia Research Laboratories, Inc.
- Sensotec Inc.
- Brüel and Kjaer
- The Modal Shop Inc.
- Kistler Instrumente AG Winterthur
- Lamerholm Fleming Ltd.
- Oceana Sensor and Electronics
- Entran Sensors and Electronics
- Orion Devices Ltd

## A.8 Loudspeakers

- Acoustic Research
- Altec Lansing
- B&W Loudspeakers
- Bose Corporation
- JBL
- KEF Audio
- Wharfedale

## A.9 Electromagnetic Actuators

- Rockwell Automation (Allen Bradley)
- Anaheim Automation
- API Motion Inc.
- BEI Sensors and Systems
- Bodine Electric Company
- DSI Magnetics CAD Group
- Franklin Electric
- Industrial Devices Co.
- Lucas Control Systems (Ledex)
- MagnetekRotech Electronics
- Northern Magnetics Inc.
- Pittman
- Servo Dynamics Corp.
- Trombetta Electromagnetics

## A.10 Hydraulic and Pneumatic Actuators

- CWS Group
- Eaton Corporation
- Fluid System
- Hydraulic Technologies
- Hydro-line Inc
- Lynair Inc.
- Mico Incorporated
- Parker Hannifin Corporation
- Sargent Controls
- Sheffer Corporation
- Vickers Inc.

## A.11 Address of Suppliers of Sensors and Actuators

**Active Control eXperts, Inc.**
215 First Street
Cambridge, MA 02142-1227, USA
Phone: (617) 577-0700
Fax: (617) 577-0656
www.acx.com

**Allen-Bradley**
**Rockwell Automation**
Halifax
21 Frazee Avenue
Dartmouth, NS B3B 1Z4
Phone: (902) 468-2454
Fax: (902) 468-3606
www.ab.com

**Anaheim Automation**
910 E. Orangefair Lane
Anaheim, CA 92801
Phone: (714) 992-6990
Fax: (714) 992-0471
www.anaheimautomation.com

**API Motion Inc.**
Amherst, NY 14228, USA
Phone: (716) 691-9100
Fax: (716) 691-9181
www.apicorporate.com

**BEI Sensors & Systems**
**Kimco Magnetics Division**
804-A Rancheros Drive
San Marcos, CA 92069
Phone: (760) 744-5671
Fax: (760) 744-8815
www.beisensors.com

**Brüel & Kjær**
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Skodsborgvej 307
2850 Naerum
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Phone: +45 45 800 500
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**Columbia Research Laboratories, Inc.**
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Woodlyn, PA 19094, USA
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Fax: (610) 872-3882
www.columbiaresearchlab.com

**DYNALLOY, Inc.**
3194-A Airport Loop Drive
Costa Mesa, CA 92626-3405, USA
Phone: (714) 436-1206
Fax: (714) 436-0511
www.dynalloy.com

**DynaOPtic Motion**
23561 Ridge Route, Suite U
Laguna Hills, CA 92653, USA
Phone: (800) 991-1420
(949) 770-9911
Fax: (949) 770-2492
www.dynaoptics.com

**Eaton Corporation**
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Cleveland, OH 44114, USA
Phone: (800) 386-1911
(216) 523-4400
Fax: (216) 479-7014
www.eaton.com

**Endevco**
Dalimar Instruments, Inc.
193, Joseph Carrier
Vaudreuil-Dorion, Quebec J7V 5V5
Phone: (514) 424-0033
Fax: (514) 424-0030

**Face International Corporation**
427 W. 35th Street
Norfolk, VA 23508, USA
Phone: (757) 624-2121

Fax: (757) 624-2128
www.face-int.com

**G. L. Collins Corporation**
5875 Obispo Avenue
Long Beach, CA 90805, USA
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Fax: (562) 633-9030
www.lvdtcollins.com

**G.W. Lisk Company, Inc.**
Clifton Springs, NY, USA
Phone: (315) 462-2611
www.gwlisk.com

**Hydraulic Technologies**
P.O. Box 672
585 N. Bank Lane
Lake Forest, IL 60045, USA
Phone: (847) 615.8290
Fax: (847) 615.8291
www.hydraulictechnologies.com

**Kavlico Corporation**
14501 Los Angeles Avenue
Moorpark, CA 93021, USA
Phone: (805) 523-2000
Fax: (805) 523-7125
www.kavlico.com

**Lord Corporation**
111 Lord Drive
P.O. Box 8012
Cary, NC 27511, USA
Phone: (919) 468-5979
www.lordcorp.com
www.mrfluid.com
www.rheonetic.com

**Lucas Control Systems**
**Ledex Actuation Products**
801 Scholz Drive
P.O. Box 427
Vandalia, OH 45377-0427, USA
Phone: (937) 454-2345
Fax: (937) 898-8624
www.ledex.com

**Lucas Control Systems**
**Schaevitz Sensors**
1000 Lucas Way
Hampton, Virginia 23666, USA
Phone: (757) 766-1500
Fax: (757) 766-4297
www.schaevitz.com

**Macro Sensors**
815 Hylton Road, Unit #8
Pennsauken, NJ 08110-1334, USA
Phone: (609) 662-8000
Fax: (609) 317-1005
www.macrosensors.com

**MICO Incorporated**
1911 Lee Boulevard (Zip Code 56003-2507)
P.O. Box 8118 North Mankato, MN
U.S.A. 56002-8118
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Fax: (507) 625-3212
www.mico.com

**Midé Technology Corporation**
56 Rogers Street
Cambridge, MA 02142, USA
Phone: (617) 252-0660
Fax: (617) 252-0770
www.mide.com

**North Atlantic Instruments, Inc.**
170 Wilbur Place
Bohemia, NY 11716-2416, USA
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Fax: (516) 567-1823
www.naii.com

**Northern Magnetics Inc.**
www.normag.com

**PCB Piezotronics, Inc.**
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Depew, NY 14043-2495, USA
Phone: (716) 684-0001

Fax: (716) 684-0987
www.pcb.com

**Penny & Giles Controls Ltd**
15 Airfield Road,
Christchurch, Dorset, BH23 3TJ
United Kingdom
www.penny-giles-controls.co.uk

**Piezo Systems, Inc.**
186 Massachusetts Avenue
Cambridge, MA 02139, USA
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Fax: (617) 354-2200
www.piezo.com

**Polytec PI, Inc.**
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Fax: (508) 832-0506
www.polytecpi.com

**Sargent Controls**
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Fax: (520) 744-9494
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**Sensor Technology**
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Collingwood, Ont., Canada
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**Sensotec Inc.**
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www.sensotec.com

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**Servo Dynamics Corp.**
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Fax: (818) 718-6719
www.servodynamics.com

**Shape Memory Applications, Inc.**
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Santa Clara, CA 95054, USA
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Fax: (408) 727-2778
www.sma-inc.com

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Fax: (315) 798-6860
www.specialmetals.com

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Fax: 860-872-4211
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This study arose from the need to control noise in naval vessels to reduce their detectability and hence vulnerability to enemy attack. In this report a detailed review of sensor and actuator technologies that could be used for active noise and vibration control in ship structures has been provided. The review focused on a wide range of sensor and actuator materials, such as piezoelectric and electrostrictive materials; magnetostrictive materials; shape memory alloys (SMAs); optic fibers; electrorheological and magneteoheological fluids, microphones; loudspeakers; electrodynamic actuators; and hydraulic and pneumatic actuators. The diesel engine noise problem was the focus of the study, and consideration was given to the noise transmission paths, namely, the engine mounting system; the exhaust stacks and piping systems; the drive shafts and mechanical couplings; and the air borne radiated noise. Sensor and actuator technologies that are suitable for active control of noise through these paths (or similar structural systems) were reviewed. Based on factors, such as cost, frequency of the disturbance, the operating (marine) environment, experience in other applications, ease of implementation, and the expected performance, detailed recommendations on sensors and actuators for the paths were provided. For the engine mounting system and the drive shafts and mechanical couplings, sensors and actuators, such as accelerometers, force transducers, hydraulic actuators, piezoelectric materials, and electrodynamic actuators, were recommended. On the other hand, acoustic sensors and actuators, such as microphones and loudspeakers, were recommended for the exhaust stacks and piping systems, and the air borne noise. It was also recommended that the active control strategies be combined with passive treatments whenever possible, to increase the robustness of the control system and to provide a fail-safe design. A systematic and pragmatic program, based on combined experimental and numerical investigations, was suggested in order to implement the sensor and actuator technologies recommendations made in the study.

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