ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE ENGINEERING AND TECHNOLOGY

ACTIVE VIBRATION CONTROL OF A CANTILEVER BEAM

M.Sc. THESIS

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Department of Mechanical Engineering

Solid Mechanics Programme

JANUARY 2013

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ANKASTRE BİR KİRİŞİN AKTİF TİTREŞİM KONTROLÜ

YÜKSEK LİSANS TEZİ

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OCAK 2013

Mehmet Volkan KAZANCI, a **M.Sc.** student of ITU **Graduate School of Science Engineering and Technology**, student ID **503091538**, successfully defended the thesis entitled **"ACTIVE VIBRATION CONTROL OF A CANTILEVER BEAM"**, which he prepared after fulfilling the requirements specified in the associated legislations, before the jury whose signatures are below.

Thesis Advisor :	**Prof. Dr. Kenan Yüce ŞANLITÜRK** İstanbul Technical University	....
Jury Members :	**Prof. Dr. Metin GÜRGÖZE** İstanbul Technical University	....
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To my father,

FOREWORD

First and foremost, I would like to express my deepest gratitude and appreciation to my supervisor, Prof.Dr. Kenan Yüce ŞANLITÜRK, who shared his wise experiences with me and encouraged me through the most intricate times. His invaluable devotion has been the biggest motivation to successfully accomplish this project. Thus far, it has been a great honour and privilege to be his student; equally as it will always be.

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In loving memory of my father, I would like to commemorate him with all my heart and respect. In his short beloved life, he bequeathed me the utmost understanding and awareness of knowledge, learning, and education. Without my loving FAMILY and dear FRIENDS, none of this work would have come to reality. Their patience, sacrifices, influences, and above all, their existence enabled me to strive through any obstacles in life and enriched me with outstanding experiences and memories.

January 2013

Mehmet Volkan KAZANCI
(Mechanical Engineer)

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ABBREVIATIONS

FRF	**:** Frequency Response Function
PVDF	**:** Polyvinylenedifluoride
SMA	**:** Shape Memory Alloys
PZT	**:** Lead Zirconate Titanate
SHM	**:** Structural Health Monitoring
LQR	**:** Linear Quadratic Regulator
PID	**:** Proportional Integral Derivative
PI	**:** Proportional Integral
P	**:** Proportional
RC	**:** Resistor and Capacitor Circuit
SISO	**:** Single Input Single Output
SIMO	**:** Single Input Multi Output
MIMO	**:** Multi Input Multi Output
SDOF	**:** Single Degree of Freedom

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LIST OF SYMBOLS

ε	: Strain vector
S	: Matrix of compliance coefficients
D	: Vector of electric displacement
g	: Matrix of piezoelectric constants
σ	: Stress vector
E	: Vector of applied electric field
ξ	: Permitivity
d	: Matrix of piezoelectric strain constant
β	: Impermitivity component
f_p	: Primary force
M	: Mass
C	: Viscous damping
K	: Stiffness
x	: Displacement
$\dot{x}$	: Velocity
$\ddot{x}$	: Acceleration
f_c	: Control force
g_d	: Displacement gain constant
g_v	: Velocity gain constant
g_a	: Acceleration gain constant
F_p	: Laplace transform of primary force
F_c	: Laplace transform of control force
[M]	: Mass matrix
[C]	: Viscous damping matrix
[K]	: Stiffness matrix
{q(t)}	: Generalized structural displacement vector
{u(t)}	: Control output actuation vector
$G(s)$	: Structure, s-domain
$C(s)$	: Controller, s-domain
E	: Tracking error
R	: Desired response value
K_p	: Proportional gain
K_i	: Integral gain
K_d	: Derivative gain
K_{PID}	: Transfer function of PID controller
$\mathrm{K_d}$	: Displacement feedback gain
$\mathrm{K_v}$	: Velocity feedback gain
$\mathrm{K_a}$	: Acceleration feedback gain
ω	: Angular velocity in terms of rad/s
ω'	: Angular velocity of the controlled structure in terms of rad/s
H	: Frequency response of the SDOF system

ζ	: Viscous damping ratio
ζ'	: Viscous damping ratio of the controlled structure
$\lvert H \rvert$	: Normalized modulus of the closed loop frequency response function
$\{X\}$	: Displacement vector
$\{\psi\}_r$	: $\mathrm{r^{th}}$ mode shape
N	: Number of modes
ω_r	: $\mathrm{r^{th}}$ natural frequency
λ	: Eigenvalue
η	: Modal damping
ζ	: Damping ratio
ϕ	: Mass normalized mode shape
α	: FRF matrix
m_r	: Mass for the $\mathrm{r^{th}}$ mode
p	: Principle coordinates as the vector of modal coordinates
η	: Loss factor
f_i	: Modal force

ACTIVE VIBRATION CONTROL OF A CANTILEVER BEAM

SUMMARY

In the several last decades, a tremendous amount of interest has been generated among the academic and industrial studies dealing with active vibration control technology which offers an alternative approach when the passive vibration isolation techniques are not sufficient especially at lower frequencies. Not only the simple structures with relatively lower natural frequencies but also quite complicated structures and parts are good candidates for the application of active vibration control methods with the rapid development of this technology.

The main purpose of this thesis is to investigate the active vibration control of a cantilever steel beam via performing both time domain and frequency domain analyses. Then, the performance of experimental setups as well as the close loop vibration response characteristics of the cantilever beam is studied.

First of all, a literature survey which is about the active vibration control studies and advancements of methods is elaborately presented. Following this, in the second part, the background theory which is concerned with the physical principles and the theory of active vibration control are stated. Then, the dynamics of a single degree of freedom system is described in this section. Furthermore, fundamentals of active vibration control studies are explained in detail and it is shown theoretically that it is possible to modify systems by applying an additional control force proportional to displacement, velocity and acceleration or the combination of those individually so as to cause stiffness, damping and mass modification to the system as desired. In essence, the experiments for active vibration control performance are conducted via altering the actuation gains of feedback signals and the effects on modal parameters of the structure are investigated.

Various active control test cases are investigated using the experimental test rig. Electrodynamic shaker is used as an actuator and laser measurement device is used as a sensor. The first case is for the examination of the controller designed for displacement feedback control in order to apply an additional controlled stiffness effect to the structure. The second case is basically active vibration suppression by adding an artificial damping to the system using velocity feedback signal. Similarly the third case is conducted with the aim of applying an additional negative stiffness or adding a controlled mass to the structure. Finally the use of an analogue phase shifter is introduced in some detail. It is utilized to enhance the controller systems by adjusting the phase angle between the measurement signal and feedback signal.

Briefly, in this thesis, an electrodynamic shaker is utilized as an actuator to apply feedback signals on the cantilever beam structure. Stiffness, damping and mass effect of the controller systems on the structure are observed individually. The performances of feedback controllers are investigated for the first mode of the beam. Although the experimental results have been found to be quite promising, some

aspects of these experiments need further development and research. An analogue phase shifter circuit is enhanced to feedback system so as to modify the phase angle between input and output signals. The results of experimental studies are compared and discussed in time and frequency-domains, respectively.

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ÖZET

Aktif titreşim kontrolü teknolojisi, pasif kontrol tekniklerinin bir sınıra ulaşması ve bilhassa düşük frekanslarda yetersiz kalması ile konu üzerinde son yıllarda hem akademik hem de endüstriyel pek çok çalışma gerçekleştirilmesine sebep olmuştur. Teknolojinin de hızlı gelişmesi ile sadece göreceli olarak düşük doğal frekanslara sahip basit yapılar değil, karmaşık parçalar veya yapılar da aktif titreşim kontrolü uygulamaları için uygun adaylar haline gelmiştir.

Bu tez çalışmasında, tek tarafından sabitlenmiş diğer tarafından serbest bırakılmış çelik bir çubuğun aktif kontrolü uygulamasının zaman ve frekans tabanında incelenmesi amaçlanmıştır. Böylelikle, aktif titreşim kontrolü amacıyla farklı geri besleme sinyalleri içeren deneysel düzenekler kurulmuş, denetçi performanslarının ilgili çubuğun birinci modu üzerine etkileri incelenmiştir

İlk bölümde, geniş bir literatür taraması sonucunda daha önce yayınlanmış çalışmalar ve kullanılan metotlar incelenmiştir. Süreç içerisindeki tarihsel gelişmelerden bahsedilmesinin yanı sıra, akıllı yapıların ve akıllı malzemelerin aktif kontrol düzeneklerinde kullanılması ile gerçekleşen hızlı ilerleme gösterilmiştir. Akıllı malzemelerin, bilhassa piezoelektrik malzemelerin, hem algılayıcı hem de eyleyici olarak deneysel düzeneklerde kullanılmasıyla daha gelişmiş kontrol algoritmalarının geliştirmesine olanak sağlanmıştır. Endüstriyel uygulamaların yanı sıra, yapılan akademik çalışmaların pek çoğunun basit, esnek ve hafif çubuk ve plaka yapılar üzerinde gerçekleştirildiği belirlenmiş ve bu konudaki çalışmalar sınıflandırılmıştır.

Bölüm 2'de, çalışmanın altında yatan teorik altyapı sunulmuş ve böylece yapılacak sayısal ve deneysel çalışmaların temeli oluşturulmuştur. Orantısal kontrolcüler başta olmak üzere temel kontrol teorisi anlatılmıştır. Sayısal ve deneysel modal analiz titreşim modelleri anlatılmasının yanı sıra aktif titreşim kontrolünün sayısal uygulamalarını çalışmak amacıyla tek serbestlik dereceli bir sistemde sırasıyla deplasman, hız ve ivme geri besleme kapalı çevrimleri kurulmuş, sistemlerin frekans tepki fonksiyonları karşılaştırılmıştır.

Sayısal çalışmalarda, beklenildiği üzere deplasman ölçümlerinin tek serbestlik dereceli sisteme geri beslenmesiyle sisteme ek bir direngenlik kazandırılmış, geri beslenen sinyalin genliklerinin orantısal olarak arttırılması, aktif sistemin de orantısal olarak direngenliğinin artmasına ve doğal frekansının artmasına sebep olmuştur. Aynı şekilde sisteme geri beslenen ölçülmüş hız sinyalleri sistemin rezonans genliklerinin azalmasına sebep olmuş, artan geri besleme sinyallerinin genliklerinin artmasıyla sisteme aktif olarak uygulanan sönüm orantısal olarak artmıştır. Öte yandan, sisteme geri beslenen ölçülmüş ivme sinyalleri tek serbestlik dereceli yapıya ek kütle etkisi oluşturmuş ve doğal frekansının geri beslenen sinyal genliklerine oranla azalmasına sebep olmuştur. Böylece, bu sayısal çalışmalar üzerinden sonraki bölümlerde incelenecek deneysel çalışmalara temel hazırlanmıştır.

Bölüm 3'de, Aktif titreşim kontrolü çalışmalarında kullanılan deney düzenekleri, bu çalışmaların başarıya ulaşması ve uygulanacak denetleyici düzeneklerinin kararlılığı açısından önem arz etmektedir. Bu sebeple düzenekte görevlendirilmiş her bir elemanın özellikleri ayrı ayrı incelenmelidir. Hem kapalı hem de açık döngülerde görevlendirilmiş her bir eleman ayrıntılarıyla işlenmiştir.

Deneysel çalışmalarda eyleyici olarak elektrodinamik modal sarsıcı kullanılmış, algılayıcı olarak ise lazer ölçüm cihazı sistemde görevlendirilmiştir. Lazer ölçüm cihazı kullanılmasının temassız ve kararlı bir ölçüm sağlanmasının dışında, ölçülen sinyaller, lazer ölçüm denetleyicisi ile deplasman ve hız olarak çıkış sinyallerine çevrilebilmektedir. Bu şekilde eyleyiciye geri beslenecek olan sinyal kolayca deplasman, hız veya ivmeye dönüştürülmüştür. Kullanılan eyleyicinin çubuk yapısına olumsuz etkileri incelenmiş ve raporlanmıştır. Eyleyiciye beslenecek sinyallerin güç yükselticisinden geçirilerek sisteme geri uygulanması planlanmıştır. Bu sayede, geri beslenecek sinyalin orantısal genliği müdahale edilebilir bir hale getirilmiştir. Ayrıca, çubuğun serbest tarafının uç noktasına bir ivmeölçer yerleştirilmiş ve kullanılan kontrol döngülerinin getirdiği faydaların zaman tabanında belirlenmesi için kullanılmıştır. Bu ivmeölçer sayesinde anlık olarak ölçülmüş sistem tepkisiyle herhangi bir algılayıcının birbirleri arasındaki faz açılarını belirlemekte kullanılmış, frekans tabanında yapılacak çalışmalarında referans algılayıcısı görevini üstlenmiştir.

Deneysel çalışmaların tümü darbe çekici eylemiyle gerçeklemiş, bu çekiçten okunan darbe kuvveti ve algılayıcılardan okunan çıkış sinyalleri hesaplamalarıyla frekans tepki fonksiyonları oluşturulmuştur. Buna ek olarak, sistem niteliklerinin ve tepkilerinin deneysel olarak belirlenmesi için gerçekleştirilmiş temel çalışmalar açıklanmıştır. Çelik çubuğun deneysel modal analizi tamamlanmış ve ilk üç rezonans frekansı ile beraber modal sönüm değerleri ve rezonans genlikleri deneysel olarak belirlenmiştir.

Deneylerde kullanılması amacıyla basit bir kapasite ve direnç devresi olan analog alçak geçiren süzgeç devre tasarlanmış, devreye giren ve çıkan sinyaller arasında oluşan faz açısı deneysel olarak belirlenmiştir. Bu devrenin tasarlanma amacı, yapıya geri beslenecek sinyal genliklerinin yükseltilmeden önce devrede bulunan ayarlanabilir direnç ile istenilen faz açısını sağlayabilmektir. Böylece deneysel sistemden kaynaklı istenmeyen faz açılarının düzenlemesinin yanı sıra geri besleme kontrolcü türünün de ayarlanabilir hale getirilmesi amaçlanmıştır.

Bölüm 4 ise aktif titreşim kontrolü deneysel çalışmaları, zaman tabanında yapılan çalışmalar ve frekans tabanında yapılan çalışmalar olarak iki alt başlığa ayrılmıştır. Böylelikle, ayrı ayrı gerçekleştirilen deplasman, hız ve ivme geri beslemeli kapalı çevrimlerin sonuçları detaylıca incelenebilmiştir. Frekans tabanında yapılan çalışmalarda ek olarak analog faz kaydırıcı devre kullanılmasıyla geri beslenecek sinyalin faz açılarının istenilen değere getirilmesi amaçlanmıştır.

Zaman tabanında yapılan açık döngü çalışmalarında, herhangi bir geri beslemeye maruz bırakılmamış çubuğun darbe çekici uyarısıyla birlikte verdiği tepki ölçülmüş ve geri besleme kapalı döngü deneysel düzeneklerinin aynı uyarılma ile verdiği tepkiler karşılaştırılırmıştır. Çubuğun serbest ucundan açık döngü durumundaki titreşim genlikleri ölçülmüş, ayrı ayrı uygulanan deplasman, hız ve ivme geri beslemeli kapalı döngü sistemlerin titreşim genliklerinin çok daha düşük seviyelere getirilebildiği gözlemlenmiştir.

Bu sonuçlarda, deplasman ve ivme geri beslemeli düzenek durumlarının bu seviyede sönümleme etkisi oluşturması, sistem üzerine uygulanan kontrol sinyallerinin istenilen faz açılarında uygulanamadığını işaret etmektedir.

Frekans tabanında yapılan çalışmalarda ise yapının kontrol sinyallerine maruz kalmadığı açık döngü durumunda frekans tepki fonksiyonları çıkartılmıştır. Böylelikle herhangi bir geri besleme kontrolcüsünün sistem üzerine oluşturduğu değişikliklerinin gözlemlenmesine referans sağlamıştır. Ayrıca sisteme geri uygulanan sinyallerin zaman tabanında anlık olarak ölçülmesi ve birbirleri arasındaki faz ilişkilerinin belirlenmesi amacıyla, frekans tepki fonksiyonlarının ölçümlerinde, ilgili algılayıcıların ve geri beslenen sinyallerin zaman tabanında kaydı tutulmuştur.

İlk olarak gerçekleştirilen deneysel çalışmalarda, deplasman geri beslemeli deney düzeneğinde lazer deplasman ölçümleri ile sisteme geri beslenen eyleyici sinyallerinin arasındaki faz ilişkisi incelenmiştir. Hesaplanan değerin, ölçülen faz açısının gerisinde olması, kapalı döngü sisteminde hem ek bir direngenlik sağladığı hem de ek sönüm uyguladığı sonucunu çıkartmıştır. Geri besleme sinyallerinin arttırılması ile kademeli olarak ölçümler alınmış, frekans tabanında sırasıyla karşılaştırmalar yapılmıştır. Buna göre, sistemin birinci modu %15,1 yükseltilmiş ancak istenilmeyen bir sönüm etkisinin de uygulandığı raporlanmıştır.

İkinci olarak yapılan deneysel çalışmada lazerden ölçülen hız sinyalleri sisteme geri beslenmiş, frekans tabanında yapılan ölçümlerde anlık alınan zaman sinyallerinin faz açısı ilişkisi incelenmiştir. Beklenildiği üzere eyleyici ucunda bulunan kuvvet sensörü ile lazer hız ölçümü sinyalleri arasında tam 90 derece faz farkı belirlenmiştir. Frekans tabanında yapılan çalışmalarda, çubuğun birinci modundaki titreşim genliklerinin %42.1 düşürüldüğü tespit edilmiştir.

Lazer algılayıcıdan ölçülen deplasman sinyalinin sisteme ivme geri besleme sinyali olarak uygulanması için lazer algılayıcı kafasının yeri eski yerinin tam karşısına geçirilmiş böylelikle ölçülen deplasman sinyalinin tersi olan ivme sinyalleri sisteme geri beslenebilmiştir. Yani yapıya uygulanacak kontrol kuvveti çubuğun hareket ettiği doğrultu ile aynı doğrultuda olacaktır. Böylece, sistem üzerinde ek bir kütle etkisi uygulanmış olacaktır.

Sonuçlar beklenildiği üzere ivme geri besleme sinyallerinin uygulandığı kapalı döngü sistemi ile çubuğun birinci doğal frekansını daha düşük frekanslara çektiğini göstermiştir. Ancak, eyleyici sinyali ile algılayıcı sinyalinin aynı faz açısında olması beklenirken, eyleyici sinyali algılayıcı sinyalinin gerisinde olduğu tespit edilmiştir. Zaman tabanında da öngörüldüğü üzere ek olarak beklenmeyen sönüm etkisinin gerçekleştiği görülmüştür. Öte yandan güç yükselticisinden geri beslenen sinyal genliği arttırıldıkça, fazlar istenildiği gibi aynı faz açısı değerine getirilebildiği gözlemlenmiştir. Çalışma sonucunda çubuğun ilk rezonans frekansı uygulanan geri besleme sinyallerinin oluşturduğu kütle etkisiyle %5,5 düşürülmüştür.

Kullanılan ve üzerinde çalışılan deney düzeneklerinde istenilmeyen faz açılarının oluşması, analog alçak geçiren süzgeç devre tasarlanmasının fikrini oluşturmuştur. Lazer deplasman çıktısının hemen sonrasına yerleştirilen devre, hesaplandığı üzere ek bir faz açısı farkı sağlamıştır. Kuvvet algılayıcısı ölçüm sinyalinin, lazer ölçüm sinyalinin gerisinde olması durumunu, 90 derece gerisinde olması durumuna çevirmiş, deplasman ölçümleri ile aktif titreşim sönümleyici düzeneği tasarlanmasına olanak sağlamıştır.

Bu yeni düzenek üzerinde gerçekleştirilen darbe çekici testleri frekans tepki fonksiyonlarının, kademeli olarak arttırılan geri besleme sinyalleri sonuçlarına göre, birinci moddaki titreşim genliklerini %44.37 düşürülmüştür.

Sonuç olarak, bu tezde elektrodinamik bir sarsıcının eyleyici olarak kullanılması ile orantısal geri besleme kontrol düzenekleri kurulmuş ve bir tarafından tutturulmuş öteki tarafından serbest bırakılmış esnek bir kirişin birinci titreşim modu üzerindeki dinamik etkileri incelenmiştir. Ek bir devre kullanılarak ölçülen ve geri beslenen sinyaller arasındaki faz açılarına müdahale edilebilmiş, bu devrenin sağladığı faydalar incelenmiştir. Bu çalışmalarda uygulanan denetçi düzeneklerinin, yapı üzerinde ayrı ayrı ya da birlikte hem direngenlik, hem negatif direngenlik hem de sönüm etkisi oluşturabildiği gözlemlenmiştir.

1. INTRODUCTION

Traditionally, vibrations that occur in mechanical systems are controlled or suppressed via springs, dampers and balance masses which are also called as passive control elements of a vibration isolation system. In many industrial and mechanical applications, although passive control of vibrations and advanced isolation technologies can reduce the vibration levels, there are various applications where passive control systems are unable to provide the desired dynamic characteristics of the system.

In general, the traditional vibration isolation techniques are based on passive control approach. The passive vibration control approach cannot adequately control the vibrations arise in variable speeds, different loading conditions and resonance issues, especially at lower frequencies. From mechanical point of view, the resonance occurs when the frequency of dynamic loading matches the natural frequency of the structure and this can causes severe damage on mechanical components of structures. In essence, lower frequency vibrations and resonances of industrial products coupled with the associated noise levels are probably one of the most important parameters adversely affecting the quality of the product. Excessive vibration and noise levels are also very undesirable from costumers point of view. Furthermore, undesirable vibrations can also cause wasting energy, reduced fatigue life and noise.

Substantial amount of research has been performed about respect to the application of active control technologies to vibration control of mechanical structures during the last few decades. Recently, active vibration control techniques have been increasingly drawn attention since these technologies have become cost efficient due to rapid development of electronic technologies.

Motivation of the studies of researchers are to design and implement active control of vibration technologies to various structures that have flexible components, resonance modes with low level of damping since passive damping treatments or isolation techniques are not conventionally efficient especially at lower frequency vibration suppression of lightly damped structures. Therefore, the design and performance goals of the active controlled closed loop studies are often relevant to the first few vibrational modes in the literature since their effect on structural failure is usually dominant.

The main purpose of active vibration control is to decrease the vibration levels of any mechanical system by direct and automatic modification of the structural response of the system. Active structures are comprised of set of sensors to measure and detect the responses and set of actuators to influence the structural responses of the system which are coupled by controllers to manipulate the acquired signals from the sensors so as to modify the responses of the system in the required manner. Moreover to this, in the literature, structures that have different and unique feature of distributed and high level of integration capability actuators and sensors are frequently called as smart structures. The main component of an active system is the actuator which behaves as an external excitation to affect the controlled system intelligently. Typical actuators used in active control systems are hydraulic, pneumatic, electrodynamic or smart material actuators as piezoelectric materials. In this thesis, experimental studies are conducted by an electrodynamic shaker as the controller actuator.

Electrodynamic actuators or shakers have the sufficient displacement and force capabilities for many applications especially for modal testing experiments. Their implementation in active controlled structures, however, is commonly not suggested for practical applications because of their electrical demands, limited bandwidths, and relatively larger weights. Therefore, it can be pointed out for the practical applications that the electrodynamic actuators would be extremely heavy and affect the structural characteristics of controlled component which make this type of actuators poorly suited to control the vibration of high amplitudes and wide bandwidths.

1.1 Literature Survey

The literature survey is conducted to investigate and compile the past researches and the development process of active vibration control technology. In the literature, there are experimental, analytical and numerical studies about active vibration control and their applications. Active vibration attenuation or suppression comprises the use of an active system so as to reduce the structural vibration or its transmission from one or more structures to others. Furthermore, a wider definition would also include the reduction of structural vibration by using an active vibration absorber. In a general manner, passive vibration control or isolation is covered adequately in the literature thus will not be discussed in this thesis.

It is a fact that active vibration control strategies for vibration attenuation are much more complex and expensive than their passive control strategy counterparts which usually consist of metal or viscoelastic springs and dampers that are being used in practice for many years.

In what follows, the literature survey first describes the advances and trends for the use of piezoelectric materials for vibration control is described first. It is seen that significant amount of published work on this subject can be found in the literature. Then a review of the work done on optimal location of controller actuator, mostly piezoelectric elements and the vibration attenuation of smart beam and plate like structures are described. It is noticed that most of the research on active vibration control is utilising piezoelectric actuators and sensors. Therefore, more attention is devoted to this topic in the literature survey.

1.1.1 Piezoelectricity and piezoelectric materials

The first scientific publication identifying the phenomenon, later termed as piezoelectricity, appeared in 1880. It was co-authored by Pierre and Jacques Currie who were conducting a variety of experiments on a range of crystal at the time [1]. Then, Woldemar Voigt, who did forerunning studies in 1884, expressed the relationship between the material structure and the piezoelectric effect [2]. After the piezoelectric effect had been discovered, Lippman who is awarded Nobel Prize in Physics found that, on the basis of thermodynamic arguments, imposition of electric charges causes mechanical deformation for piezoelectric materials. The first serious application of this effect was developed by Langvin, during the World War I, for generation of sound waves in water [3].

The discovery of piezoelectricity generated a considerable amount of interest and has encouraged many researchers to work on this field. Piezoelectric materials can be used as sensors that measure the physical quantities such as strain and they can also be used as actuators since they respond as mechanical strain when an electrical voltage is applied to such materials. At the present time, piezoelectric materials, by using their sensing and actuating behaviour, have been widely accommodated in different areas of technology and industry as well as in nano to macro scale systems [4].

Development of piezoelectric materials during and after World War II helped modernize the piezoelectricity studies, hence significant research was performed in United States, Japan and the former Soviet Union for determining the materials with very high dielectric constants for construction of capacitors. By the time, in 1969, a powerful piezoelectric response was discovered in polyvinylenedifluoride (PVDF). The piezoelectric coefficient of the poled thin films of PVDF was 10 times larger than the other related polymers.

As far as active vibration control systems are concerned, the piezoelectric materials are manufactured as thin plates to be bonded on or embedded into structures to obtain intelligent or smart structures. They can also be used in discrete or distributed locations to determine the performance of the system [5, 6].

According to the assumed operating conditions of the active structure, the selection of the sensors and actuators differ significantly in practice. The smart materials that can be considered in smart technologies can be classified as piezoelectric materials, electrostrictive materials, magnetostrictive materials, shape memory alloys (SMA), optical fibers, magnetorhelogical and electrorhelogical fluids. An extensive literature survey in relation to various smart materials and their applications, especially the piezoelectric ceramics and their applications in smart aerospace structures, can also be found in PhD thesis of T.Çalışkan [7]. In this study, he focused on the vibration control aspects by using piezoelectric ceramics, Lead-Zirconate-Titanate (PZT) type, as actuators and sensors.

Although PVDF have found diverse uses in industrial applications, for instance in vibration damping experiments [8], their low stiffness and electromechanical coupling coefficients have limited their use when compared to piezoelectric ceramics like PZT type piezoelectric ceramics. PZT type piezoelectric ceramics have excellent properties to specific applications by compensating the appropriate composition of zirconate-titanate ratio. They are widely used as actuators and sensors for broad range of frequencies [9], including accelerometer, force transducers, ultrasonic applications, high accuracy applications as well as nano positioning applications [4].

Piezoelectric transducers are widely used as sensors and actuators in vibration control studies. For this purpose, transducers are bonded to a flexible structure such as beams or plate like structures, and utilized as a sensor to monitor structural vibrations or as actuators to apply stiffness, damping, and mass effect to the structure. The dynamics of a structure with incorporated transducers can be derived from physical principles where these transducers are linear devices whose properties are governed by a set of tensor equations [10].

Considering the literature on piezoelectric materials, it can be said that, piezoelectric ceramic patches for the purpose of active vibration control provide cheap, reliable as well as high integration and good broadband actuation and sensing capabilities. It should be noted that, however, that is important to optimize the locations of patches on the structure to improve their control efficiency. This is the subject addressed in the next section.

1.1.2 Optimal placement of sensors and actuators

Determination of the optimal performance locations for the sensors and actuators in active vibration control of beams and plates is addressed here. Various studies have been done on this subject and a literature search for optimal actuator or optimal sensor placement methods yields a large number of publications from different engineering disciplines.

One of the limitations of the actuators and sensors is the amount of force it can exert, hence it is important to optimize the location and sizes of transducers so that the required control effort is minimal. In behalf of active vibration control by using sensors and actuators studies; to improve the control performance of the related system or structure frequently focuses on the actuator and sensor optimal location for using controller outputs efficiently. In order to use the actuating and sensing capabilities effectively, optimal location for beam and plate structure, various studies are conducted and reported by many researches in the literature [11]. Hence, the positions of sensors and actuators play an important role; a misplaced sensor/actuator couple may cause lack of observability, controllability, and spill over.

Some of the references describe small optimization problems and employ manual “cut and try” optimization techniques or spontaneous placement recipes rather than systematic optimization methods. Other references discuss challenging numerical optimization problems and most often use genetic algorithms as the optimization method. It is reported in the literature that misplaced sensors and actuators lead to problems such as the lack of observability, controllability and the instability effects. [12, 13, 14, and 15]

V.Gupta et al. [11] ,in their review paper, created a table of optimal locations of surface bonded piezoelectric sensor and actuator patches on a smart beam and plate structures. They presented the result of the survey in a tabular form to demonstrate the sensor and actuator locations. In their literature survey, they took account of the boundary conditions and the modes of the system to be controlled. In addition, authors also pointed out that optimal placement of actuators have greater significance than that of sensors by comparing the optimization criterions.

D.Halim et al. [13] established a methodology by using actuator and sensor pairs on a thin flexible plate to determine the optimal locations and to acquire these results by applying maximum modal controllability approach. This approach claims that the optimal location for piezoelectric actuator and sensor pair is in the middle of the thin rectangular plate. At the end of this study it is stated that optimization method should be studied for selected/individual modes for complex structures.

R.Barboni et al. [16] considered an analytical approach sequentially to find the best patch location as well as the geometry of piezoelectric material which was bounded on the flexural cantilever beam. In this study, the researchers excited the beam via piezoelectric patches, which were bounded on two sides of the beam, to create bending moment. The conclusion of this article is that the optimal circumstances and locations exist on each mode of the passive structure for maximizing the effect of piezoelectric patches.

I.Bruant [17] developed a new methodology to optimize piezoelectric actuator location and those of sensor locations by minimizing an expression about the mechanical energy integral of the system. The primary objective of the work was to develop a methodology for beam like structures. It was also pointed out in the paper that it can directly be usable for more complex structures. Consequently, it is reported that adding one actuator and one sensor to the system gives more efficient active control performance for the simple beam for the case of sinusoidal loads.

Padula et.al. [18] reported a survey which consists of the publication years up to 1999, including aerospace and non-aerospace applications by problems and their solution methods for sensor placement. The proposed studies are referenced to the experienced problems and their solutions in NASA Langley Research Centre.

I.Frecker et.al. [19] classified the results of the literature survey and presented the outcome in tabular form in terms of design variables, constraints, solution methods, actuator types, and target applications by referring to the paper by Padulla as well as the latest researches of piezoelectric sensors/actuators placement. They stated out that most of these optimization problems have been addressed by the authors referenced in those papers via using various approaches and solution techniques.

F.Peng et.al. [20] developed the placement optimization methodology for piezoelectric patches which was very effective in determination processes for the optimal actuator locations to minimize the energy requirement of control strategies. They finalized their study by demonstrating the computer simulations performed on a thin rectangular plate with four patches at optimized locations. Results show that their method is effective for reducing the vibration control power requirements as well as increasing the control affectability.

Şahin et.al. [21] and T.Çalışkan [7] focused their studies on cantilevered beam and plate like structures respectively. Theoretical and numerical studies were conducted in order to optimize piezoelectric material locations to determine maximum tolerable actuation value for piezoelectric actuators. The influence of the size of piezoelectric patches is also considered. Most of the work reviewed here is mainly targeted on simple beam and plate structures. In essence, number of the optimization solutions for the placement of actuators and sensors are usually close to fixed ends of the beams and plates due to the strain characteristics of the first three modes of forced vibrations of cantilever beam or plate like structures.

As stated in many publications, it is also important to keep the thickness of the piezoelectric patches less than the thickness of the controlled structure. Due to the fact that many piezoelectric patches are thin and light weight compared to the controlled structure properties. It can be reasonably assumed that patches keep the mass and stiffness properties of the structure unaffected.

It can also be reported that, studies on optimal piezoelectric sensor actuator placement of real life complex structures are hardly available in the literature.

1.1.3 Smart systems and structures

Mechanical systems are generally subjected to internal and external disturbances that may cause undesirable mechanical vibrations which in some cases put the structural integrity of the system at risk. A structure is an assembly that serves an engineering function. The term smart structure or system refer to the integration of actuators, sensors and the usage of some kind of control unit or enhanced signal processing [22]. The active smart systems and structures are estimated to provide innovative capabilities in advanced industrial applications; hence this can be done by specific implementations in the sense of system functional enhancements like active vibration control or health monitoring.

According to Chopra et.al. [23], a smart structure has the capability to respond to a varying external environment such as loads, geometry changes and to a changing internal environment such as damage or failure. Therefore, a smart structure involves three basic elements: Actuators, Sensors and Controllers, to analyse the response obtained from the sensors as well as use special control logics to command the actuators to apply localized strains to interested structure to obtain the desired equilibrium of the system.

A prominent study was conducted by Matsuzaki [24] to review the intelligent/smart structure researches in Japan. This review was focused on motion, shape, vibration control and micro precise control of space and non-space structures. Also the smart reinforced composites and shape memory alloys design approaches are offered by using new ideas and future work suggestions.

In earlier times, the words intelligent, adaptive, organic, sensory etc. were also used to characterize the smart systems and materials. For instance, Tzou et.al. [25] defined smart structures as intelligent, adaptive or structronic systems that imply an ability to be clever, sharp, active, fashionable and sophisticated. Also, they further pointed out that materials and structures can never achieve true intelligence without the addition of artificial intelligence. In the same spirit, the paper also focused on a wide range of smart material and appliance domain of smart structures review.

In the literature, many researchers referred to reference the work of C.Fuller et.al. [26] when referring to the categorization of smart structures. They defined a framework of smart systems with additional terms which are meant to classify smart structures further, based on the level of complexity. Moreover, smart structures by using piezoelectric materials alter the dynamic characteristics of the structure to a great extent. On the other hand, researchers generate a relationship between intelligent structures and organisms or biological systems. For example, S.Kamle [27] termed smart structures that can assess their own health, perform self-repair or can make critical adjustments in their behaviour as the structure condition changes. Furthermore, it is indicated that the human is a self-regulating network of cells controlled by our mind since he tried to clarify that the body is a mechanical structure which consists of feedback controllers via using its sensors and actuators.

J.Elliot [28] considered a detailed diagrammatic presentation of mechanical structure of the inner ear in which the spiral structure of the inner ear has been straightened out as mass and stiffness of the individual parts. Although they have an imperfect knowledge about the exact mechanism of human ear, they assumed that the outer hair cells acts as sensors and generate feedback loops. As a result, they tried to demonstrate automatic tuning process of active control of structural vibrations. However the smart structures are much inferior to living beings because of their primitive level of intelligence.

1.1.3.1 Structural health monitoring via piezoelectric materials

Due to their exceptional mechanical and electric coupling properties, piezoelectric materials hold many potential applications in the field of structural health monitoring and repair. In essence, the structure is sensed during in service life and measurements are compared with reference response levels. Thus, the process provides effective solutions to the health monitoring of interested structures on its local parts when compared with the non-destructive inspection methods. Structural health monitoring (SHM) applications can be used to prevent catastrophic failures via smart system integration which consist of both passive and active sensing monitoring.

As stated by Doebling et.al. who reviewed a comprehensive literature survey about damage identification and health monitoring of structural and mechanical systems from changes in their vibration characteristics, active sensing monitoring is used to localize and determine the magnitude of an existing damage [29]. To illustrate, a recent research about the structural health monitoring using piezoelectric materials was presented by G.Park et.al. [30]. By monitoring the current and voltage applied to surface bonded piezoelectric transducers, the impedance of structures measured via applying high frequency structural excitations under consideration of the composite reinforced concrete wall. Thus the method is based on indicating the variations in impedance which in turn can indicate damage has occurred. Moreover, it is reported that earlier than the cracks were physically visible, a PZT sensor detected the damage to the structure.

Also, the extreme sensitivity to presence of damage and the location of damage picked up by relatively wide sensing regions of each PZT sensor. In brief, multiple cracks of the damage in different locations of the structure at different periods of time are acquired accurately.

A wide ranging review of recent applications of piezoelectric materials in structural health monitoring and repair conducted in the literature can be found in the paper which was handled by Duan et.al. [31]. In this research paper, the analysis of piezoelectric sensors and actuators applications in beam, plate and pipe structures were reviewed in detail. Besides, an overview is presented on recent advances. It is also indicated that the basic principles and the current developments of the process can also be found in this study. It is concluded that the piezoelectric materials have the capability and promising features for structural health monitoring with reasonable efficiency.

1.1.3.2 Industrial applications of smart structures

Smart materials and structures have attracted a great awareness due to their potential advantages in a wide range of industrial applications, such as aeronautical and aerospace engineering, civil and mechanical engineering, precision instruments and health monitoring, etc. For this purpose, among the other smart materials, piezoelectric materials received most attention because of their features in the field of structural shape and vibration control as indicated before.

A number of Japanese researches focused on the smart vibration control system of buildings subjected to horizontal and vertical seismic excitation as described by Yoshida et.al. [32]. A new mechanism is presented to compare five different scenarios to obtain numerical and experimental results of the active dynamic vibration absorbers. In addition, the paper draws attention to active control of the external excitations which are in both horizontal and vertical directions. In addition to this paper, F.Ross [33] paid attention on active isolation and damping of space structures in order to examine the best use of active control of smart structure. They described the reduction of the level of internal and external excitations as well as increasing robustness of attitude control. Furthermore, they also concentrated on independent modal control of mentioned space structure due to its relative simplicity, yet it will not be essential to control all of the modes. Thus, as the controller system complexity is increased for a given number of modes, they implied the smart space structure becomes more robust. Last but not least, in order to determine a smart system for a textile bobbin, H.Freidmann et.al. [34] paid particular attention to predict and measure all disturbing forces in number of cases and generated a method for the control of vibration resonances of a bobbin rotor. Limited numbers of actuators are used to control an infinite number of vibration modes. Although the active vibration reduction is not appropriate for one frequency, it is reported that it is necessary to cover vibrations of the first three bending modes. In an economic aspect, although active methods can improve the quality and fatigue life, the installation and the maintenance of a smart system enhanced to a textile bobbin increase the system cost. Nevertheless, there are possibilities for the application of smart systems and structures for active vibration reduction where passive systems did not lead to a minimization of structural vibrations. Active technologies are more effective than passive technologies where the passive methods are of limited use if several vibration modes are excited simultaneously. Researchers frequently discussed economical features that active methods are not applied until all passive methods failed although active approaches are extremely flexible than passive approaches [35].

As a conclusion of this part of the literature survey, in an appliance area aspect, intelligent/smart systems and structures are used in bridges, trusses and buildings, mechanical systems, space and aeronautical appliances, telescopes and so on.

1.1.4 Active vibration control

Engineers have been controlling vibrations for quite a long time by modifying mass, stiffness and damping of the structure. As stated before, a structure in which external source of energy is used to control structural vibrations is called smart structure and the method is called active vibration control. Also, as mentioned, the use of piezoelectric materials in the field of active vibration control has interested an immense deal attention in the last few decades. Within the perfection of high levels of piezoelectric activity, broad dynamic response, high efficiency and fast reaction, low energy consumption, extremely wide frequency range and low impedance, piezoelectric patches are considered to be optimal and attractive for actuator and sensor applications.

The most effective way to reduce unwanted vibration is to stop or modify the source of the vibration. D.J.Inman paid attention for the methods of designing systems so that they suppress vibration, in Engineering Vibration book [36]. As he indicated, it is sometimes possible to design a vibration isolation system to isolate the source from the system of interest or isolate the device from the source of vibration. The choice of the physical parameters m, k and c determines the response of the system thus the passive control can be achieved by using highly damped materials such as rubber to change the stiffness and damping between the source of vibration and the structure that is to be protected from the vibrations. If the constraints on physical parameters are such that the desired response cannot be obtained by changing them, active approach may provide an effective alternative. In the book Vibration Control of Active Structures, A.Preumont [37] summarized the process as; active control uses an external active or adaptable device, called an actuator, to provide a force to the device, structure, or machine whose vibration properties are to be changed. The actuator is used to apply the force, together with the sensor used to measure the response of the structure, and also the electronic circuit required to read the sensor's output and apply the appropriate signal to the actuator is called the control system where the mathematical rule used to apply the force from the sensor is called control law.

The active control of vibration reduction of flexible structures like beam and plates by using smart materials such as piezoelectric transducers attracted a lot of research interest. Since these beams, plates and complex thin structures that are lightweight and under-damped, are more and more used in industrial applications, there is increased need for active vibration control of such structures. One of the simpler ways to accomplish active vibration damping is using piezoelectric materials as sensors and actuators. The sensors are used to perceive the vibration state of the simple beam/plate structure or other intelligent structure while the actuators generate excitation in obedience to the controllers' output after related processing. Briefly, J.Xiaojin et.al. [38] listed the main points in a brief and comprehensive manner that the active vibration control using piezoelectric patches as the signal taken from the piezoelectric sensors to relative controllers and the controller outputs the control signal to piezoelectric actuators and according to the control output, actuators controls the controlled subject. Besides, they also generalized the active control procedure that the vibration of the concerned structure should be controlled or suppressed if the actuators' output is equal to the vibration response of this structure. A modal analysis procedure conducted in behalf of the basic rule suggested by NASA [19], which is to place the piezoelectric transducers in regions of high strain, together with away from zero strain areas and a proportional-integral-derivative (PID) controller proposed to confirm sensor and actuator correlation.

1.1.4.1 Active control of beam structures

Karagülle et.al. [39] studied on the integrations of finite element method products to control vibration suppression of flexible cantilever beams with piezoelectric materials by applying PID controller. A comparison of analytical, finite element method and experimental result are presented. As a result, they indicated that in order to design a suitable control technique, finite element method approach was verified by numerical results and can be successfully associated with vibration measurements.

Song and Sethi [40] demonstrated multimodal vibration suppression of a cantilever beam by using pole placement controller with an observer. The concerned system dynamics was gathered via non-parametric and parametric model approaches. They evidently proved that the effectiveness of these type of controllers are effective in multimodal vibration damping by comparing the power spectrum density plots and frequency response functions of vibrations with and without control.

On the other hand, engineering structures operate frequently under dynamic excitations and these types of excitations may vary. However the outputs are generally in the form of mechanical vibrations. In the study titled, “Active control of residual vibrations of a cantilever smart beam”, Kıral et.al. [41] aimed to control the dynamic response of a cantilever beam subjected to moving load with constant amplitude and uniform velocity. The moving load is applied along the beam by the pressurised air which was obtained via a nozzle on industrial robot manipulator. Piezoelectric actuators are used for acquiring displacement feedbacks yet as a sensor a laser displacement sensor is employed. The air nozzle was moved from the clamped end to the free end of the cantilever beam and the responses during the action are recorded. Consequently, the residual vibrations of the controlled beam were damped effectively by use of proportional control and also the results show that the finite element method results and experimental results are in very good agreement. Similarly, H.Hongsheng et.al. [42] analysed the vibration characteristics of a cantilever beam under a moving mass. They referenced that the vibration control for cantilever beam under a moving mass belongs to a time-varying and non-linear problem. The active vibration control approach is aimed to suppress its vibrations via self-sensing piezoelectric materials, where a piece of piezoelectric element simultaneously acts as both a sensor and an actuator in an adaptive fuzzy control strategy. In the final decision, the paper reported that the experimental results showed that vibrations are suppressed effectively.

Ülker and Nalbantoğlu [43] designed a H_∞ controller to suppress the free vibrations and the forced vibrations of a cantilever smart beam by utilizing piezoelectric patches. In this study, eight piezoelectric patches were bonded to the root of the cantilever beam in bimorph condition for both sensing and actuating applications. In bimorph configuration, when one piezoelectric patch extends in one side of the beam, the other patch shrinks in the exact opposite side of the beam. It is also reported that the bimorph configuration doubles the actuation performance of piezoelectric patches. In case of designing the controller, the required system model was obtained by experimental data obtained from the structure. In addition, finite element analysis program was used in the numerical studies. Finally, the closed loop experimental results of forced vibrations showed that first and second resonance frequency vibration levels were suppressed as anticipated before.

Not only in his book but also in the articles, D.J.Inman [44] prepared the active modal control of smart structures review which is also pointed out the basic idea is that the structural designer often looks at the frequency responses of a system and detects a troublesome mode or group of modes. The paper presented illustrative numerical examples and experimental verifications that modal control is a simple and effective solution to problem associated with control of flexible structures such as thin cantilever beams. Since the modal model is an approximation, the independent control of individual modes is difficult. However, this paper demonstrated that the independent control of modes can be accomplished with a large number of piezoelectric sensors and actuators for managing the control spill over problems.

In his MSc. thesis, F.Kırcalı [45] studied on about the smart beams consisting of a passive aluminum beam with surface bonded Lead-Zirkonate-Titanate (PZT) piezoelectric material patches used as actuator and besides, a laser displacement sensor was used as the sensor. Experimental system identification work was executed in order to obtain the modal resonance frequencies, damping ratios and uncertainty on associated with them. Furthermore, analytical model of the structure under transverse vibration was obtained via assumed modes method. Finally, a point wise H_∞controller, which was considered for suppressing the first two flexural vibration modes of the structure, was designed and experimentally compared by the spatial controllers by additionally applying simulations.

With the extension of explaining the usage of piezoelectric transducers efficiently in active vibration control domain, U.Arıdoğan et. al. [46] studied on the vibration characteristics of a smart beam by using impact hammer and piezoelectric patches as actuators. Besides that in order to investigate the sensing performance of piezoelectric patches, single axis accelerometers and a laser displacement sensor are used as reference sensors. It is reported that the effects of the piezoelectric sensor locations on the frequency response of the system are presented by positioning the sensor to different locations along the smart beam. As a result it is directly shown that, since the accelerometers are heavier than the patches, the natural frequencies of the system with the accelerometer determined to be lower than those when the piezoelectric patches or laser displacement sensors are used.

Xiongzhu et.al. [47] planned a study on active vibration suppression of a flexible beam by using system identification approach experimentally. The passive beam bonded with the piezoelectric patches which were assigned as actuators and a set of strain gauges as sensor. The examination focused on the relationship between the input control voltage applied to the actuators and the influenced strain measured by the sensors. All in all, usage of different input voltages in order to propose a control algorithm is reported. The results revealed that efficient vibration damping can be achieved by using higher input control voltages.

1.1.4.2 Active control of plate and complex structures

As outlined in previous section, experiments on simple and light weight plate like structure are conducted by a number of experimentalist and academics. One of the present studies conducted by S.Carra et.al. [48] is an experimental and theoretical approach of active vibration control of a thin walled rectangular aluminum plate. The plate is bolted to a wall of a rectangular Plexiglas container and the experiments focused on the empty, different levels of fluid as well as the water filled tank which were investigated by five piezoelectric patches as control actuators. They reported that the first three complete modal analyses show that the progressive increment of the fluid level produces a progressive decrease of the natural frequencies but not very significant changes in mode shapes. As a matter of fact, the use of multiple sensors and actuators is important to effectively control the complex structures with several vibration modes where the use of single sensor and actuator can result in inadequate observability and controllability properties for some of the modes. G.Caruso et.al. [49] addressed to the problem of damping flexural vibrations of an elastic cantilever plate and impulsive transversal force acting on a free corners of the plate. Three couples of piezoelectric patches were used as sensors and actuators. Many different H_2 control laws have been designed and compared by simulation, in order to evaluate the performance obtained using different patch location combinations. As a final point, the experimental results showed that both from analysis and simulation, the increase in performance attainable through the use of multiple transducers is conditioned to use of a properly accurate model for the design of the controller for obtaining effective vibration suppression in complex structures.

R.L.Clark et.al. [50] reported a comparison between experimental and theoretical results of the simply supported, elastic, rectangular plate which was excited by multiple piezoelectric patches bonded to the specified locations of on the surface of the structure. It is shown that, the multiple actuators yield the capability of generating an almost unlimited range of simple supported plate response since a new parameter is introduced as relative actuator phasing. Finally, the results verified that modes can be selectively excited depending on the chosen phasing of voltage supplied to each actuator. The analytical model can be accurately used to predict the forcing function of piezoelectric actuator patches by using the correlation results.

Trojanowski and Wiciak [51] developed the implementation design of LabVIEW software and PID controllers for the attenuation of sinusoidal excited forced vibrations of cantilevered aluminum plate by attaching two piezoelectric sensor patches and three piezoelectric actuator patches. One of the piezoelectric actuators was used for primary disturbance with the frequency range from 100 to 3000 Hz where the other two actuators for active vibration control. This research paper only presented the introductory results of the designed data acquisition system and also reported that the developed controller provided satisfactory results.

Generally, in the literature, it is indicated that the surface damping treatments are often effective at for suppressing higher frequency vibrations in beams, plates and shells. However, the efficient damping of lower frequency modes usually requires the addition of active vibration control scheme to enhance the passive treatment. For instance, Chantalakhana and Stanway [52] proposed a numerical and experimental study of active control of vibrations of clamped-clamped plate by using PZT patches. The experimental configuration yields both active and passive damping treatments; thus the bending and torsional modes of the plate were effectively suppressed through active control using one sensor and one actuator piezoelectric patches in the feedback algorithm. Vibrations corresponding to the higher modes were suppressed by constrained passive damping layer. Due to control forces exciting the truncated modes and unmodelled dynamics problems namely spill over problems, the desired poles are obtained via linear quadratic regulator (LQR) design to achieve higher damping levels. This study drawn attention to the presence of the passive layer introduces sufficient damping to avoid major problems when using the minimum amount of active control hardware.

One of the modal analysis based technique was presented by S.Kalaycıoğlu et.al. [53] who developed a new dynamic modelling procedure for vibration excitation and suppression of plate structures with surface bonded PZT actuators. They justified their work both experimentally and numerically by using the time delay procedures and finally showed the efficiency of this technique on active vibration control.

An analytical solution and a finite element method approach is complicated and time intensive since the complex geometry of the structures, as an alternative, an experimental modal analysis can be used to obtain modal parameters such as eigenfrequencies ω_i and the mode shapes ψ_i from the measured data. Moreover, the mode shapes include the required information for positioning and placement of piezoelectric transducers. Also the modal input and output matrices can be calculated for the aim of implementing the modal state-space controllers.

From active control strategy of a complex structure point of view, S.Hurlebaus et.al. [54] presented a successful implementation of active modal control to arbitrary curved panels by using experimentally evaluated mode shapes technique. However, the numerical evaluation of modal parameters of such structures, for instance a car body, is complicated since an analytical solution does not exist and the results generally depend on boundary conditions of truncated structure. In this case, the PVDF type piezoelectric materials are used as actuators and sensor in which they were located on the maximum curvatures of modes. However, it is also stated that actuators are most effective for controlling just one mode shape due to the geometric shapes of such a curved panel. As a conclusion, a significant reduction obtained in vibrations of a complex structure, which also guided to a reduction in acoustic radiation.

Thanks to Y.Yaman and his students [55], there are number of studies conducted in Aerospace Engineering Department of Middle East Technical University on smart structures with particular attention given to the structural modelling characteristics, H_∞ and μ-synthesis controllers for further applications and active suppression of in-vacuo vibrations, as reviewed in this literature survey previously. One of the recent researches conducted by him focused on theoretical and experimental results of a smart structure consisting of a rectangular aluminum plate in cantilever configuration with symmetrically surface bonded PZT patches. First of all, the paper reported the influences of actuator sizes and placements on the plate as well as the maximum acceptable actuation voltages on them. Secondly, the research aimed to design single input single output (SISO) H_∞ controller to attenuate the first two flexural modes of the smart rectangular plate. It is shown that the structural modes within the interested frequency range successfully suppressed via designed controller in the presence of uncertainties and also guarantied the robust performance of the concerned system.

1.2 Procedure of the Study

The general purpose of this thesis is to develop and implement an active vibration control experiment in order to manipulate the dynamic behaviour of a cantilever steel beam.

During this process, an electrodynamic shaker is employed to active control system as controller actuator. The experimental studies are focused on the comparison of four different experimental setups to investigate the results of variances of structural responses of closed loop systems in time-domain and frequency-domain respectively.

The following procedures in developing active vibration control system are stated as follows;

- Preparation of experimental setup to generate active control system.
- The dynamics of the flexible cantilever beam is analysed and measured experimentally in order to examine vibration characteristics of the structure both in time and frequency domain.
- All the open loop and closed loop experiments are conducted via impact hammer excitations.
- A rubber hammer tip is selected to excite lower natural frequencies precisely as well as to measure higher tip displacements of the cantilever beam.
- A single point laser sensor head is employed to increase the performance of feedback control setups by using its contactless signal measurement property.
- An electrodynamic modal shaker is utilized as the controller actuator and located as close as the root of the cantilever beam.
- The frequency range of interest only covers the first bending mode of the cantilever beam.
- Laser vibrometer controller is used to derivate/integrate the signals measured so as to generate a controlled feedback signal to the actuator.
- Proportional feedback gains are tuned by the power amplifier of control shaker and an analogue low pass filter is built to implement the setup.
- Four different experimental setups are designed on four different feedback signals: displacement, velocity, acceleration and phase modified feedback signals.
- An analogue low pass filter as a phase shifter is designed to perform another alternative experimental architecture to adjust the phase angles of the feedback signals manually to desired levels.
- The performance and behaviour of four active control architectures are compared individually.

1.3 Objectives of This Thesis

In this particular study, it is aimed to perform an active vibration control in order to observe the dynamic responses of a cantilever beam by altering the first natural frequency. Thus, this thesis focused on the closed loop responses of a cantilever beam which are subjected to displacement, velocity, acceleration and phase adjusted feedback signals individually.

The objectives of this thesis so as to generate an active vibration control are stated as follows;

- Investigation of dynamic behaviour of the test structure both in open loop and closed loop configurations
- Investigation of different control feedback signals on the structure, namely; displacement, velocity, acceleration feedback signals
- Performance demonstration and comparison with reference system of the controller architectures by using impact hammer excitation
- Design of a low pass filter in order to utilize it as an analog phase shifter circuit to adjust phase angles of the feedback signals
- Extract the modal parameters of closed loop responses so as to investigate the effectiveness of the closed loop
- Assessment and comparison of each experimental configuration whether the results are as expected.

1.4 Outline of the Thesis

After the introduction given to this chapter, Chapter 1 presents a literature review of the advances for active vibration control via smart materials and the trends in the application of the smart structures. Although the study in thesis is not focused on the use of advanced control algorithms and advanced materials, modelling and performance evaluation of piezoelectric materials as well as choosing the best performance controller and optimal positioning is also included. The research studies and industrial applications of active vibration control strategies explained briefly.

In Chapter 2, the background theory of governing equation of motion for active control and PID control is considered. Additionally, a numerical simulation for displacement, velocity and acceleration feedback control on single degree of freedom system is studied and presented. The fundamentals of the theory of experimental and numerical vibration models for modal analysis are outlined.

In Chapter 3, an experimental setup and its members that are employed for active vibration control experimental studies are introduced. The modal analysis of a cantilever beam is performed in order to define the dynamic behaviour of reference system in time and frequency domain. The effect of the electrodynamic controller shaker on the structure is also examined during the process. A low pass filter designed in the form of resistor and capacitor (RC) filter circuit which has the cut-off frequency is equal to one divided by the multiplication of capacitance and resistor value. The design process of analog low pass filter as a phase shifter and its input-output relationship is presented in this chapter.

Chapter 4 explains the experimental studies performed on active vibration control test rig based on a cantilever beam. This chapter first describes the determination of various factors influencing the dynamic behaviour of closed loop setups in time and frequency domains. Then, the responses and effects of displacement, velocity and acceleration feedback control experiments are compared with the reference system and variations in vibration characteristics of the first bending mode of the structure are examined. The investigation of the controller architectures are divided into three sections namely; time-domain analysis, frequency-domain analysis and the use of the analog phase shifter circuit for the phase shift adjustment of the closed loop system. Finally, the benefits of implementing an analog phase shifter circuit on feedback closed loop systems are described in detail. Hence, modification of the dynamics of the first bending mode of the structure by applying controlled feedback signals proportional to displacement, velocity, acceleration or combination of these are also examined.

In Chapter 5, general conclusions are drawn and recommendations for further studies are discussed.

2. BACKGROUND THEORY

This chapter describes the physical principles and the theory of piezoelectricity by representing the governing equations of piezoelectric materials for the aim of analytical modelling whose properties are governed by a set of tensor equations. Moreover, the generalized relationships of piezoelectric coefficients are briefly explained and the piezoelectric sensing and actuation functions are presented. Governing equations of motion of the flexible structures as well as the dynamics of a structure with bonded piezoelectric transducers are studied. Furthermore, modal analysis and the basic control theory are investigated. The concepts developed in this chapter by using feedback control of single degree of freedom systems also constitute an introductory review of fundamentals of active vibration control methodology to more complex systems in experimental studies in further sections.

2.1 Theory of Piezoelectricity

In the literature there are several guiding text books and articles that can be used to reference the piezoelectricity fundamentals and nomenclature concerning to the piezoelectric relations. The symbols and units which are widely accepted as an excellent representation of piezoelectric materials are determined by a standards committee in 1958 which is called the IEEE IRE Standards on Piezoelectric Crystals [56].

As explained in literature survey section, piezoelectric devices or transducers are utilized as sensors or actuators in active vibration control systems. Although the IEEE Standards assume that piezoelectric materials behave linearly, these materials may show considerable nonlinearity under high electric field or high mechanical stress levels. In this section, the linear behaviour of piezoelectric materials is presented, thus it is assumed that the transducers are being operated at low electric field levels as well as under low mechanical stresses which are called linear constitutive equations.

The piezoelectric effect is also very non-linear in nature, thus the usage of piezoelectric materials exhibit a strong hysteresis and creep characteristics [57] that will not be addressed here.

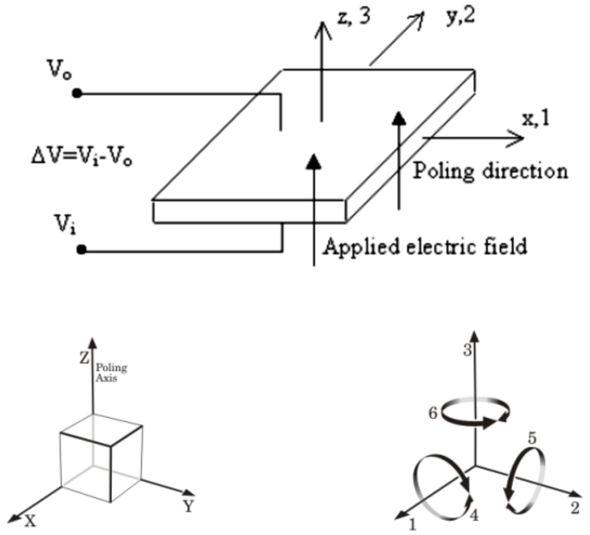


Figure 2.1 : Piezoelectric film actuator diagram and related coordinate systems [7].

The linear constitutive equations [4, 56 and 58] which describe the piezoelectric property are based on the statement that the total strain in sensors and actuators is the sum of mechanical strain induced by mechanical stresses and the controllable actuation strain caused by electrical voltage. Among other things, positive sign convention as well as the actuation voltages to inner and outer electrodes of the piezoelectric material characterized by V_0 and V_i respectively. Electromechanical constitutive equations for a piezoelectric material can be recast in the following form:

$$\varepsilon_i = S_{ij}^E \sigma_j + d_{mi} E_m \quad \textbf{(2.1)}$$

$$D_m = d_{mi} \sigma_i + \xi_{ik}^{\sigma} E_k \quad \textbf{(2.2)}$$

where the indices $i, j = 1,2, \ldots, 6$ and $m, k = 1,2,3$ refer to different directions as indicated in Figure 2.1 within the material coordinate system. Here, the superscripts D, E, ε and σ represent measurements taken at constant electric displacement, constant electric field, strain vector as well as constant stresses relatively. Additionally, S indicates the matrix of compliance coefficients, where d and g relates the matrix of piezoelectric strain constants and matrix of piezoelectric constants. The superscript σ in ξ_{ik}^{σ} point out to constant, zero stresses or open circuited condition for the permittivity matrix. The piezoelectric constant d is defined as the ratio of developed free strain to the applied electric field. The subscript d_{ij} implies that the electric field is applied in the i direction for a displacement force in the j direction. The above equations can be rewritten in the following form, which is frequently used for in applications that involve sensing:

$$\varepsilon_i = S_{ij}^{D} \sigma_j + g_{mi} D_m \qquad \textbf{(2.3)}$$

$$E_m = g_{mi} \sigma_i + \beta_{ik}^{\sigma} D_k \qquad \textbf{(2.4)}$$

The superscript g represents the matrix of piezoelectric constants and β indicates the impermitivity component. The first relationships in equations **(2.1)** and **(2.3)** describe the converse piezoelectric effect, in other words, when the device is being used as actuator. Alternatively, the second relationships in equations **(2.2**) and **(2.4)** dictate the direct piezoelectric effect, so, when the device is used as a sensor.

Piezoelectric ceramics are referred to as transversely isotropic, thus it is generally assumed that z–axis is along the polarization direction which also coincides with the axis of transverse isotropy. Besides that, for these piezoceramics which belong to this class of materials, their matrices can be reduced, therefore, better visualizing the material constants expressed above, the piezoelectric linear constitutive equations can be written in matrix form as [58]:

$$\begin{Bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \varepsilon_3 \\ \varepsilon_4 \\ \varepsilon_5 \\ \varepsilon_6 \end{Bmatrix} = \begin{bmatrix} S_{11} & S_{12} & S_{13} & 0 & 0 & 0 \\ S_{12} & S_{11} & S_{13} & 0 & 0 & 0 \\ S_{13} & S_{13} & S_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & S_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & S_{44} & 0 \\ 0 & 0 & 0 & 0 & 0 & 2(S_{11} - S_{12}) \end{bmatrix} \begin{Bmatrix} \sigma_1 \\ \sigma_2 \\ \sigma_3 \\ \sigma_4 \\ \sigma_5 \\ \sigma_6 \end{Bmatrix} + \begin{bmatrix} 0 & 0 & d_{31} \\ 0 & 0 & d_{31} \\ 0 & 0 & d_{33} \\ 0 & d_{15} & 0 \\ d_{15} & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} \begin{Bmatrix} E_1 \\ E_2 \\ E_3 \end{Bmatrix} \tag{2.5}$$

$$\begin{Bmatrix} D_1 \\ D_2 \\ D_3 \end{Bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 & d_{15} & 0 \\ 0 & 0 & 0 & d_{15} & 0 & 0 \\ d_{31} & d_{31} & d_{33} & 0 & 0 & 0 \end{bmatrix} \begin{Bmatrix} \sigma_1 \\ \sigma_2 \\ \sigma_3 \\ \sigma_4 \\ \sigma_5 \\ \sigma_6 \end{Bmatrix} + \begin{bmatrix} \xi_{11} & 0 & 0 \\ 0 & \xi_{11} & 0 \\ 0 & 0 & \xi_{33} \end{bmatrix} \begin{Bmatrix} E_1 \\ E_2 \\ E_3 \end{Bmatrix} \tag{2.6}$$

The equations **(2.5)** and **(2.6)** above, for transversely isotropic piezoceramics, it can be clearly observed that, there are five elastic constants, three piezoelectric strain constants and two dielectric or permittivity constants. Therefore, representing these definitions using the indicial and matrix notations, it is also stated out that equation **(2.5)** is the actuation matrix and equation **(2.6)** is the sensing matrix.

2.2 Governing Equation of Motion for Smart Structures

In this section, the equation of motion of a multi degree of freedom active structural system will be considered. The given equations will be the backbone of the related control theory and the experimental studies addressed later in the thesis. The equation of motion of a single degree of freedom system can be written in time-domain as [59],

$$f_p = M\ddot{x}(t) + C\dot{x}(t) + Kx(t) \quad \textbf{(2.7)}$$

where M indicates the mass, C is the viscous damping, K is the stiffness and f_p defines the primary force. Also, $\ddot{x}$ symbolizes the acceleration where $\dot{x}$ and x indicates velocity and displacement respectively. The applied external forces lead the controller to generate the electrical force, which is defined by f_c, and the time-domain response of the control force is,

$$f_c = g_a\ddot{x}(t) + g_v\dot{x}(t) + g_d x(t) \quad \textbf{(2.8)}$$

here, by defining control force, it has three components which are proportional to acceleration, velocity and displacement of mass, accompanied by the gain constants g_a, g_v and g_d. The Laplace transforms of these differential equations yield,

$$F_p(s) = Ms^2X(s) + CsX(s) + KX(s) \quad \textbf{(2.9)}$$

$$F_c(s) = g_a s^2X(s) + g_v sX(s) + g_d X(s) \quad \textbf{(2.10)}$$

Furthermore, the related transfer functions of mechanical response and the applied response of the related system can be written as,

$$G(s) = \frac{X(s)}{F_p(s)} = \frac{1}{Ms^2 + Cs + K} \quad \textbf{(2.11)}$$

$$H(s) = \frac{F_c(s)}{X(s)} = g_a s^2 + g_v s + g_d \quad \textbf{(2.12)}$$

Now, these two open loop transfer functions can be used to generate closed loop transfer function with the aim to modify the effective mass, damping and stiffness via applying the effect of feeding back acceleration, velocity and displacement of the related mechanical system.

$$\frac{X(s)}{F_p(s)} = \frac{G(s)}{1 + G(s)H(s)} = \frac{1}{(M + g_a)s^2 + (C + g_v)s + (K + g_d)} \qquad \textbf{(2.13)}$$

By the way, equation **(2.13)** implies the use of three separate transducers to measure the acceleration, velocity and displacement; however, in practice usually a single transducer is available to measure the response either in acceleration, velocity and displacement.

As seen extensively in the literature that the finite element method can effectively be used in modelling of smart structures. The governing differential equation of motion for a multi degree of freedom externally controlled structure subjected to the any measured control force can be represented as [7],

$$[F_{control}]\{u(t)\} = [M]\{\ddot{q}(t)\} + [C]\{\dot{q}(t)\} + [K]\{q(t)\} \qquad \textbf{(2.14)}$$

Here, the global mass, damping and stiffness matrices consecutively $[\mathrm{M}]$, $[\mathrm{C}]$ and [K] defined $\mathrm{N_{dof}}$ x $\mathrm{N_{dof}}$ matrices, where $\mathrm{N_{dof}}$ is the number of degrees of freedom of finite element model and $\{\mathrm{q(t)}\}$ is the generalized structural displacement vector and $\{\mathrm{u(t)}\}$ represents the control output actuation vector of $\mathrm{N_{dof}}\,\mathrm{x}\,1$ matrix. By signifying r as the number of controlled feedback force $[\mathrm{F_{control}}]_{\mathrm{Nxj}}$ is the unit output generalized force transformation matrix from $\mathrm{r^{th}}$ actuator related to each node $(\mathrm{j} = 1,2, \ldots, \mathrm{r})$, thus $\{\mathrm{x(t)}\}_{\mathrm{rx1}}$ and $\{\mathrm{u(t)}\}_{\mathrm{rx1}}$ can also be associated with $\mathrm{r^{th}}$ controller actuator output.

2.3 Control Theory

Although this thesis focuses on the closed loop control systems, it is appropriate to describe briefly both open loop and closed loop systems here. In depth discussion of the control theories which are applied on the flexible structures can be found in the literature [60 and 61].

In essence, there are two distinct approaches for control system design; one is called the frequency domain approach of classical control techniques and the other one is the time-domain approach mainly adopted in modern control techniques. Each method has its own features; the frequency domain approach yields plenty of analytical tools and results, where the modern control theory is easy to implement with the help of abundant computational software available and estimate of all degrees of freedom with a limited number of sensors, a feature which is also attractive for the purpose of this thesis.

D.J.Inman [60], in his book, divided the control methods into three categories as; single input single output (SISO) frequency domain methods, namely classical control, state space methods which allows multi input multi output (MIMO) time-domain control as well as modern control theory which is mainly covers MIMO control in the frequency domain. It is also stated that control systems refers to one or more actuators which are used to control forces to be applied to the structure (also called plant) and the rule or algorithm that determines how the force is applied. It is also worth stating here is that the structure is usually called as the open loop system, while the structure along with the control loop is called the closed loop system.

One of the foremost textbooks on the subject, namely Modern Control Engineering [61], deeply introduced and examined the control theory and its appliance area. K.Ogata, the author of the book, described the open loop and closed loop systems by comparing the advantages and disadvantages in the closed loop feedback control systems. According to the definitions described above, an open loop system can be described as the system for which the output has no effect on the control action. To state the matter differently, in an open loop system the output response is neither measured nor fed back for the comparison with the input.

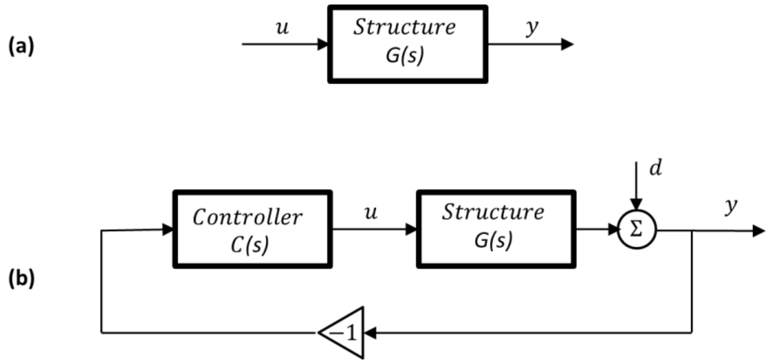


Figure 2.2 : The generic block diagram representation of open and closed loop controllers. (a) Open Loop and (b) Closed Loop.

A closed loop system, on the other hand, is a system that is often referred as feedback control system. In this type of control systems, the actuating error signal, which is the difference between the command signal and the output signal, is fed to controller with the aim of reducing the error as well as bringing the system output to the desired level. The transfer functions that can be used to describe the performance of the related closed loop system as shown in the figure that are the transfer function relating between the reference signal to the output,

$$G_{yu} = \frac{Output}{Input} = \frac{y(s)}{u(s)} = \frac{G(s)C(s)}{1 + G(s)C(s)} \qquad \textbf{(2.15)}$$

As expected, the closed loop systems have advantages as they are capable of dealing with unpredictable disturbances and unpredictable variations in system components. In the long turn, the stability of an open loop controller system is not a major problem, whereas stability is the major problem in closed loop systems.

It can be concluded that, from active control point of view, the main advantage of closed loop control is due to the fact that the use of the feedback composes the system response relatively insensitive to the external disturbances at any possible internal variations in the system parameters [62].

2.3.1 Proportional integral derivative compensators

Proportional derivative integral (PID) control is a control strategy that has been successfully used over many years due to its simplicity, robustness, a wide range of applicability and a near optimal performance are some reasons that have made PID control so accepted in industrial and academic sectors.

In the literature survey section, it is noted there are many applications of PID controllers which applied on active vibration control systems. The controllers related to literature survey indirectly related with proportional (P), proportional integral (PI), proportional derivative (PD) and proportional integral derivative (PID) compensators with the influences of gain margins and variations on performances on feedback control systems. Mainly, such controller designs are considered as the modification of open loop response of the smart structure to obtain the desired response [63].

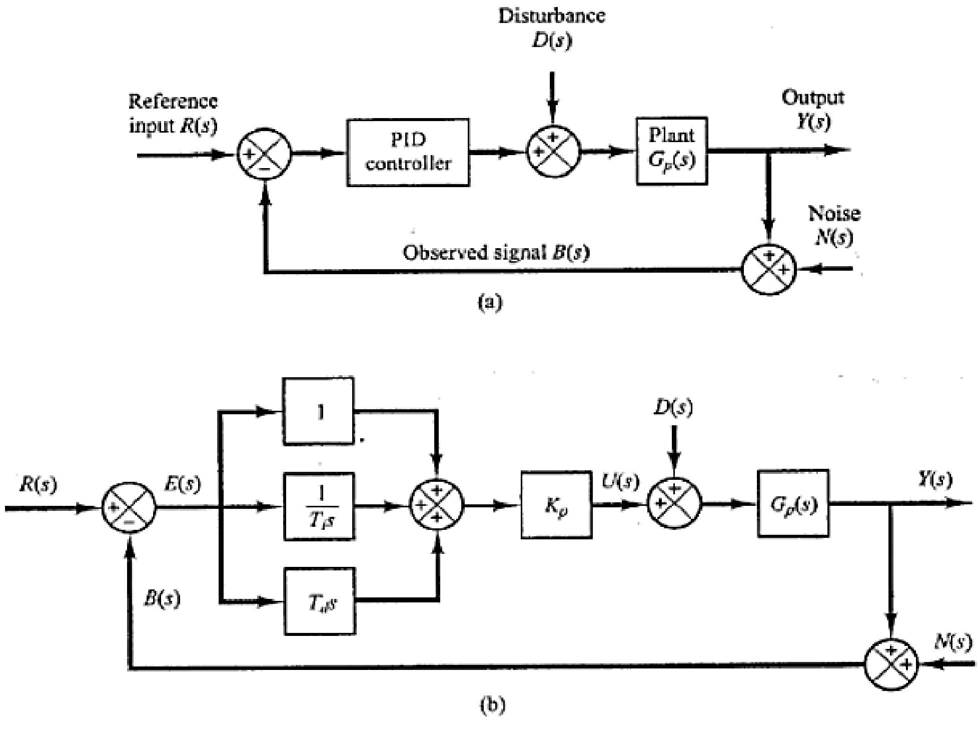


Figure 2.3 : The block diagram representation of PID controller.

Figure 2.3 shows schematically that a PID controller in a closed loop where the variable E(s) represents the tracking error, the difference between the desired value R(s) and the actual outputY(s). The signal U(s) just pass the controller is now equal to the proportional gain K_p times the magnitude of the error plus the integral gain K_i times the integral of the error additionally the derivative gain K_d times the derivative of the error. Thus, the signal U(s) will be sent to the plant $G_p(s)$ so the new output Y(s) will be sent back to the sensor again to find the new error signal. It can be observed that the transfer function of the PID controller has three components,

$$K_{PID}(s) = K_p + \frac{K_i}{s} + K_d s \quad \textbf{(2.16)}$$

The proportional controller will have the effect of reducing the rise time; however never eliminate the steady state error. An integral control will have the effect of attenuating the steady state error yet it may take the transient responses worse. A derivative control will have the effect of increasing stability reducing the overshoot, improving the transient responses as well. Consider the signal u will be sent to the plant to obtain new output y. In this case the signal u is obtained as;

$$u = K_p e + K_i \int e\, dt + K_d \frac{de}{dt} \quad \textbf{(2.17)}$$

One must pay attention to coherence between $K_p, K_{i,}\, K_d$ may not be accurate since they are independent of each other. For this reason, some of the most used PID tuning methods are discussed and the most promising tuning techniques are recommended in [64]. It should be stated that it is not essential to implement all three controllers (proportional, derivative, and integral) into a single system. L.Malgaca [65] stated that in some cases the PI (Proportional, Integral) controller may provide acceptable response by comparing the effects of other controllers with the aim of keeping active controller as simple as possible.

2.4 Fundamentals of Active Vibration Control

As stated, there are two fundamentally different control approaches which have been used in the past for implementing active vibration suppression systems; feedforward and feedback control strategies.

Briefly, feedforward control includes feeding a signal related to the disturbance input into the controller which then generates a control output to derive a control exciter in such a way in order to attenuate the input excitation. In contrast, feedback control uses a control signal output derived from the system response to a disturbance which is mainly amplified, passed through a compensator circuit and used to derive the control exciter output to diminish the residual effects taking place after the initial disturbance has passed. An inherent disadvantage of feedback control systems is their tendency to go unstable if the feedback gain is set high enough, however a high feedback output signal is reduced in amplitude, hence limiting the potential performance of the response controlled system. Also, the feedforward systems do not manipulate the dynamic response of the structure being controlled. Nevertheless, a feedback system is usually the only feasible type and care must be taken to limit the feedback gain so as to stabilize the system or related structure over a whole range of possible inputs and variations in the system dynamics being controlled.

In the reminder of this section, first of all simple feedback active control isolators are discussed, beginning with a single degree of freedom system. The vibration response of a single degree of freedom system including a mass supported on a spring and a dashpot linked to a rigid foundation is described. This system is excited by simple harmonic force acting on a mass and a model as a second order differential equation is obtained.

2.4.1 Feedback control of single degree of freedom system

In this section, based on the assumption that a passive isolation system may not be sufficient to produce desired response of the system, especially at lower frequencies and at frequencies corresponding to resonance frequencies, feedback control of a single degree of freedom system is discussed. It is also clear in the literature that even with an active controlled system; it may not be possible to derive the system response to zero at each sensor location [7]. The dynamics of the system studied here can be modified by adding a control force proportional to displacement or velocity or acceleration or the combination of those to the vibrating mass. This is called as feedback control of the single degree of freedom system and consequently the effect on the response of applying various types of feedback force on the response of the mass the mass is examined.

In order to implement the control force to the equation of motion, the acceleration, velocity and displacement feedback applied through gains K_a, K_v and K_d to obtain control force $f_c(t)$. Then, it may be written as,

$$f_c = -[K_a\ddot{x}(t) + K_v\dot{x}(t) + K_d x(t)] \tag{2.18}$$

A block diagram illustrating the physical system modelled as single degree of freedom and the feedback control method arrangement is shown in Figure 2.4 (a,b);

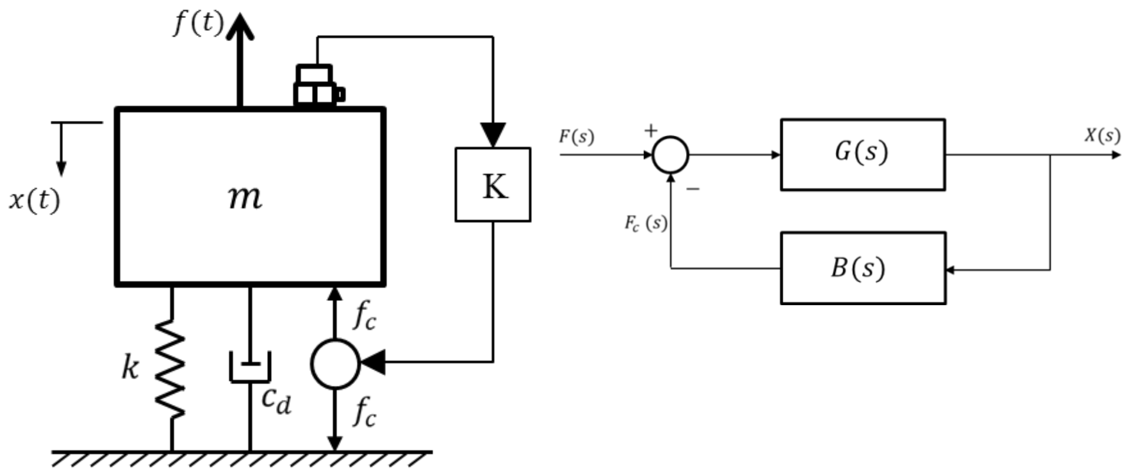


Figure 2.4 : Feedback Control of a SDOF isolation system. (a) Physical Model (b) Block Diagram.

In practice, typical feedback control methods use some mixture of acceleration, velocity and displacement of the output as a feedback signal. The dynamics of the mass can be represented by $G(s)$ and the feedback control force in the s domain is established by taking the Laplace Transform of the equation **(2.18)** with applying zero initial conditions. Thus,

$$F_c(s) = (K_a s^2 + K_v s + K_d)X(s) = B(s)X(s) \quad \textbf{(2.19)}$$

The frequency response of the modelled system is given by,

$$H(s) = \frac{X(s)}{F(s)} = \frac{G(s)}{1 + G(s)B(s)} = \frac{1}{(m + K_a)s^2 + (c + K_v)s + k + K_d} \quad \textbf{(2.20)}$$

The time-domain equivalent of the equation **(2.20)**,

$$f(t) = (m + K_a)\ddot{x}(t) + (c + K_v)\dot{x}(t) + (k + K_d)\, x(t) \quad \textbf{(2.21)}$$

In further sections, the control of vibration of the mass via using assigned vibration observer tuned by acceleration feedback, velocity feedback and displacement feedback will be discussed. In real physical system, as will be discussed in the experimental study sections, there occurs a finite time delay between acquiring the signal from vibration sensor and feeding it back to the structure via the controller actuator after the necessary processing. The finite time delay affects the system stability it is shown that for the reason that of this phenomenon, inherently velocity feedback control methods are usually the most stable [66].

2.4.1.1 Displacement feedback

In order to examine the new natural frequency and damping ratio of the displacement feedback controlled system, the acceleration and the velocity feedbacks are not considered in this section. Applying a displacement feedback is actually means adding stiffness proportional to displacement signal to the related system, thus,

$$\omega' = \sqrt{\frac{k + K_d}{m}} = \omega_o \sqrt{1 + K_d/k} \quad \textbf{(2.22)}$$

$$\zeta' = \frac{c_d}{2\sqrt{(k + K_d)m}} \quad \textbf{(2.23)}$$

By substituting $j\omega$ to s used in the equation **(2.20)**, the frequency response of the system with displacement feedback non-dimensional form can be written as,

$$kH(j\omega) = \frac{1}{1 + K_d/k - (\omega/\omega_o)^2 + 2j\zeta(\omega/\omega_o)} \quad \textbf{(2.24)}$$

In this way, in favour of displaying the effect of varying displacement feedback gain for various values of $K_d\ /k$ with applying $\zeta = 0.05$, the normalized modulus of the closed loop frequency response function $|kH(j\omega)|$ is plotted in the Figure 2.5.

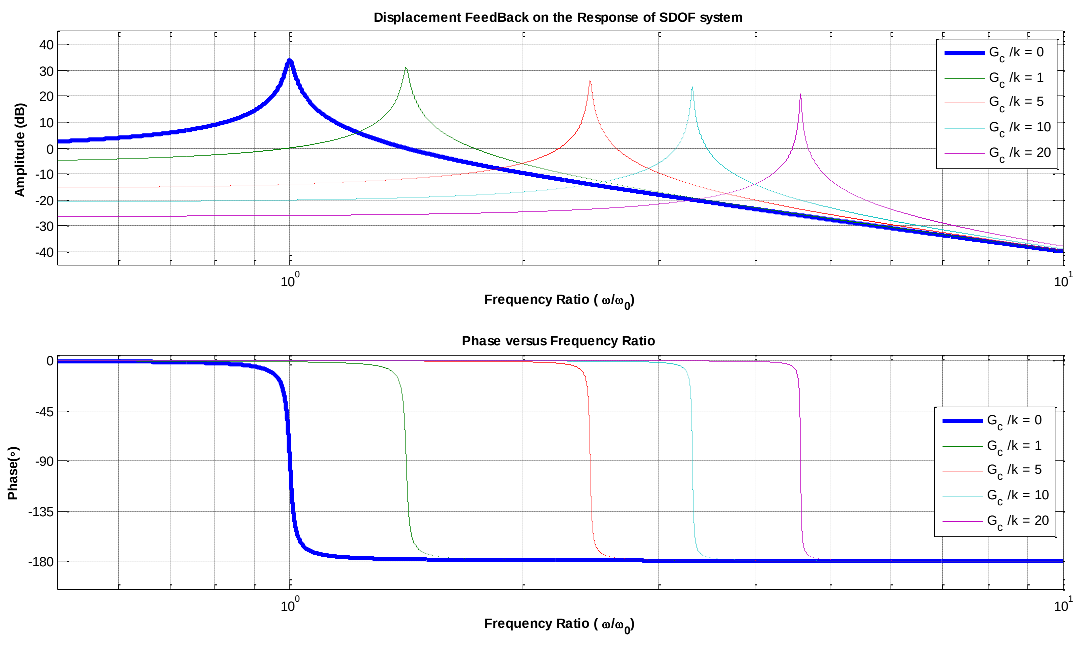


Figure 2.5 : The effect of displacement feedback on the response of a SDOF system.

It can easily be seen from the frequency response function plot that increasing displacement feedback gains increases the low frequency isolation while reducing the high frequency isolation. Moreover, it is observed that the system behaviour depends on the applied feedback signals to the single degree of freedom structure.

2.4.1.2 Velocity feedback

In this case, so as to observe the new natural frequency and damping ratio of the velocity feedback controlled system, the acceleration and displacement feedback are not considered in this section. Applying a velocity feedback signal basically means adding a controlled damping to related system proportional to velocity feedback signals, therefore,

$$\omega' = \sqrt{\frac{k}{m}} = \omega_o \qquad \textbf{(2.25)}$$

$$\zeta' = \frac{c_d + K_v}{2\sqrt{km}} = \zeta(1 + K_v/c_d) \qquad \textbf{(2.26)}$$

Substituting $j\omega$ to s used in the equation **(2.20)**, the frequency response of the system in non-dimensional receptance can be written as,

$$kH(j\omega) = \frac{1}{1 + (\omega/\omega_o)^2 + 2j\zeta(1 + K_v/c_d)(\omega/\omega_o)} \qquad \textbf{(2.27)}$$

In order to display the effect of varying velocity feedback gains for various values of K_v/c_d with $\zeta = 0.05$, the Figure 2.6 is plotted for the normalized modulus of the closed loop frequency response function $|kH(j\omega)|$.

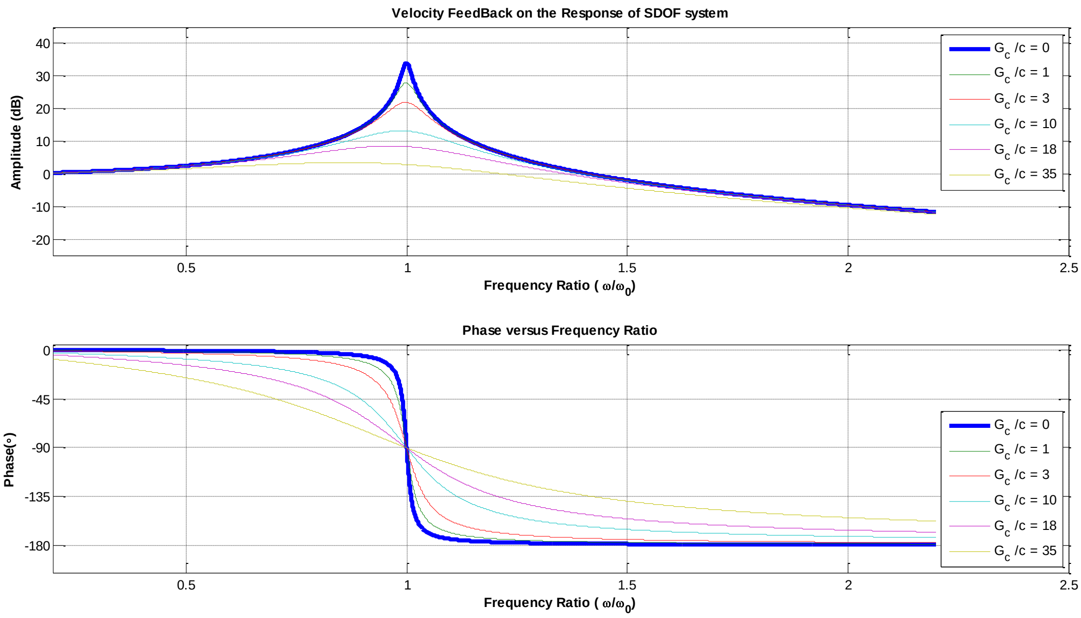


Figure 2.6 : The effect of velocity feedback on the response of a SDOF system.

It can easily be seen from the frequency response function plot that increasing velocity feedback gain increases the system damping. Any applied velocity feedback to a single degree of freedom structure provides a controllable damping to the structure proportional to the related gains. Furthermore, velocity feedback increases the effectiveness of the active vibration control in the region of system resonance with minimal effect at low and high frequencies.

2.4.1.3 Acceleration feedback

In this part, the output signal of the acceleration is re-applied to the system with no derivation, thus no phase shift between the signals is expected. As in the previous sections, the feedback gain increased gradually. In a physical manner applying the same output to the structure means adding additional mass to the location control excitation force. It should be noted that if the frequency kept constant, applying output acceleration signal to single degree of freedom system is equivalent to adding negative displacement to the structure.

$$\omega' = \sqrt{\frac{k}{m \ + K_a}} \quad \textbf{(2.28)}$$

$$\zeta' = \frac{c_d}{2\sqrt{k(m + \ K_a)}} \quad \textbf{(2.29)}$$

Again substituting $j\omega$ to s used in the equation **(2.20)**, the non-dimensional frequency response of the system in receptance form can be written as,

$$kH(j\omega) = \frac{1}{1 - (\omega/\omega_o)^2(1 + \ K_a/m) + 2j\zeta(\omega/\omega_o)} \quad \textbf{(2.30)}$$

The normalized modulus of the closed loop frequency response function $|kH(j\omega)|$ is plotted in the Figure 2.7 showing the effect of varying acceleration feedback gain for various values of $K_a \ /m$ when $\zeta = 0.05$

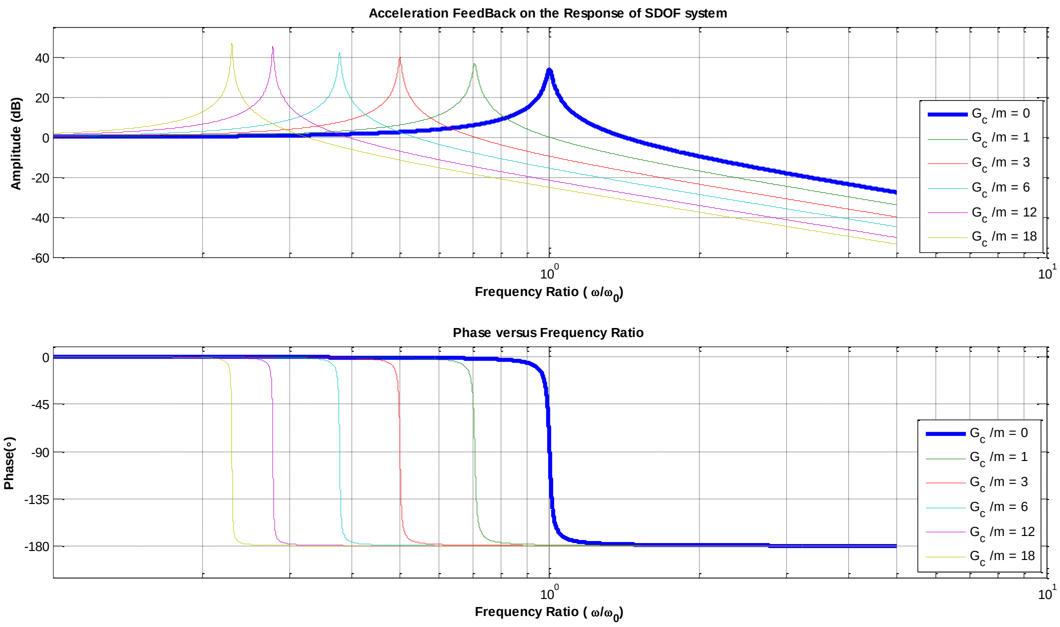


Figure 2.7 : The effect of acceleration feedback on the response of a SDOF system.

In Figure 2.7, it is estimated that externally adding acceleration feedback gain to the single degree of freedom system essentially changes the system parameters such as natural frequency and damping of the structure. In other words applying acceleration feedback decreases the system natural frequency, a situation which is also expected by adding additional mass or negative stiffness to the control location of the system.

As a summary of the results presented above, it is seen that applying displacement feedback provides additional stiffness to the system. However, applying velocity feedback to the system causes additional damping to system. Applying acceleration feedback to the system, on the other hand, alters the system response as if additional mass is attached to the location of the excitation. It is obvious that all types of feedback control applied to a single degree of freedom system can either change the natural frequency or damping, and these in turn affect the frequency response of the system. Consequently, these types of feedback control approaches can be used as valuable tools for controlling vibration levels under operating conditions, including avoidance of resonance situations.

2.5 Modal Analysis

At the same time as the development of modern computer technology, experimental and numerical modal analyses have become the foremost solution for solving complicated structural vibration problems. Modal analysis aims to determine the modal properties of the system including natural frequencies, mode shapes and damping levels. For an existing experimental structure, the modal analysis techniques provide vital information about the dynamic properties of structures. Estimating the flexible mode shapes is critical for designing an active control system in frequency domain since any implementation of an active control law to a structure stands for the derivation of dynamic properties of the structure.

Before embarking on both numerical and experimental theory of modal testing procedures, it is better to briefly introduce modal testing. Integration of the theoretical basis and the accurate measurement of vibrations as well as the realistic and detailed data analysis are the major requirements of the subject of modal testing.

D.J.Ewins [67], who is the author of one of the prominent book on the subject, described the physical characteristics of structures, in terms of its mass, stiffness and damping properties which are often called in modal testing as spatial model. Additionally, the dynamic behaviour of the structures can be described using the so-called modal model which comprises a set of vibration modes and natural frequencies with corresponding modal damping factors. The so-called response model is another way of describing the dynamic behaviour of structures and this type of models consist of a set of frequency response functions (FRFs). Interested readers may directly refer to [67] for details of the theoretical and experimental modal analysis.

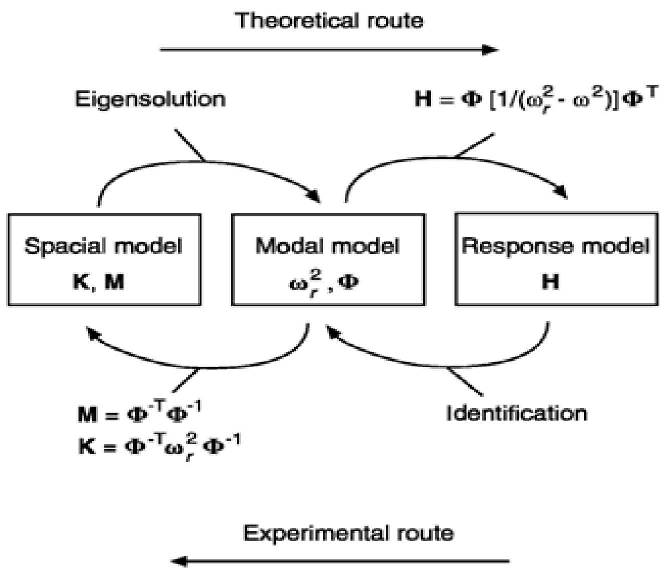


Figure 2.8 : Theoretical route for modal analysis [67].

One may define modes which represent each component of overall dynamic responses as well as they are essential in describing the nature of vibration characteristics, motion and provide physical understanding of the dynamic behaviour of the structure or the system. Vibration modes are obtained by solving the eigenvalue problem derived from the mathematical model. Further subsections include solution techniques for eigenvalue/eigenvector problems, which are commonly called as the theoretical modal analysis.

2.5.1 Theoretical model

As indicated before, the spatial model comprises m, c and k to generate the modal model. For a given n degree of freedom system, the governing differential equation of motion is described via the second order matrix equation. The governing equation of multi-degree of freedom system is given by,

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = 0 \tag{2.31}$$

In order to find the free vibration of the system response without any forcing function, the form of response can be assumed by translating the equation **(2.31)** into an eigenvalue problem. The eigenvalues (λ) and the eigenvectors (ψ) can easily be calculated as,

$$([K] + i\omega[C] - \omega^2[M])\{X\}e^{i\omega t} = 0 \tag{2.32}$$

Here, the solution of the equation **(2.32)** returns eigenvalues as the squares of natural frequencies. Hence substituting any of the natural frequency back into the eigenvalue equation yields a corresponding set of relative values for $\{X\}$, the so-called mode shape $\{\psi\}_r$ equivalent of related natural frequency. Therefore, eigenvectors presents the mode shapes of the matching natural frequency of related structure, and then the solution of equation **(2.32)** returns the vibration characteristics of the structure. Eigenvalues and eigenvectors obtained from the solution can then be used to obtain FRF as,

$$\alpha_{ij}(\omega) = \frac{X_i(\omega)}{F_j(\omega)} = \sum_{r=1}^{N} \frac{(\phi_{ir})(\phi_{jr})}{{\omega_r}^2 - \omega^2 + i\eta_r{\omega_r}^2} \tag{2.33}$$

where, N indicates the number of modes, ω_r indicates the square root of eigenvalue (λ), η is the structural damping term and ϕ indicates the mass normalized mode shapes.

2.5.2 Experimental model

Basically, understanding the natural frequencies and modes shapes helps us to design structural systems for noise and vibration applications. This knowledge is essential for designing control systems for dynamic structures. The modes are further characterized as either rigid body or flexible body modes. All structures have six rigid body modes in free-free conditions in which three of which are translational and three of which are rotational modes.

In modal testing, frequency response function measurements are usually performed under controlled conditions; where the test structure is artificially excited by using impact hammer or using one or a more shakers driven by controllable broadband signals. Experimental modal analysis methods using frequency response function techniques are the most commonly used approach for the estimation of the modal parameters.

Depending on the number of excitation and measurement coordinates in FRF, the process of acquiring the FRFs are named as; single input single output (SISO) or single input multi output (SIMO) and multi input multi output (MIMO) measurements. These types of measurements provide the elements of the FRF matrix. In SISO measurement case, since the excitation and the measurement points are fixed, in every measurement it returns only one element of the following FRF matrix as inertance;

$$[H(\omega)] = [\alpha_{ij}(\omega)] \tag{2.34}$$

In SIMO measurement case, the excitation coordinate is fixed and the measurements are made at more than one coordinate. Such measurements, it returns one column of the FRF matrix since the individual rows of the FRF matrix are related to individual .excitation coordinates

$$[H(\omega)] = \begin{bmatrix} \alpha_{11}(\omega) \\ \alpha_{21}(\omega) \\ \alpha_{31}(\omega) \\ \vdots \\ \alpha_{i1}(\omega) \end{bmatrix} \quad \textbf{(2.35)}$$

Lastly, as its name implies, the MIMO measurement has multiple excitation coordinates and multiple measurement coordinates, hence it provides FRF matrix corresponding to excitation and measurement coordinate as;

$$[H(\omega)] = \begin{bmatrix} \alpha_{11}(\omega) & \alpha_{12}(\omega) & \alpha_{13}(\omega) & \dots & \alpha_{1j}(\omega) \\ \alpha_{21}(\omega) & \alpha_{22}(\omega) & \alpha_{23}(\omega) & \dots & \alpha_{2j}(\omega) \\ \alpha_{31}(\omega) & \alpha_{32}(\omega) & \alpha_{33}(\omega) & \dots & \alpha_{3j}(\omega) \\ \vdots & \vdots & \vdots & \dots & \vdots \\ \alpha_{i1}(\omega) & \alpha_{i2}(\omega) & \alpha_{i3}(\omega) & \dots & \alpha_{ij}(\omega) \end{bmatrix} \quad \textbf{(2.36)}$$

It can easily be observed from the equations **(2.34)**, **(2.35)** and **(2.36)** that multi-input multi-output (MIMO) measurements return more information than the other measurement techniques. Furthermore, the use of the FRFs obtained via MIMO measurements leads to more reliable parameters modal parameters.

Mode shapes can be normalised using one of the few normalisation methods. The so-called mass normalisation approach is the most widely used one in experimental modal analysis. The mass normalised mode shapes can be obtained as;

$$\{\phi_r\} = \frac{1}{\sqrt{m_r}} \{\psi_r\} \quad \textbf{(2.37)}$$

where, ϕ indicates the mass normalized mode shapes, m_r is the mass for the r^{th} mode. The eigenvector which are the results of experimental modal analysis, are become more convenient after the process of mass normalization. Once the eigenvectors are normalized, the following coordinate transformation can be proposed;

$$\{x\} = [p]\{\phi_r\} \tag{2.38}$$

At that point, p symbolizes the principle coordinates as the vector of modal coordinates. One can transform the equation **(2.15)** into the modal coordinates, then the equations of motion are decouples as into [59];

$$\ddot{p}_i + 2\zeta_i\omega_i\dot{p}_i + \omega_i{}^2 p_i = f_i \tag{2.39}$$

where $i = 1,2,3,\ldots n$ thus the input function stands for i^{th} modal coordinate, f_i is the modal force that represents how much the mode is excited from the external input and ζ_i represents the modal damping ratios. The equation above corresponds to the modal coordinate form of the equations of motion, for which independent vibrational modes are described by a second order differential equation. The modal coordinate equations are useful since they can also provide the analytical solution for each mode.

3. EXPERIMENTAL ANALYSIS

In this chapter, the experimental setup which is used for the baseline experimental studies of active vibration control is presented. The data acquisition system and various elements of the experimental active control systems are introduced. The modal analysis of the structure is performed to obtain a baseline time-domain signal and frequency-domain modal parameters for comparison purposes. The dynamic effect of control actuator on the structure is also investigated and benefits and drawbacks are discussed. After introducing the control actuator and sensor pairs, a low pass filter as a phase shifter is utilised in the experiment. It is role as well as the effectiveness of this RC circuit is investigated.

3.1 Experimental Setup

For the rest of the thesis, the measurements and the feedback control signals will be applied on a cantilever beam. The structure is a steel beam with the dimensions of 420mm x 40mm x 3mm and clamped from one side with another structure with high inertia. The clamping structure in the experiment was quite solid due to its relatively higher mass compared to the mass of the beam. This was important because if the clamped end of the beam was not held rigidly, the baseline and further control application studies would have been invalidated. The response of the cantilever beam will be described in terms of displacement, velocity and acceleration in order to explain the feedback signals and the procedure of different setups for active control closed loops. Although accelerometers are one of the most common forms of the measurements of relatively large structures, considerable mass and local stiffness effects of accelerometers may have negative effects on the response of the structures especially for light weight structures such as cantilever beam in our experimental setup. Therefore, as a non-contacting transducer, laser based measurements system is considered for the measurement of the responses of the beam in order to supply the control feedbacks to closed loop system by a negligible efficiency loss [7].

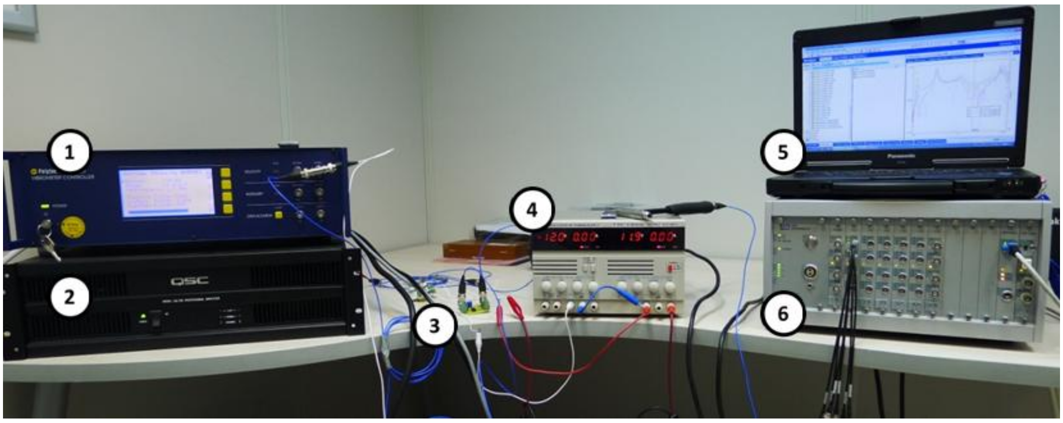


Figure 3.1 : The measurement system 1) laser vibrometer controller 2) power amplifier 3) analog low pass filter 4) power supply 5) computer 6) analyzer.

Measurements of the tip displacement of the beam were made via the use of linear accelerometer. The triaxial ICP type accelerometer was rigidly attached to tip of the beam via thin layer of adhesive wax [68]. As this transducer is always positioned at the tip of the beam, its mass effect is considered as an integral part of the system itself.

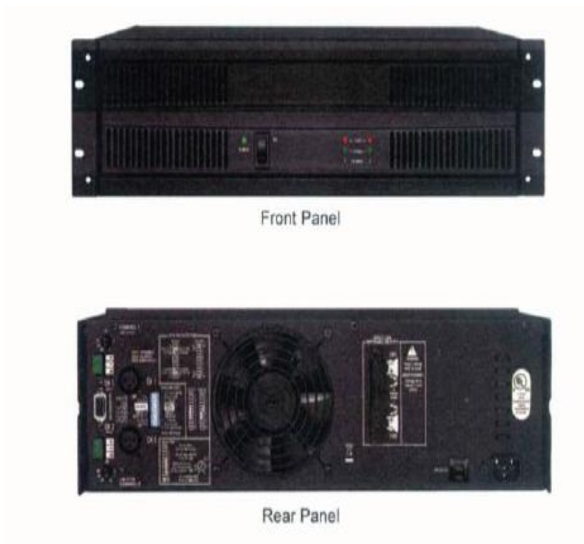
Figure 3.2 : The PCB Piezotronics Model 356A24 triaxial ICP type accelerometer.

As a control output actuator, an electrodynamic vibration exciter is implemented on the active control system. These types of modal shakers convert the alternating signal into an oscillatory motion via moving the armature of the shaker, and the motion is generated by electrodynamic forces created by current passing through a coil buried in a permanent magnetic field.

For the purpose of decoupling and eliminating cross-axis force inputs and measurement errors while using modal shaker, a stinger is used to connect the force transducer and the shaker. In order to control the voltage or current supplied to the modal shaker, a power amplifier is required. In this thesis, the feedback excitation force generated by modal shaker is controlled by using this amplifier. The force applied to accelerate the structure is proportional to the drive current. The TMS Model 2100E18 power amplifier is attached to the modal shaker to amplify and control the output signals generated by sensors [69].

Providing adequate input force to test structure and obtaining accurate and reliable input force measurements is vital for the satisfactory performance of the control system and also achieving good results from modal analysis.

(a)



(b)

Figure 3.3 : (a) The modal shaker and its stinger. (b) The power amplifier.

Driving signal to the control shaker is fed via Laser Doppler Vibrometer and Vibrometer Controller. The Polytec sensor head OFV-505 is used for non-contact vibration measurement of the cantilever beam which moves in a transparent surrounding media [70]. Thus, this vibrometer measures the amount of vibration at a single point on the surface of the structure.

(a) (b)

Figure 3.4 : (a) The laser sensor head. (b) The vibrometer controller.

The laser vibrometer controller uses the principle of the interferometer in order to acquire the mechanical vibration signals and its characteristics. Velocity and displacement amplitude of any vibration object generate a phase or frequency modulation of the laser light due to the doppler effect. In order to perform sufficient amplitude resolution and cover the entire dynamic range, the measurement range of vibrometer controller is set to 1 m/s/V. According to its operating principle, the velocity information is recovered from the frequency modulation of the doppler signal and the displacement signal is reconstructed from the phase modulation of the signal [71]. The main reason for choosing a single point leaser sensor head is to increase the performance of feedback control setups by using its contactless velocity measurement property. Additionally, by using laser sensor, the outputs of the measured velocity signal can be converted into displacement signal via vibrometer controller.

Endevco Modal Hammer Model 2302 is used to excite and measure the impact forces applied to the structure [72]. Using an impact hammer in modal testing, the selection of the hammer tip can have a significant effect on the measurement quality as will be discussed in further sections.

Figure 3.5 : The endevco modal hammer Model 2302.

The signals acquired from excitation and the response sensors are all sent to the LMS SCADAS [73] front-end and the time-domain and frequency-domain analyses are performed by using LMS TestLab 12A modal, signature and impact hammer analysis modules. Moreover, some of the post-processing of the frequency response functions and time-domain signals are performed via using LMS TestLab 12A such as modal damping and loss factor calculations as well as phase shift comparison between the time-domain signals.

3.1.1 Experimental modal analysis of the cantilever beam

In this section, the vibration characteristics of a clamped beam are investigated with the aim of generating a baseline reference time-domain and frequency-domain results for active control applications. The calculation of time-domain analysis and frequency-domain transfer functions are obtained via impact testing.

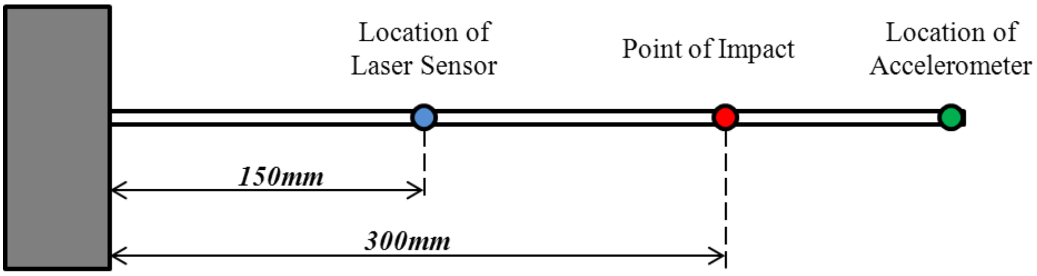


Figure 3.6 : Location of impact point and sensors.

The frequency range of impact excitation is controlled mainly by the hardness of the tip selected. The modal hammer excites the structure with approximately a constant force over a frequency range of interest. In these studies, the impact hammer is supplied with a rubber tip so as to excite relatively lower frequencies and obtain higher tip displacements of the cantilever beam. For low bandwidth of the excitation and relatively long duration of impact, rubber tip is utilized and used with the impact hammer.

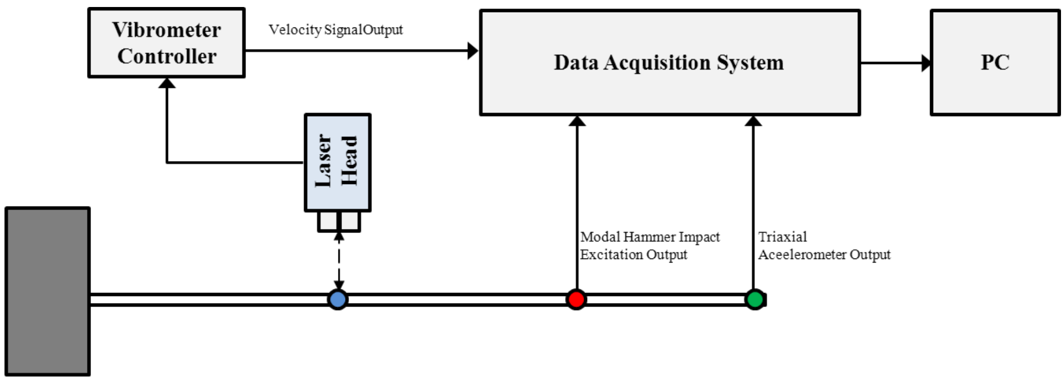


Figure 3.7 : Impact hammer modal analysis diagram of the cantilever beam.

The cantilever beam is divided into 14 sections from root to tip of beam sequentially, the distance between two grid points being defined at 30 mm intervals. Thus, it can alternatively be described as the laser is at grid point 5, the impact point is at grid point 10 and the accelerometer is attached to grid point 14 from impact hammer modal testing point of view.

The laser vibrometer and the accelerometer are used individually as sensors to perform modal analysis to measure the response of the structure to impact hammer excitation with their corresponding locations in Figure 3.6. LMS SCADAS front-end is used to record the outputs of the signals of transducers.

LMS TestLab 12A software is used for modal analysis with the frequency bandwidth of 0-512 Hz and frequency resolution of 0.125 Hz with 4096 spectral lines. An exponential window is set automatically for transient responses of the beam. In order to have reliable and accurate response measurements, impact hammer excitation is applied three times and then FRFs are obtained by linear averaging.

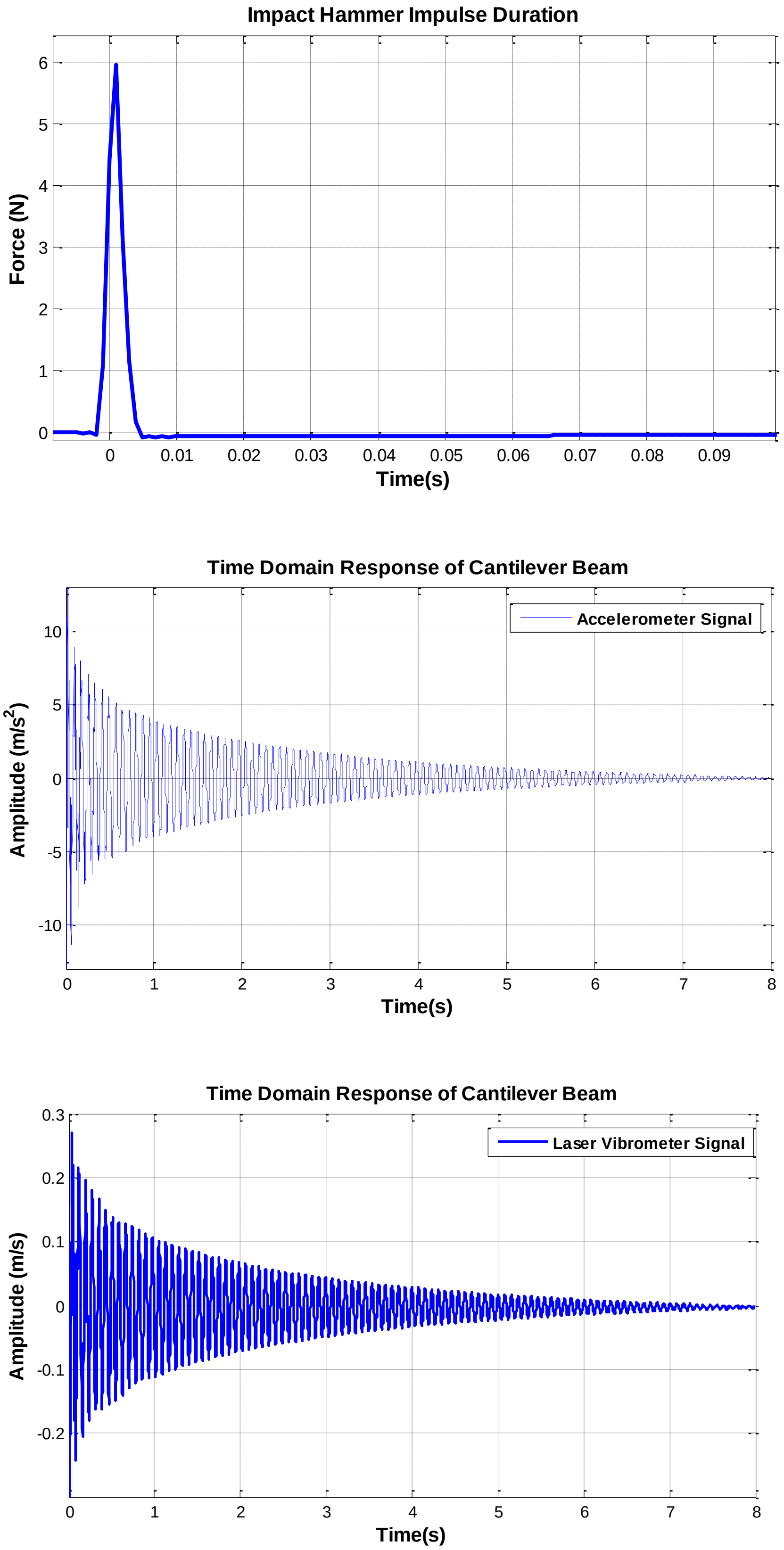


Figure 3.8 : Time records of hammer impulse, accelerometer and laser signal respectively.

The response of the beam is monitored via accelerometer and laser vibrometer sensors, thus the frequency response functions are derived by the recorded signals. Figure 3.8 shows the impact hammer impulse, accelerometer measurement signal, laser vibrometer velocity output signal respectively. As stated before, the tip displacement measurements are recorded via accelerometer signals while the laser sensor acquires data from 300 mm away from the root of the cantilever beam.

As expected, the phase angles between the accelerometer signal and the laser velocity signal is 90 degrees. Figure 3.9 shows a transfer function between the output acceleration response measured by accelerometer and the input impact hammer force. Thenceforth, the frequency response function will be used as a baseline FRF between node 6 as impact point and node 14 as accelerometer of the cantilever beam.

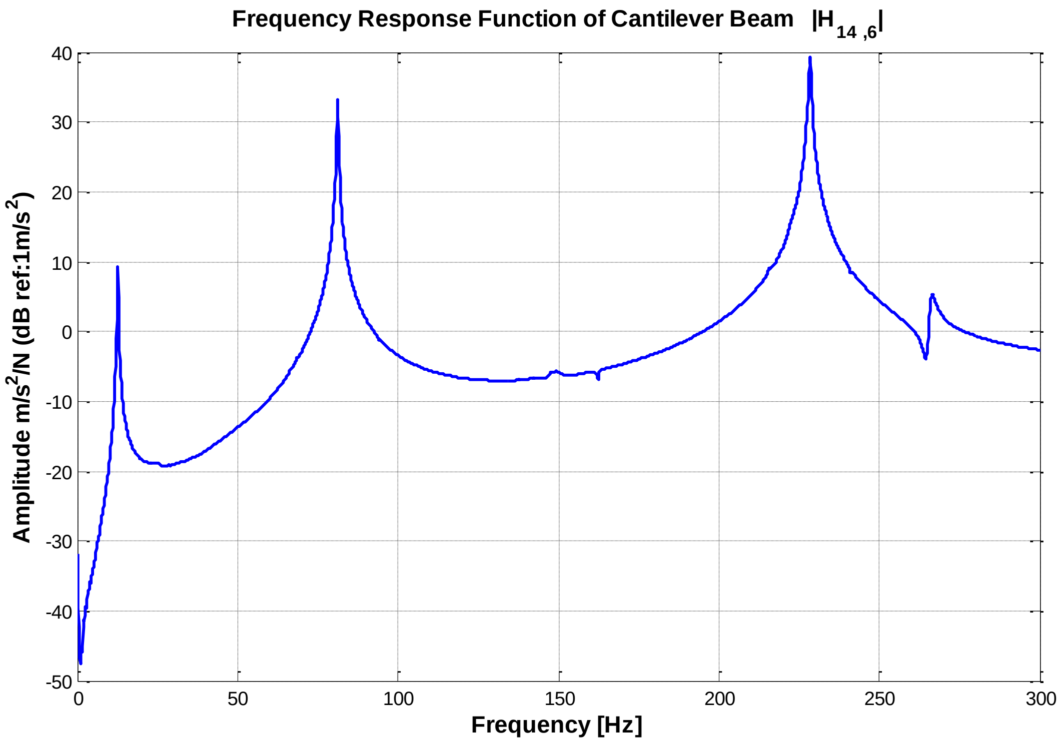


Figure 3.9 : The inertance FRF of the cantilever beam by impact hammer modal analysis.

It is observed from peaks in the measured inertance FRF that natural frequencies for the first three bending modes of the cantilever beam are at 12.75 Hz, 81.2 Hz and 228.4 Hz. The modal damping levels are calculated by using aforementioned software. The experimentally measured first three resonance frequencies, amplitude and calculated modal damping by using accelerometer measurement are listed in Table 3.1.

Table 3. 1 : The parameters of the structure via impact hammer excitation $|\boldsymbol{H}_{14,6}|$.

Mode #	Resonance Frequency (Hz)	Amplitude (dB)	Damping Ratio ζ (%)
1^{st}bending	12.75	9.25	1.30
2^{nd}bending	81.20	33.12	1.08
3^{rd}bending	228.42	39.19	0.25

3.1.2 The effect of the control shaker on the structure

One of the most important aspects of this study is the investigation of the effects of the control shaker on the cantilever beam. The dynamic behaviour of shaker and the effects on the structure is observed by applying modal analysis with and without control shaker to the structure. It is worth to explain here is that the control shaker is non-operational in this experimental condition.

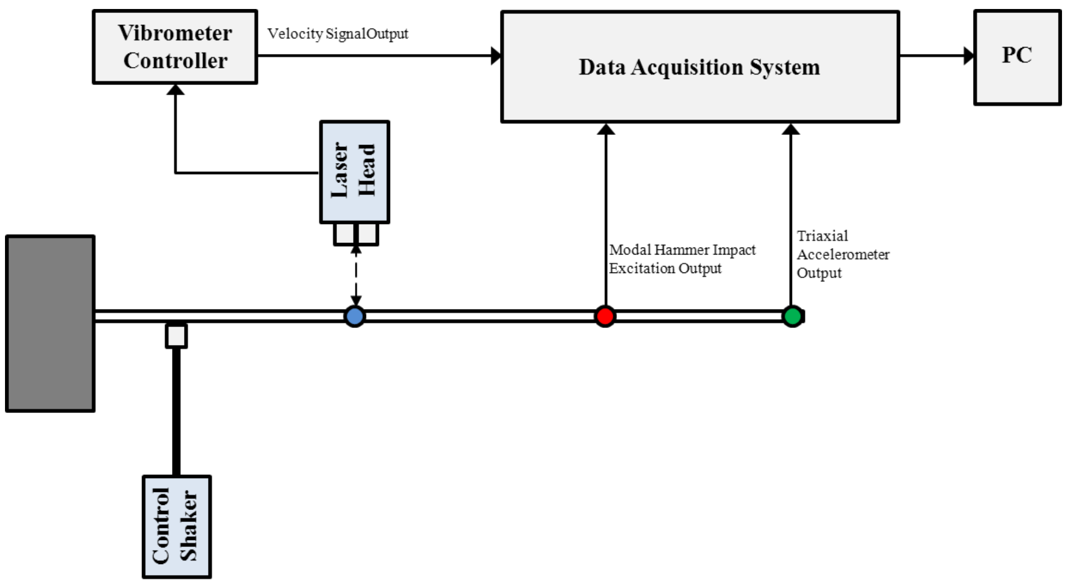


Figure 3.10 : The control shaker attached configuration of the experimental setup.

It is anticipated that the control shaker should be attached as close as to the root of the beam so as to minimize its adverse effect of stiffness, mass and damping modifications. The distance between the root of the beam and control shaker attachment point is set to 50 mm. Shaker attachment of different locations of the beam revealed that minimal effect is achieved by attaching the control shaker as close as possible to the root of the cantilever beam. It should also be noted that attachment of the shaker too close to the root results in poor energy transfer from shaker to the test beam. It is also noted that as the distance between the control shaker location and the root of the beam increases, the dynamic behaviour of the beam changes unexpectedly.

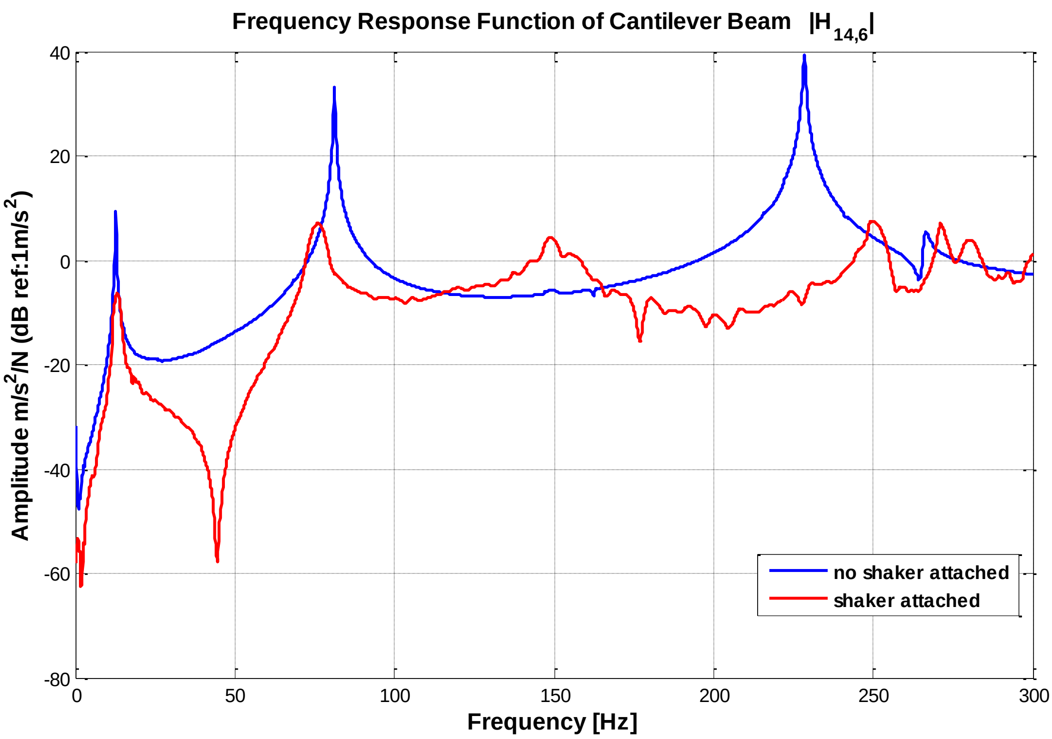


Figure 3.11 : The FRF comparison of shaker attached and unattached configurations.

Results in Figure 3.11 show the shaker attachment causes an additional damping for first bending mode. However, due to the shaker attachment the second natural frequency is shifted from 81.2 to 76 Hz and highly damped. It is also seen from Figure 3.11 that it is hard to get reliable FRFs at frequencies corresponding to the third and higher bending modes due to the adverse effects of shaker attachment. This investigation showed that although the electrodynamic actuator has the sufficient displacement and force capabilities, the main disadvantage of using such a shaker is the undesirable dynamic effects of the shaker itself. It should be stated here that, ideally, a piezoelectric actuator is well suited for closed loop control applications of structures. However, due to financial limitations, such an actuator could not be used in this investigation. Due to the limitation summarized above it is decided that the closed loop control investigation in this thesis should focus on the first bending mode and the first mode is not affected as much as the others in terms of frequency shift.

As a final statement here, it is not surprising that, most of the active vibration control studies presented in the literature survey section is conducted by using light weight actuator materials such as piezoelectric actuator and sensor pairs.

3.2 Analog Low Pass Filter Circuit as a Phase Shifter

The performance of feedback control system depends entirely on the control signals which are used to bring the system to desired states. The analog low pass filter circuit designed to improve the performance of control system is actually for more accurate control of the phase between the input and the output signals.

Low pass filters attenuate the high frequency signal components which are higher than the tuned cut-off frequency and pass the lower frequency components. The input signal is passed through the low pass filter designed in the form of RC (resistor and capacitor) filter. This filter has a cut-off frequency which is equal to 1/RC where R is the equivalent resistance and C is the capacitance.

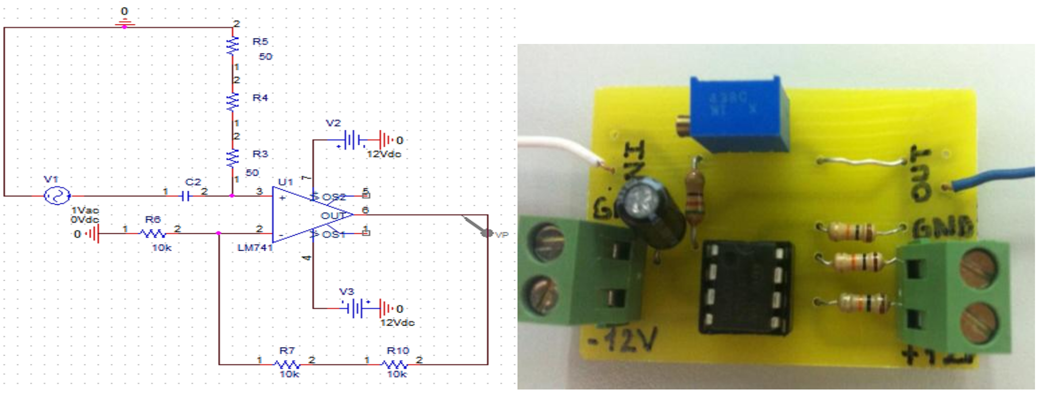


Figure 3.12 : The circuit diagram of the phase shifter low pass filter and the analog low pass filter circuit.

The aim of designing this low pass filter is to perform a controlled phase shift at predefined frequencies during some experimental studies. It is worth stating here that as the filter comprises of a capacitor and resistances, the output signal lags behind the input signal. Due to the time required to charge and then discharge the capacitor as the input signal oscillates. An adjustable resistor is implemented in the circuit of the phase shifter to alter the phase angle manually between input and output signals.

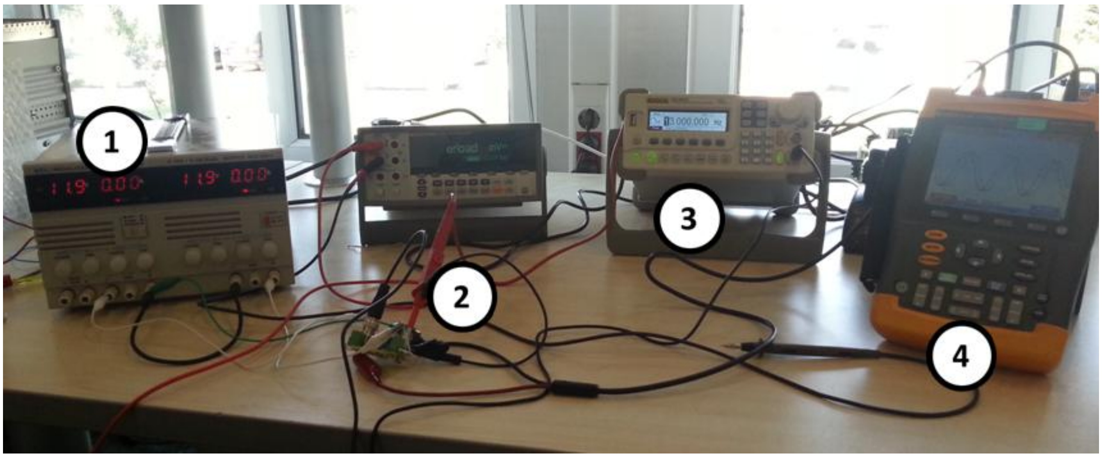


Figure 3.13 : The phase shifter experimental setup. 1) Power Supply 2) Low Pass Filter 3) Signal Generator 4) Oscilloscope.

As it can be seen in Figure 3.13, a signal generator is set to 13 Hz which simulates the first bending mode of the cantilever beam and the power supply provides harmonic voltage to the circuit with a peak level of 12V. An oscilloscope is used so as to observe the phase angle between the input and output signals.

Due to the stability problems of phase lags, the proportional feedback controllers can be applied at lower frequency ranges. In order to employ only one controller parameter and to avoid the need for advanced control algorithm parameter, it is decided to employ a proportional closed loop controller utilizing the low pass filter as a phase shifter.

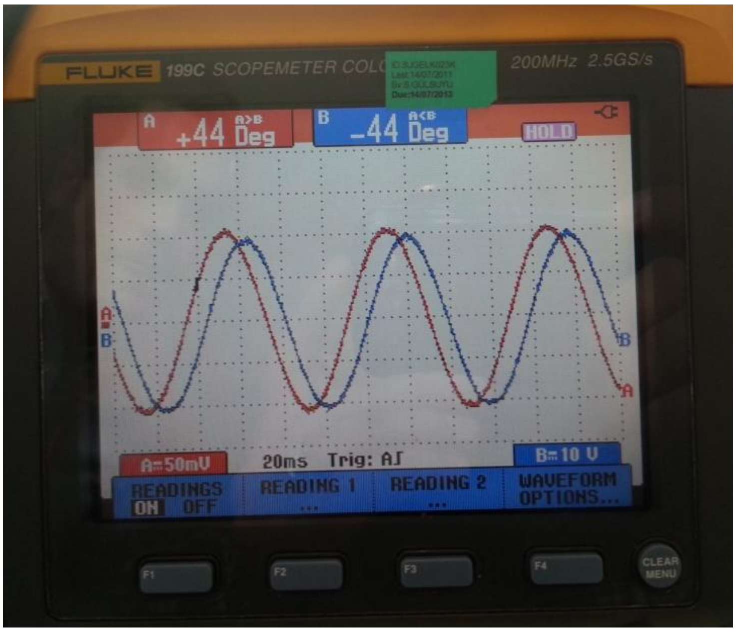


Figure 3.14 : Oscilloscope result.

In Figure 3.14 shows that the phase lag between the output and the input signal is approximately at -45° once the signal supplied to the circuit is at 13 Hz. Moreover, the analog circuit reduces the output amplitudes by 200 times the input signal amplitudes since the designed resistance parameter configurations are set to decrease the output signal amplitudes.

3.3 Summary

In this chapter, the experimental setup and its members are introduced. An impact hammer test is performed to investigate the dynamic behaviour of the cantilever beam. The schematic diagram of controller setups are presented and will be used in further sections as the system varies. The frequency response function is obtained by impact hammer testing and modal parameters of the first three modes of open loop cantilever beam are extracted using suitable software. Besides, the adverse effects of attaching an electromagnetic shaker to the cantilever beam structure as a feedback actuator is studied and reported from dynamic response point of view. Furthermore, the time-domain signals are recorded to investigate the phase characteristics of the output signal. Lastly, an analog low pass filter circuit is designed to control the phase angles between the input and the output signals manually.

4. ACTIVE VIBRATION CONTROL: EXPERIMENTAL STUDIES

In order to study the effects of control feedback signals on the dynamics of the structure, four different experiments were conducted. During the first three experiments, different controllers were utilized namely: displacement feedback controller, velocity feedback controller, acceleration feedback controller. The fourth and the last experiment also comprised, the analog phase shifter described before. In each case, an electrodynamic shaker is utilized as an actuator. As these experiments require somewhat different setups, each test case will be discussed individually. Time-domain and frequency-domain analyses of each system are completed and the final configurations and resultant behaviour of the structure is investigated. In all cases, the excitation and response signals are recorded simultaneously and the frequency response functions are derived from those recorded data. The frequency-domain studies are presented as open-loop versus closed-loop comparisons so as to define the controller performance and its efficiency. The results are processed and presented in various form to demonstrate changes in dynamic behaviour of the test structure due to the control action.

4.1 Time Domain Analysis

In this section, the signal outputs and the performance of the closed loop configurations is discussed. Here, in this section, all the data acquired from cantilever beam setup in time-domain and damping characteristics of the system is investigated using the decaying transient vibration signals.

The feedback control-loop configurations are investigated in time-domain to examine the efficiency of the setups. Time-domain vibration response characteristics under various proportional gains recorded and examined as described next.

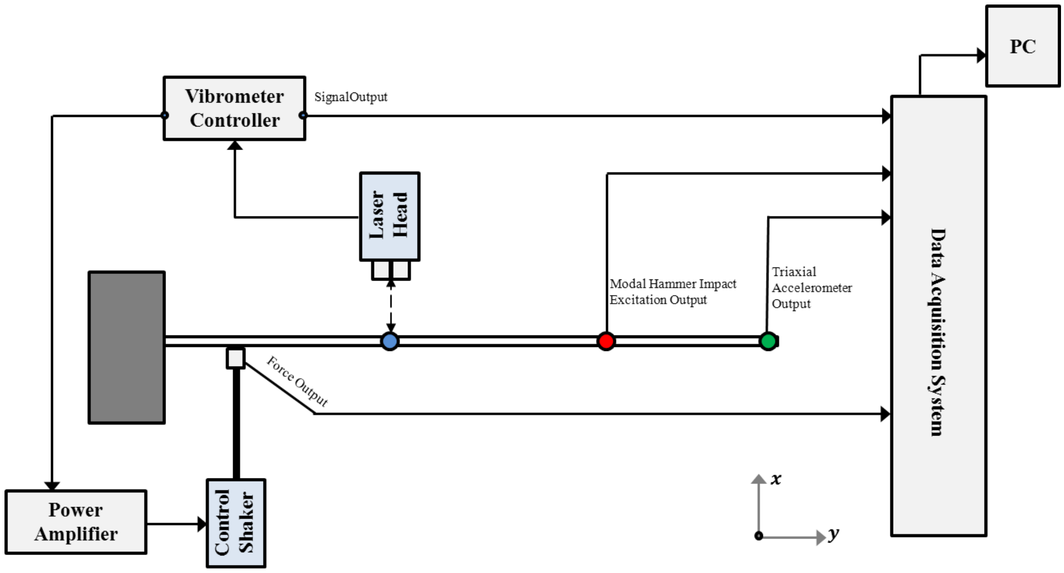
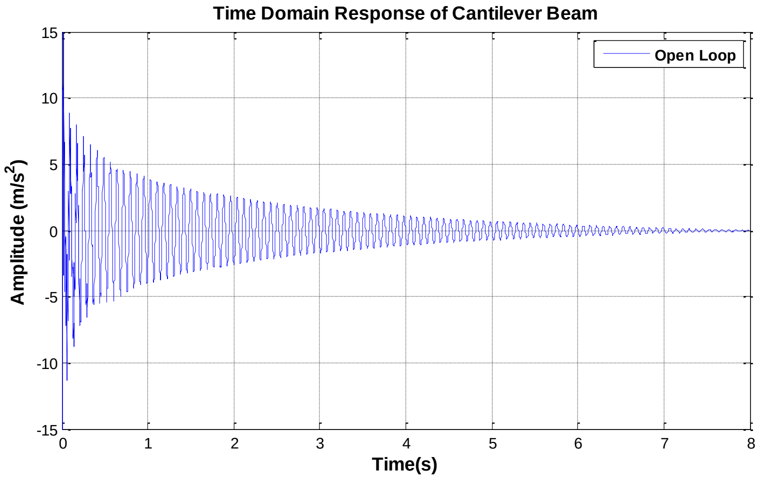


Figure 4.1 : The experimental closed loop configuration for displacement and velocity feedback control setups.

First of all, cantilever beam is excited from the same location, point 10, by impact hammer. Then, the laser sensor vibration signal is sent to the vibrometer controller to create a desired output i.e. velocity or displacement. One of the outputs of the laser controller is sent to the analyser where the other is directly sent to the power amplifier. The output of the amplifier is fed the control shaker to excite the system. Thus, the closed loop setup is completed via excitation of the control shaker with desired feedback signals whose amplitude proportional with the laser signal and the phase angle of this feedback signal is controlled. On the other hand, accelerometer is used to measure the tip acceleration of the beam, this signal is directly sent to the analyser. It should be stated explicitly here that, for the rest of the experiments, acceleration signals is not used in the closed loop feedback control system. Instead, it is used for monitoring purposes only both in time- and frequency-domains. It is also worth stating that the gain of the proportional control is adjusted using the gain control button of the power amplifier manually. There were 8 fixed amplification levels of the power amplifier and for the rest of the experimental studies the proportional amplification of the power amplifier, i.e. the gain of the controller excitation input, will be presented in amplification level 1 to 8.

4.1.1 Displacement feedback control

After the impact impulse applied to the structure with a modal hammer with rubber tip, the resultant vibration signal measured via laser sensor head is sent to the laser vibrometer. The laser vibrometer output is set to displacement output and this output is fed to the power amplifier the output of which is sent to the control shaker. The proportional gain of the power amplifier is set to amplification level 2.



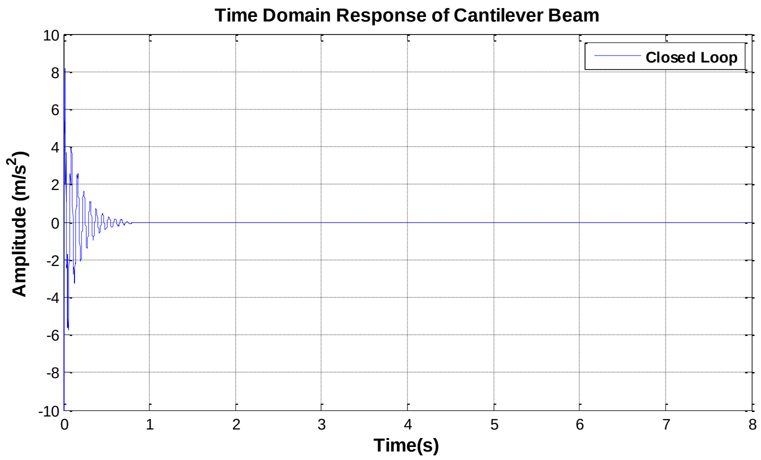


Figure 4.2 : The comparison of open loop response and displacement feedback loop time domain response.

Integrating the measured velocity signal yields displacement signal, and ideally, the phase angle between the vibrometer displacement output signal and the accelerometer output signal should be 180°. Under ideal circumstances, it means that while the beam moves in the positive x direction while the control shaker pulls the structure in negative x direction. In general, it can be said that the control shaker excitation acts like a positive stiffness at the point of the attachment of the shaker. However, as can be seen from Figure 4.2, the feedback control system also introduces an additional damping to the cantilever beam and this means that the phase angle between the control excitation and acceleration is not precisely at 180°. The time domain responses with and without the feedback controller system are compared in order to examine the effects of displacement feedback controller. Briefly, before the controller actuator is activated the tip displacements of the cantilever beam decayed in 8 seconds. However, after feeding back the displacement signal to the control shaker amplitudes decayed to zero approximately in 1 second. In frequency-domain experimental analysis section, the stiffness and the damping effect will be examined in detail. In order to observe an exact stiffness effect of the controller, unexpected phase lag will be modified via the analog phase shifter circuit.

4.1.2 Velocity feedback control

Same time domain are made in section 4.1.1, but this time feedback signal is set to velocity output from the vibrometer controller. Again, the power supply amplified the signal and sent it to the control shaker. The proportional gain of the power amplifier is set to amplification level 2 and 3 respectively.

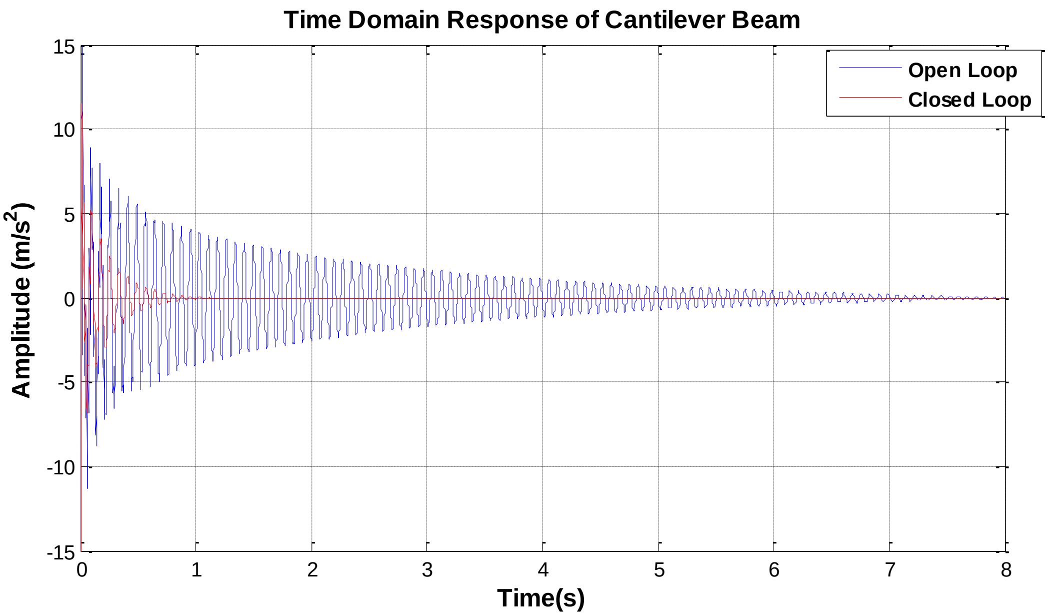


Figure 4.3 : The comparison of open loop response and velocity feedback loop time domain response.

It can easily be observed from Figure 4.3 that decay rates of the time-domain responses are quite different. As expected, applying a force proportional to the velocity causes damping effect, forcing the structure to decay faster. It has seen that the settling time is reduced from 8 seconds to approximately 1 second.

In Figure 4.4, power amplifier is tuned for the amplification level 3 to increase the velocity feedback control performance; hence the effect of the different controller gains on the time domain behaviour of the cantilever beam tip acceleration is investigated.

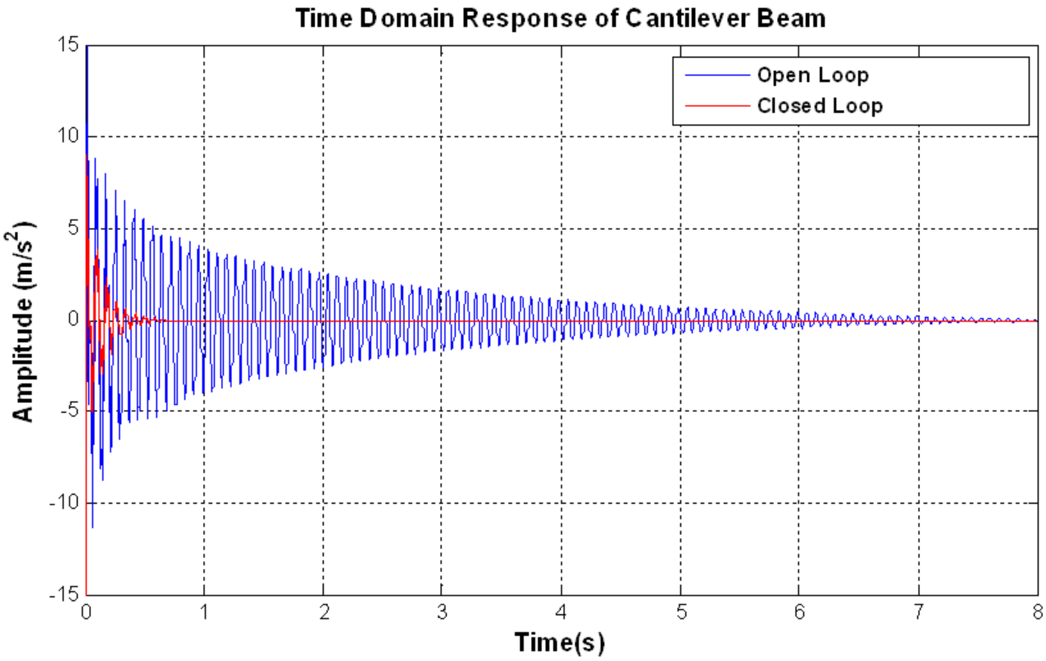


Figure 4.4 : The comparison of open loop response and velocity feedback loop time domain response with higher control gain.

It is obvious that increasing the actuator signal gains decreases the closed loop vibration amplitudes considerably. At first sight, increasing gain leads to increasing damping factor, thus the decay time is reduced from 8 seconds to approximately 0.5 second. In addition to this, signal that measured by the laser sensor is sent to the vibrometer controller and converted into velocity output. Amplifying the velocity signal and feeding that to the shaker generates a force proportional to the velocity. Ideally, the phase angle between the accelerometer measurement and the velocity signal should be 90°. This means that the shaker excitation simulates an artificial damping at the shaker attachment coordinate and the ground.

4.1.3 Acceleration feedback control

In this time domain experiment, it is aimed to apply a feedback control signal which is in phase with acceleration output. Again, the power supply amplified the acceleration signal and sent it to the control shaker. The proportional gain of the power amplifier is set to level 2.

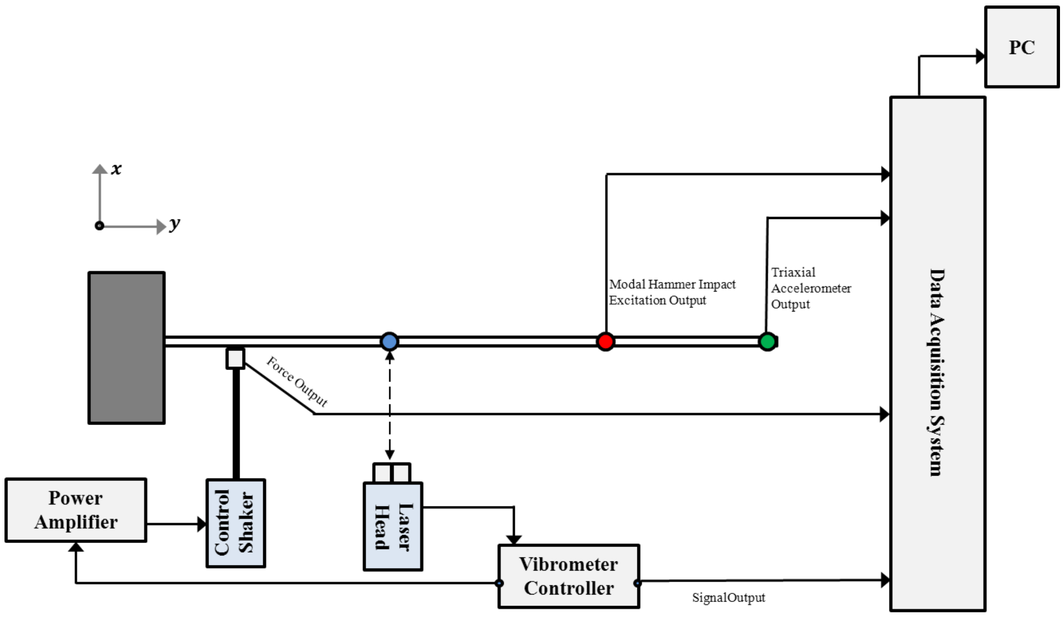


Figure 4.5 : The experimental closed loop configuration with acceleration feedback.

Since the vibrometer controller has velocity and displacement output options, it is decided to use displacement output again. However, the laser sensor is located at the opposite direction of its previous direction so as to simulate a negative displacement measurement which is phase with acceleration output. This means that the control shaker applies excitation in positive x direction while the laser measures the cantilever displacement along positive x vibration. Ideally, under harmonic vibration, by feeding back negative displacement signal to the system means that the control signal applied to the structure and the acceleration signal measurement are all in same phase and along the same direction. In such configuration, the displacement output of the vibrometer controller is fed to the power amplifier.

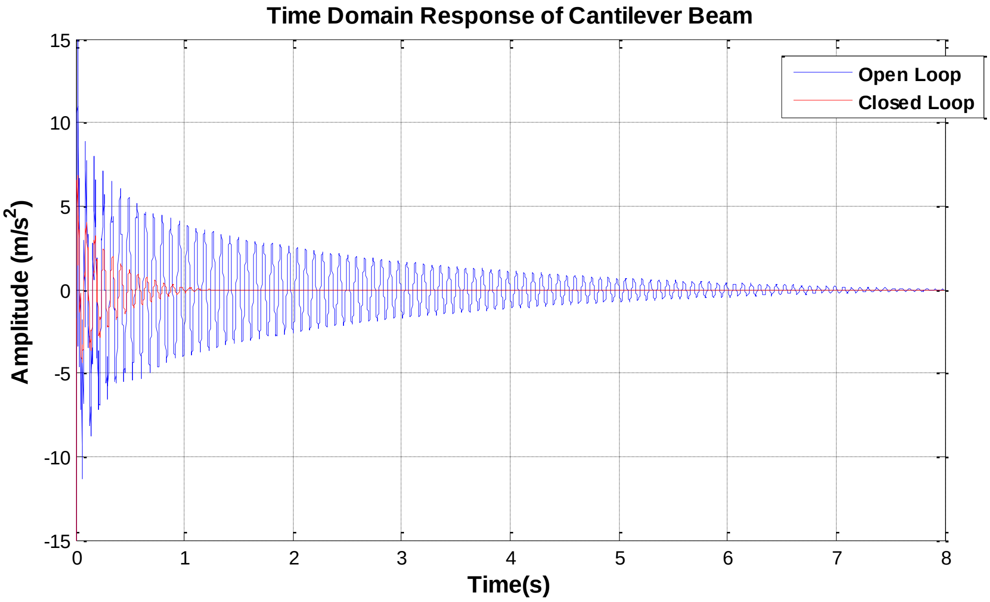


Figure 4.6 : The comparison of open loop response and acceleration feedback closed loop time domain response.

As a comparison, Figure 4.6 presents the tip acceleration level of the cantilever beam and also shows that oscillation duration of the tip acceleration is decreased from 8 seconds to approximately 1.5 second.

The location of the laser is changed to the opposite side of its previous location so as to ideally perform a same phase angle between the accelerometer measurement and the velocity signal. Therefore, in this configuration the shaker excitation simulates a virtual mass response on the related attachment location. It can clearly be seen that, acceleration feedback controller decreases the amplitude levels of vibrations at the tip of the beam, particularly at its fundamental frequency. It is not expected since relatively lower decay time means that the controller applies an additional damping effect on to closed loop controlled cantilever beam. It is clear that this type of feedback control is introducing very significant levels of damping to the system. The reason for this will be addressed in Frequency Domain Analysis section. In frequency-domain acceleration feedback analysis section, the mass effect and the additional damping effect will further be analysed in detail.

4.2 Frequency Domain Analysis

In this section, the effects of different experimental setups and different proportional controller gains on the modal behaviour and modal parameters of the cantilever beam are investigated. The open and closed loop analyses of the structure are completed and the frequency response functions of open and closed loop configurations are compared. Notice that the experimental configurations and the setups for frequency domain analysis are the same with the time domain analysis.

Due to the aim of the control feedback signal observation, the time domain signals are all recorded and the phase differences between the sensors and control actuator are examined. Controller gain is adjusted gradually in order to get the best performance for vibration at first mode of the cantilever beam. For further analyses of the controller setups, different gains are applied and changes in vibration characteristics of the closed loop configurations are examined.

In contrast to expectations, as it is reported in the time domain analysis section, even with the smallest possible displacement feedback control efforts, the controller is able to increase the damping of the closed loop system in the cases of displacement and acceleration feedback configurations particularly. On the other hand, a phase shifter analog circuit is employed additionally to experimental configuration as mentioned before in order to have better control of the phase angles manually for the displacement output control setups.

4.2.1 Displacement feedback control

In this subsection, frequency domain analysis of the controller designed for displacement feedback control is investigated. Moreover, time domain signals are recorded so as to observe the phase angle relationship between sensors and the control shaker excitation. Control gains are increased gradually to examine the efficiency of applying controlled stiffness via displacement feedback signal. Modal parameters are investigated and reported as the system behaviour varies.

The first point here is that time domain analysis shows additional damping effect on closed loop system and it was higher than the expected numerical simulation levels. Nevertheless, it is a must that in the beginning of the analysis, signals from the sensors and the force transducer should be examined.

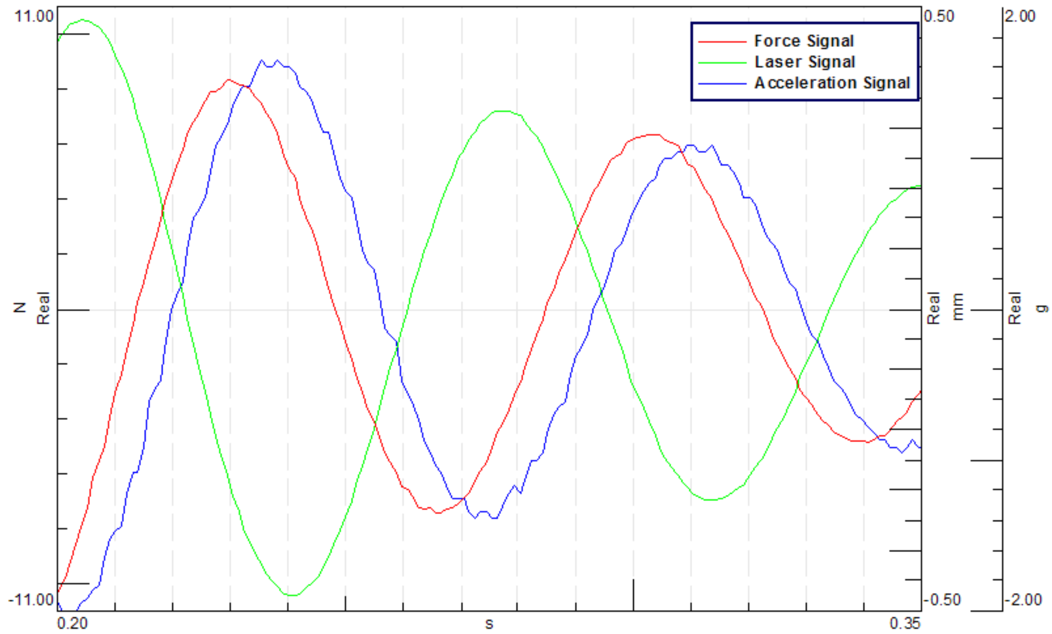


Figure 4.7 : Time domain comparison of sensor signals and displacement feedback excitation.

In order to examine the time domain signal comparison of the sensors and the displacement feedback control excitation, it is validated that the laser displacement signal and the accelerometer signals have 180° phase angle between them as it should be. In order to cause stiffness modification effect, the amplified displacement feedback signal applied to the control shaker should be in the same phase with accelerometer measurement signal. In other words, the force applied to the structure must be out of phase with the displacement in order to cause stiffness effect. However, it is observed that the signal acquired from the force transducer is approximately 45° behind the accelerometer signal, or 135° in front of the displacement signal. Consequently, it is possible to say that the closed loop frequency domain responses will show not only the stiffness effect but also additional damping effect due to the phase angle is between 90° and 180°.

The effect of the five different controller gains on the modal parameters of the cantilever beam is investigated. Notice that experimental setups for open and closed loops of the frequency domain investigation are the same with those configurations in section 5.1.1.

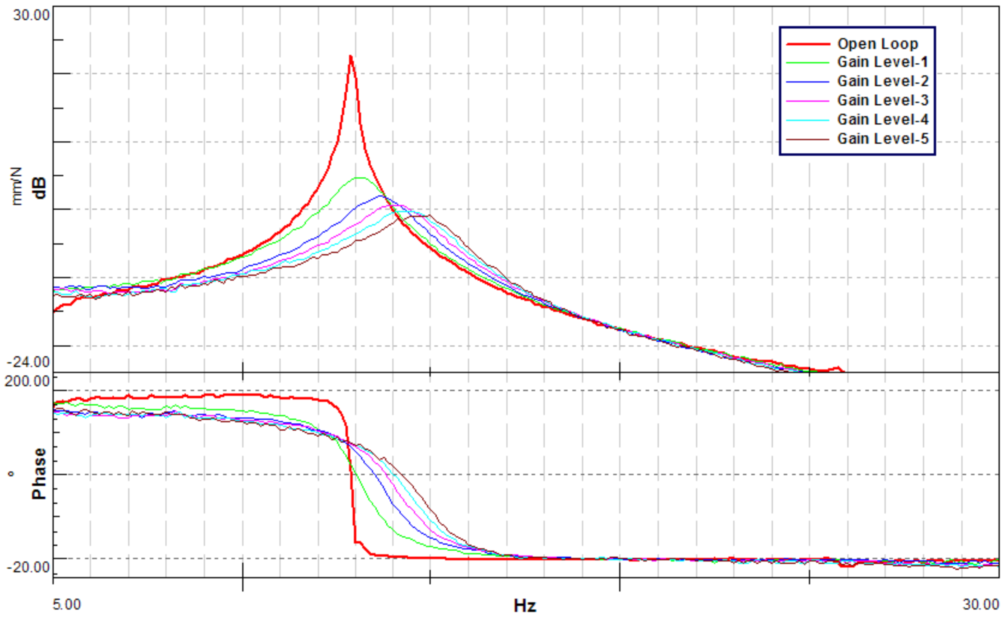


Figure 4.8 : Frequency response functions for open loop and displacement feedback closed loop system for the first mode of the cantilever beam

As stated before, the feedback gains are tuned by the power amplifier and frequency response functions in receptance form are measured with laser sensor head displacement signals and the impact hammer force. In Figure 4.8, the results of the open and closed loop frequency domain analyses are completed for each controller. As expected, the results show both the stiffness and the damping effects are increased as the level of the controller force is increased.

The results also summarized in Table 4.1 that the natural frequency for the first mode of the cantilever beam is shifted gradually from 12.90 Hz to 14.85 Hz via the stiffness effect of the control feedback excitation. In addition to frequency shift observations, effect of the controller on the amplitudes of the FRFs is also obvious. By applying the same feedback excitations, additional damping effect is observed and the amplitudes showed significant decrease from 22.27 dB to -1.05 dB sequentially as the controller gain increased.

Table 4.1 : The modal parameters of the displacement feedback closed loop FRFs.

Control Input Gains	1^{st} Bending Mode (Hz)	Amplitudes (dB)	Damping Ratio ζ (%)
Open Loop	12.90	22.27	0.75
Gain Level-1	13.13	4.64	5.44
Gain Level-2	13.62	1.94	6.73
Gain Level-3	14.12	0.65	7.29
Gain Level-4	14.48	-0.61	7.72
Gain Level-5	14.85	-1.05	7.81

The modal damping and loss factor values are also estimated using modal analysis software and results are listed in Table 4.1. It can be said that, although all the controller gains are efficiently effective on the cantilever beam closed loop configurations, the unexpected phase lag should be altered to its expected level so as to observe a full stiffness effect of the controller setup.

4.2.2 Velocity feedback Control

Here, frequency domain analysis of the feedback controller setup designed for velocity feedback is investigated. In time domain analysis, it is reported that the phase angle between sensors signals and the controller excitation is 90° as anticipated. As explained in the literature survey section, Fuller [26] reported that the velocity feedback systems are often the most inherently stable due to there being a finite time delay between measuring the response by a transducer, processing it and propagating the feedback signal again to the actuator in real physical systems. Therefore, it is obvious that the frequency domain analysis via increasing the control gain gradually will show only the damping effect on the structure. It should be noted that for the closed-loop frequency domain analysis presented here, signal characteristics of the sensors, controller signal and frequency domain analysis procedure remained the same.

Furthermore, modal parameters are examined and especially the modal dampings are listed so as to explain the relationship between the control gains and its effects to first mode of the cantilever beam. It should be noted that, there is no stiffness or mass effect on the closed loop feedback system in this case as predicted in time domain analysis.

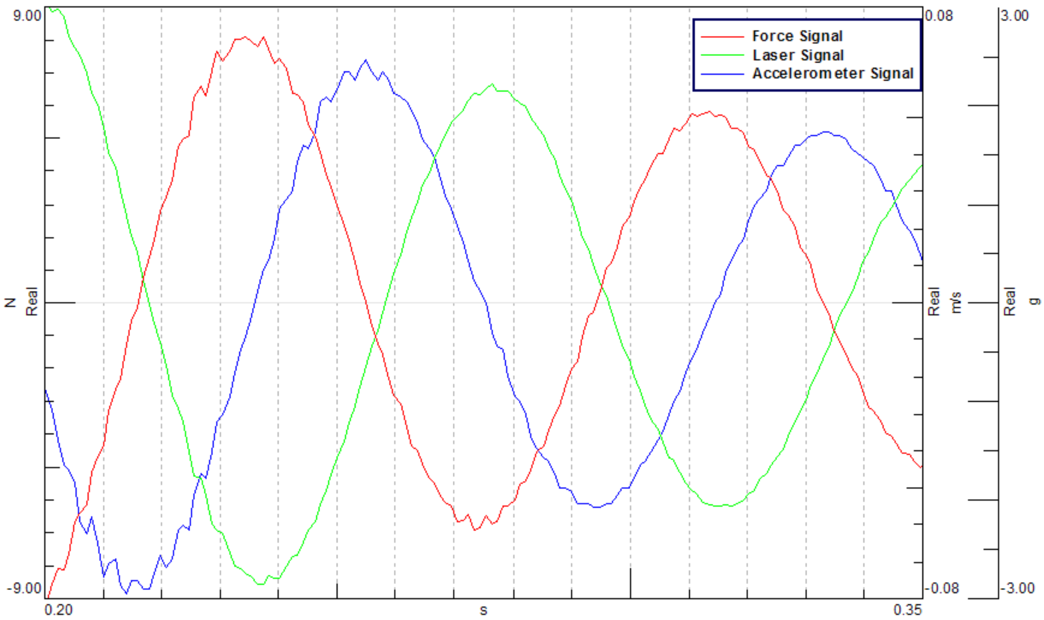


Figure 4.9 : Time domain comparison of sensor signals and velocity feedback excitation.

Fortunately, the time domain comparison of sensors and the velocity feedback signals shows the phase angle between laser and accelerometer is precisely 90° and the phase angle between laser and force signal is 180° as expected. Then, the resultant frequency behaviour of the cantilever beam is examined and analysed separately for each control gain level so as to investigate the damping effect as a function of velocity feedback control gain.

Applying the signal 90° behind the acceleration stands for applying a damping excitation at the control actuator location. Hence, it can be said that the control forces are applied to the system suppress the first mode vibrations of cantilever beam with externally controlled excitation or damping very successfully.

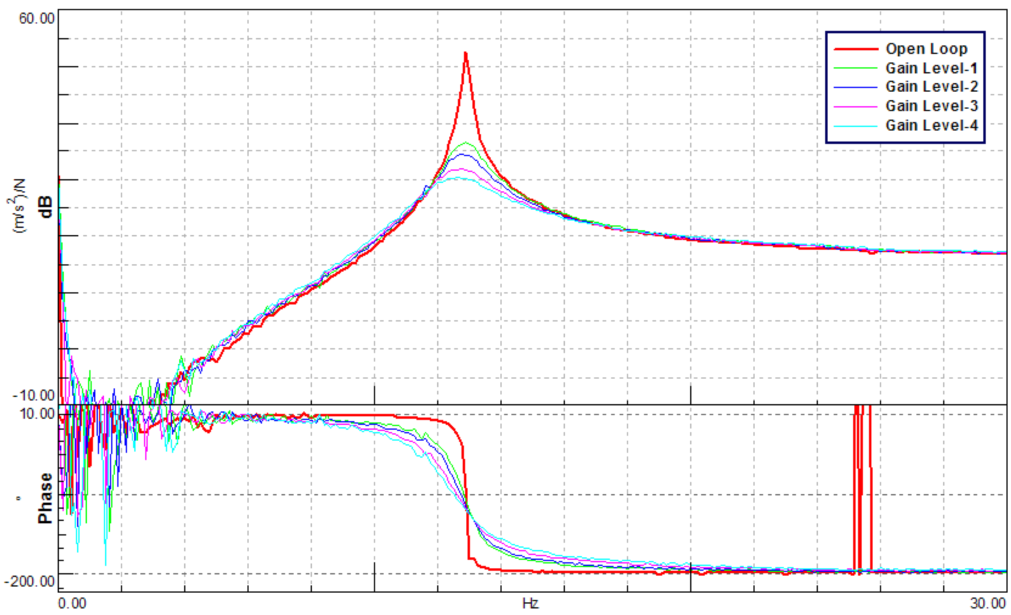


Figure 4.10 : Frequency response functions for open loop and velocity feedback closed loop system for the first mode of the cantilever beam.

The effect of the four different controller gains on the frequency domain behaviours of the cantilever beam is examined. Notice that experimental setups for open and closed loops of the frequency domain investigation are the same as those configurations in section 4.1.2.

At a first glance, time domain phase angle comparison of sensor signals and velocity feedback signal is verified by the frequency domain characteristics of the controller experimental setups. The inertance frequency response functions are measured by using the accelerometer signal and the impact hammer force. As expected, the frequency domain analysis show damping effect since the velocity feedback signals, i.e., the force transducer, is almost 90° behind the accelerometer signal. In addition, the amplitude of the frequency response functions corresponding to the first mode decreased gradually as the controller signal gain is increased. Therefore, in Figure 4.10, amplitudes of the first bending mode of the cantilever beam are falling down gradually from 51.98 dB to 30.14 dB as the control input gain is increased.

Table 4.2 : The modal parameters of the velocity feedback closed loop FRFs.

Control Input Gains	1^{st} Bending Mode (Hz)	Amplitudes (dB)	Damping Ratio ζ (%)
Open Loop	12.90	51.98	0.68
Gain Level-1	12.82	36.43	4.30
Gain Level-2	12.78	34.42	5.32
Gain Level-3	12.70	31.57	7.73
Gain Level-4	12.65	30.14	8.66

The modal damping values are estimated using modal analysis software again. In Table 4.2, it can be said that the damping ratio at related bending resonance frequency value increased step by step from 0.68 per cent to 8.66 per cent as the control input gains increased.

As predicted again, natural frequency change due to velocity feedback control is quite negligible. It is observed from the modal frequencies that as the gain of controller excitation is increased, insignificant frequency shifts are detected. Slight decrease in resonance frequencies is believed to be due to the slight phase angle shift between the control excitation signal and the velocity signal.

4.2.3 Acceleration feedback control

The results of frequency domain analysis of the controller designed for acceleration feedback closed loop system are presented here. The same procedure for the displacement and velocity feedback control experiment followed. Time domain signals are recorded instantly and the frequency domain analysis is performed in order to observe the phase relationship between the control shaker excitation to the system and laser vibrometer output. Again, control gains are increased steadily to examine the efficiency of applying controlled virtual mass or negative stiffness via controlled feedback signal. For further analysis of the frequency response characteristics of controlled system, different gains are applied and changes in modal parameters of the cantilever beam are studied. Modal parameters are investigated and tabulated as the system behaviour varies.

As described acceleration control feedback diagram shown in Figure 4.5, location of laser sensor is switched to simulate a system with mass modification. The main purpose of changing the excitation position is basically stands for inversing the measured displacement signal. Therefore, the laser displacement signal measures the vibrations in –x direction which is defined in a coordinate system in Figure 4.5. While the control shaker applies excitation in positive x direction, the laser measures the cantilever positive x vibration movement. In order to not to be misapprehended by the time domain sensor and control excitation signal comparison figures, it should be noted that the force transducer measures the negative of laser sensor because of their inherent measurement behaviour once they measure on the same side of the cantilever beam.

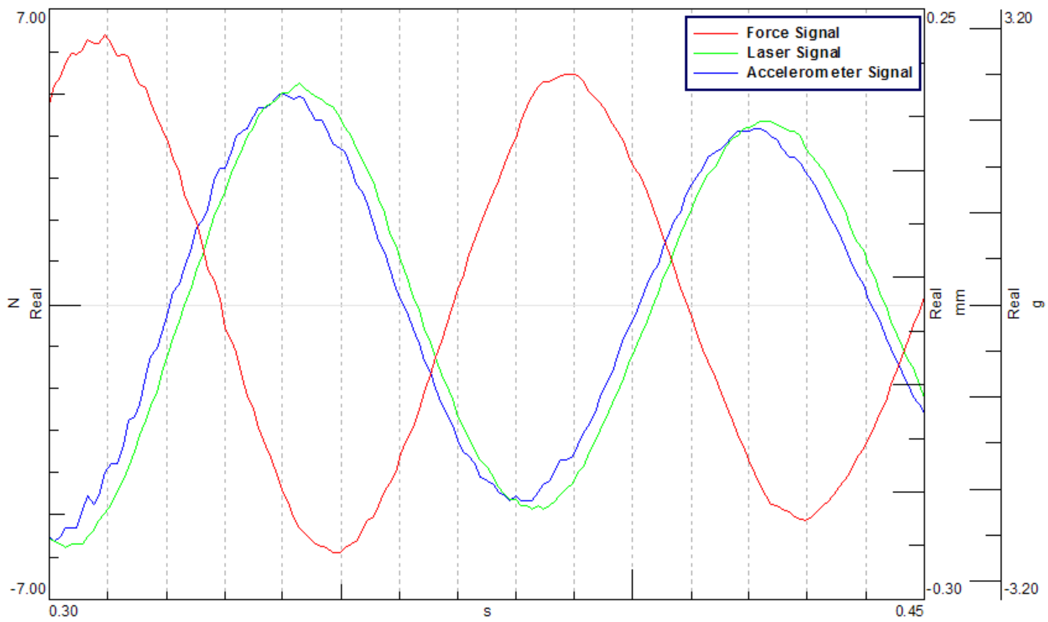


Figure 4.11 : Time domain comparison of sensor signals and acceleration feedback excitation.

However, as stated before, feeding negative displacement control signal to the closed loop system means that the control signal applied to the structure and the acceleration measurement should be in same phase and along the same direction. It is noticed in the time domain analysis of acceleration feedback control that there is unexpected damping on the response of the cantilever vibrations, thus it indicates that the phase between laser sensor signal and the control excitation is not quite the same.

In Figure 4.11, it is seen that laser vibrometer output and the accelerometer signal is approximately in same phase once gain level 1 is supplied to control shaker. On the other hand, control excitation force signal is expected to be out of phase with vibrometer output. However, the time domain signal investigation shows that the control force is 45° behind the laser displacement signal. Thus, this explains why the closed loop time domain responses show the additional damping effect on the response of the cantilever beam since the excitation signal phase somewhere in between 0° and 90°, causing damping as well.

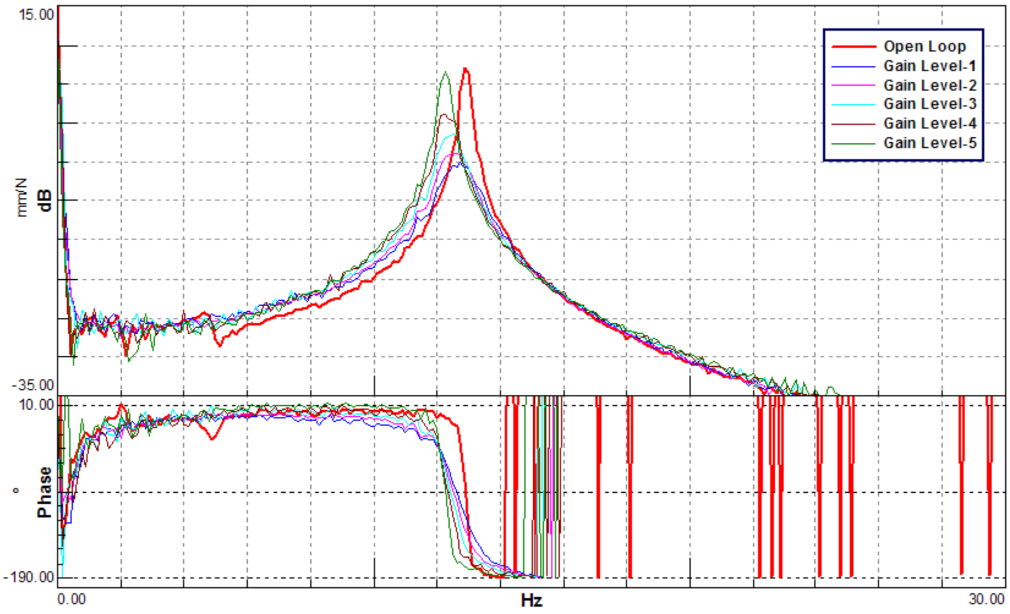


Figure 4.12 : Frequency response functions for open loop and acceleration feedback closed loop system for the first mode of the cantilever beam.

The frequency responses of the acceleration feedback control applied to cantilever beam are presented in Figure 4.12. Its controlled mass effect is seen as the feedback input gains increased gradually, however, especially at lower gains, damping effect of the controller is also seen in Figure 4.12. Notice that by reversing the laser sensor location, the displacement output of the laser vibrometer acted like acceleration output in terms of phase for the closed loop system. Henceforth, as the beam moves in positive x direction, the shaker excitation applies force in the same direction via its stinger. For this situation, this is why the amplitude responses of the closed loop FRFs increase as the feedback gains rise gradually from amplification level 1 to 5.

Table 4.3 : The modal parameters of the acceleration feedback closed loop FRFs.

Control Input Gains	1^{st} Bending Mode (Hz)	Amplitudes (dB)	Damping Ratio ζ (%)
Open Loop	12.90	6.87	1.18
Gain Level-1	12.76	-4.97	5.25
Gain Level-2	12.54	-3.90	4.42
Gain Level-3	12.46	-1.53	3.38
Gain Level-4	12.38	0.56	2.56
Gain Level-5	12.20	6.48	1.40

In Table 4.3, it is seen that the natural frequency for the first bending mode of the cantilever beam is decreasing step by step from 12.90 Hz to 12.20 Hz by the mass effect of the control excitation as the gain is increased. At lower controller excitations, the feedback control system generates both additional negative stiffness and damping on the structure. Fortunately, it can also be examined that as the controller gain is increased manually, the modal dampings of the first bending mode of the cantilever beam increase.

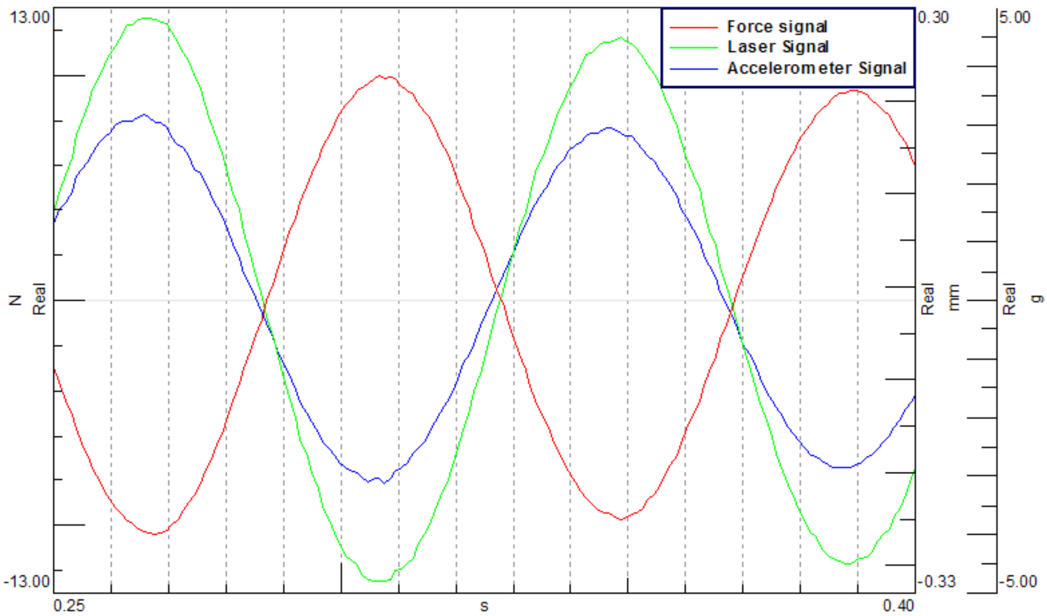


Figure 4.13 : Time domain comparison of sensor signals and acceleration feedback excitation with amplification level 5.

Another thing here is that as the feedback gains rise, better negative stiffness on the response of the structure is achieved. It means that the phase angle between the laser response and the control excitation is getting closer to the desirable value as the as the feedback gain rises. Therefore, the amplitudes of FRFs rise gradually and overcome the undesired damping effect itself. Due to this observation, it is better to observe the time domain characteristics of sensors and the controller input excitation in order to understand the phase angle relationship whether the signals are in same phase angle of each other as expected.

In closed loop frequency-domain analysis, it is pointed out that relatively higher feedback gain conditions show no damping effect for the cantilever beam first bending mode. Figure 4.13 presents the time domain signals of closed loop sensor measurements and control feedback excitation signal. Although the figure shows that the force measurement and the laser signal are become the opposite signed of each other, the control shaker applies excitation in positive x direction while the laser measures the cantilever positive x vibration movement. Thence, it can easily be observed that the force signal, which represents the control feedback excitations, and laser displacement output signal measures and excites in same direction. In other words, the phase angle between the control excitation and laser signal is approximately 0° as it should be.

4.3 Frequency Domain Analysis with Analog Phase Shifter Circuit

In this section, active vibration suppression of the natural frequency for the first mode of the cantilever beam by using analog low pass filter as a phase shifter/adjuster is investigated. In addition to the previous experimental configurations, a low pass filter is employed after the vibrometer displacement output so as to adjust and regulate the undesired additional phase angle between sensors and controller excitation signal. Additionally, time-domain and frequency-domain studies of open and closed loop configurations of the cantilever beam are evaluated and reported.

In subsection 4.2.1, it is pointed out that relatively higher amount of damping effect is included to displacement and acceleration feedback control inputs since the signal is not exactly out of phase with the laser displacement output signal. It is noticed that the signal acquired from the force transducer is approximately 45° behind the accelerometer signal, 135° in front of the laser signal.

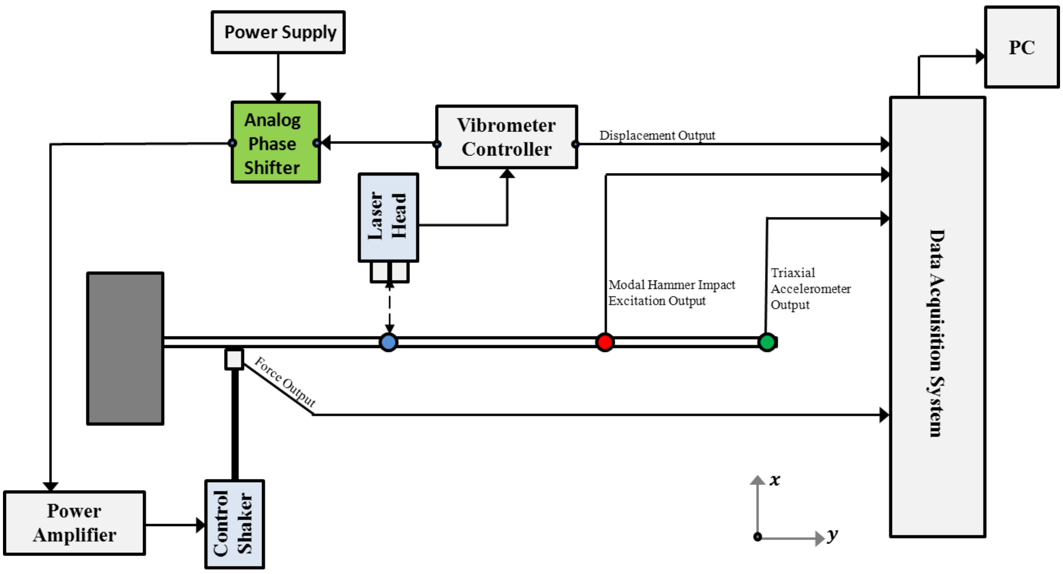


Figure 4.14 : The experimental closed loop configuration with analog phase shifter.

Due to this discrepancy, an analog low pass filter is employed as phase shifter, thus Figure 4.14 shows the closed loop setup with analog low pass filter as a phase shifter. By applying an additional 45° phase lag to the feedback excitation, it is aimed to achieve the desired phase angle between the input and control signals. In other words, via using analog phase shifter, velocity feedback control is performed by laser displacement measurements. Moreover to this, the analog circuit reduces the output amplitudes by 200 fold due to the input signal attenuation within the analogue phase shifter as a consequence of the resistive element. Additionally, the phase angle between output and input signals of the phase shifter is approximately at -45° when the signal supplied to the circuit is at 13 Hz which is the natural frequency of the first bending mode of the cantilever beam.

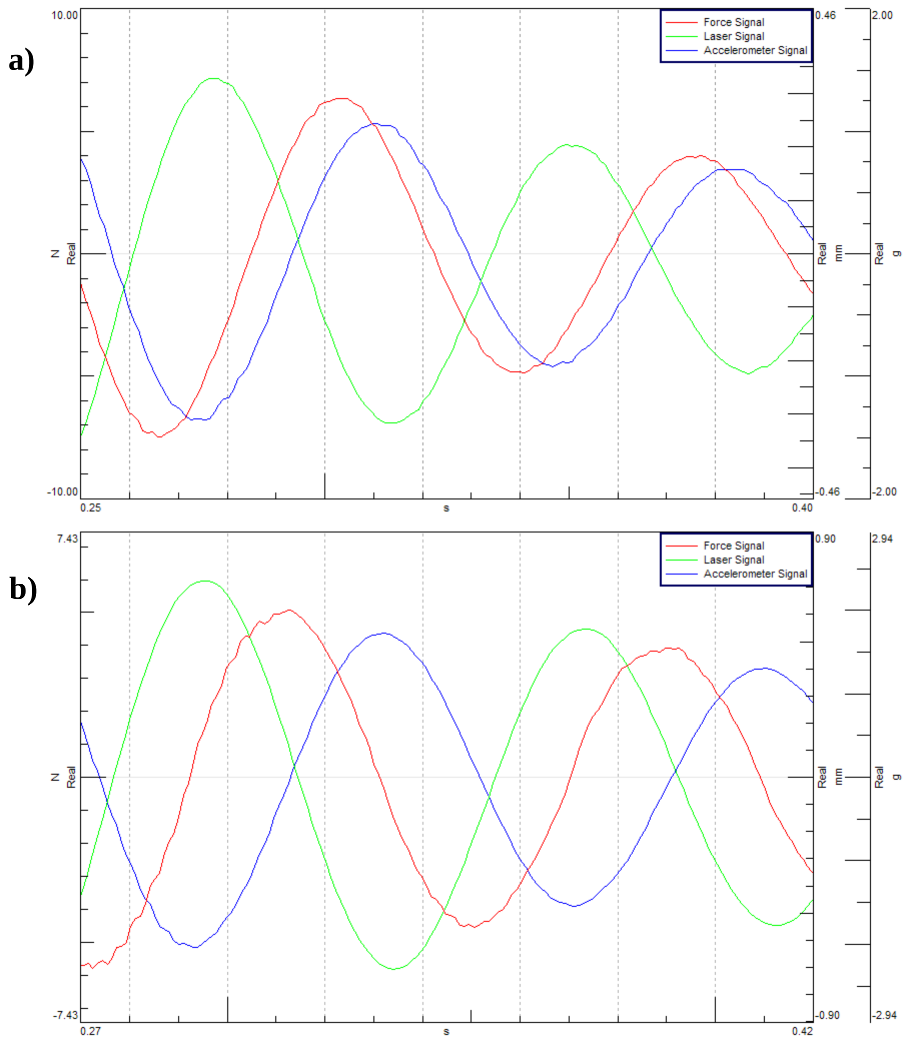


Figure 4.15 : Time domain comparison of sensor signals and acceleration feedback excitation. a) without analog phase shifter b) with analog phase shifter.

Following the time domain phase relationship investigation, Figure 4.15 shows time domain comparison of signals with and without phase shifter closed loop time domain comparison results. Briefly, Figure 4.15.a presents the previous condition of the displacement feedback closed loop system and here the force signal is approximately 45° behind the accelerometer measurements. Applying additional negative 45° phase angle to the system via analog low pass filter supplies 90° phase shifted controller signal which is presented in Figure 4.15.b. Then in that case, the system acts like a perfect velocity feedback controller (adding damping to the system) and it is presumed in this case that the controlled feedback excitation will apply only damping to the cantilever beam in closed loop.

As indicated before, the analog circuit reduces the output amplitudes by 200 fold the input signal amplitudes for the reason that designed configurations of resistance parameters are set to decrease the output signal amplitudes.

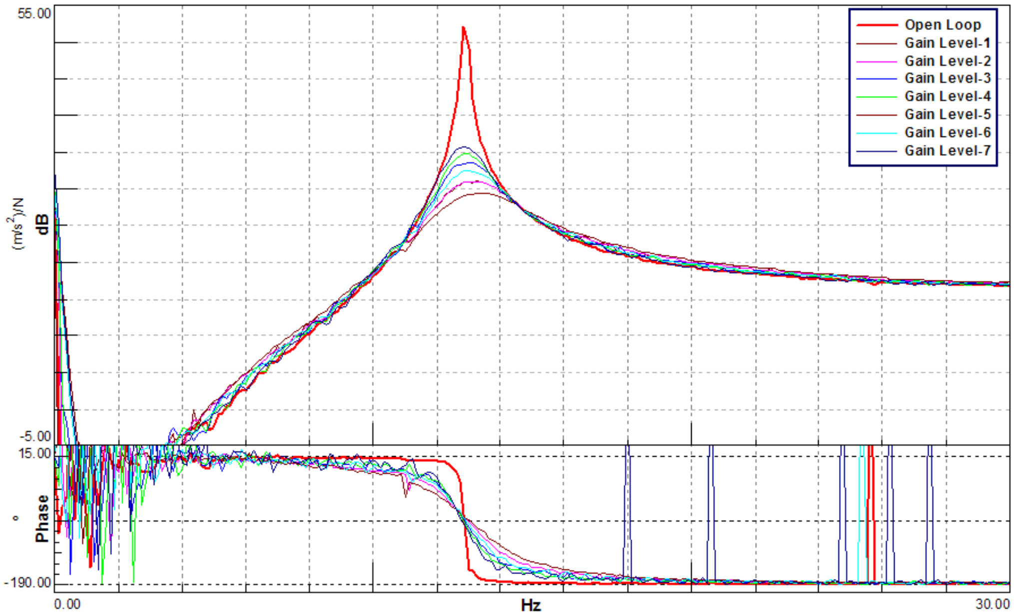


Figure 4.16 : Frequency domain open loop and displacement feedback with analog phase shifter closed loop FRF of the first mode of the cantilever beam.

Once again, the frequency response functions are measured by computing the displacement output signal of laser vibrometer signal and the impact hammer force. Inertance FRFs of open loop and closed loop system are also measured and presented in Figure 4.16. The results show, as anticipated, that the controller feedback gains only apply damping to the cantilever beam.

The amplitudes of first bending mode of the beam fall steadily as the controller input gain is incremented step by step. In other words, it is obvious that as the controller input gain increases, vibration suppression or damping effect is also increased. Effect of the different control gains on the modal behaviour of the cantilever beam is studied and in this configuration seven different feedback control gains are applied to the closed loop system.

At a first glance, time domain phase angle comparison of sensor signals and velocity feedback signal verified by the frequency domain characteristics of the controller experimental setups. The frequency response functions are measured between computing the displacement output signal of laser vibrometer signal and the impact hammer force.

Table 4.4 : The modal parameters of the FRFs using analog phase shifter.

Control Input Gains	**1^{st} Bending Mode (Hz)**	**Amplitudes (dB)**	**Damping Ratio ζ (%)**
Open Loop	12.90	52.17	0.77
Gain Level-1	12.90	35.56	5.42
Gain Level-2	12.90	34.81	5.56
Gain Level-3	12.90	33.51	6.84
Gain Level-4	12.90	32.53	7.71
Gain Level-5	12.90	30.93	9.17
Gain Level-6	12.90	29.02	11.89

As shown in Table 4.4, it is pointed out that the amplitudes fall significantly from 52.17dB to 29.02 dB as the control gains increased. Therefore, the modal dampings show an upward increment from 0.77 per cent to 11.89 per cent via controlled damping effect of the controller.

The first bending frequency of the open loop and closed loop feedback systems are stayed at the same frequency thus all the feedback control configurations perform effective damping on the structure and the phase angle between laser and the control excitation signal is always at 90°. It is worth to explain here is that the phase angle between the displacement and the force transducer signal is 180°. Besides, it can also be examined that modal damping of the natural frequency for the first mode of the cantilever beam is increased as predicted.

4.4 Summary

In this chapter, active vibration control by using an electrodynamic shaker as a control actuator and an analog low pass filter as a phase shifter to manually control the output phases are presented. The time-domain and frequency-domain analyses are performed for all experimental configurations.

Firstly, the time domain analyses of closed loop systems are studied in order to examine the effectiveness of the individual experimental setups. Then, the frequency response functions are measured for each experimental control setup. Also open-loop versus closed-loop comparisons is made so as to define the controller performance, efficiency and to observe the structural dynamic behaviour variations in terms of amplitude, modal damping and modal frequency respectively.

It is observed that both of the time-domain and the frequency-domain displacement and acceleration feedback control setups also exhibit some additional damping on the closed loop cantilever experimental conditions. It is identified that this is because the phase angles are not as expected as in ideal conditions, particularly those of which use displacement feedback signals from the vibrometer output in its closed loop system. However, in acceleration feedback control section, it is worth to report here is that the phase angle between the laser response and the control excitation are become closer to the desirable value as the as the feedback gain rises. Then, it is said that a controlled mass effect without any additional damping is applied to the structure.

Finally, a new experimental controller setup is introduced that includes an analog low pass filter as a phase shifter, thus laser displacement measurement feedback controller setup turned into a velocity feedback controller setup via adjusting the output phase angle. As predicted, the results show that feedback control excitation of this control system performs perfectly and provides gradually increased damping levels to the structure as the control input gain is increased.

5. CONCLUSIONS AND SUGGESTIONS FOR FUTURE WORK

5.1 Achievements and Conclusion

In active vibration control, sensors and actuators are used to implement a feedback control on a structure in order to suppress the vibrational responses or adjust the structural resonances via shifting the natural frequencies to other frequencies and/or applying additional damping. In this study, it is aimed to perform an active vibration control in order to observe the dynamic responses of a cantilever beam by altering the first bending natural frequency and also by adding additional damping to the system. There were four cases which were the subject of experimentation. Thence, this thesis devotes effort on the closed loop responses of a cantilever beam subjected to displacement, velocity, acceleration feedback and phase adjusted feedback signals individually.

In Chapter 1, a detailed literature survey of advancement of active control of vibrations technology is presented in academic and industrial studies manner. Chapter 2 explains the background of control theory and active control theory so as to present the analytical manner behind the numerical and experimental procedures used in this thesis.

Chapter 3 gives the detailed information of experimental setup and its members utilized for active vibration control experiments. The impact hammer modal analysis is performed to observe the vibration characteristics as well as to generate a baseline response in time-domain and frequency-domain studies for further closed loop analyses. Briefly, a laser sensor and its vibrometer controller are employed to perform measurements of beam responses and derive output signals to displacement or velocity output. As an actuator, an electrodynamic shaker is utilized to generate feedback control signals at the attachment location of the beam. A power amplifier is used during the study so as to supply power to the control shaker and to adjust output gains manually.

During the experimental studies, the impact excitation is used to excite the structure using a modal hammer with a rubber tip. After investigating the vibration characteristics, the negative effects of actuator shaker are reported. Also, a low pass filter which is designed to alter phase angles of laser measurement outputs is presented. To summarize, the preparation of control loop experimental setups and the baseline analysis are performed and examined.

Chapter 4 explains the experimental efforts on studies for the active vibration control of the cantilever beam which is individually subjected to displacement, velocity, acceleration and phase adjusted feedback signals. The factors that influence the dynamic behaviour of closed loop system are studied in two parts as time domain and frequency domain. Consequently, the benefits and effects of implementing an analog phase shifter circuit on displacement feedback closed loop system are also described in frequency domain. The responses and relative performances of designed experimental setups are compared and variations of dynamic characteristics of the closed loop structure due to the feedback signals are examined.

Effects of the experiments on modal behaviour of the cantilever beam are analysed via a number of different control gain. The results of the time domain experiments revealed that the feedback closed loop systems are effective in suppressing the tip displacements of the beam. The open loop tip displacements decay in 8 seconds. However, when the displacement feedback closed loop is in action, it takes approximately 1 second. Similarly, in velocity feedback control loop the tip displacements are suppressed in approximately 1 second and the acceleration feedback control returned a settle time in 1.5 seconds. These results showed the effectiveness of the designed controlled loop systems. However, another thing here is that an additional and undesired damping effect is also observed in displacement and acceleration feedback control results.

In order to explain the damping issue, frequency-domain experiments are conducted and instant time domain measurements are recorded for phase angle investigation between input and output signals. The results show that phase angle between the controller excitation and the measured response is not as predicted. In displacement feedback controller setup, the signal acquired from the force transducer is approximately 45° behind the accelerometer signal, 135°in front of the laser signal. By feeding back 135°instead of feeding back 180°means that supplying the control inputs to the system with both stiffness and damping effect. The results of the acceleration feedback control setup revealed that the control force is 45° behind the laser displacement signal which yields negative stiffness (or additional mass) and damping effect. Fortunately, by increasing the control input gain of acceleration feedback the phase angles rises to the desired angles and the measured frequency response function returns the expected results.

In displacement feedback controller setup, the natural frequency of first mode is changed from 12.90 Hz to 14.85 Hz gradually via stiffness effect of the control feedback force. Moreover, amplitudes of frequency responses showed a significant decline from 22.27 dB to -1.05 dB and damping ratios rose dramatically as controller gains are increased.

On the other hand, the velocity feedback controller setup resulted in very effective damping for the first flexural mode as confirmed in the time domain observations. The amplitudes of the resonance frequency are reduced gradually from 51.98 dB to 30.14 dB. The damping ratios of the first mode showed a considerable surge from 0.68 per cent to 8.66 per cent. An additional but negligible frequency shift is also noticed.

In order to perform the acceleration feedback control, the experimental setup is prepared and described in detail. It is reported that the natural frequency of the first mode of the cantilever beam is modified from 12.90Hz to 12.20 Hz gradually by additional mass effect or so-called negative stiffness effect of the applied feedback excitations. As explained previously, as acceleration feedback control input gains are increased, the phase angle between the feedback signal and the laser signal is levelled off, in other words the phases of actuator signal and sensor signal became as expected. Then the modal damping value is returned approximately to its original level of 1.40 per cent.

As a result, it can easily be pointed out that all types of feedback control closed loop system can either reduce the frequency response over first bending mode of the cantilever beam or reduce the responses by shifting the natural frequency to other frequencies by increasing the damping ratio of the first structural mode. It is also reported that the dynamics of the structure is modified by applying controlled feedback signals proportional to displacement velocity, acceleration or combination of these.

In the interest of adjusting the phase angles of measured signals and the controller excitation signals, an analog phase shifter is designed with the aim of generating an additional -45°when the signal is supplied to the circuit is at about 13 Hz which is the first bending natural frequency of the cantilever beam. After implementing the designed low pass filter in experimental setup, the phase angle is set to 90° between controller excitation and the laser signal from undesired 135° in displacement feedback situation. The results of the frequency domain analyses showed that the amplitudes of the first bending mode which is at 12.90 Hz is significantly decreased from 52.17 dB to 29.02 dB as the actuator feedback signal gains increased. As a final comment, analog phase shifter implementation performed effective active vibration suppression up to 22 dB via displacement measurements of the laser sensor.

5.2 Suggestions for Future Work

The author believes that studies and results of this thesis may serve sufficient fundamental information and motivation for further studies.

Experimental setups can be improved further. During this study, most striking aspect of the experiments is that the control actuator is very bulky and it is not an integral part of the controller system. Therefore, utilizing advanced actuators such as piezoelectric materials or magnetostrictive materials would provide a wider range of control capabilities to active control studies. Then, more research would also be generated on the optimization sensor-actuator pair placement on the structure.

One thing here is that the power amplifier of the controller shaker in this study is not capable of performing precise amplification, thus more controllable power amplifiers can be employed to generate much more appropriate proportional feedback gains.

Another suggestion that deserves further investigation is more advanced and effective low pass filters can be designed so as to adjust the phase angle between input-output signals more efficiently. A particularly good example here is that adjustable capacitance and resistance parameters would serve both amplification and phase shifting modification opportunities.

In addition, one may employ a digital controller so as to observe and utilize it with complex and advanced controller algorithms. Therefore, the results that reported in this thesis may be developed for controllability of a number of modes at the same time. Advanced algorithms can provide simultaneous modification opportunity for desired number of modes within the selected frequency range. Thus, optimization studies of sensor actuator pair locations can be generated with no limitation.

Last but not least, instead of using lumped parameter models for numerical analysis, finite element modelling or distributed parameter models can be studied to design a controller for active vibration control of flexible structures.

Consequently, an alternative future study would be the adaptation of these studies into complicated geometries or more complex structures such as washing machine or automotive applications so as to observe the efficiency of such active vibration control methodologies in real life applications.

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