ROBUST VIBRATION CONTROL OF CLAMPED PLATE USING PIEZOMATERIAL CERAMICS

A Thesis

submitted in partial fulfillment of the requirements for the award of degree of

MASTER OF ENGINEERING

IN

CAD/CAM & ROBOTICS

to

THAPAR INSTITUTE OF ENGINEERING AND TECHNOLOGY

(DEEMED UNIVERSITY)

PATIALA (PUNJAB) -147004

Submitted by

KANWAR PREET SINGH

UNDER GUIDANCE OF

Mr. J.S.SAINI
Lecturer
TIET, PATIALA

MECHANICAL ENGINEERING DEPARTMENT

THAPAR INSTITUTE OF ENGINEERING AND TECHNOLOGY

(DEEMED UNIVERSITY)

PATIALA-147004, INDIA

2006

CERTIFICATE

This is to certify that the Thesis report entitled "**Robust vibration control of clamped plate using piezomaterials ceramics"** submitted by **Mr. Kanwar Preet Singh,** in the partial fulfillment of the requirement for the award of the degree of **Master of Engineering in Mechanical (CAD/CAM and ROBOTICS) Engineering** to **Thapar Institute of Engineering and Technology (Deemed University), Patiala,** is a record of candidate's own work carried out by him under our supervision and guidance. The matter embodied in this report has not been submitted in part or full to any other university or institute for the award of any degree.

MR. J.S SAINI
Lecturer,
Department of Mechanical Engineering,
Thapar Institute of Engineering and,
Technology,
Patiala.

Countersigned by:

(Dr. S.K. MOHAPATRA)
Professor and Head,
Affairs, Mechanical Engineering Department.
Engg. &Technology
Thapar Institute of Engg. &Technology
Patiala (Punjab)-147004

(Dr. T.P. SINGH)
Dean of Academic
Thapar Institute of
Patiala (Punjab)-147004

ACKNOWLEDGEMENT

I express my sincere gratitude to my guides**, Mr.J.S.Saini,** Lecturer, Mechanical Engineering Department at Thapar Institute of Engineering & Technology, for their valuable guidance, proper advice, painstaking and constant encouragement during the course or my work on this thesis.

I feel very much obliged to **Dr. S.K Mohapatra,** Professor & Head, Department of Mechanical Engineering for his encouragement and inspiration for execution of the thesis work.

I am deeply indebted to my parents for their inspiration and ever encouraging moral support, which enabled me to pursue my studies.

I am also thankful to the entire faculty and staff members of Mechanical Engineering Department for their direct–indirect help and cooperation.

Dated: (Kanwar Preet Singh)

Roll.No.8048110

ABSTRACT

Active vibration control (AVC) is not very old, although the control theory started to develope quite earlier, its applications were limited to the processes, which were quite slow. Slow chemical processes is the example, where the control theory showed its effectiveness in early days. Due to the lack of fast, dedicated processors, control theory remained only in theoretical frame, for active vibration control problems. Also the AVC is an interdisciplinary problem, in which use of intrumentation and control field is extensively involved. So the practical utility of AVC remained in research domain only, for a long period of time. With the emmergence of embedded automation, fast processors became the practical reality. Some dedicated software firms, collaborated with hardware firms and extended the technology of 'computer based instrumentation' or 'computer based measurements'. Interdisciplinary gap is thus reduced. Using these 'computer based instrumentation' techniques along with 'real time engines', AVC emerged as a practical reality few years back. Also with the development of piezoelectric ceramics, 'Intelligent' or 'smart' structures using low cost sensors and actuators, mounted on the surface of the flexible structure, AVC came into existance. Combining all these technologies, the practical applications of AVC are now increasing day by day.

Several advanced applications, such as those in jet fighters, automobiles and spacecrafts, require structures that are highly strong, lightweight and possess high structural damping property. A great difficulty is faced in the design of such structures due to the fact that the reduction in weight results in low rigidity and poor vibration characteristics. Unless the vibration is effectively controlled, it may destabilize the system and may, very often, result in complete failure of the system. Therefore, there is need to develop structures that are equipped with suitable vibration control features. Smart structure technology using robust control system may provide a solution to this problem. These structures use piezoelectric materials, electro-rheological (ER) fluids and shape memory alloys as sensors and actuators for providing effective vibration control.

Active vibration control requires high sampling rate and fast processors for doing control. Using the computer based measurements active vibration control is now a practical reality.

A detailed literature survey carried out for setting objectives for the present work suggests further work on the development of an Robust controller for estimating the parameters of a flexible structure and then using these parameters for the design of an effective controller. So,

the present work deals with a rectangular steel plate modeled in a cantilever configuration with surface bonded with piezoelectric patches. The study uses ANSYS (vs 5.6) software to derive the finite element model of the steel plate. By using the results from ANSYS a single – input/single-output H_∞ controller is designed to suppress the vibrations due to the first two flexural modes of the plate. And it has been shown that the designed controller guaranties robust performance.

LIST OF FIGURES

LIST OF TABLES

Page

CONTENTS

CHAPTER

ONE INTRODUCTION

1.1 INTRODUCTION

Usually satellites and other space structures have low flexural rigidity and are lightly damped due to the presence of small structural damping. Besides, it is difficult to provide other forms of damping, such as air damping, in space. Therefore vibrations once produced grow to large amplitudes and take a lot of time to decay. This may leads instability and failure of these structures.

Passive dampers, due to their heavy weight and fixed parameters, are not suitable for these applications. Also, for large space structures mathematical modeling is difficult and differs from on-orbit behavior. Corrosion and other environment factors result in the change of structural characteristics at a slow rate. In certain cases the quick change in parameters can also occur due to sensor/actuator failure causing the long range of change in frequencies where the designer has to deal with lot of uncertainty. Thus all type of parameters must be taken in advance in these situations. Use of high loss factor visco-elastic materials is an important damping technique. However, these are effective only in small frequency range and are not adaptive to intelligent control. In active control, piezoelectric materials have been given special attention due to their high stiffness, low weight and wide frequency range of applications

Adaptive and robust controllers provide control signals to a structure in an attempt to cause the structure to exhibit a desired behavior. Together, the structure, the controller, and their interconnection comprise the system. The design of controllers is complicated by system instability which results in at least improper system operation, and possibly, significant damage to equipment and/or injury to people. Fortunately, a properly designed controller prevents the system from operating in a dangerous, unstable mode. In adaptive control technique, the system parameters are identified on line, and hence the controller parameters are modified. But in robust control design, the controller parameters are designed in such a way that the resulting controller is valid for a large range of system parameters.

It is, therefore, imperative that the controller be engineered with stable operation as a primary goal, performance is a secondary design consideration to be pursued only after stability is assured. Using robust control, a good control function can be maintained, confirming stability as well as optimal performance.

It is desired that once the vibration is induced, it should be damped out as quickly as possible. This task can be easily accomplished using higher control voltages. Since the control voltage that can be applied at the actuators is limited in amplitude and hence limited control energy is available for functioning of these controllers. Therefore, there is need to develop a controller which requires minimum control effort and provides minimum settling time to the system.

1.1.1 USED DEVICES

Technical Manufacturing Corporation has produced a device called **Stacis 2000**, based on active piezoelectric vibration control system.

When it comes to isolating highly sensitive equipment from troublesome vibrations, the TMC STACIS 2000 system stands alone. It offers the worlds most advanced vibration control solution. With isolation starting at 0.6 Hz, the Stacis 2000 system provides the widest band of vibration isolation available, capable of isolating sensitive equipment from vibration at any relevant frequency.

With the Stacis 2000 system, we can locate our sensitive machines and equipment virtually anywhere within a fabrication site without suffering the effects of minute vibration such as those from passing workers, air handling systems or nearby machinery, for example, chip manufacturers can produce higher-density chips and achieve higher yields, along with reduced process cycle times and lower cost per chip. This robust, advanced system is easy to use and simple to install.

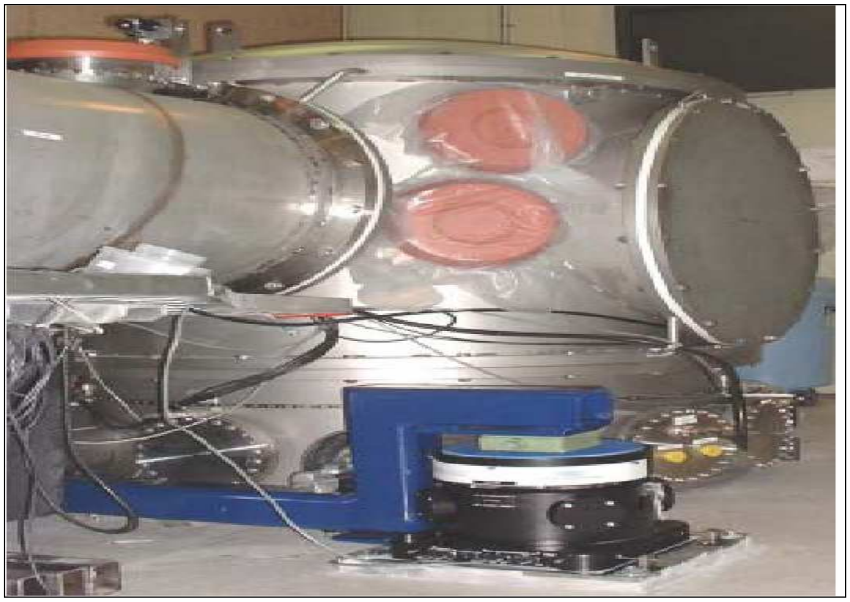


Figure 1.1 Vibration Control Device (STACIS 2000)

- **Design and Structure**

The revolutionary design of the Stacis 2000 system combines active vibration control based on piezoelectric actuators with proven passive isolation technology. The main components of the system are a user interface controller and three to four active, independent isolators positioned under the equipment they serve. Each isolator houses five piezoelectric actuator and a passive rubber mount 100 times stiffer than standard pneumatic isolators. The actuators control vibration along all three axes. The piezoelectric actuators receive information on disturbances through absolute velocity sensors. This information from the sensors causes the actuators to expand or contract as required, moving out of the way of vibration and diminishing its impact.

Stacis is ideal for supporting semiconductor inspection and manufacturing equipment both at the point of use and OEM applications

- **Features**

➢ Provides greater than 90% isolation in all axes at frequency greater than 2Hz.

- Has a "hard mount" feel which minimizes the effects of payload disturbances.
- Installs easily- robust control system requires minimal on-site adjustments.
- Maximizes uptime through extensive, easy-to-use diagnostics and its modular design.
- Supports computerized data acquisition and control from its user-friendly software interface.
- Non-magnetic, clean room compatible, and requires no air.

- **Benefits**

- Helps your equipment meet its resolution specifications.
- Stretches the limits of our old equipment.
- Increases throughput, quality, and yield.
- Reduces downtime through quick installation, minimal calibration and reliability.
- Meets tough vibration specifications of next-generation equipment.

1.2 BASIC CONTROL THEORY

Figure (1.2) shows the basic components of a typical robust control system. The structure, represented by system matrices **A**, **B** and **C**, is the device to be controlled. A controller, **K**, produces the control signal, **u**, which is used to modify the behavior of the plant. The plant output, **y** mimics an external, time-varying, input signal called the reference or excitation input, $\mathbf{u_x}$. Performance is measured by the error signal, **e**, which is the difference between the reference signal and the plant output: $\mathbf{e} = \mathbf{u_x} - \mathbf{y}$.

The vast majority of research in control theory is dedicated to systems which are linear time-invariant (LTI). The simple mathematics of LTI systems enables the application of the mature and extensive body of linear systems theory. Consequently, the design of stable controllers is straightforward. With the passage of time, as the system parameters **A**, **B** and **C** are changing, the robust controller which is designed considering all parameters in advance calculates these system matrices and calculates the control gains **K** for optimal performance of the system.

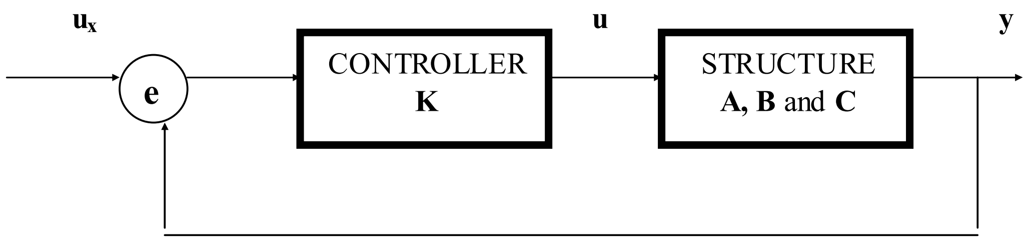


Figure 1.2 Robust Vibration Control

1.3 OVERVIEW OF SYSTEM AND CONTROL SYSTEM DESIGN

A significant amount of the current research activity in the field of structural mechanics has focused on the study of smart, intelligent structures. Specific studies have considered definition, analysis and synthesis of smart structures. The long term objective of the design of a smart and intelligent structure is to develop a structural system that can execute selected functions like an intelligent person. There objective, however, is to be able to design a structural system that can be described as adaptive structure. These adaptive structures have their own sensors, detection circuits, controllers and actuators. The sensors and actuators are made of a class of materials known as ‘smart materials’. Piezoceramic materials, Polyvinylene Difluoride (PVDF) films, shape memory alloys, electrorheological fluids, fiber optic devices and electro restrictive devices can be considered in the class of smart materials.

1.3.1 VIBRATION CONTROL

To reduce the amplitude of vibrating structures, two types of control strategies are possible; first by using dampers or shock absorbers; the vibration amplitude can be reduced, in a short span of time. By using the layers of viscoelastic materials, the same effect can be produced. Composite materials made by sandwiching the layers of viscoelastic materials between the layers of parent material, are quite common. Reducing the vibration amplitude by this technique is called **Passive control**.

The second type of control is obtained by using the structure mounted with smart materials. Due to the vibrations, the strain is developed in smart sensors fixed on the surface of the structure. This strain produces electrical voltage in these materials. This voltage is used as sensor signal. Using the information of the sensor voltage, suitable feedback electrical voltage is applied to the actuators made of smart materials. The applied actuator voltage produces strain in opposite direction and reduces the vibration amplitude. This type of vibration control is called **Active Control** (AC).

Due to the heavy weights of the passive controllers and their fixed parameter nature, these are not suitable for lightweight structures. Active vibration control finds extensive usage for these types of light weight structures. Applications of these structures using AC have been in the following fields:

- Vibration control of flexible structures
- Optical surfaces maintenance
- Improvement in performance of systems such as aircrafts, helicopters and space vehicles by changing the shape and configuration.

1.3.2 SMART MATERIALS AS SENSOS AND ACTUATORS

“**Piezoelectric (PE)”** materials are those materials which when subjected to electric field get deformed and vice – versa. This change in configuration may be used to nullify the effect of vibrations induced by external disturbances. These are made by the combination of three elements i.e. lead, zirconium and titanium. Electrical properties are shown in table (1.1).

Smart materials play the main role in above mentioned applications. Piezoceramic materials are less costly and have higher electromechanical coupling, therefore, these are widely used as smart, distributed sensors and actuators. Piezoelectricity offers many advantages for active vibration excitation and isolation over a wide frequency range, especially for small amplitudes. Most conventional piezoelectric materials such as Barium Titanium Oxide, Lead Zirconium Oxide and Quartz are, however, brittle and difficult to fabricate in complex shapes. This has limited their applications up to some extent. Polyvinylidene fluoride (PVDF) is a piezo-polymer with unique properties that make it attractive for active vibration control applications. PVDF is tough, pliant and can be easily formed in sheets and complex shapes. Lead Zirconiated Titanate (PZT) is another ceramic with favorable mechanical and electrical properties.

TABLE (1.1) ELECTRICAL PROPERTIES OF PZT

Property		**Value**
Piezoelectric charge constant (m V^{-1})		$171x10^{-12}$
Piezoelectric charge constant(m V^{-1})		$171x10^{-12}$
Poisson's ratio		0.28
Permittivity (Fm^{-1})		$106x10^{-12}$

1.3.3 MATHEMATICAL MODELING

To study the effectiveness of control algorithms, accurate modeling of electro – elastic interactions of piezoelectric sensors and actuators with substrate is necessary.

- **Mathematical Modeling by Finite Element Methods**

With the increasing complexities of the structures and emergence of digital computers, the need was felt to develop the numerical solution of the static and dynamic problems. With this, the techniques of Finite Element Methods emerged. The subtle idea behind this was to explore the power of modern digital computer for irregular shaped products. In the finite element method, the equations of motion are obtained by deriving first the element equations of motion for a particular element and then assembling the equations for all the elements.

Any framed structure, vibrating in a single plane can be assumed to be made of 2D beam or frame elements. Each of these elements may be loaded with axial as well as transverse forces, along with bending moments. At each axial end of the element, called a node, there are horizontal and vertical deflections and slope.

Exact closed form solution of these dynamic problems is limited to extremely simple structures. Very simple cases of geometry and boundary conditions can be accommodated in this method. Finite Element Methods (FEM) can be used for analyzing the static and dynamic performance of intelligent structures. FEM analysis is normally based on first order shear deformation theory, Damle et al. (1997). However, such models suffer from the problem of

Spurious shear stiffness, which need special consideration. So there is a need for improved FE approximation techniques for analyzing ‘Intelligent’ structures of general configurations. When ordinary FEA techniques are used for laminated composite plates with piezoelectric layers acting as sensors and actuators, the theoretical results deviate from experimental results. Also a lot of computational effort and time is needed in modeling these structures. Due to manufacturing inaccuracies, non – linearity, non – isotropy and structure degradation; theoretical results given by FE analysis deviate from experimental results.

- **System models**

To properly understand the system theory, different system models must be studied. Various system forms, normally used in control theory are discussed in following:

(a) System Categorization Based On Time

Depending upon the relationship between time and system parameters, following two categories are possible

(i) **Time Varying Systems:** These are the systems whose parameters vary with the passage of time.

(ii) **Time In-varying Systems:** These are the systems whose parameters do not vary with the passage of time.

(b) System Categorization Based On Input – Output Relationship

Depending upon the relationship between input and output, following two categories are made

(i) **Linear Systems:** Outputs are linear function of inputs

(ii) **Nonlinear Systems:** Outputs are related with inputs by non-linear relationships.

To deal with different sub-categories of the system, respective mathematical treatments are developed. In normal practice, the linear and time invariant systems are encountered. These are combined together to form a class of systems called Linear Time Invariant **(LTI)** systems.

(c) System Categorization Based On Number Of Input/Outputs

Depending upon the number of inputs and outputs, following categories are made

(i) SISO: i.e. Single Input Single Output Systems

(ii) SIMO: i.e. Single Input Multiple Output Systems

(iii) MISO: i.e. Multiple Input Single Output Systems

(iv) **MIMO:** i.e. Multiple Input Multiple Output Systems. These systems are also called Multivariable Systems.

(d) System Categorization Based On Continuity

Based on the form of data available, following two categories are made

(i) Continuous Systems: The data is continuously available from the sensors regardless of the sampling frequency. This means that at each instant of time t, the signal is available.

(ii) Discrete Systems: Using the modern digital computers it is easy to handle the system in discrete form. Even if it is available in continuous form, it is possible to convert it into discrete form. In this case the signal is available at certain discrete steps of time. i.e. at $t_k=t_{k-1}+\Delta t$, where Δt is the sampling interval. Discrete system is easy to handle and a lot of mathematical background is available.

(e) System Categorization Based On Structure Of Parameters

Depending on the structure of system parameters and the relationship with each other, following two categories are made:

(i) Transfer function form

The dynamic SISO LTI system in continuous form can also be presented in transfer function form. These transfer functions are presented as the ratio of two polynomials. Writing in mathematical form

$$H(s,P) = \frac{N(s,P)}{D(s,P)} = \frac{\sum_{k=0}^{n} \alpha_k s^k}{\sum_{k=0}^{n} \beta_k s^k} = \frac{\alpha_0 s^0 + \alpha_1 s^1 + \dots\dots\dots + \alpha_n s^n}{\beta_0 s^0 + \beta_1 s^1 + \dots\dots\dots\dots + \beta_d s^d}$$

Where s is the Laplace complex number; α and β is the coefficients of system outputs and inputs respectively. This forms the part of classical control theory. Various control techniques like pole zero placement, root locus method, etc., can be used for control implementations of the system in this form. However using certain transformations, these continuous systems can be presented in discrete form given below

$$H(z,P) = \frac{N(z,P)}{D(z,P)} = \frac{\sum_{k=0}^{n} \alpha_k z^k}{\sum_{k=0}^{n} \beta_k z^k} = \frac{\alpha_0 z^0 + \alpha_1 z^1 + \dots\dots\dots + \alpha_n z^n}{\beta_0 z^0 + \beta_1 z^1 + \dots\dots\dots\dots + \beta_d z^d}$$

Where z is a shift forward operator.

For MIMO systems, these polynomials form the part of a matrix. This type of matrix is called Polynomial Transfer function matrix. But as the number of inputs and outputs are increased, it becomes quite difficult to handle the system in this form. To avoid this problem for MIMO systems, state space models are quite useful and are presented below

(ii) State - Space form

Any continuous, LTI system can be written in the following form, Mohammed et al. (1994)

$$\mathbf{X}(k+1) = \mathbf{A}\,\mathbf{X}(k) + \mathbf{B}\,\mathbf{u}(k)$$
$$\mathbf{y}(k) = \mathbf{C}\,\mathbf{X}(k)$$

Where **X** (k), **u** (k) and **y** (k) is the state vector, input vector and output vector at kth instant of time respectively. **A**, **B** and **C** are called state coupling matrix, input coupling matrix and output coupling matrix respectively. Using the standard methods, Meirovitch (1986), it is easy to transform these methods to discrete form.

(iii) Difference Equation Form

Although state space models are best suited for control applications, the identification in this form is comparatively complex. Complexity increases tremendously as the system order is increased. Difference equations are the best form, in which identification can be carried out from experimental data. Certain methods are available in literature which can be used to transform the system from state-space form to difference equation form and vice versa. So it is desirable to identify the system in difference equation form and then convert this into state- space form. The difference equation form can be written as

$$\mathbf{I}\,\mathbf{y}(t) + \mathbf{A}_1\,\mathbf{y}(t-1) + \ldots\ldots + \mathbf{A}_n\,\mathbf{y}(t-n) = \mathbf{B}_0\,\mathbf{u}(t) + \mathbf{B}_1\,\mathbf{u}(t-1) + \ldots\ldots + \mathbf{B}_n\,\mathbf{u}(t-n)$$

The matrices $\mathbf{A}_0$, $\mathbf{A}_1$, $\mathbf{A}_2$,……., $\mathbf{A}_\mathbf{n}$ are the coefficient matrices of the output vectors and $\mathbf{B}_0$, $\mathbf{B}_1$, $\mathbf{B}_2$, ……., $\mathbf{B}_\mathbf{n}$ are coefficient matrices of the input vectors.

(f) System Categorization Based On Feedback

Depending upon the presence or absence of the control signal, following two categories are made:

(i) Open- loop Systems and Identification

Any natural system can be presented in terms of input output relationships. These parameters represent the system in parametric form. This form is quite useful in control applications. Since the free excitation of the system is normally possible, the flexible structures can be excited with specially designed input signals for system identification of these structures. The only condition for free excitation is that the system should not turn unstable. So for open-loop identification, system stability is the first requirement. Stable systems are those systems which turn to their equilibrium position after excitation. From the output data thus obtained, parametric identification can be made. This is called open-loop identification or offline- identification.

(ii) Closed -loop Systems and Identification

Some times the systems are not stable i.e. if they are disturbed from their equilibrium position, they never return to their original position. So these systems are always run under feedback conditions. Using specially designed excitation signal along with feedback signal, system parameters can be identified from input-output data by using various techniques Ljung (1999), in closed-loop conditions. Some times system parameters are time varying. In these situations, it becomes necessary to identify the changed parameters in the presence of the control action. This type of identification is called closed-loop identification or online- identification.

CHAPTER

TWO LITERATURE REVIEW

2.1 GENERAL INTRODUCTION

The extraction of information from observations of the surrounding world is of primary importance for the design and control of a robust system. Often this information is used to develop theories for describing and understanding reality rather more accurately. The process of observing and modeling is the key to the present technical culture. The selection of one out of a class of possible models and determination of the values of the unknown model parameters using a limited amount of information is the aim of system identification and parameter estimation.

Smart structures use piezoelectric materials, electro-rheological (ER) fluids, magneto-rheological (MR) fluids and shape memory alloys as sensors and actuators. Results of several investigations are available on the use of such materials for vibration control. The literature on **Robust Vibration Control of Structures using Piezoelectric Ceramics**, in general, may broadly be divided into two main categories, viz. **piezoelectric sensors/actuators for vibration control** and the other is **robust**

vibration control. A brief survey of literature falling under these categories is presented in the following paragraph.

2.2 ROBUST VIBRATION CONTROL OF FLEXIBLE STRUCTURES USING PIEZOELECTRIC CERAMICS

The literature on system identification and robust control theory forms the backbone of instrumentation and control engineering. It is very difficult to discuss these in such a small space. However, to discuss the applications of these theories for vibration related problems is manageable.

2.2.1 VIBRATION CONTROL OF CONTINUA: PIEZOELECTRIC FINITE ELEMENT FORMULATION AND ANALYSIS

Normally the structures used in aircrafts and spacecrafts are made lightweight for the optimal use of limited available energy. Due to this lightweight, however, inherent damping is decreased. As a result, these structures once excited may go on vibrating for a long time and in several cases this may result into instability. Active control of these structures becomes necessary. Using the powerful finite element methods the structural modal parameters can be identified. But, unfortunately, these estimated parameters differ from the on-orbit conditions. Therefore, there is a need to develop methods, for estimation of parameters as well as for finding the relation between input-output from the data collected for on-orbit conditions, so that the same may be used for control of vibration.

Because of the limitations of the passive control with the help of dampers there emerged a need to develop active vibration control. Several authors applied the linear control theory to the vibration reduction problem of cantilever, beams, plates and shells. For these simple structures it is easy to find the transfer function of the system, so conventional control laws could easily be applied. But for complex structure transfer function is not easy to find with mathematical modeling and when it could be obtained it may have serious errors.

Conventional elastic continua usually are “**passive**” in nature, i.e. they do not process any sensation and action /reaction capability. In recent years, “**smart**” continua with integrated sensors/actuators which provide self-monitoring and control capabilities are of importance due to the rapid development of “**smart**” or “**intelligent**” space structure and mechanical systems (Tzou and Grade,1989; Crawly and De Luis,1987).Since these continua are, in general ,distributed and flexible in nature ,distributed dynamic measurement and active vibration suppression are essential to their high-demanding performance ,e.g. precision ,accuracy , shape control etc.

In 1991 Tzou et.al developed the theory on **Distributed Modal Identification and Vibration Control of Continua**: In this theory, Continua (shells or plates) integrated with distributed piezoelectric sensors and actuator has been studied using a finite element technique. Thin piezoelectric layers are coupled with conventional elastic continua and are used as distributed sensors and distributed actuators in “smart” continua.

In his theory, Tzou had taken the continuum as a general elastic shell or plate configuration with one piezoelectric layer servicing as a distributed sensors and the other layer servicing as distributed actuator. The direct effect is used in distributed sensing and the converse effect in distributed active vibration control .The piezoelectric material used in the finite element analysis of the smart continua is a 3-phase polymeric piezoelectric **Polyvinylidene fluoride** (PVDF). Mono- and bi-axially oriented piezoelectric actuators were used with the plate and their control efficiencies has been evaluated and compared. It has been observed that the damping ratio, in general, increases when the feedback voltage /gain increases.

In 1987 Plump et al, studied the use of the flexible piezoelectric PVDF in vibration control and active damping of beam structures. Crystalline piezoelectric materials were also investigated in a longitudinal actuation (Fanson et al. in 1988) and a distributed sensors /actuators for beam structures (Crawley et al. in 1987).

In 1987 pandita et al. applied the **direct piezoelectric effect**, a charge/voltage generated by an imposed force/pressure to a piezoelectric, to a variety of transducers designs, e.g., accelerometers, pressure transducers etc.The **converse piezoelectric effect**, an induced stress/strain due to an externally applied voltage/charge, has been applied to flexible mirrors, piezoelectric exciter/vibration isolator by Grade et al in 1990.

It may be noted that normally the flexible structures are not always so complex. As an example, one or two link robotic arms, which change their parameters with the passage of time due to various changing conditions, are not that complex. Therefore, there is need of algorithms, which are less computationally burdened and are suitable for real-time implementation. Copper et al. (1990), checked the capability of various time domain algorithms like "**Least-Square Method (LS),**" "**Double Least-Square Method (DLS)**", "**Instrumental Variable Method (IV)**", **"Certainty Factor (CF)"** and "**Maximum Likelihood (ML)"** method for estimating the time varying modal parameters. With simulation results the authors concluded that DLS and ML methods give genuine results from noisy sensor data. They also proposed a method to calculate the modal parameters from "**Infinite – Impulse Response (IIR)"** filters like co-efficient of **"Auto Regressive Moving Average (ARMA)"** model. The entire work of Cooper was based on the idea that it is difficult to find the on-line solution of "**Eigenvalue Problem (EV)"**; because the normal solution of EV problem is computational extensive. Because of the noise present in the measured data the statistical methods were used in conjunction with ERA. Copper (1990) extended his own work and used data co-relation methods with ERA to remove the affect of presence of noise, for estimating the modal parameters.

Based on the difficulties present in time domain, like un-consistent sampling rate, under modeling, etc, Tzes et al. (1990), extended the work towards frequency domain identification techniques. They used input-output relationship instead of modal characteristics or modal parameters for a simple cantilever beam using displacement sensor at free end and an electric motor at the fixed end. The "**Time Varying Transfer Function Estimate (TTFE)"** is obtained from the spectra of output and input. **TTFE** updates the frequency component in time domain through recursive - least square method. Then the time varying poles and zeros are calculated to design the controller. Almost on parallel lines Bayard et al. (1991), used frequency domain identification technique for large space structures with multiple sensors and multiple end thrusters. Authors identified the system non-parametrically by correlation and spectral estimation techniques using windowing techniques. Then by complex curve fitting technique parametric transfer function in the form of IIR filter was estimated from non-parametric data. Model order is determined by successively increasing the number of modes so that there is minimum error between obtained data and produced curve. For these purposes **"Product Moment Matrix (PMM)**" was used. The natural frequencies and damping ratios were calculated from the denominator of the polynomials. The author also noted that correlation techniques are not quite suitable for structural vibration problems due to the presence of low frequency modes and low sampling rates.

In the next contribution to the field of ERA, Copper (1992) used recursive form of EV to the on-orbit spacecrafts, for identification of modal parameters with reduced computational burden. Afterwards Wimmel et al. (1992) used adaptive filters to estimate the system parameters. Because of the recursive nature these were capable of on–line identification purposes.

With the development of **"Neural Network (NN)"** technology, some researchers used **NNs** for system identification and control. Snyder et al. (1993), used neural network and adaptive algorithm suitable for driving on-line feed forward control systems with smart structures to actively control sound and vibration. The neural network/ LMS algorithm combination was viewed as an extension to the commonly used transversal filter/filtered -x LMS algorithm. In combination, the neural network was essentially a transversal filter with a non-linear hidden layer placed between the input delay chain and output accumulator. The algorithm accommodated the transfer function and the time delay between the control signal output and error signal input. The arrangement was shown, in simulation, to be capable of suppressing a primary disturbance, which was a non-linear function of the reference signal fed to the control filter, and of suppressing harmonics introduced into a system by a control actuator exhibiting non-linear performance characteristics.

With the development of **"Neural Network (NN)"** technology, some researchers used NN based hardware for system identification and control. Yen (1997) used **frequency domain** NNs for identification of complex structures. Afterwards, Yen (1997) and Damle et al. (1997), developed **"Electronically Trainable Analog Neural Networks (ETANN)"** chip for robust control of cantilever plate with PZT patches as actuators and PVDF film as sensors. Controller dynamics is copied into the chip, which is then used to control the system

Luzardo et al. (1997), used a new **"Neural Network Controller (NNC)"** whose parameters were adjusted on line, to control a class of multivariable linear systems. The plant to be controlled was assumed to be square (p-inputs, p-outputs). The NNC was applied to a linear model of a large segmented space reflector and the authors presented simulation results.

A theory on **multi-layered shells** coupled with piezoelectric shells actuators was derived and evaluated by Tzou and Grade in 1989.Distributed active vibration control of a shell coupled ,with PVDF was investigated by Tzou and Tseng in 1988. New distributed sensing/control theory for a generic **shell continuum** was also proposed and evaluated by Tzou in 1991.Isopaametric piezoelectric finite elements were develop and applied to piezoelectric transducers designs by nailon et al. in 1983.

In 1991 E.K.Dimitriadis developed the theory for the excitation of **two-dimensional thin elastic structures** by **patch type piezoelectric actuators** bonded to the structure surface. And the theory has been briefly applied to excitation of a simply supported rectangular thin plate by a single rectangular actuator consisting of two piezoelectric actuators bonded symmetrically to both sides of the plate .The results demonstrated that modes can be selectively excited and that the geometry of the actuator shape markedly affects the distribution of the response among modes. Also the location of the actuator strongly influences the ability of the actuator to excite certain modes as well as the spillover.

A rigorous study of the **stress/strain- voltages** behavior of piezoelectric elements bonded to and imbedded in one dimensional beam has been done by Crawley et al. in 1987 ,and analyzed the stress, strain and loads generated on the cantilever beam when piezoelectric segments were bonded symmetrically on both sides . A number of important results were demonstrated such as increased effectiveness of the actuator for stiffer and thinner bonding layers, as well as for stiffer piezoelectric material. And also observed that the effective moments resulting from the piezoactuators can be seen as concentrated on the two ends of the actuator when the bonding layer is assumed infinitely thin.

In 1987 Baz et al. performed significant research on the use of the piezoelectric actuators,they work, on control of motion in beams, and have demonstrated the potential of piezoelectric as distributed vibration actuators by simultaneously controlling a number of modes with reduced spillover.

The Eigenvalue problem is computational extensive for small sized computers, generally used in spacecrafts, with limited computational capability. In such a situation some type of recursive algorithm with less computational burden would be better suited. Keeping this into mind Longman et al. in 1989 proposed an algorithm for **recursively** calculating the **Minimum Realization** of a linear system from sampled impulse response data. This technique was used to generate an orthonormal basis for factorization of modal parameters, i.e., natural frequencies, mode shapes and damping factors. This method is fast when compared with the simple Eigen system realization method proposed by Pappa et al. in 1985.

Bayard et al. in 1994 developed multivariable frequency domain curve fitting for multivariable flexible structures. This algorithm is specially tailored to the identification of complex systems having large number of parameters. This algorithm is well suited for open-loop identification of lightly damped systems. But the present system is limited for offline identification purposes. Also the main limitation of this is that it can be used for frequency domain data.

In 1996, Butler developed a system identification technique for multivariable smart structures based on the measurement of eigenvalues and eigenvectors of the given structure using analog linear quadratic regulator. The identification procedure utilizes n – measurement variables of the structure system with n/2 modes to produce an n-th order model. A sensor array consisting of shaped and segmented **"Polyvinylidene Fluoride Film (PVDF)"** was used as a practical measurement device. Author presented a procedure for generation of distributed sensors for the state-variable measurement. The identification and control were successively implemented on a multivariable cantilever plate.

Extending the work on frequency domain system estimation for vibrating structures Ashokanathan et al. (1996), developed an on – line frequency domain identification technique for a slewing cantilever beam of varying tip loads. Authors developed the transfer function estimation similar to the one by Bayard (1991) and developed the state-space representation of the IIR filter to design the controller. Frequency weighing was applied for proper curve fitting at low as well as high frequencies.

Callafon et al. in 1996 presented an approach to estimate a linear multivariable model on thebases of frequency domain data via a curve fitting procedure. The multivariable model was presented in either a left or a right polynomial fraction description and the parameters were computed by using a two norm minimization of a multivariable output error. The procedure was demonstrated on experimental data obtained from a three input three output Wafer Stepper system.

2.2.2 ROBUST VIBRATION CONTROL BY A CONTROLLABLE CONSTRAINED DAMPIG LAYER

Normally the structures used in aircrafts and spacecrafts are made lightweight for the optimal use of limited available energy. Due to this lightweight, however, inherent damping is decreased. As a result, these structures once excited may go on vibrating for a long time and in several cases this may result into instability. Active control of these structures becomes necessary. Using the powerful finite element methods the structural modal parameters can be identified. But, unfortunately, these estimated parameters differ from the on-orbit conditions. Therefore, there is a need to develop methods, for estimation of parameters as well as for finding the relation between input-output from the data collected for on-orbit conditions, so that the same may be used for **robust control of vibration.**

In 2001, S.L.Xie et al. investigated the theory for the robust vibration control of the thin plate (a type of thin-walled structure) covered with a controllable constrained damping layer. The thin walled structure covered with a controllable constrained damping layer is made up of four layers: the **piezoelectric laminate**, the **elastic cover sheet**, the **viscoelastic damping layer**, and the **structure under control**, respectively. For such a complicated multilayer structure, it is difficult to describe accurately its electromechanical dynamic characteristics, which leads to uncertainties between the theoretical model and the actual structure. In the past research on the controllable constrained damping layer technique, only displacement proportional feedback strategy has been investigated. Unfortunately, this strategy may cause spillovers of higher-frequency modes of the structure and thereby lower stability of the control system due to lack of the ability to treat the uncertainties such as model errors and external disturbances.Therfore, it seems that **robust control strategy** should be considered in the controllable constrained damping layer technique to improve the stability of the control system.

The uncertainties between the theoretical model and the actual structure are analyzed by means of numerical computation and experimentation, which lays the foundation of robust control design .The mixed –sensitivity method in **H∞** control theory is thereafter employed to synthesize the robust vibration controller, where the disturbance/output characteristic are considered and the structured and unstructured uncertainties are all treated as the additive uncertainties of nominal plant. The robust stability and control performance of **H∞** control are verified experimentally and compared with those of displacement proportional feedback control.

There results indicate the effectiveness of **H∞** control to attenuate modal vibration and reject modal spillovers, which is thereafter validated experimentally. In contrast, the displacement proportional feedback control causes significant spillovers of the truncated modes because of the lack of the ability to consider the model uncertainties. Therefore, **H∞** control can provide more **robust stability** then those by the displacement proportional control.

Constrained damping treatment with viscoelastic materials has been widely used in passive vibration control of thin-walled structures in the past. It can effectively suppress the structural vibrations in the high-frequency domain but has less effect on those in the low-frequency domain due to damping characteristics of the viscoelastic materials. In addition, this technique lacks the flexibility to various external loads.

To overcome the above shortcomings the hybrid damping techniques that combine active control with constrained damping treatment have been fully investigated by Baz et al. in 1997.Shen et al. in 1994, 1997has proposed a hybrid control method called **the active constrained damping layer**, where the piezoelectric layer serves as both the cover sheet of the **viscoelastic damping layer** and **the actuator**. An active constrained damping layer can introduce active control by the use of the inverse piezoelectric effect of the piezoelectric layer and may consequently control the low-frequency vibration of the structure well. However, it suffers from such disadvantages as the requirement of a piezoelectric layer with bigger area, difficulty in applying distributed electric field and low economic efficiency in vibration control of larger structures.

Based on the classical techniques of adaptive control, Ishitobi et al. (1991), developed MRAC mixed with Delta Operator to neglect the effect of unknown disturbances on the system. Considering the non- linear characteristics of the vibrating structures and, in general the non-linear properties of the actuators Bozich et al. (1991), used neural networks for non-linear control of the vibrating structures. Since the disturbance is a non-linear function of reference signal fed to the control filter and also since the actuators exhibit non-linear characteristics, the linear adaptive control strategy gives poor results. Knospe et al. (1992), used pulse response formulation of acceleration measurements to estimate the parameters of linear controller for vibration suppression. By taking into account the statistical nature of the disturbance Pan et al. (1992), developed a method in which reference signal correlated with disturbances, calculated the parameters of the feed-forward controller for predicting and canceling the effect of disturbance. They used the LMS algorithm, and the controller was designed on the principle of time varying system.

For multi-input systems using several numbers of actuators, there are some times, the problem of failure of some of the actuators. Taking this into consideration, Flowers et al. (1992) used adaptive centralized control method for vibration control. Authors integrated various sub-systems with known parameters but unknown inter-connections. The main advantage of this system was that number of actuators can be decreased or increased without redesigning the whole system. Snyder et al. (1992), developed **"Multi-Input Multi-Output (MIMO)"** systems with multiple sensors and actuators. They developed adaptive control system based on the LMS algorithm and used mean square error as a convergence criterion.

With the emerging trend of neural networks, several other investigators used neural networks for vibration control problems. Tanaka et al. (1993) showed that neural networks are also able to compensate for introduction of harmonics generated by the actuators. It was also shown that the **NNs** are capable of compensating the distorted signal. They found that the neural networks are equally good for linear as well as for non-linear problems. They used adaptive non-linear **NNs** for the suppression of the flow-induced vibrations in the aircrafts. Employing the linear control theory Prakah et al. (1993), developed a two methods based system. The first method investigated linear constant gain displacement feedback and the second method investigated linear quadratic gain/transfer recovery method with augmented state to accommodate disturbances. The LMS algorithm was used for **H-infinity** optimization.

Due to the development of model-less controllers like the systems based on **Fuzzy logic** Peters et al. (1993), used Fuzzy controller based on some parametric errors and certain **if–then rules**, for vibration reduction problems. Librescu et al. (1993), suggested that piezoelectric actuators change the dynamics of the system, localized strain fields are produced by injecting the current, so there is necessity of using adaptive systems which can compensate these affects during the control process. Spanos et al. (1994), worked on identifying troublesome modes off-line and canceling them by direct inversion.

Gopinathan et al. (1995), integrated linear quadratic optimal control with classical self- tuning regulator to get better results. Nam et al. (1995), integrated linear quadratic optimal control law and linear programming techniques to optimize a certain performance index. This method optimizes the size, shape, and location of piezoelectric patches for flutter suppression of vibrating structures. Sutton et al. (1995), developed a real time harmonic controller to control magnostrictive actuators. With varying DC voltage, authors were able to compensate for, the seven harmonics of the system, and to overcome inherent non-linearity of the actuator at fundamental frequencies in excess of 3 KHz. In their work, the steepest- decent algorithm was used to adapt the controller parameters and making optimum waveform at the input to the system.

Prakah et al. (1995), developed a fuzzy controller based on Fuzzy rules together with LMS updating algorithm. Then Snyder et al. (1995), used NNs for active vibration control of smart structures. Some other researchers like Clark et al. (1995), used other types of updating algorithms such as time averaged gradient descent algorithm, for adaptive feed-forward controller with direct velocity feed-back control. An improvement was observed with this type of system. Viperman et al. (1995), suggested that even with collocated filters, actuators and sensors it is not always possible to produce minimal phase control. So the authors developed a method by which low order filters acting as feed-forward compensators can serve the purpose of large order filters needed for broadband vibration control.

Zhang et al. in (1999) developed a new hybrid control method called **the controllable constrained damping layer**. Where the piezoelectric actuator is adhered to the elastic cover sheet of the viscoelastic damping layer. It can overcome the inherent shortcoming of the active constrained damping layer because the piezoelectric laminates with smaller areas, instead of the piezoelectric layer with a bigger area, are enough for the vibration control of larger structures. Especially, from an economical point of view, it provides a preferable hybrid method for vibration control of structures that have been previously covered with the constrained damping layer.

2.3 CONCLUSION FROM THE LITERATURE REVIEW

From the literature review presented above, following observations can be made

- Much work has been done on the adaptive vibration control using gain scheduling, self-tuning regulators and LMS algorithms, further investigations using piezoelectric materials as sensors/actuators for intelligent adaptive control are desirable.
- Although the use of frequency domain methods are quite popular in control system design, but little work about the use of these methods is reported in literature for vibration control of lightweight structures. Frequency domain methods are quite attractive as they require less computation power and are robust towards sensor noise.
- There is a need to carry out the work on the improved and fast methods of parametric and non-parametric estimation of transfer function.
- Very less amount of work is available on the implementation of Auto – Regressive Moving Average methods of parametric estimation in identification and control of structural vibrations. Whatever information is available, it is limited to Single – Input Single – output (SISO) systems. So there is a need to carry out further work to include Multi – Input Multi – Output (MIMO) systems
- Reported literature in the direction of closed - loop identification is secant.
- Adaptive control using Linear Quadratic Regulator (LQR) based controllers is not reported in literature.
- Reported literature in the direction of Minimum Time optimal control is secant.

- Most of the work reported in literature is on cantilever beams or plates which are one of the easier configurations for control effectiveness measurements. Very less amount of literature is available for other types of boundary conditions and on built up structures.
- Reported literature in the direction of robust vibration control is secant.

CHAPTER

THREE MATHEMATICAL MODELLING

3.1 GENERAL INTRODUCTION

Using different types of distributed sensors and actuators based on piezoceramics, with higher electromechanical coupling, sensing and control has become practically feasible for large and complex shaped flexible structures. With these developements in the control hardware, Active vibration control (AVC) has now became a practical reality. A lot of practical applications of AVC are emerging day by day.

Because of their coupled mechanical and electrical properties, piezoceramics have recently attracted significant attention for their potential application as sensors for monitoring and as actuators for controlling the response of structures. The concept of using a network of actuators and sensors to form a self-controlling and self monitoring “smart” system in advanced structural design has drawn considerable interest among the research community. This new technology could possibly be applied to the design of large-scale space structures, aircraft structures, satellites, and so forth. Finite element analysis is the fundamental tool for modeling these flexible structures. A large amount of literature is available till date for mathematical modeling of these structures using finite element techniques. For controlling the vibrating structures using piezoelectric materials, the dynamics of these structures must be known. Electromechanical coupling of piezoelectric materials can easily be related with the dynamics of these structures using finite element methods

TABLE 3.1 GEOMETRICAL AND MECHANICAL PROPERTIES OF STEEL AND PZT

Material Property	STEEL	PZT
Length of plate(mm)	L_H= 200	-------
width of plate(mm)	L_V=200	--------
Thickness(mm)	t_s=0.54	t_p=1
Length(mm)	--------	l_p=20
Width(mm)	--------	b=20
Young's Modulus(Mpa)	E_s=210	E_p=64
Density(Kg/m^3)	ρ_s=7800	ρ_p=7650

ELECTRICAL PROPERTIES OF PZT

Property	Symbol	Value
Piezoelectric charge constant (m V^{-1})	d_{31}	171x10^{-12}
Piezoelectric charge constant(m V^{-1})	d_{32}	171x10^{-12}
Poisson's ratio	υ_p	0.28
Permittivity (Fm^{-1})	ε	106x10^{-12}

3.2 DESCRIPTION OF THE STRUCTURE USED

In the present work, a plate clamped from one side,with dimensions (200 x 200 x 0. 54) mm is to be studied. Piezoelectric materials, bonded on a part of the surface of the structure, are to be used as sensors and actuators. The strain developed due to vibrations can be converted into electrical voltages, and is measured at the output sensors. The observed signal can be in the range –10 volt to +10 volt , as this does not need amplification. To remove the effect of higher modes the signals are pre – filtered. Then the signal goes to data acquition card. Analog-to-digital (A/D) convertion of the signal is done in this card. Based on these digitized, input signals, control signals are calculated by the designed controller. These signals can also be in the range of –10 V to +10V. After digital-to-analog (D/A) convertion in the data acquition card, control signal goes to the amplifier. By the amplifier, it is amplified 10 to 15 times, so that the voltage after amplification is in the range –150 V to +150 V. This high voltage signal is then sent to the PZT actuators. This voltage develops the strain in the structure such that vibration damping is achieved.

For controlling the vibrating structure using piezoelectric materials, the dynamics of these structures must be known. The relationship of electromechanical coupling of piezoelectric materials with the dynamics of these structures will be established by using finite element methods (FEM). Modal parameters, i.e. frequencies, damping ratios and mode shapes of the clamped plate are to be calculated using the finite element model. Any flexible, vibrating system can be modeled into state space form using modal parameters. The system models based on actual input - output data are more suitable for identifying these system parameters. Difference equation models based on ‘Auto Regressive Moving Average Method (ARMA)’ are easy to construct and estimate from practical data. The system can be estimated in difference equation form and then can be transformed to unique state space form for designing the controller. The FEM models can be used for initial estimate of these parameters.

It is obvious that as the payload attached at the tip of the clamped plate or the relative lengths or orientations of clamped plate changes, the system natural frequencies and mode shapes are changed. The system matrices depend upon systems natural frequencies, damping ratios and mode shapes. Hence with the change in structural parameters, system matrices are changed. Depending upon different geometric configurations of the structure and tip loads attached, system matrices can be calculated offline.

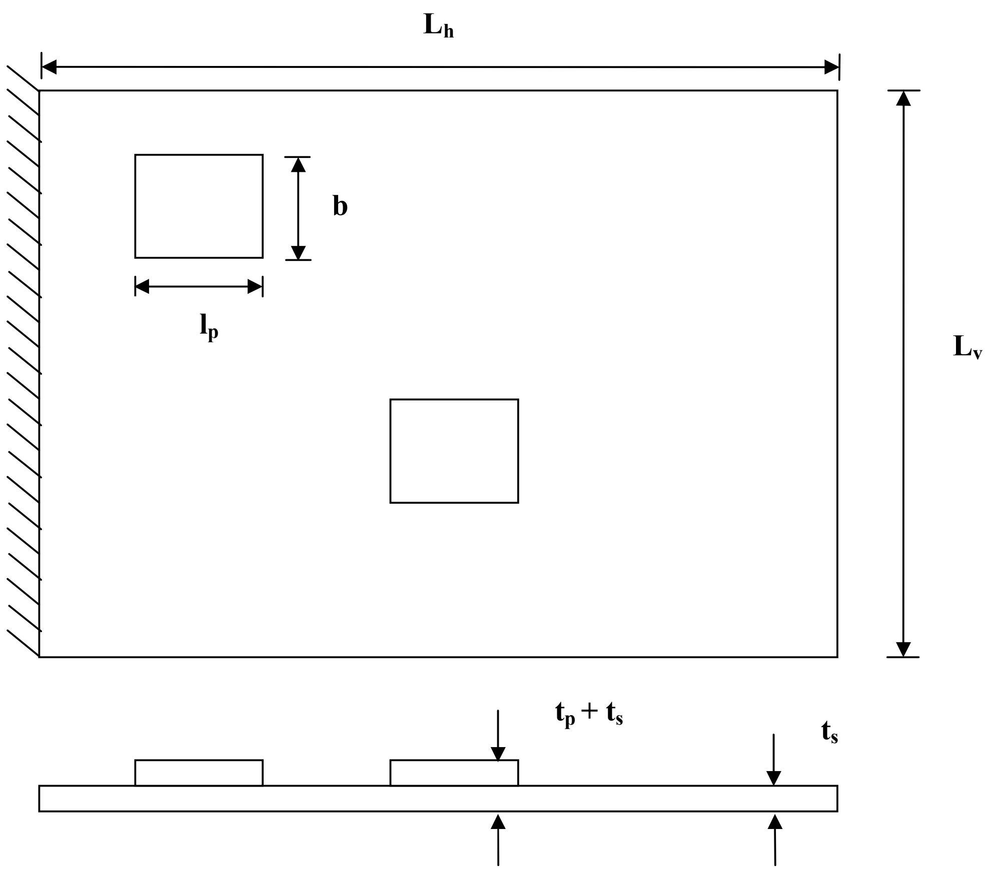
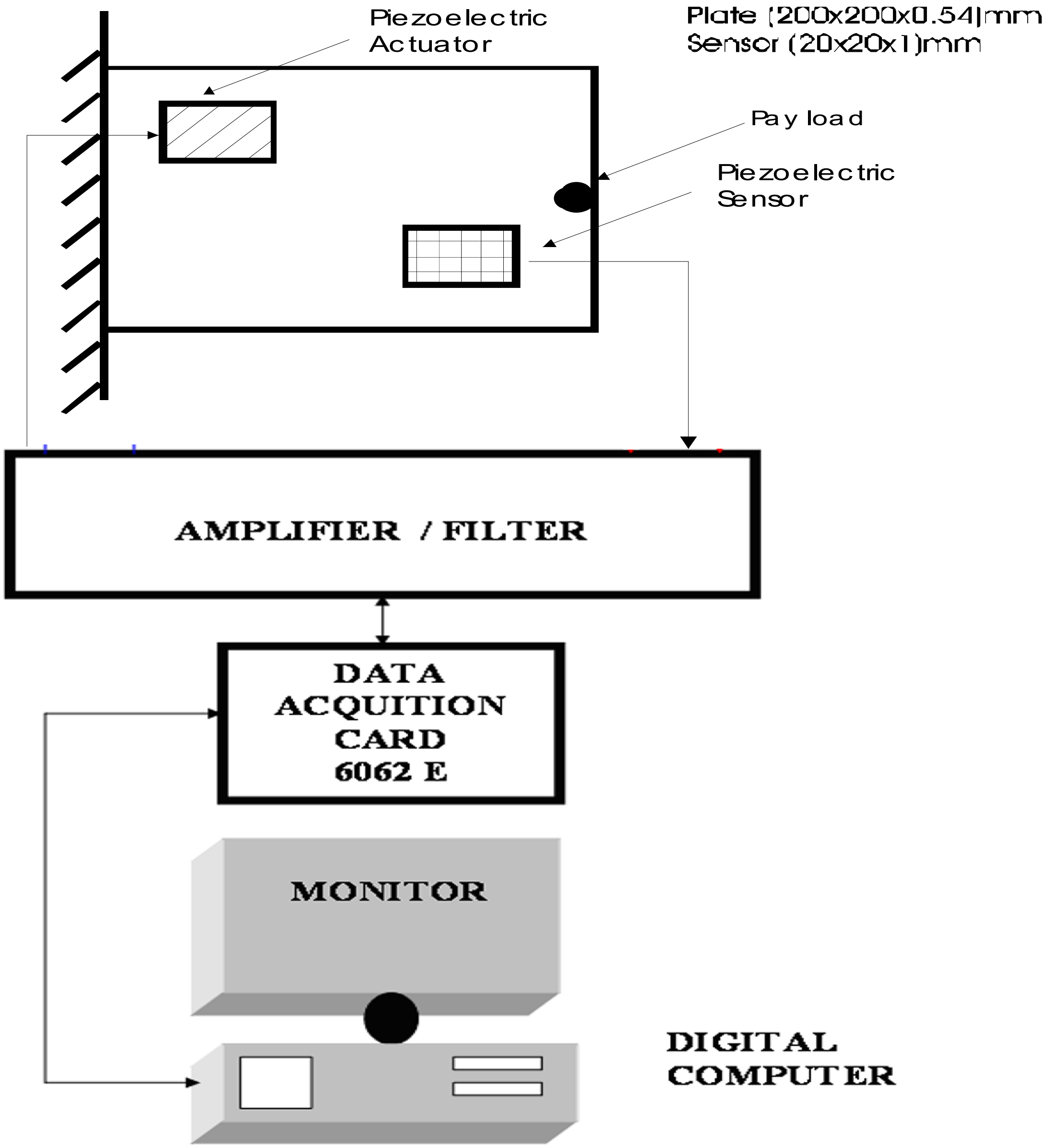


Figure 3.1 Schematic Diagram of the Clamped plate



Piezoelectric Actuator
Plate (200x200x0.54)mm
Sensor (20x20x1)mm
Pay load
Piezoelectric Sensor
AMPLIFIER / FILTER
DATA ACQUITION CARD 6062 E
MONITOR
DIGITAL COMPUTER

Figure 3.2 Schematic Diagram of the theoratical setup

3.3 EQUIPMENT CHARACTERISTICS

The following are the characteristics of the different equipments

3.3.1 Software 'Lab VIEW RT'

The National Instruments 'Lab VIEW' is a highly productive graphical development environment for building data acquition, instrumentation and control systems. With 'Lab VIEW' it is easy to create interfaces that give interactive control of software system. The tight integration of 'Lab View' with measurement hardware facilities, rapid development of data acquisition and control is possible. This software contains powerful built – in measurement analysis and a graphical compiler for optimum performance.

3.3.2 Simultaneous I/O Data acquition card (DAQ 6062 E)

National Instruments 'E-Series' technology is a complete data acquisition (DAQ) hardware architecture that takes advantage of the latest in electronics and technological innovations and advances the capabilities of PC – based DAQ solutions. 'E-Series' is a standard architecture in instrumentation class, for multichannel data acquisition.

3.3.3 Amplifiers

The control signals generated from the computer via data acquition card, are in the range –10 volt to +10 volt. Since for getting lesser and lesser settling times, the control voltage has to be increased. This job is done by the amplifiers. These amplifiers are capable of increasing the control signal voltage up to 200 volt, withiout any phase change.

3.3.4 Pre – filter

Any vibrating flexible strucure, contains infinite number of modes of vibration. Although for practical purposes only few modes are of importance. The dynamics of these structures can be modelled mathematically, using finite number of modes using modal analysis. For better sensing the signal obtained from these vibrating structures, higher modes need to be elliminated. This work is done by higher order low pass filters. They pre-filter the input signal, to remove the high frequency signals. So some times these low pass filters are also called 'pre-filters'.

3.3.5 PZT patches as Sensors/Actuators

"**Piezoelectric (PE)**" materials are those materials which when subjected to electric field get deformed and vice – versa. This change in configuration may be used to nullify the effect of vibrations induced by external disturbances. These are made by the combination of three elements i.e lead, zirconium and titanium. The geometrical and mechanical properties are discussed in table (3.1). Electrical properties are shown in table (3.2).

3.4 MATHEMATI CAL MODELLING

The finite element method works by discretising the domain of interest (for example, a car) into elements, where all events, reactions etc are assumed to happen at the nodes (the vertices of elements), and the behaviour of the structure within an element is assumed to be described by simple functions of the coordinates within the element. These functions are termed the shape functions, and often describe the relationship between the displacements at the nodes and the displacements within an element.

The strain-displacement and stress-strain relationships can be defined for each element, and the element stiffness matrix derived which describes the relationship between the displacements of the element nodes and the nodal forces. Since the nodes are shared between several elements, the element stifness matrices of all the elements must be combined to produce the global stifness matrix. Hence a governing equation can be derived describing the relationships between the displacements of all the nodes and all the forces on the nodes. The boundary conditions of the problem may be applied to the governing equation, and the equation solved to determine the unknown displacements of the nodes, and hence the displacements, strains and stresses within each element and the whole domain.

Finite element method is useful for the determination of natural frequencies and mode shapes of these structures. These parameters (i.e. natural frequencies and mode shapes) are the pre-requisites for designing an active vibration control system. Classical control systems are normally based on transfer functions.

The developments in the field of piezoelectric materials have motivated many researchers to work in the field of smart structures. The smart structure can be defined as the structure that can sense external disturbance and respond to that with active control in real time to maintain the mission requirements. Smart structures consist of highly distributed active devices and processor networks. The active devices are primarily sensors and actuators either embedded or attached to an existing passive structure. Bailey and Hubbard (1985) initiated the research on the application of the smart structures in active vibration control. The utilization of discrete piezoelectric actuators has been shown to be a viable concept for vibration suppression of one dimensional structure by Crawley and de Luis (1989) . The vibration excitation of thin, flat plates by using piezoelectric patches has been analyzed by Dimitridis and Fuller (1991).. The application of the finite element modeling techniques in the smart materials technologies has been in continuous growth during the last decade. Hence some piezoelectric elements have become available in commercial finite element codes like ANSYS.

3.4.1 FINITE ELEMENT MODEL OF PLATE : GENERAL

The effcient analysis of plate structures is of fundamental importance in many branches of engineering, and continues to be an area of active research. While the governing differential equations for particular models of plate analysis are well-established, it is generally not possible to obtain analytical solutions of the equations except for simple geometries. Hence, numerical solutions are required for practical problems. By far the most popular method in use for plate analysis is the finite element method .Its strength lies in its generality and its ability to easily deal with complex geometries and loading conditions. Bending of plates can be considered as an extension of beam theory, the word "plate" is used to indicate bodies that are bounded by two parallel planes whose lateral dimensions are large as compared to the separation between these plates. Geometrically, plates are similar to plane elastic bodies which can be loaded transversely (i.e. perpendicular) to the plane of the plate as shown in figure 3.3 and figure 3.4. Under certain simplifying assumptions bending moment $M_{x,}$ twisting moment $M_{xy,}$ shear force Q_x can be given as

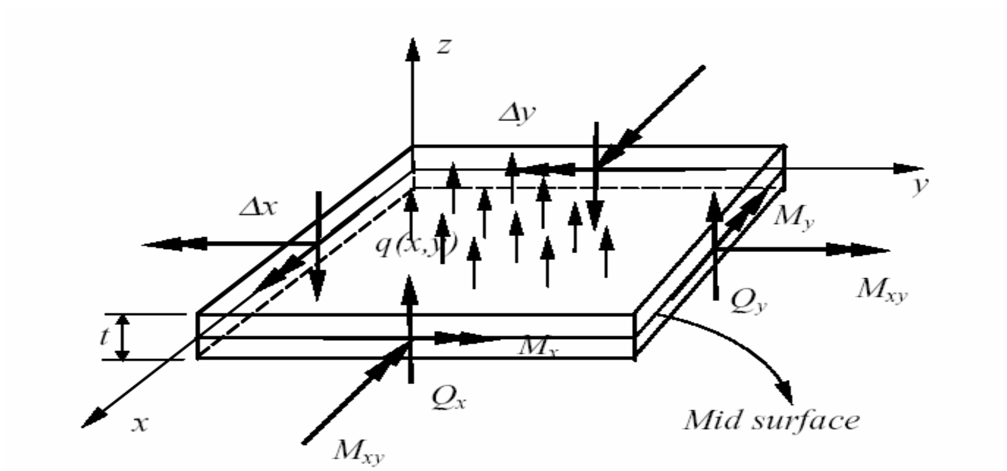


Figure 3.3 Forces and Moments acting on the Plate

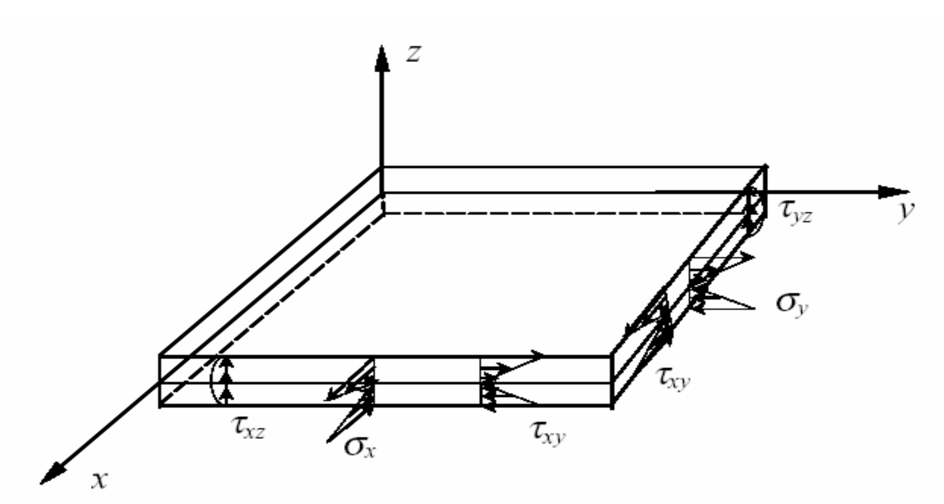


Figure 3.4 Stresses acting on the Plate

Bending Moment (per unit length)

$$M_x = \int_{-t/2}^{t/2} \sigma_x z dz, \qquad (N \cdot m/m)$$

(3.1)

Twisting Moment (per unit length)

$$M_{xy} = \int_{-t/2}^{t/2} \tau_{xy} z dz, \qquad (N \cdot m/m)$$

(3.2)

Shear Forces (per unit length)

$$Q_x = \int_{-t/2}^{t/2} \tau_{xz} dz, \qquad (N/m)$$

(3.3)

$$Q_y = \int_{-t/2}^{t/2} \tau_{yz} dz, \qquad (N/m)$$

(3.4)

Maximum Bending Stresses

$$(\sigma_x)_{\max} = \pm \frac{6M_x}{t^2}, \qquad (\sigma_y)_{\max} = \pm \frac{6M_y}{t^2}.$$

(3.5)

Note that Maximum stress is always at $z = \pm\ t/2$

3.4.2 THIN PLATE THEORY(Kirchhoff Plate Theory)

Assumptions (similar to those in the beam theory): A straight line along the normal to the mid surface remains straight and normal to the deflected mid surface after loading, that is, these is no transverse shear deformation:

$$\gamma_{xz} = \gamma_{yz} = 0.$$

Displacement:

$$
\begin{aligned}
w &= w(x, y), \qquad (deflection)\\
u &= -z\frac{\partial w}{\partial x},\\
v &= -z\frac{\partial w}{\partial y}.
\end{aligned}
$$

(3.6)

Strains:

$$
\begin{aligned}
\varepsilon_x &= -z\frac{\partial^2 w}{\partial x^2},\\
\varepsilon_y &= -z\frac{\partial^2 w}{\partial y^2},\\
\gamma_{xy} &= -2z\frac{\partial^2 w}{\partial x \partial y}
\end{aligned}
$$

(3.7)

Stresses:(plane stress state)

$$
\begin{Bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{Bmatrix} = \frac{E}{1-\nu^2} \begin{bmatrix} 1 & \nu & 0 \\ \nu & 1 & 0 \\ 0 & 0 & (1-\nu)/2 \end{bmatrix} \begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{Bmatrix},
$$

(3.8 a)

or

$$
\begin{Bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{Bmatrix} = -z\frac{E}{1-\nu^2} \begin{bmatrix} 1 & \nu & 0 \\ \nu & 1 & 0 \\ 0 & 0 & (1-\nu) \end{bmatrix} \begin{Bmatrix} \dfrac{\partial^2 w}{\partial x^2} \\ \dfrac{\partial^2 w}{\partial y^2} \\ \dfrac{\partial^2 w}{\partial x \partial y} \end{Bmatrix}.
$$

(3.8 b)

Main variable deflection w = w(x,y)

Governing equations

$$D\nabla^4 w = q(x, y)$$

(3.9)

where

$$\nabla^4 \equiv (\frac{\partial^4}{\partial x^4} + 2\frac{\partial^4}{\partial x^2 \partial y^2} + \frac{\partial^4}{\partial y^4})$$

(3.10)

and the bending rigidity of the plate

$$D = \frac{Et^3}{12(1-\nu^2)}$$

(3.11)

q = lateral distributed load

3.5 FINITE ELEMENT MODELLING OF PLATE USING ANSYS

ANSYS is a sophisticated and comprehensive finite element program that has capabilities in many different physics fields such as static structural, nonlinear, thermal, implicit and explicit dynamics, fluid flow, electromagnetic, and electric field analysis. It can also perform coupled field analysis combining one or more of these different physics. ANSYS is an integrated program with all operations performed under one GUI. Creating the model, running it, and post processing the results are all done without leaving the ANSYS environment.

ANSYS has a large library of element types. Elements are organized into groups of similar characteristics. These group names make up the first part of the element name (BEAM, SOLID, SHELL, etc). The second part of the element name is a number that is more or less (but not exactly) chronological. As elements have been created over the past 30 years the element numbers have simply been incremented. The earliest and simplest elements have the lowest numbers (LINK1, BEAM3, etc) , the more recently developed ones have higher numbers. The “18x”

series of elements (SHELL181, SOLID187, etc) are the newest and most modern in the ANSYS element library. There are several solvers in ANSYS that differ in the way that the system of equations is solved for the unknown displacements. The two main solvers are the sparse solver and the PCG solver. If the choice of solvers is left to “program chosen” then generally ANSYS will use the sparse solver. The PCG (preconditioned conjugate gradient) solver works well for models using all solid elements. From a practical perspective one thing to consider is that the sparse solver doesn’t require a lot of RAM but swaps out to the disk a lot. Disk I/O is very slow. If you have a solid model and lots of RAM the PCG solver
could be significantly faster since the solution runs mostly in core memory. The basic equation of the finite element method that ANSYS is solving is, [K]{ U}= { F}, where [K] is the assembled stiffness matrix of the structure, {u} is the vector of displacements at each node, and {F} is the applied load vector. This is analogous to a simple spring and is the essence of small deflection theory.

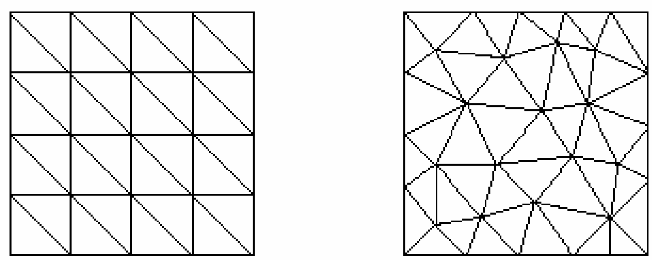
3.5.1 MODELLING OF PLATE

In the theoretical analysis, finite element code ANSYS (v5.6) was used. During the development of the smart plate, a model with parametric design capability is created. The most suitable element having piezoelectric capability in three-dimensional coupled field problems is the shell type 2-D element SHELL99. Similar to other structural solid elements, that element has three displacement degrees of freedoms per node. In addition to these degrees of freedoms, the element has also potential degrees of freedoms for the analysis of the electromechanical coupling problems. Piezoceramic actuators inherently exhibit anisotropy and yield three-dimensional spatial variation in their response to the piezoelectric actuation. Therefore, the models developed for the passive portion should include consistent degrees of freedoms with the actuator degrees of freedom at the locations where these elements interface. Theoretically, the plate elements (shell or solid) can be used in the modeling of the passive portion of the smart structure. While the shell elements can be used in accordance with the thin plate theory, whereas the solid elements work with the three dimensional elasticity theories.

The influence of the placement of (20x20x1) mm piezoelectric actuators and sensors on the steel plate is considered. The steel plate being anisotropic material, its Geometrical and Mechanical properties along with PZT are shown in Table (3.1). Its Electrical properties are shown in Table (3.2). In this work, the identically polarized patches are assumed to be bonded symmetrically on top surface of the plate. The smart plate is considered to be clamped along one edge and a Single Input Single Output system is considered.

3.5.2 MESHING OF PLATE

There can be two different types of mesh as shown in figure 3.5



Firgure 3.5 Structured (n = 6) and unstructured (n ≠ constant) mesh

Structured Mesh

1) The number of elements n surrounding an internal node is constant.
2) The connectivity of the grid can be calculated rather than explicitly stored.
3) Simpler and less computer memory intensive.
4) Lack of geometric exibility.

Unstructured mesh

1) The number of elements surrounding an internal node can be arbitrary.
2) Greater geometric exibility
3) Crucial when dealing with domains of complex geometry or when the mesh has to be adapted to complicated features of computational domain or to complicated features of the solution (eg. boundary layers, internal layers, shocks, etc).
4) Expensive in time and memory requirements

A triangulation, or triangular mesh, is a collection of non-overlapping connected triangles. Such a triangulation is said to be conforming if every triangle's side is

shared by exactly one neighbour or zero neighbours on the boundary. Figure 3.6 illustrates both non-conforming and conforming triangulations. Conforming triangulations (and meshes in general) are required to get a correct solution, although corrections can be made to deal with non-conforming meshes. The finite element method is a method of solving partial differential equations, particularly those arising in solid mechanics and heat transfer. An accurate solution usually requires accurate refinement of the mesh. ANSYS usually uses triangular or quadrilateral elements for meshing as in our case triangular element mesh generation has been done with help of ANSYS as shown in Figure 3.5.

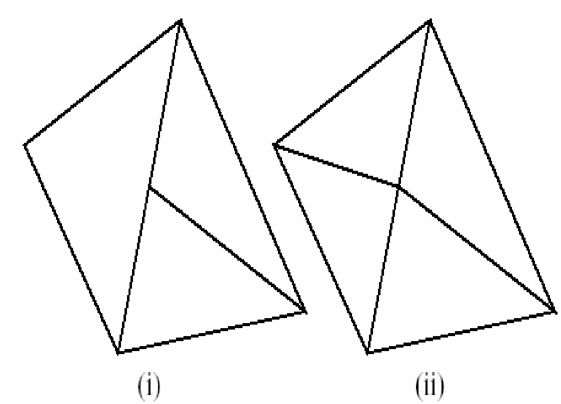


Figure 3.6 Triangular Mesh (i) Non Conforming (ii) conforming

Various results of ANSYS like Triangular Meshing of plate , Final Mesh formed ,Actuator patch, Sensor patch etc are shown below from figure 3.7 – 3.10

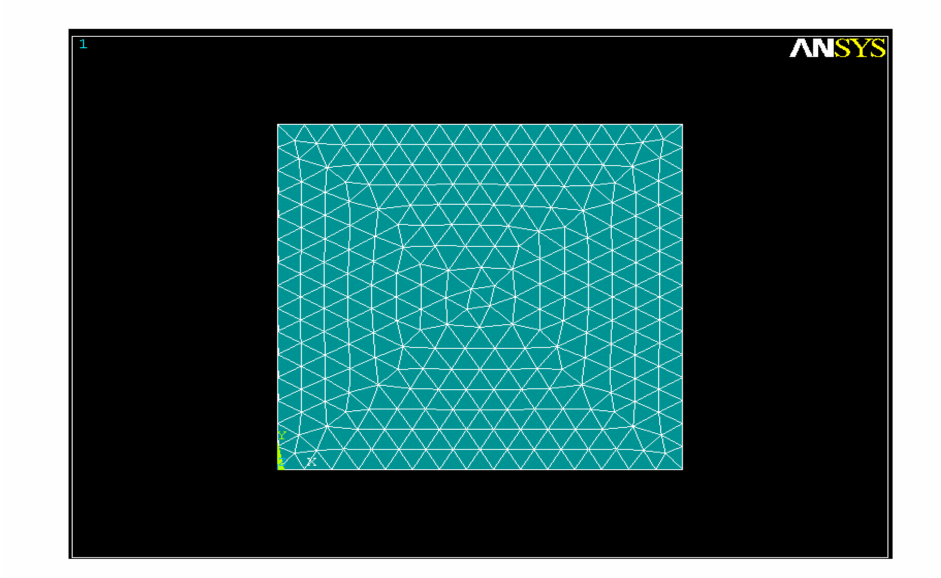


Figure 3.7 Plate With Triangular Mesh

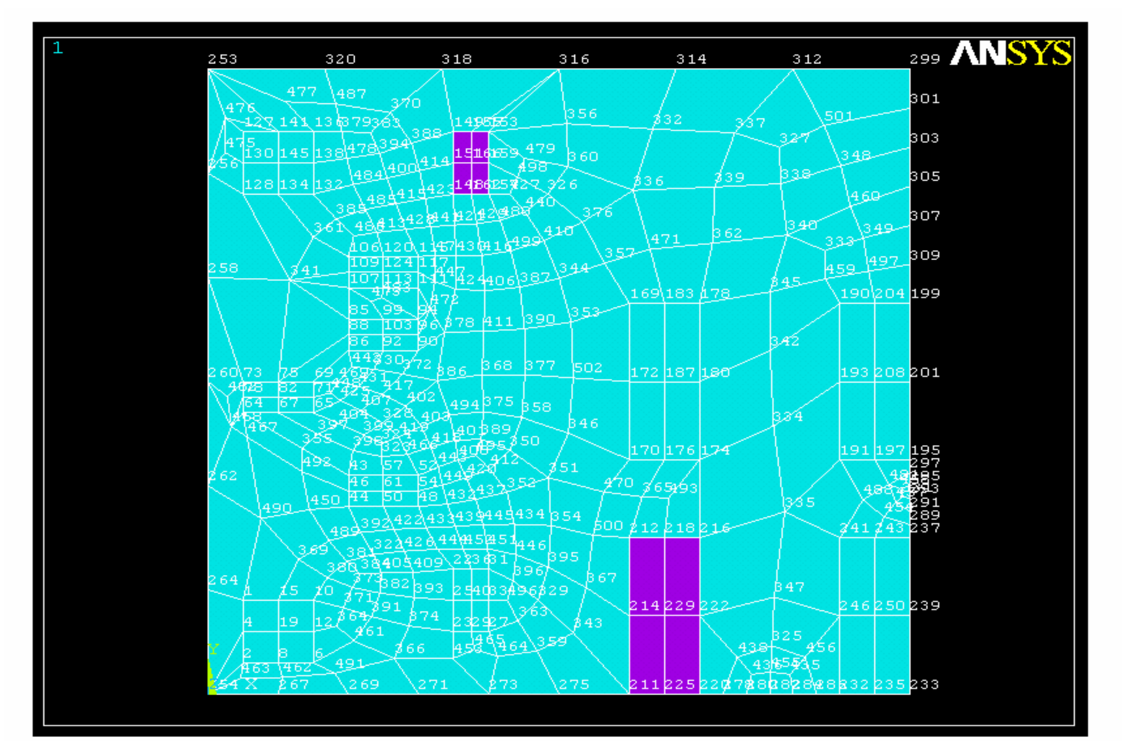


Figure 3.8 Final Mesh Formed after Mesh Refinement Total No. of Nodes 950 and Elements 307

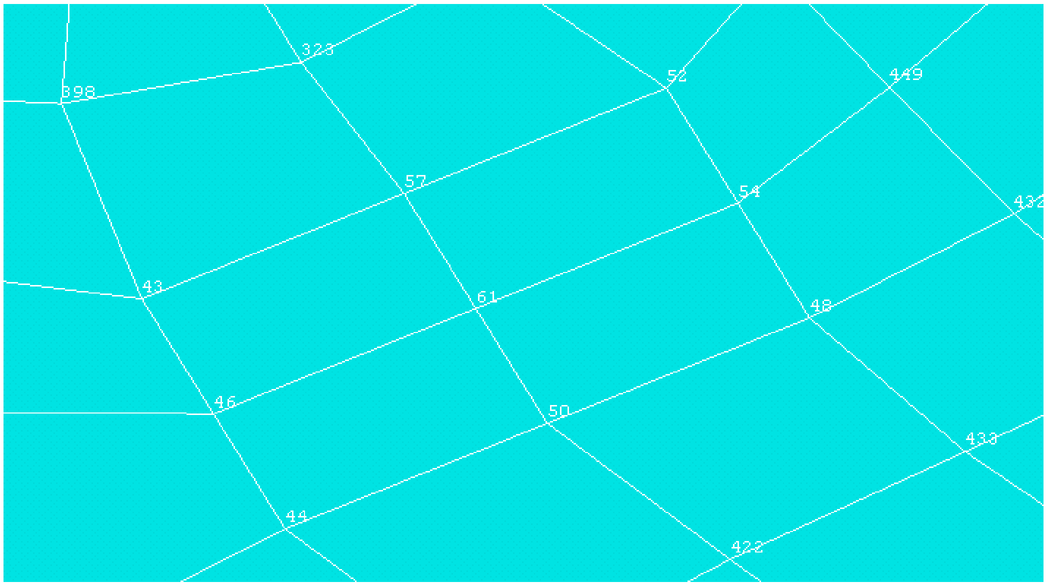


Figure 3.9 Piezzoceramic Sensor (nodes 43,44,48,52)

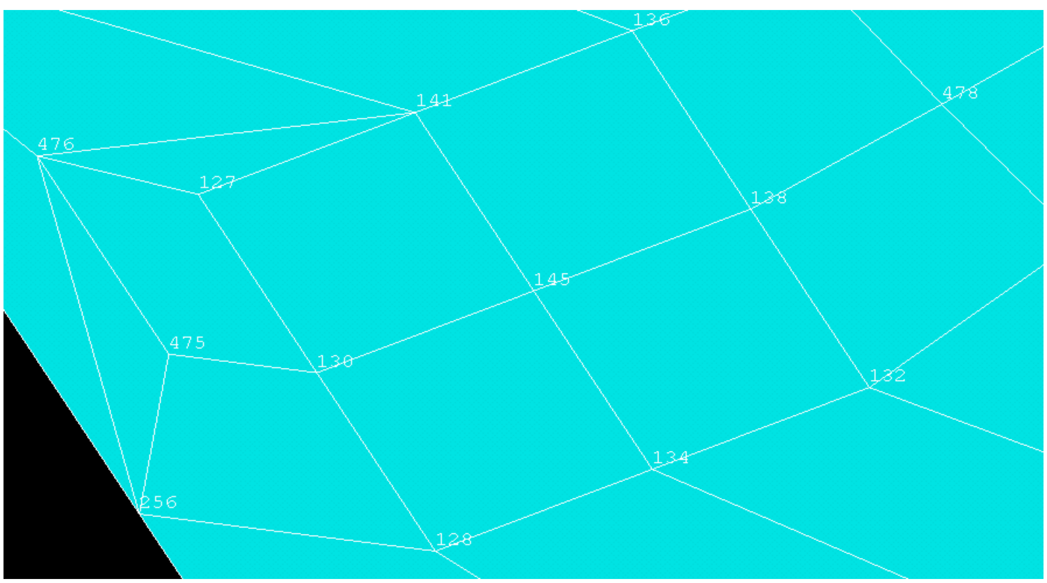


Figure 3.10 Piezzoceramic Actuator(nodes 127,128,132,136)

3.6 RESULT SUMMARY

1) Total number of elements formed is 370 and total number of nodes formed is 950.

2) Actuator embeded on the plate have its ends on the nodes 127, 128, 132, 136.

3) Sensor embeded on the plate have its ends on the nodes 43, 44,48,52, .

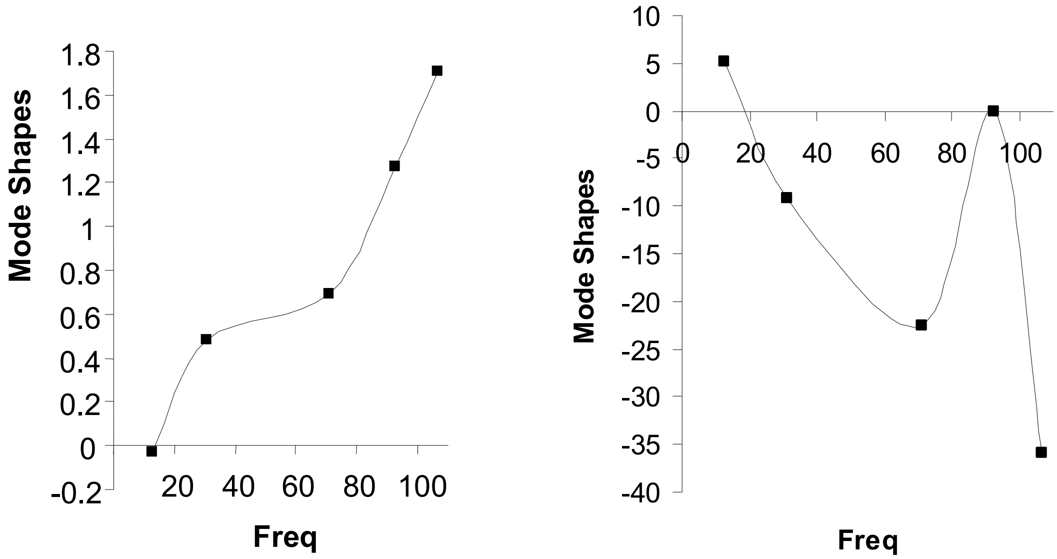
4) Frequencies obtained through ANSYS for the plate has been validated against the experimental values of Yavuz Yaman .et .al (2000)

5) From the comparison of experimental results of Yavuz Yaman .et .al (2000) ,we are assuming value of damping ratio as 0.0061 for the first and second mode for our plate.

6) Variation of mode shapes with frequencies of first two nodes of actuator and sensors is shown graphically from Figure 3.11 – 3.14.

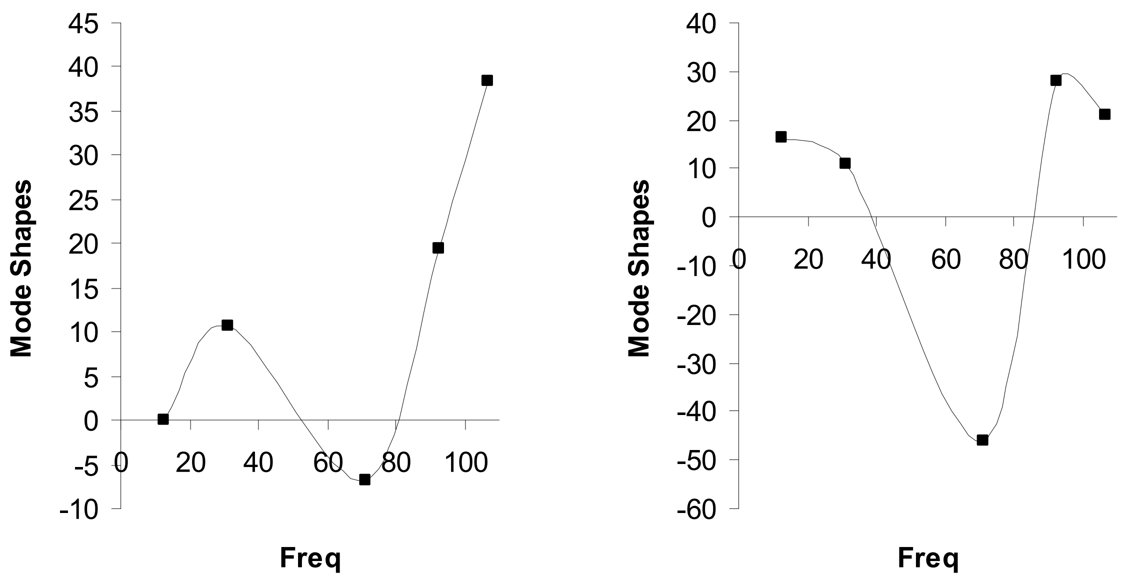
TABLE 3.2 COMPARISON OF THEORATICAL AND EXPERIMENTAL FREQUENCY

Frequency in (Hz)	Ansys	Experimental
f1	12.83	11.88
f2	30.96	41.17
f3	70.88	71.62



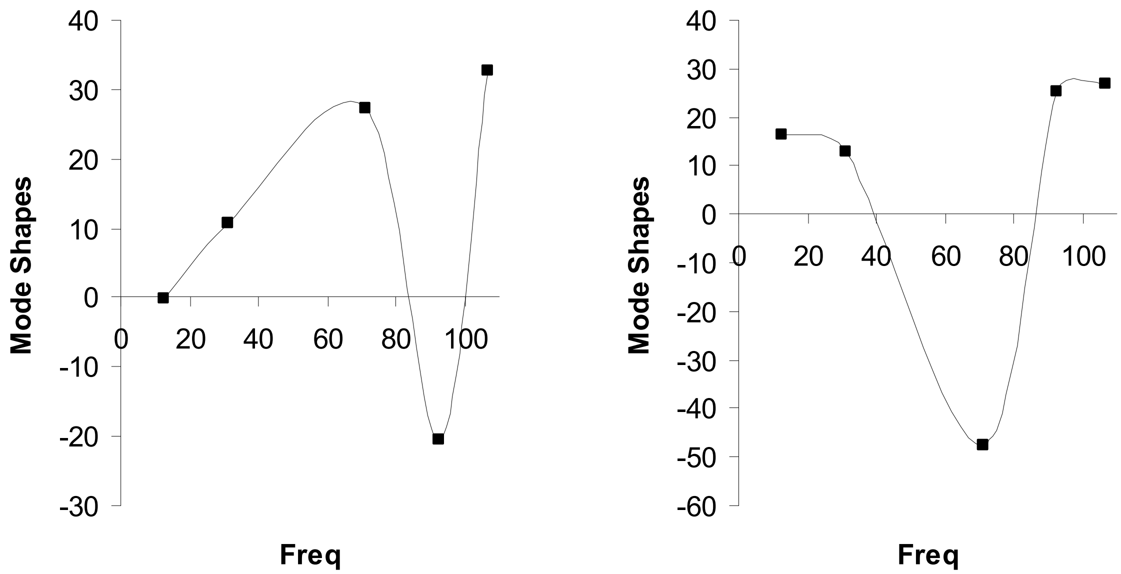
(i) (ii)

Figure 3.11 Variation of Mode shape (i) X-dir (ii) Y-dir with Frequency at 1st actuator node127



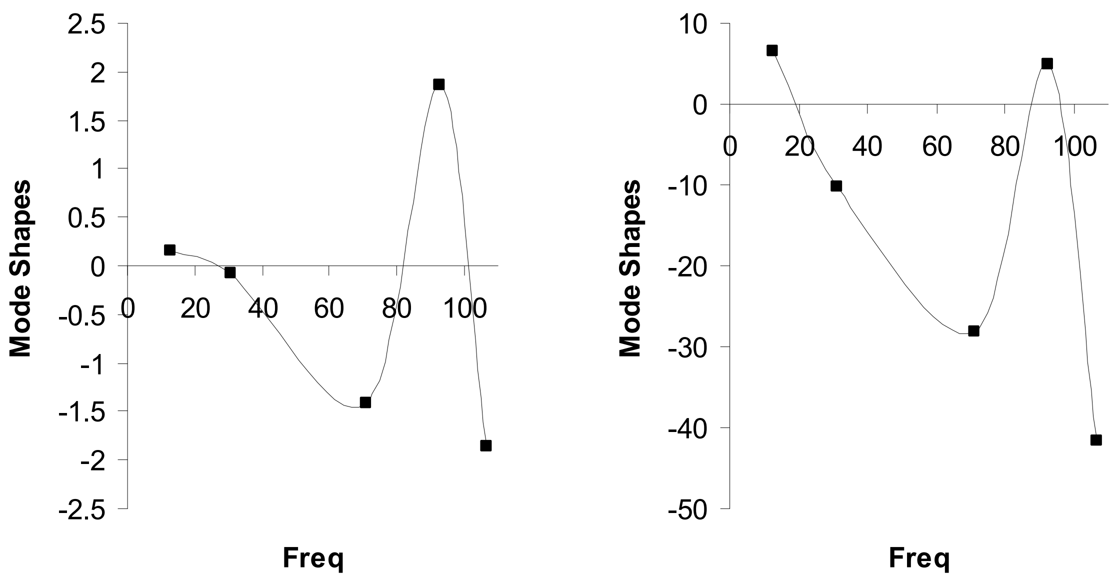
(i) (ii)

Figure 3.12 Variation of Mode shape (i) X-dir (ii) Y-dir with Frequency at 2nd actuator node 128



(i) **(ii)**

Figure 3.13 Variation of Mode shape (i) X-dir (ii) Y-dir with Frequency at 1st Sensor node 43



(i) **(ii)**

Figure 3.14 Variation of Mode shape (i) X-dir (ii) Y-dir with Frequency at 2nd Sensor node 44

3.7 STRUCTURED ELEMENT MOUNTED WITH PZT

In case the element is mounted with a piezoelectric element, the mass and stiffness elements are modified as below

$$\mathbf{m}^{(e)} = \mathbf{m}^{(e)}{}_{\text{Steel}} + \mathbf{m}^{(e)}{}_{\text{PZT}} \qquad \text{---(3.12)}$$

$$\mathbf{k}^{(e)} = \mathbf{k}^{(e)} \qquad \text{---(3.13)}$$

But for the composed structure Moment of inertia is choosen such that

$$E\,I = E_{steel}\, I_{steel} + E_{PZT}\, I_{PZT} \qquad \text{---(3.14)}$$

3.8 LAGRANGE'S EQUATIONS OF MOTION FOR LINEAR SYSTEMS

The motion of a general linear system is given by, Meirovitch (1986)

$$\sum_{i=1}^{n} \left[m_{ji}\, \ddot{\Delta}_i(t) + c_{ji}\, \dot{\Delta}_i + k_{ji}\, \Delta_i(t) \right] = Q_j(t) \qquad i, j = 1,2,\ldots\ldots, n \qquad \text{---(3.15)}$$

where Δ_i (t) is the physical displacement, $\dot{\Delta}_i$ (t) is physical velocity and $\ddot{\Delta}_i$ (t) is the acceleration at time instant t for the particular degree of freedom i. The vector of externally applied forces is denoted by $\mathbf{Q}_j$ (t). Also m, c and k are the elements of mass, damping and stiffness matrices respectively.

Relation (3.15) represents a set of n simultaneous second – order ordinary differential equations in generalized coordinates, and are called Lagrange's equations of motion. By this relation, the infinitely many degree-of-freedom distributed system is approximated by an n-degree of freedom system. This relation can be written in matrix form as

$$\mathcal{M}\ \ddot{\mathbf{\Delta}}(t)+\mathcal{C}\,\dot{\mathbf{\Delta}}(t)+\mathcal{K}\,\mathbf{\Delta}(t)=\mathcal{Q}\,(t) \qquad \text{---}(3.16)$$

where **M**, **C** and **K** are the global mass, damping and stiffness matrices respectively, and **Q** (t) is the vector of physical applied forces at various degrees of freedom on instant of time t. The column vector $\mathbf{\Delta}(t)$ is the nodal (also called physical) displacements at time t.

3.9 MODAL ANALYSIS

The eigenvalue problem associated with undamped free-vibration system represented by relation (3.16) is

$$(\mathcal{K} - \omega^2\ \mathcal{M})\,\mathbf{\Phi} = \mathbf{0} \qquad \text{---}(3.17)$$

where the transformation matrix $\mathbf{\Phi}$ is known as **mass normalized modal matrix**.
The following transformations relates the physical displacement 'Δ' to the modal displacement 'q' at time instant t

$$\mathbf{\Delta}(t) = \mathbf{\Phi}\,\mathbf{q}(t) \qquad \text{---}(3.18)$$

This mode superposition method is used to transform the 'coupled equations of motion' in physical co-ordinates to a set of 'uncoupled equations of motion' in the modal co-ordinates.

$$\mathbf{\Phi}^T \mathcal{M} \mathbf{\Phi} \ddot{\mathbf{q}}(t) + \mathbf{\Phi}^T \mathcal{C} \mathbf{\Phi} \dot{\mathbf{q}}(t) + \mathbf{\Phi}^T \mathcal{K} \mathbf{\Phi} \mathbf{q}(t) = \mathbf{\Phi}^T \mathcal{Q}(t)$$

where

$\mathbf{\Phi}^T \mathcal{M} \mathbf{\Phi} = \mathbf{I}$, is unity matrix

$\mathbf{\Phi}^T \mathcal{K} \mathbf{\Phi} = \boldsymbol{\omega}^2$ is a diagonal matrices containing the square of natural frequencies and

$\mathbf{\Phi}^T \mathcal{C} \mathbf{\Phi} = \widetilde{\mathbf{C}}$ is a symmetric damping matrix.

In case of proportional damping, the normalized damping matrix is given as, Rao JS et al (1984)

$$\widetilde{\mathbf{C}} = \mathrm{diag}[\ 2\ \xi_r\ \omega_r\ ] \qquad \text{---(3.19)}$$

where ξ_r is the damping ratio associated with r'th particular mode. Thus, the uncoupled system of equations with proportional damping takes the following form

$$\ddot{\mathbf{q}}(t) + 2\ \boldsymbol{\xi}\ \boldsymbol{\omega}\ \dot{\mathbf{q}}(t) + \boldsymbol{\omega}^2 \mathbf{q}(t) = \mathbf{\Phi}^T \mathcal{Q}(t) \qquad \text{---(3.20)}$$

The modal matrix $\mathbf{\Phi}$ for an N degree of freedom system may be written in component form as

$$\mathbf{\Phi} = \begin{bmatrix} {}^{(1)}\Phi_1 & {}^{(2)}\Phi_1 & \cdots & {}^{(N)}\Phi_1 \\ {}^{(1)}\Phi_2 & {}^{(2)}\Phi_2 & \cdots & {}^{(N)}\Phi_2 \\ \vdots & \vdots & \ddots & \vdots \\ {}^{(1)}\Phi_N & {}^{(2)}\Phi_N & \cdots & {}^{(N)}\Phi_N \end{bmatrix} \qquad \text{---(3.21)}$$

where ${}^{(j)}\Phi_k$ is the modal co-ordinate at k^{th} degree of freedom for j^{th} mode, and each column in the matrix represents the eigenvectors.

The natural frequencies are collected to form the $\boldsymbol{\omega}^2$, diagonal matrix of modal frequencies squared, as

$$\boldsymbol{\omega}^2 = \begin{bmatrix} \omega_1^2 & 0 & 0 & \cdots & 0 \\ 0 & \omega_2^2 & 0 & \cdots & 0 \\ 0 & 0 & \omega_3^2 & \cdots & 0 \\ \vdots & \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & 0 & 0 & \omega_N^2 \end{bmatrix} \qquad \text{---(3.22)}$$

where ω_i is the natural frequency of the i^{th} mode and i=1 to N.

For most of the structure systems under practical loading, only first few modes need to be considered. Thus the mode super position method is used to form a **reduced order dynamic system**. Depending upon the number of modes to be considered, the dimensions of the system are changed from N to R where R (R<N) is the reduced order model of the complete system.

3.10 MODAL STATE SPACE CONTROL

It has been mentioned earlier that for a physical structure only the first few modes are important from vibration control considerations. If such a structure is controlled by using discrete distributed actuators numbering 'a' and discrete distributed sensors numbering 's', the following relations may be written.

For 'r' modes and 'a' actuators, equation (3.20) takes the form

$$\ddot{q}_k(t) + 2\xi_k \omega_k \dot{q}_k(t) + \omega_k^2 q_k(t) = \sum_{i=1}^{i=a} \left[{}^{(k)}\Phi_i \quad Q_i(t) \right] \qquad k = 1,2,....r \qquad \text{---(3.23)}$$

where Q_i is the actual force acting at i^{th} degree of freedom and q_k is the modal displacement at k^{th} mode.

And the sensor output at the ith degree of freedom, by the contribution of 'r' modes is given by

$$\Delta_i(t) = \sum_{k=1}^{k=r} \left({}^{(k)}\Phi_i \quad q_k(t) \right) \qquad\qquad i = 1,2,....,s$$

--- (3.24)

where ${}^{(k)}\Phi_i$ is the mode shape, at i th degree of freedom and for k th mode.

Relation (3.23) for single actuator and single mode, in matrix form, is written as

$$\begin{Bmatrix} \dot{q}(t) \\ \ddot{q}(t) \end{Bmatrix} = \begin{bmatrix} 0 & 1 \\ -\omega^2 & -2\xi\,\omega \end{bmatrix} \begin{Bmatrix} q(t) \\ \dot{q}(t) \end{Bmatrix} + \begin{Bmatrix} 0 \\ \Phi_{actuator} \end{Bmatrix} \left\{ Q_{actuator} \right\}$$

where q (t), $\dot{q}(t)$ and $\ddot{q}(t)$ are the modal displacement, modal velocity and modal acceleration respectively.

By making the substitution $w_1 = q\,(t)$ and $w_2 = \dot{q}\,(t)$ for the single particular mode, the above equations may be written as

$$\begin{Bmatrix} \dot{w}_1(t) \\ \dot{w}_2(t) \end{Bmatrix} = \begin{bmatrix} 0 & 1 \\ -\omega^2 & -2\xi\,\omega \end{bmatrix} \begin{Bmatrix} w_1(t) \\ w_2(t) \end{Bmatrix} + \begin{Bmatrix} 0 \\ \Phi_{actuator} \end{Bmatrix} \left\{ Q_{actuator} \right\}$$

so that $w_1(t)$ is the modal displacement and $w_2(t)$ is the modal velocity.

This type of representation is known as state space form. For the system having 'r' modes, equations (3.23) and (3.24) can be written in matrix state space form as

$$\dot{\mathcal{W}}(t) = \mathcal{F}\,\mathcal{W}(t) + \mathcal{G}\,\boldsymbol{u}(t)$$
$$\mathcal{y}(t) = \mathcal{H}\,\mathcal{W}(t)$$

---(3.25)

where the modal state vector is defined as

$$\mathcal{W}(t) = \begin{bmatrix} w_1(t) \\ w_2(t) \\ w_3(t) \\ w_4(t) \\ \vdots \\ w_{2r-1}(t) \\ w_{2r}(t) \end{bmatrix}$$

Such that $w_{2r-1} = q_r$ and $w_{2r} = \dot{q}_r$ and the other matrices, such as **F**, **G** and **H** are called system matrices. The matrix **F**, known as the state coupling matrix, and is given as

$$\mathcal{F} = \begin{bmatrix} 0 & 1 & 0 & 0 & \cdots & \cdots & 0 & 0 \\ -\omega_1^2 & -2\xi_1\,\omega_1 & 0 & 0 & \cdots & \cdots & 0 & 0 \\ 0 & 0 & 0 & 1 & \cdots & \cdots & 0 & 0 \\ 0 & 0 & -\omega_2^2 & -2\xi_2\,\omega_2 & \cdots & \cdots & 0 & 0 \\ \vdots & \vdots & \vdots & \vdots & \ddots & \ddots & \vdots & \vdots \\ \vdots & \vdots & \vdots & \vdots & \ddots & \ddots & \vdots & \vdots \\ 0 & 0 & 0 & 0 & \cdots & \cdots & 0 & 1 \\ 0 & 0 & 0 & 0 & \cdots & \cdots & -\omega_r^2 & -2\,\xi_r\,\omega_r \end{bmatrix}$$

---(3.26a)

$$\mathcal{G} = \begin{bmatrix} 0 & 0 & 0 & 0 & \cdots & 0 \\ {}^{(1)}\Phi_1 & {}^{(1)}\Phi_2 & {}^{(1)}\Phi_3 & {}^{(1)}\Phi_4 & \cdots & {}^{(1)}\Phi_a \\ 0 & 0 & 0 & 0 & \cdots & 0 \\ {}^{(2)}\Phi_1 & {}^{(2)}\Phi_2 & {}^{(2)}\Phi_3 & {}^{(2)}\Phi_4 & \cdots & {}^{(2)}\Phi_a \\ \vdots & \vdots & \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & 0 & 0 & \cdots & 0 \\ {}^{(r)}\Phi_1 & {}^{(r)}\Phi_2 & {}^{(r)}\Phi_3 & {}^{(r)}\Phi_4 & \cdots & {}^{(r)}\Phi_a \end{bmatrix}$$

---(3.26b)

$\mathcal{G}$ is known as the input coupling matrix, and is given as

$\mathcal{H}$ is known as output coupling matrix, and is given as

$$\mathcal{H} = \begin{bmatrix} {}^{(1)}\Phi_1 & 0 & {}^{(2)}\Phi_1 & 0 & \cdots & {}^{(r)}\Phi_1 & 0 \\ {}^{(1)}\Phi_2 & 0 & {}^{(2)}\Phi_2 & 0 & \cdots & {}^{(r)}\Phi_2 & 0 \\ {}^{(1)}\Phi_3 & 0 & {}^{(2)}\Phi_3 & 0 & \cdots & {}^{(r)}\Phi_3 & 0 \\ {}^{(1)}\Phi_4 & 0 & {}^{(2)}\Phi_4 & 0 & \cdots & {}^{(r)}\Phi_4 & 0 \\ \vdots & \vdots & \vdots & \vdots & \ddots & \vdots & \vdots \\ \vdots & \vdots & \vdots & \vdots & \ddots & \vdots & \vdots \\ {}^{(1)}\Phi_s & 0 & {}^{(2)}\Phi_s & 0 & \cdots & {}^{(r)}\Phi_s & 0 \end{bmatrix}$$

---(3.26c)

3.11 PIEZOELECTRIC SENSING AND ACTUATION

3.11.1 PIEZOELECTRIC SENSING

When a piezoelectric patch, attached to the distributed structure, is subjected to a change in slope at its two edges, electric charge is developed in the system. This charge developed in the PZT patch mounted on the steel structure is given by, Butler et al (1996)

$$\delta(t) = \frac{1}{2}\left(t_s + t_p\right)\left(d_{31} + \upsilon_p d_{32}\right)\frac{E_p}{1-\upsilon_p^2} b\left(\theta_2(t) - \theta_1(t)\right)$$

---(3.27)

where $\theta_1(t)$ and $\theta_2(t)$ are respectively the slopes of end1 and end2 of the PZT patch at t instant of time. The thicknesses of the steel plate and of the PZT patch are denoted by t_s and t_p respectively. The dielectric constants of the PZT patch are denoted by d_{31} and d_{32}. The width piezoelectric patch is represented by letter b. E_s and E_p denote respectively the Young's Modulus of Elasticity for the steel and piezoelectric material used. The value of these parameters is given in tables (3.1) and (3.2).

The voltage developed due to this charge is given by the relation, Butller et al (1996)

$$\mathcal{V}(t) = \frac{\delta(t) t_p}{\varepsilon_p A_p} \qquad ---(3.28)$$

where A_p is the area of PZT patch ε_p is relative permeability.

Since all values except θ_1 (t) and θ_2 (t) are constant in equation (3.16), this equation may be written as

$$\mathcal{V}(t) = \Gamma \left(\theta_2(t) - \theta_1(t)\right) \qquad ---(3.29)$$

where Γ denotes a conversion coefficient.

3.11.2 PIEZOELECTRIC ACTUATION

A PZT patch may be mounted over the vibrating structure to provide a controlling action. When a voltage 'V(*t*) ' volt is applied across the piezoelectric patch, at the instant of time t, the bending moment 'M_f (t)' of opposite sense is produced at both the edges. Value of this bending moment is given by, Baz et al (1988)

$$M_f(t) = \left[\frac{d_{31} \; b \; E_p \left(E_s t_p t_s + E_s t_b^2 \right)}{2\left(E_p t_p + E_s t_s \right)} \right] V(t) \qquad ---(3.30)$$

Since all the parameters except V (t) are constant in relation (3.19), the following relation may be written

$$M_f\,(t)_{\,(Nm)} = \Psi \quad V\,(t)_{\,(Volts)}$$

---(3.31)

where Ψ denotes the conversion coefficient.

3.12 FORMING A , B , C MATRICES THROUGH MATLAB

Using mode shapes and frequencies of the piezoelectric actuator and sensor at 0gm the following program has been made in MATLAB for forming A, B ,C,D matrices ,which can be further used to design the controller for the plate.

```
clear all;

f1= [    12.83     ];
f2= [    30.96     ];
f3= [    70.88     ];
f4= [    92.51     ];
f5= [    106.91    ];

vx_127_1= [   -2.91E-02  ];
vx_128_1= [   0.16049    ];
vx_132_1= [   1.0211     ];
vx_136_1= [   1.2023     ];

vy_127_1= [   5.1053    ];
vy_128_1= [   6.5367    ];
vy_132_1= [   8.7009     ];
vy_136_1=[    8.2208     ];

wx1_43_1= [  -1.22E-03   ];
wx1_44_1= [  -0.26479    ];
wx1_48_1= [  -8.72E-02   ];
wx1_52_1= [  0.14261     ];

wy1_43_1= [  16.3425     ];
wy1_44_1= [  16.4938     ];
wy1_48_1= [  18.3947     ];
wy1_52_1= [  18.3698     ];
```

```
vx_127_2= [   0.48705     ];
vx_128_2= [   -8.14E-02   ];
vx_132_2= [   1.6634      ];
vx_136_2= [   1.0246      ];

vy_127_2= [   -9.2496      ];
vy_128_2= [   -10.1645     ];
vy_132_2= [   -12.6551     ];
vy_136_2= [   -15.8567     ];

wx1_43_2= [  10.6984       ];
wx1_44_2= [  10.6732       ];
wx1_48_2= [  14.2274       ];
wx1_52_2= [  14.4517       ];

wy1_43_2= [  10.9188       ];
wy1_44_2= [  12.9616       ];
wy1_48_2= [  14.1875       ];
wy1_52_2= [  12.0483       ];

vx_127_3= [   0.68834      ];
vx_128_3= [   -1.4279      ];
vx_132_3= [   -2.3555      ];
vx_136_3= [   -2.6602      ];

vy_127_3= [   -22.589      ];
vy_128_3= [   -28.201      ];
vy_132_3= [   -34.053      ];
vy_136_3= [   -35.268      ];

wx1_43_3= [  -6.8594       ];
wx1_44_3= [  -7.1981       ];
wx1_48_3= [  -8.6894       ];
wx1_52_3= [  -8.0005       ];

wy1_43_3= [  -46.024       ];
wy1_44_3= [  -47.532       ];
wy1_48_3= [  -43.737       ];
wy1_52_3= [  -43.345       ];


vx_127_4=[    1.2727       ];
vx_128_4=[    1.8683       ];
vx_132_4= [   9.619        ];
vx_136_4= [   8.7818       ];

vy_127_4= [   -0.11861      ];
vy_128_4= [   4.8862        ];
vy_132_4= [   5.9312        ];
vy_136_4= [   -4.1868       ];
```

```
wx1_43_4= [   19.346            ];
wx1_44_4= [  -20.464            ];
wx1_48_4= [  -26.092            ];
wx1_52_4= [  -24.428            ];

wy1_43_4= [  28.105             ];
wy1_44_4= [  25.282             ];
wy1_48_4= [  24.316             ];
wy1_52_4= [  27.565             ];

vx_127_5= [   1.7192             ];
vx_128_5= [-1.8784               ];
vx_132_5= [   2.8003             ];
vx_136_5= [   0.6272             ];

vy_127_5= [   -35.948            ];
vy_128_5= [   -41.686            ];
vy_132_5= [   -48.788            ];
vy_136_5= [   -57.011            ];

wx1_43_5= [  38.435             ];
wx1_44_5= [  38.797             ];
wx1_48_5= [  47.135             ];
wx1_52_5= [  47.259             ];


wy1_43_5= [  20.937             ];
wy1_44_5= [  26.979             ];
wy1_48_5= [  22.652             ];
wy1_52_5= [  18.814             ];


d1= [ 0.0061    ];

d2= [ 0.0061    ];

d3= [ 0.0061    ];

d4= [ 0.0061    ];

d5= [ 0.0061    ];


for i=1:1:5


FF(:,:,i)=[

     0           1                   0              0

   -f1(i)^2     -2*f1(i)*d1(i)  0              0
```

```
    0          0                0           1
    0          0               -f2(i)^2     -2*f2(i)*d2(i)
    ];


GG (:,:,i)=[            0
(-
(vy_127_1(i)/2+vy_128_1(i)/2)+(vy_132_1(i)/2+vy_136_1(i)/
2)+ (vx_127_1(i)/2+ vx_136_1(i)/2)-
(vx_128_1(i)/2+vx_132_1(i)/2))
                        0
(-
(vy_127_2(i)/2+vy_128_2(i)/2)+(vy_132_2(i)/2+vy_136_2(i)/
2)+ (vx_127_2(i)/2+ vx_136_2(i)/2)-
(vx_128_2(i)/2+vx_132_2(i)/2))

          ];




HH(:,:,i)=[
-
(wy1_43_1(i)/2+wy1_44_1(i)/2)+(wy1_48_1(i)/2+wy1_52_1(i)/
2)+ (wx1_43_1(i)/2+wx1_52_1(i)/2)-
(wx1_44_1(i)/2+wx1_48_1(i)/2)                          0


-
(wy1_43_2(i)/2+wy1_44_2(i)/2)+(wy1_48_2(i)/2+wy1_52_2(i)/
2)+   (wx1_43_2(i)/2+wx1_52_2(i)/2)-
(wx1_44_2(i)/2+wx1_48_2(i)/2)                           0
           ];


KK(:,:,i)=[0 ];


sys=ss(FF(:,:,i),GG(:,:,i),HH(:,:,i),KK(:,:,i));
SYSTEM(:,:,i)=sys;
```

```
sysd=c2d(sys,(1/250));
[A(:,:,i),B(:,:,i),C(:,:,i),D(:,:,i)]=ssdata(sysd);
SYSTEM_discrete(:,:,i)=sysd;
end



for i=1:1:5

[NUM1(:,:,i),DEN1(:,:,i)] =
ss2tf(FF(:,:,i),GG(:,:,i),HH(:,:,i),KK(:,:,i),1);


end
```

CHAPTER

FOUR ROBUST CONTROL SYSTEM

4.1 GENERAL INTRODUCTION

Robustness is of crucial importance in control system design because real engineering systems are vulnerable to external disturbance and measurement noise and there are always differences between mathematical models used for design and the actual system. Typically a control engineer is required to design a controller which will stabilize a plant, if it is not stable originally, and satisfy certain performance levels in the presence of disturbance signals, noise interference, unmodelled plant dynamics and plant parameter variations. Those design objectives are best realized via the feedback control mechanism, though which brings in the issues of high cost (the use of sensors), system complexity (implementation and safety) and more concerns on stability (thus internal stability and stabilizing controllers).

Though always being appreciated, the need and importance of robustness in control systems design has been particularly brought into the limelight during the last two decades. In classical single-input single-output control, robustness is achieved by ensuring good gain and phase margins. Designing for good stability margins usually also results in good, well-damped time responses, i.e. good performance.

In vibration control of smart structures, there is continuous investigation done to have different types of active controllers for minimizing vibration of smart structures. There is development in controllers that are capable of minimizing vibration of the entire structure in a spatially-averaged sense. The proposed spatial controllers should also have sufficient robustness properties and can be implemented on real systems effectively. For these reasons, the spatial controllers would need to be implemented on real smart structure systems such as piezoelectric laminate plates and beams.

4.2 CONTROL SYSTEM REPRESENTATION

A control system or plant or process is an interconnection of components to perform certain tasks and to yield a desired response, i.e., to generate desired signal (the output), when it is driven by manipulating signal (the input). A control system is a causal, dynamic system, i.e., the output depends not only the present input but also the input at the previous time. In general, there are two categories of control systems, the open-loop systems and closed-loop systems. An open-loop system uses a controller or control actuator to obtain the design response. In an open-loop system, the output has no effect on the input. In contrast to an open-loop system, a closed-loop control system/index closed-loop system uses sensors to measure the actual output to adjust the input in order to achieve desired output. The measure of the output is called the feedback signal, and a closed-loop system is also called a feedback system.

In order to analyze and design a control system, it would be advantageous if a mathematical representation of such a relationship (a model) is available. The system dynamics is usually governed by a set of differential equations in either open-loop or closed-loop systems. In the case of linear, time-invariant systems, those differential equations are linear ordinary differential equations. By introducing appropriate state variables and simple manipulations, a linear, time-invariant, continuous-time control system can be described by the following model,

$$\ddot{y}\, x(t) = Ax(t) + Bu(t)$$

$$y(t) = Cx(t) + Du(t) \qquad (4.1)$$

where $x(t)$ € R^n is the state vector, $u(t)$ € R^m the input (control) vector, and $y(t)$ € R^p the output (measurement) vector.

With the assumption of zero initial condition of the state variables and using

Laplace transform, a transfer function matrix corresponding to the system in (4.1) can be derived as

$$G(s) = C(sI_n - A)^{-1} B + D \qquad (4.2)$$

and can be further denoted in a short form by

$$G = A, B , C, D \qquad (4.3)$$

In the case of discrete-time systems, similarly the model is given by

$$x(k + 1) = Ax(k) + Bu(k)$$
$$y(k) = Cx(k) + Du(k) \qquad (4.4)$$

with a corresponding transfer function matrix as

$$G(s) = C(zI_n - A)^{-1} B + D \qquad (4.5)$$

Or can be further denoted in a short form by

G = A, B , C, D

4.3 ROBUST STABILITY

An essential issue in control systems design is the *stability*. An unstable systemis of no practical value. That is because any control system is vulnerable to disturbances and noises in a real work environment, and the effect due to those signals would adversely destroy the expected, normal system output in an unstable system. Feedback control techniques may reduce the influence generated by uncertainties and achieve desirable performance. However, inadequate feedback controller may lead to an unstable closed-loop system though the original open-loop system is stable. In addition to nominal performance, robust stability is an important design consideration. Robust stability requires the closed-loop system to remain stable for bounded model errors. The uncertainty may be modeled in many forms such as multiplicative, inverse multiplicative, additive, and parametric, etc., and

may be located at various points in the loop. A control system interacts with its environment through
command signals, disturbance signals and noise signals, etc. Tracking error signals and actuator driving signals are also important in control systems design. For the purpose of analysis and design, appropriate measures, the norms, must be defined for describing the “size” of these signals. From the signal norms, then it is easy to define induced norms to measure the “gain” of the operator which represents the control system.

4.3.1 SMALL GAIN THEORM

The Small Gain Theorem is of central importance in the derivation of many stability tests. In general, it provides only a sufficient condition for stability and is therefore potentially conservative. The Small Gain Theorem is applicable to general operators.Consider the feedback configuration in Figure 4.1, where $G_1(s)$ and $G_2(s)$ are the transfer function matrices of corresponding linear, time-invariant systems.

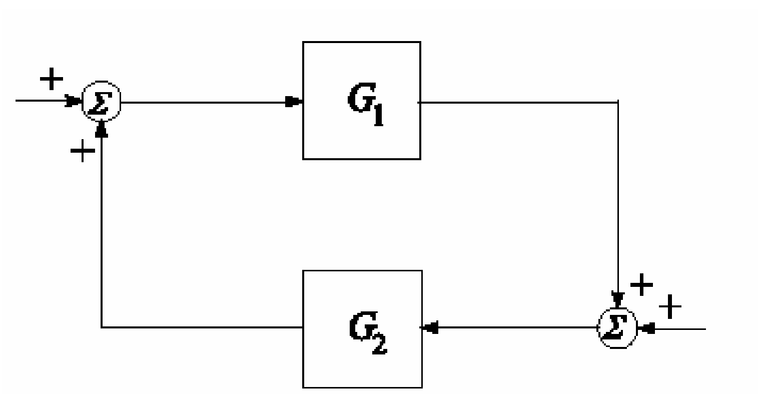


Figure 4.1 A Feedback Configuration

If G1(s) and G2(s) are stable, i.e., G1 € H_∞ , G2 € H_∞, then the closed-loop system is internally stable if and only if

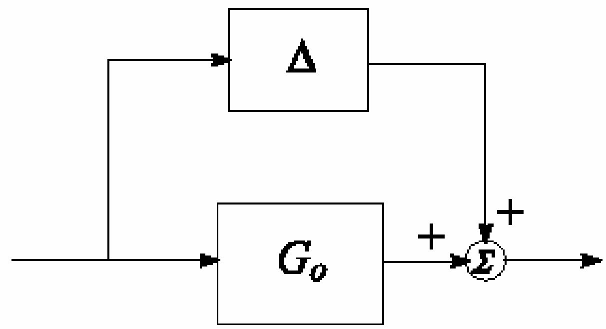
$$\| G1G2 \|_\infty < 1 \quad \text{and} \quad \| G1G2 \|_\infty < 1$$

A closed-loop system of the plant G and controller K is robustly stable if it remains stable for all possible, under certain definition, perturbations on the plant. It implies of course that K is a stabilizing controller for the nominal plant G, since it is always assumed that the perturbation set includes zero (no perturbation).

4.4 MODELLING OF UNCERTAINTIES

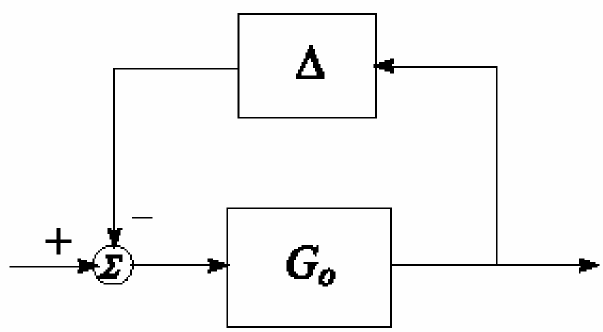
It is well understood that uncertainties are unavoidable in a real control system. The uncertainty can be classified into two categories: disturbance signals and dynamic perturbations. The former includes input and output disturbance (such as gust to an aircraft), sensor noise and actuator noise, etc. The latter represents the discrepancy between the mathematical model and the actual dynamics of the system in operation. A mathematical model of any real system is always just an approximation of the true, physical reality of the system dynamics. Typical sources of the discrepancy include unmodelled (usually high frequency) dynamics, neglected nonlinearities in the modeling , effects of deliberate reduced order models, and system parameter variations due to environmental changes and torn and worn factors. Those modelling errors may adversely affect the stability and performance of a control system.

Many dynamic perturbations which may occur in different parts of a system can however be lumped into one single perturbation block Δ, for instance, some unmodelled , high frequency dynamics. This uncertainty representation is referred to as "unstructured" uncertainty. In the case of linear, time-invariant systems, the block Δ
may be represented by an unknown transfer function matrix. The unstructured dynamics uncertainty in a control system can be described in different ways, such as shown in the following Figure 4.2 and 4.3 , where $G_p(s)$ denotes the actual, perturbed system dynamics and $G_o(s)$ a nominal model description of the physical system.



$$G_p(s) = G_o(s) + \Delta(s)$$

Figure 4.2 Additive Perturbation Configuration



$$(G_p(s))^{-1} = (G_o(s))^{-1} + \Delta(s)$$

Figure 4.3 Inverse Additive Perturbation Configuration

4.5 LINEAR FRACTIONAL TRANSFORMATION

The block diagram in Figure 4.4 can be generalized to be a standard configuration to represent how the uncertainty affect the input/output relationship of the control system under study.

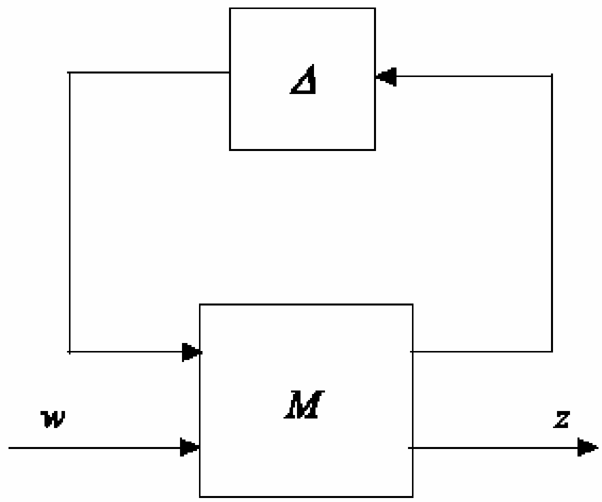


Figure 4.4 Upper LFT Configuration

When M is an interconnected transfer function M(s). The perturbation block Δ in Figure 4.4 corresponds to parameter variations and is called “parametric uncertainty”. The uncertain block Δ is not a full matrix but a diagonal one. It has certain structure, hence the terminology of “structured uncertainty”. F(M,Δ) is called a *linear fractional transformation(LFT)* of *M* and *Δ*. Because the “upper”loop of *M* is closed by the block *Δ*, this kind of linear fractional transformation is also called an *upper linear fractional transformation (ULFT)*, and denoted with a subscript *u*, i.e., $F_u(M,\Delta)$, to show the way of connection. Dually, there are also *lower linear fractional transformations(LLFT)* which are usually used to indicate the incorporation of a controller *K* into a system as shown in figure 4.5.

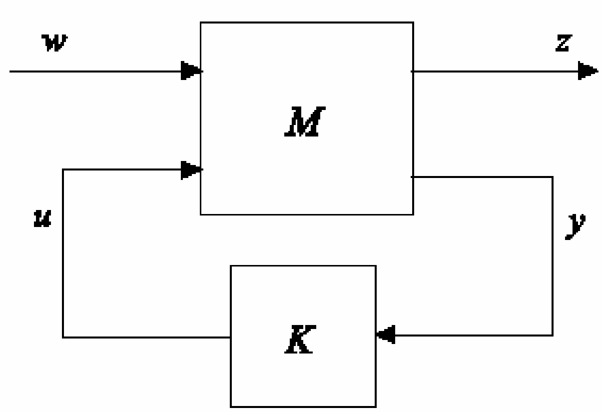


Figure 4.5 Lower LFT Configuration

4.6 H_∞ DESIGN FOR CONTROLLER

A control system is *robust* if it remains stable and achieves certain performance criteria in the presence of possible uncertainties. The robust design is to find a controller, for a given system, such that the closed-loop system is robust. The H_∞ optimization approach and its related, being developed in the last two decades and still an active research area, have been shown to be effective and effcient robust design methods for linear, time-invariant control systems. Every practising control engineer knows very well that it will never be appropriate in any industrial design to use just a single cost function . A reasonable design would use a combination of those functions. For instance, a general mixed sensitivity problem as shown in figure 4.6

$$\min_{K stabilizing} \left\| \begin{matrix} (I+GK)^{-1} \\ K(I+GK)^{-1} \end{matrix} \right\|_\infty . \tag{4.6}$$

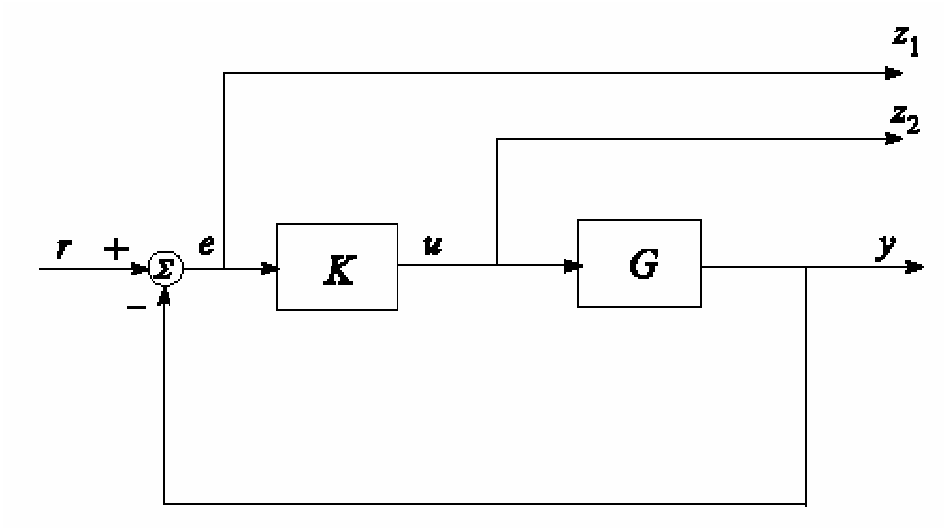


Figure 4.6 A Mixed Sensitivity Consideration

This cost function can also be interpreted as the design objectives of nominal Performance , good tracking or disturbance attenuation, and robust stabilization, with regard to additive perturbation. In order adopt a unified solution procedure, the above cost function (eq.4.6) can be recast into a standard configuration as in Figure 4.7. This can be obtained by using the (Linear Fractional Transformation) LFT technique and by specifying/grouping signals into sets of external inputs, outputs, input to the controller and output from the controller which of course is the control signal. Note that in Figure 4.7 all the external inputs are denoted by w, z denotes the output signals to be minimized/penalized which includes both performance and robustness measures, y is the vector of measurements available to the controller K and u the vector of control signals. $P(s)$ is called the *generalized* plant or *interconnected* system. The objective can be to find a stabilizing controller K to minimize the output z, in the sense of energy, over all w with energy less than or equal to 1. Thus, it is equivalent to minimize the H_∞ norm of the transfer function from w to z.

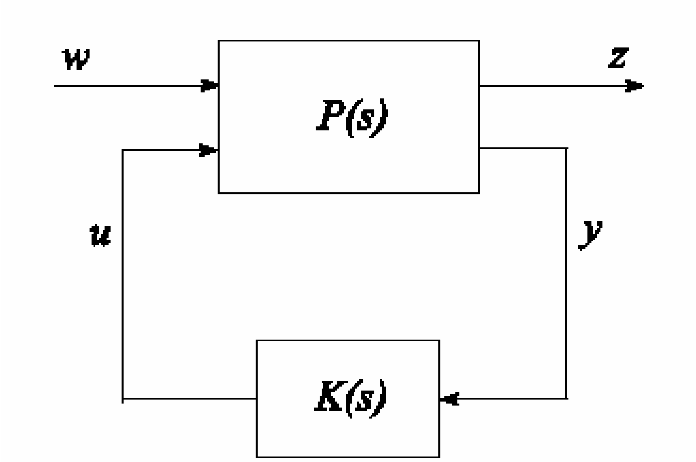


Figure 4.7 The standard H∞ Configuration

The above general mixed sensitivity problem (or, so called *S over KS*) problem, and also Other mixed cases of cost transfer function matrices such as *S over T*, *S over T over KS*, etc. can be dealt with similarly to formulate into the standard configuration. In practical designs, it is often necessary to include (closed-loop) weights with those cost functions. For instance, instead of (eq. 4.6), we may have to consider with $z_1 = W_1 e$ and $z_2 = W_2 u$, H∞ control design theory, involves defining (possibly frequency-dependent) weights on the inputs and outputs such that the performance objectives are satisfied by minimizing

$$\min_{K\ stabilizing} \left\| \begin{matrix} W_1(I+GK)^{-1} \\ W_2K(I+GK)^{-1} \end{matrix} \right\|_\infty .$$

(4.7)

Several formulations of cost function are applicable in the robust controller design.

For instances, the weighted *S/KS* and *S/T* design methods. The optimization of *S/KS*, where *S* is the sensitivity function and *K* the controller to be designed, could achieve the nominal performance in terms of tracking or output disturbance rejection and robustly stabilize the system against additive model perturbations. On the other hand , the mixed sensitivity optimization of *S/T* , where *T* is the complementary sensitivity function, could achieve robust stability against multiplicative model perturbations in addition to the nominal performance. Both of them are useful robust controller design methods. The significant benefit of H_∞ theory is that robustness to unstructured model errors is explicitly factored into the design process.

Based on the model obtained in previous chapter, an H_∞ controller is designed for the smart plate. The goal of the controller is to attenuate the vibrations of the smart plate at its first two flexural frequencies (in the range from zero to 100 Hz) and gain stabilizes the unmodeled high frequency modes. In H_∞ control design framework, the objective is to minimize the H_∞ norm of the weighted transfer functions from the input disturbance signals to the output error signals, the uncertainties in the plant model can be put in such a form that some of the disturbances and error signals correspond to the channels through which the nominal model interacts with a norm bounded uncertainty block. Δ. This generates the set of plants in which the true plant is assumed to exist. This framework is represented in Figure 4.8

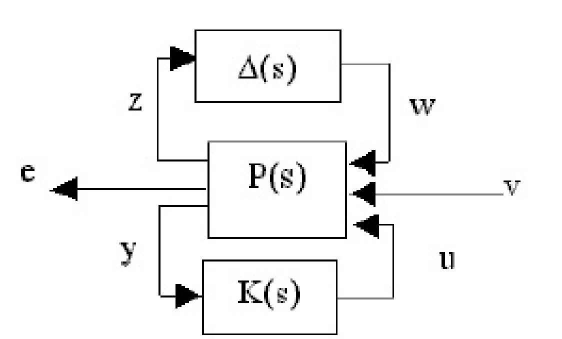


Figure 4.8 Modeling of uncertainties

Here, *P* is the nominal plant model with appropriate weights to reflect the design goals, *K* is the controller to be designed, and Δ is the norm bounded uncertainty block, v is a vector of exogenous inputs such as reference commands, disturbances and noise, *e* is a vector of error signals to be kept small, y is a vector of sensor measurements and u is a vector of control signals, w and z are the disturbance and error channels corresponding to the uncertainty block Δ respectively. For the design purposes, the Δ block is eliminated and the input-output map from *[w v]T* to *[z e]T* is expressed in lower linear fractional transformation form $F_l\ (P, K)$ as

$$\begin{bmatrix} z \\ e \end{bmatrix} = F_l(P, K)\begin{bmatrix} w \\ v \end{bmatrix} \tag{4.8}$$

Where, $F_l\ (P, K) = P_{11} + P_{12}\ K(I - P_{22}\ K)^{-1}\ P_{21}$

Assuming that plant *P* is partitioned according to the dimensions of the control, measurement, disturbance and error signals, as

$$P = \begin{bmatrix} P_{11} & P_{12} \\ P_{21} & P_{22} \end{bmatrix} \tag{4.9}$$

The objective is to find a stabilizing controller *K* that minimizes the ∞-norm of $\|\ F_l\ (P, K)\ \|_\infty$. For an uncertainty block *satisfying* $\|\ \Delta\ \|_\infty < 1$ the closed loop system in Figure 4.9 has robust performance *if* $\|\ F_l\ (P, K)\ \|_\infty \leq 1$ achieved .

This result, however, is conservative because it assumes that the delta block is a full block with no structure to it. The uncertainties in a realistic problem are due to the components of a system, and representation of such uncertainties results in a block diagonal Δ.

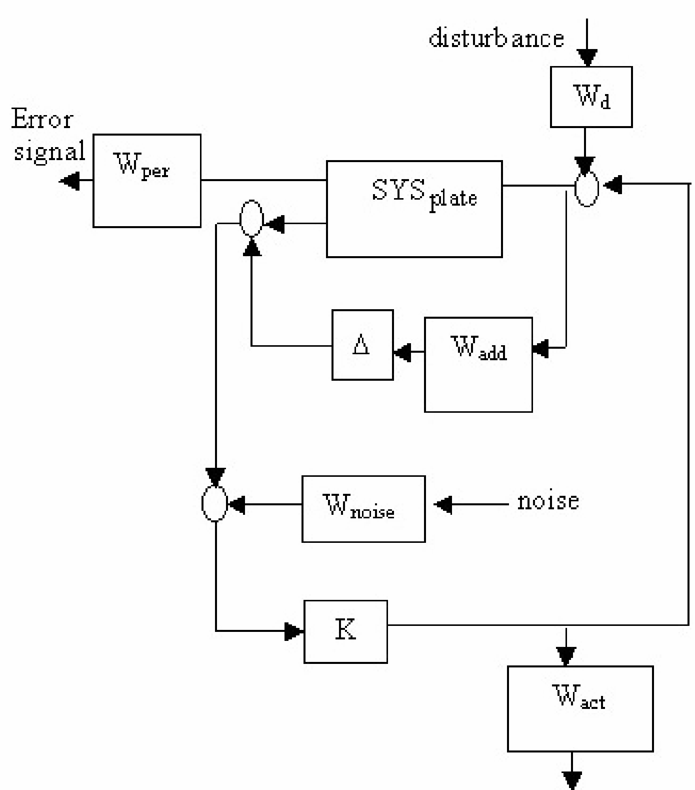


Figure 4.9 The Block diagram formulation of the control problem

In H_∞ controller design for the smart plate, the performance objective is to minimize the
maximum frequency response of the first two flexural modes of the smart plate at the sensor locations. Figure 4.9 shows the formulation of the closed loop control in H_∞ framework. *SYSplate* defines the nominal smart plate model, Wper represents a performance weight on the sensors to achieve the performance objective. This weight is selected to achieve attenuation in the peak frequency response of the closed-loop system.

An additive uncertainty is included in the problem formulation to account for the unmodeled high frequency modes and modeling errors inside the controller bandwidth. This weight is selected to have a magnitude greater than the structural modes above 500 rad/sec. If robust stability of the closed-loop system is achieved for this additive uncertainty model, the flexible modes of the structure will be gain stabilized above 500 rad/sec. The weights on the disturbance input, Wdist is taken to be 1. This indicates that the input disturbance is expected to be on the same order of magnitude as the controller signals. A 19th order controller is obtained by applying the standard solution techniques to the system formulated in Figure 4.8. For validation , this controller is compared by a 18^{th} order model of the smart plate obtained from the experimental data by Yavuz yaman et. al 2000. Open and closed loop frequency response of the channel is shown in Figure 4.10. The comparison of the time domain responses are shown in Figure 4.11

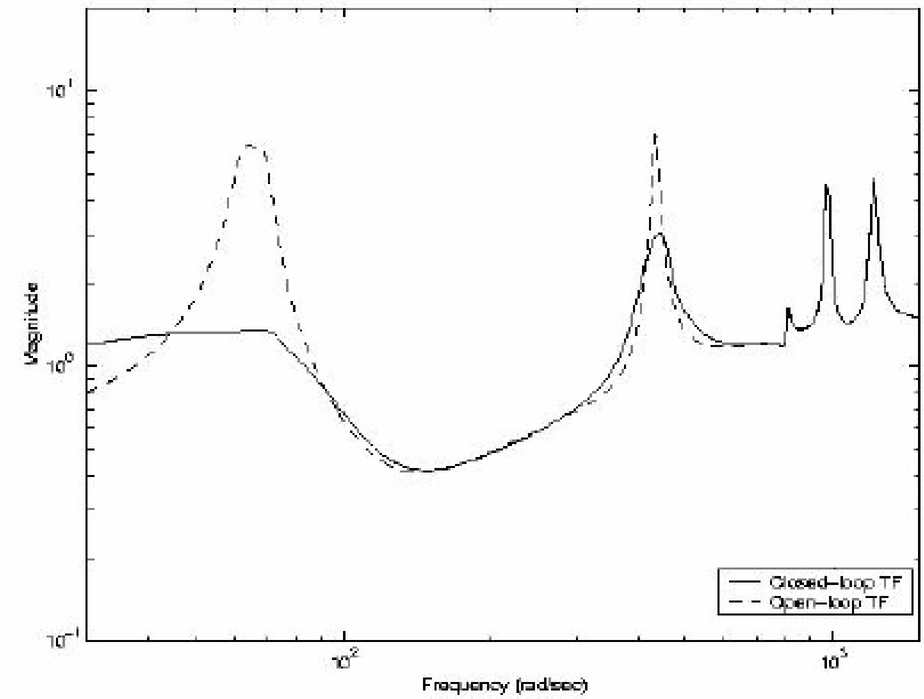


Figure 4.10 The comparison of open loop and closed loop responses of the smart plate

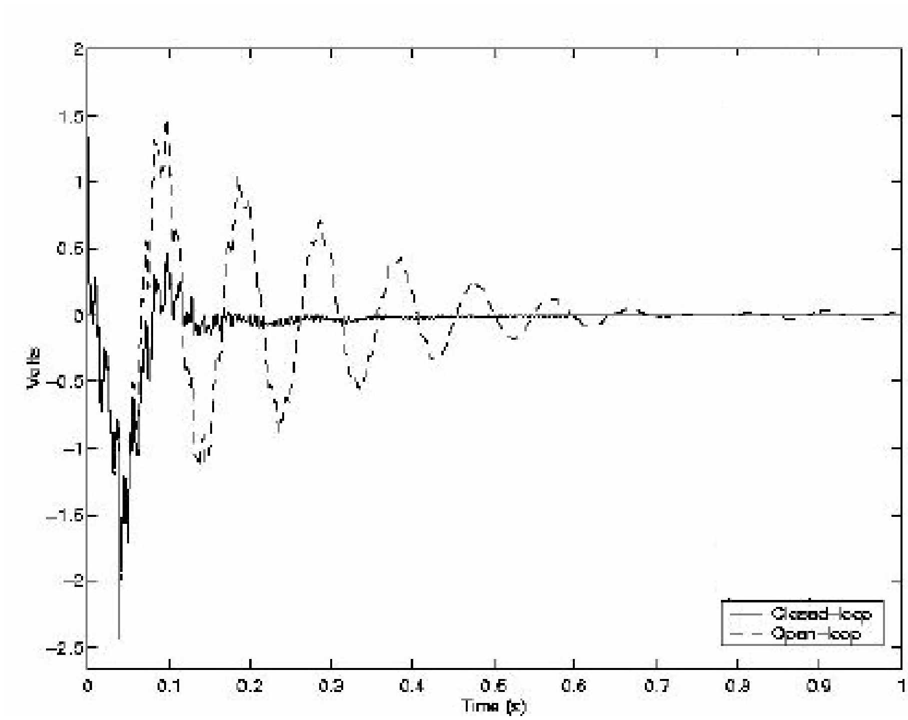


Figure 4.11 The comparison of open loop and closed loop response to a step input

If the robust stability of the closed-loop system is achieved for the additive uncertainty model, the flexible modes of the structure will be gain stabilized above 500 rad/sec. This indicates that the robust performance requirements are satisfied for the closed loop system.

4.7 CONCLUSION

A Clamped plate of dimensions (200 X 200 X 0.54) mm bonded with piezoceramic material of dimensions (20 X 20 X 1)mm is considered for study. FEM modeling of plate is done using ANSYS(vs 5.6). Results were then validated against the experimental work done by Yavuz et al 2000. Using h-infinity method, a required robust controller is designed for the plate .

REFERENCES

1984…Rao JS, Gupta K, "Theory and Practice of Mechanical Vibrations", New Age International Publishers, New Delhi.

1985…Papp. a R.S. and Juang J.N, "An Eigensystem Realization Algorithm for Modal Parameter Identification and Model Reduction", *Journal of Guidance* , 8(5), pp. . 620- 627.

1985 Bailey, T. Hubbart, J. E. , DistributedPiezoelectric-Polymer Active Vibration Control of

a Cantilever Beam. *Journal of Guidance, Controland Dynamics*, Vol.8, No. 5, pp 605-611.

1986a… Meirovitch, "Elements of Vibration Analysis", McGraw Hill Publishing Company, New York.

1987....Crawley and De Luis" Use of Piezoelectric Actuators as Elements of Intelligent Structures,"*AIAA journals*, Vol. 25, No.19, pp.1373-1385.

1987.. Plump" Nonlinear Control of a Distributed System: Simulation and Experimental results ,"ASME journals *of dynamic systems, measurement and control*,"Vol.113,no.11,pp.133-139.

1987...Pandita"A Multi-Purpose Dynamic and Tactile Sensor for Robot Manipulator ,"*J.Robotic Systems*, Vol.4-6,pp.719-741.

1987...Baz "Optimum Vibration Control of Flexible Beams by Piezoelectric Actuators ," *NASA CR_180209*.

1988...Fanson"Experimental Studies of Active Members in Control of Large Space Structure,"*AIAA journal* pp.88-2207 *proceedings of AIAA/ASME/AHS 29th structure dynamic and material conference*,pp.9-17.

1988..Tzou and Tseng"Active Vibration Control of Distributed Parameters Systems by Finite Element Methods,"*ASME Computer in Engineering* ,Vol.3,pp.599-604.

1988…Baz A and Poh S. "Performance of active Control System with Piezoelectric crystals", Journal of sound and vibration, 126(2), pp. .327-343.

1989...Tzou and Grade " Theoretical Analysis of a multilayered Thin Shell Coupled with piezoelectric Shell Actuator for distribution vibration controls", *journals of sound and vibration* ,Vol.132,No.3,pp.433-450.

1989…Long man R.W. and Juang J.N. , "Recursive Form of the Eigen-system Realization Algorithm for System Identification", *Journal of Guidance,* 12(5), pp. . 647-652.

1989... E. F. Crawley, J. Louis, Use of Piezoelectric Actuators as Elements of Intelligent Structures. *AIAA Journal*, October.

1990…Cooper J.E, "Identification of time varying modal parameters", *Aeronautical Journal of America*, 10(6), pp. 271-278.

1990…Tzes A.P. and Yurkowich S,. "A Frequency domain Identification Scheme for Flexible Structure Control", Journal *of sound and vibration*, 112(1), pp. 427-434.

1991…Ishitobi M, Fukumoto Y., Mizumoto I., Iwai Z., “Adaptive vibration control using a delta operator” *Transactions of the Japan society of mechanical Engineers, part c,* 57 (544), pp. 3782-3787.

1991...Tzou “Distributed Modal Identification and Vibration Control of Continua: Theory and Application”, ASME *journals of dynamic systems,mearurement and control*,Vol.113,no.11,pp.500-595.

1991…Bayard S and Hadaegh F.Y., “Automated On-Orbit Frequency Domain Identification for Large Space Structures”, *Automatica,* 27(6), pp. 931-946.

1991...Bozich D.J., Mackay H. B., “Vibration Cancellation at multiple locations using neuro controllers with real-time learning”. *Proceedings IJCNN- International joint conference on Neural networks IJCNN 91 Seattle,* pp. 775-780.

1991...E. K. Dimitridis C., R. Fuller, C. A. Rogers Piezoelectric Actuators for Distributed Vibration excitation of Thin Plates, *Journal of Vibration and Acoustics*, vol. 113, pp. 100-107, Jan.

1992...Cooper J.E. and Wright J.R., “Spacecraft In-Orbit Identification Using Eigensystem Realization Methods” *Aeronautical Journal of America*, 15(2), pp. 353-359.

1992…Wimmel R, Melcher J., “Application of recursive adaptive algorithms for system identification and vibration control”, *Journal of intelligent Material System and Structures,* 3 (3), pp. 519-535.

1992…Knospe C.R., Haviland J. K., “Pulse response method for vibration reduction in periodic dynamic systems”, *Journal of guidance, Control and dynamics,* 15 (3), pp. 782-784.

1992…Pan, J, Berinson D., “Adaptive Control of vibration in a multiple mount system”, *National Conference publication- institution of Engineers, Australia*, 92 (15), pp. .87-95.

1992…Flowers, George T., Manikkam, Madheswaran; Venkayya vipp. erla B, “Study of the app. lication of adaptive decenterlazied control to a multibody space structure", *American Society of mechanical Engineers, Applied mechanics Division AMD* 141, pp. 187-195.

1992…Snyder, S.D., Hansen C.H., Clark R. L., "Convergence characteristics of the multiple input, multiple output LMS algorithm", *Journal of intelligent Material System and Structures*, 3(1), pp. 115- 133.

1993…Snyder S. D., Tanaka N., "Neural Network for feedforward controlled smart structures", *Journal of intelligent material system and structures*, 4 (3), pp. 373-378.

1993…Prakah- A, Kwaku O and Craig K.C., ".Multichannel control design for vibration minimization of an active structure", *Recent Development in stability , vibration and control structural System American society of Mechanical engineers, Applied mechanics Division*, New York, NY, U.S.A., AMD 167, pp. 35-48.

1993…Tanaka N., Snyder S.D., Kikushima Y. and Kuroda M., "Active control of nonlinear vibration using a neural network", *Transactions of the Japan Society of Mechanical Engineers,* Part C 59, pp. 700-707.

1993…Peters L., beck, K. and Camposano R., "Adaptive Fuzzy controller improves comfort", *IEEE International conference on Fuzzy System Second IEEE Int. Conf. on Fuzzy Syst, published by IEEE Service center,* Piscataway, NJ, USA, pp. 512-515.

1993…Librescu L., Meirovitch L. and Song O., "Integrated Structural tailoring and adaptive control of advanced flight vehicle structural vibration", *Collection of Technical papers- AIAA/ASME structures, Structural Dynamics and Materials Conference pt 6, AIAA* Washington, DC U.S.A, pp. 3457-3462.

1994… Bayard DS, "Higher – order Multivariable transfer function curve fitting: Algorithms, Sparse Matrix Methods and Experimental Results", *Automatica,* vol. 30, no.9, pp. 1439 –1444.

1994...Shen"Bending Vibration Control of Composite and Isotropic Plates Through the Intelligent Constrained Treatments,"*Smart Materials and structures*,3,pp.59-70.

1994…Spanos J.T., Rahman Z. H., " Narrow-band control experiments in active vibration isolation" *Proceeings of SPIE- The International society for optical Engineering 2264, Society of photo-optical Instrumentation Engineers,* Bellingham, WA, USA, pp. 13-19.

1995…Sutton T.J and Elliott S.J, "Active attenuation of periodic vibration in nonlinear system using an adaptive harmonic controller", *Journal of vibration and acoustics*, 117 (3), pp. 355-362.

1995…Nam C, Kim Y. and Weisshaar T. A., "Optical sizing and placement of Piezo actutators for active flutter suppression*", Proceedings of SPIE- The International society for optical Engineering vol. 2443. Society of Photo-optical Instrumentation Engineers,* Bellingham, WA, USA, pp. 40-51.

1995…Gopinathan M, Pajunen G, Neelakanta P.S. and Arockiasamy M., "Linear quadratic distributed self-tunning control of vibration in a cantilever beam." *Proceedings of SPIE- The International society for optical Engineering vol. 2443. Society of photo-optical instrumentation Engineers,* Bellingham, WA USA, pp. 542-553.

1995…Prakah- A. K and Craig K.C., ".Development of active structural control methods for vibration and acoustic radiation reduction", *Proceedings of SPIE - The international society for optical engineering , 2441, Society of photo- Optical Instrumentation Engineers,* Bellinghan, WA, USA, pp. 305-319.

1995…Snyder S. D. and Tanaka N., "Active control of vibration using a neural network." *IEEE Transactions on Neural Networks*, 6 (4), pp. 819- 828.

1995…Vipperman J.S. and Burdisso R.A., "Adaptive feedforward control of non-minimum phase structural systems", *Journal of sound and vibration*, 183 (3), pp. 369-382.

1996…Butler R., "A State – Space Modeling and Control Method for Multivariable Smart Structural System", *Int. Journal of Smart Materials and Structures*, vol. 5, pp. 386-389.

1996…Ashokanathan S.F. and Milford R.I. , " Experimental On-Line Frequency Domain Identification and Adaptive Control of a Flexible slewing beam", *Journal of Dynamic System Measurement and Control*, 118 (1), pp. 58-65.

1996… Callafon RA, Roover DV den , Hoff PMJ, "Multivariable least squares Frequency Domain Identification using Polynomial Matrix Fraction Descriptions", *Proc. IEEE conference on Decision and control*, pp. 2030 – 2035.

1997…Yen Gary G.,"Frequency-domain vibration control using adaptive neural network" *IEEE International conference on Neural Networks- Conference proceedings vol.2 IEEE, Piscatawaqy*, NJ, USA,97CB36109, pp. 806-810

1997…Luzardo J. A., Chassiakos A. and Ryaciotaki B.H.,"Adjustable neural network controller: Application to a large segmented reflector", *Proceedings of the American Control conference vol. 1, IEEE, Piscataway,* NJ, USA, 97CH36041, pp. 227-231.

1997...Baz "Optimization of Energy Dissipation Characteristics of Active Constrained Layer Damping,"*Smart Materials and Structure* 6(3),pp.360-368.

1997....Shen"A Variational Formulation: a work-energy relation and damping mechanisms of active constrained layer treatment," *journals of Vibration and Acoustics* 119(2), pp.192-199.

1990..Zhang "Controolable Constrained Damping Layer,"*journals of Xi' an Jiaotong University* 32(8).pp.118-121

2000...yavuz et al "Active Vibration Control of a Smart Plate" ICAS 2000 Congress

2001...S.L xie"H-infinity Robust Vibration Control of Thin Plate Covered with a Controllable Constrained Damping Layer," *journals of Vibration and Control*, Vol.10, pp.115-133.