

DOKUZ EYLÜL UNIVERSITY
GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

**ACTIVE AND PASSIVE VIBRATION CONTROL OF
SINGLE-LINK FLEXIBLE SMART
MANIPULATORS**

by
Mehmet UYAR

December, 2020
İZMİR

ACTIVE AND PASSIVE VIBRATION CONTROL OF SINGLE-LINK FLEXIBLE SMART MANIPULATORS

**A Thesis Submitted to the
Graduate School of Natural and Applied Sciences of Dokuz Eylül University
In Partial Fulfillment of the Requirements for the Degree of Doctor of
Philosophy in Mechanical Engineering, Machine Theory and Dynamics Program**

**by
Mehmet UYAR**

December, 2020

İZMİR

Ph.D. THESIS EXAMINATION RESULT FORM

We have read the thesis entitled "**ACTIVE AND PASSIVE VIBRATION CONTROL OF SINGLE-LINK FLEXIBLE SMART MANIPULATORS**" completed by **MEHMET UYAR** under supervisor of **PROF. DR. LEVENT MALGACA** and we certify that in our opinion it is fully adequate, in scope and in quality, as a thesis for the degree of Doctor of Philosophy.

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ACKNOWLEDGEMENTS

I would like to thank my supervisor, Prof. Dr. Levent MALGACA for his very valuable guidance, his support and his critical suggestions throughout my doctoral studies. It was a privilege to study under his supervision.

I am grateful to the members of my doctoral committee, Prof. Dr. Hira KARAGÜLLE and Assist. Prof. Dr. Serkan GÜNEL, for their careful review and advice during the research.

I would also like to thank my colleagues, Research Assistant Şefika İpek LÖK and my friends for their inspiration.

Finally, I wish to express special thanks to dear my wife, EMİNE for her encouragement, patience and love during this doctoral work.

I wish to dedicate this thesis to my parents who have always supported to me.

Mehmet UYAR

ACTIVE AND PASSIVE VIBRATION CONTROL OF SINGLE-LINK FLEXIBLE SMART MANIPULATORS

ABSTRACT

In this thesis, a model extraction approach is proposed for vibration suppression. For this purpose, active (AC) and passive (PC) vibration controls of single-link flexible smart manipulators are studied theoretically and experimentally.

To shorten elapsed time in the finite element (FE) analyses, the mathematical models for the state-space (SS) method are extracted from ANSYS. First, the model extraction approach is tested with an analytical solution of a 2-DOF mass-spring system. After validating the results for a lumped parameter system, the SS models of the smart manipulators made of aluminum and composite materials are studied. The SS analyses are performed in MATLAB. The SS results are validated with the open and closed-loop FE simulations performed in ANSYS. The AC and PC are used with the PD and full-state feedback (FSF) controls while the PC can be only implemented to the smart manipulators. The parameters of motion profiles used in the PC are determined based on modal analysis while control design techniques are used to tune the control gains used in the AC. As the closed-loop SS analysis requires a mathematical model, the gains of FSF control are designed with different design criteria based on the settling time, overshoot, and rise time.

Experiments are performed with LABVIEW codes to verify the PC and AC results. Experimental modal analysis is done using a chirp signal. Experimental and theoretical residual vibration amplitudes are successfully reduced. The effect of motion time, angular displacement, and payload on the dynamic performance of the smart manipulators are investigated.

Keywords: Flexible manipulator, piezoelectric, active and passive vibration control, state-space model extraction

TEK EKSENLİ ESNEK AKILLI MANİPÜLATÖRLERİN AKTİF VE PASİF TİTREŞİM KONTROLÜ

ÖZ

Bu tezde, titreşimlerin azaltılması için bir model çıkarma yaklaşımı önerilmiştir. Bu amaçla, tek eksenli esnek akıllı manipülatörlerin aktif (AC) ve pasif (PC) titreşim kontrolleri teorik ve deneysel olarak incelenmiştir.

Sonlu eleman (FE) analizlerinde, geçen süreyi kısaltmak için, durum uzayı (SS) yöntemi için matematiksel modeller ANSYS den çıkarılır. İlk olarak, model çıkarma yaklaşımı, 2-DOF kütle yay sisteminin analitik bir çözümü ile test edilmiştir. Topaklanmış bir kütle sistemi için sonuçlar doğrulandıktan sonra, alüminyum ve kompozit malzemelerden yapılmış akıllı manipülatörlerin SS modelleri incelenmiştir. SS analizleri, MATLAB da gerçekleştirilir. SS sonuçları, ANSYS de gerçekleştirilen açık ve kapalı devre FE simülasyonları ile doğrulanır. AC ve PC, PD ve tam durum geri besleme (FSF) kontrolleriyle kullanılırken, PC yalnızca akıllı manipülatörlere uygulanabilir. PC de kullanılan hareket profillerinin parametreleri, modal analize göre belirlenirken, AC de kullanılan kontrol kazançlarını ayarlamak için kontrol tasarım teknikleri kullanılmıştır. Kapalı devre SS analizi matematiksel bir model gerektirdiğinden, FSF kontrolünün kazançları düzenli rejim süresi, aşma ve yükselme süresine bağlı olarak farklı tasarım kriterleriyle tasarlanmıştır.

PC ve AC sonuçlarını doğrulamak için LABVIEW kodlarıyla deneyler yapılmıştır. Deneysel modal analiz bir sinüsoidal sinyali kullanılarak gerçekleştirilmiştir. Deneysel ve teorik artık titreşim genlikleri başarıyla azaltılmıştır. Hareket süresinin, açılma ve değişim süresinin ve yükün akıllı manipülatörlerin dinamik performansı üzerindeki etkisi araştırılmıştır.

Anahtar kelimeler: Esnek manipülatör, piezoelektrik, aktif ve pasif titreşim kontrolü, durum uzayı model çıkarma

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CHAPTER ONE

INTRODUCTION AND LITERATURE REVIEW

1.1 Introduction

The manipulators are preferred due to the raise in productivity and help people in harmful works. They are made rigid from massive and stiff materials in order to carry out high precision on the endpoint motion. Therefore, these manipulators have more power consumption, higher mass to payload ratios, and limited operation speed. Unlike the rigid manipulators, the flexible manipulators (FM) offer many advantages such as requiring smaller actuators, less energy consumption, faster system response, less cost and mass (Rigatos & Busawon, 2018; Tokhi & Azad, 2017). For this reason, the usage of the FM has recently increased in industrial applications such as space exploration, robotic crane systems, and excavators, in mining and construction applications as described by Rigatos & Busawon (2018); Tokhi & Azad (2017); Xu & Yuan (2017); He et al. (2017); Gao & Wang (2003). The flexibility of the FM based on their nature is quite complex and difficult to control the residual vibrations occurring at the tip point. As a consequence, the dynamic modeling of the FM is considered as a critical step to perform better performance of controller design. The control techniques are applied to flexible manipulators for the position accuracy and the elimination of residual vibrations at the tip point (Yang et al., 2018; Yang & Tan, 2018; Zhang et al., 2017; Kiang et al., 2015; He et al., 2015; Rahimi & Nazemizadeh, 2014; Benosman & Le Vey, 2004).

A smart structure is a configuration consisting of the actuators and sensors placed at important points of a structure to create the moments and forces for the control system (Wagg et al., 2007). Thus, the components of a smart manipulator can be expressed as actuators, sensors and a control system, together with a manipulator (Preumont, 2011). Schematic block diagram of a smart manipulator is shown in Figure 1.1.

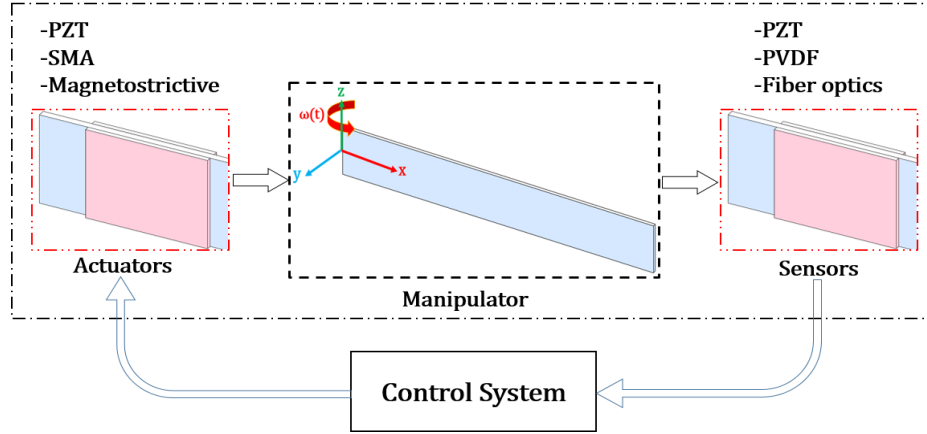


Figure 1.1 Schematic block diagram of a smart manipulator (Preumont, 2011)

The actuators and sensors can be used in vibration suppression. The sensors are selected from the PZT materials, Polyvinylidene Fluoride sensors (PVDF), and fiber optic materials while Shape Memory Alloys (SMA), the PZT and Magnetostrictive materials are used as the actuators in the control system.

The active and passive control methods are used to minimize the residual vibrations of the FM. For the active control (AC) method, the smart structures occurred from piezoelectric (PZT) actuators are utilized. The PZT actuator is used as an actuator in the AC while there is no need for an external effect in the PC method. In this chapter, the literature on vibration control and modeling of single-link FM are summarized. Numerical and experimental studies on the active and passive vibration controls of flexible smart manipulators are summarized. Also, the originality and organization of the thesis are presented.

1.1.1 Modeling and Simulation Methods of Flexible Manipulators

As seen from Figure 1.2, there is a rising influence in research studies on the mathematical modeling and vibration control of the FM.

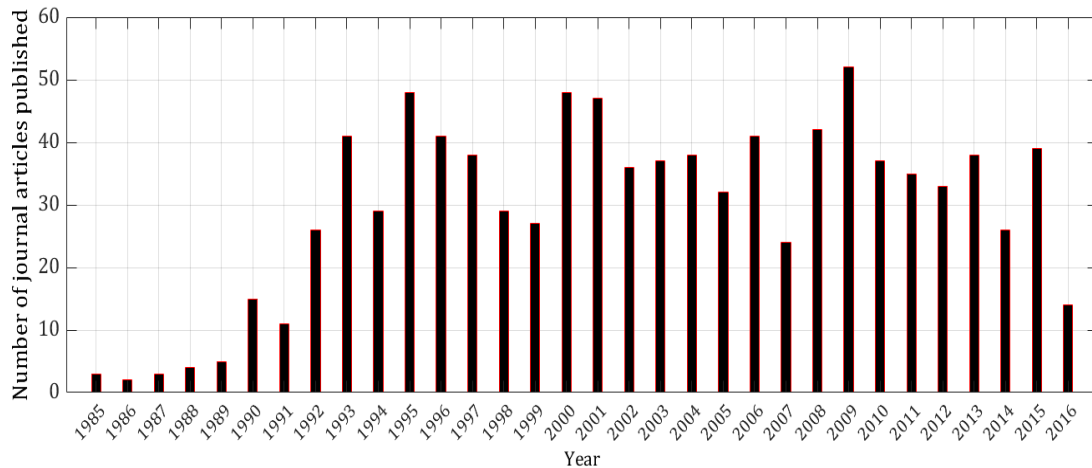


Figure 1.2 Published articles in journals until mid 2016 (Tokhi & Azad, 2017)

The modeling of the FM with a suitable mathematical model is important for the selected control method to eliminate the residual vibrations. Therefore, the mathematical modeling can be considered with various methods such as finite element (FE) model, modal expansion, Euler-Newton Equation, Lagrange Equation and frequency domain techniques. Especially, FE method is used extensively to model the FM as described by Yavuz et al. (2020); Yavuz (2019); Garcia-Perez et al. (2019); Sun et al. (2018); Karagülle et al. (2017); Gilardi et al. (2009); Piras et al. (2005); Tokhi et al. (1999).

The flexibility takes an important role in modeling and vibration control. Dynamic modeling of the FM is presented by Dwivedy & Eberhard (2006) in a research study with a literature review. Also, the modeling, sensors and smart control techniques for the applications of the FM are recently summarized by Sayahkarajy et al. (2016); Alandoli et al. (2016); Kiang et al. (2015); Rahimi & Nazemizadeh (2014).

The first step in order to improve the control performance of FM is creating an accurate dynamic model. Dynamic modeling can be realized with the finite element methods (FEM) and assumed modes (AMM) as well as finite-dimensional models. Briefly, the studies related to the modeling of the FM are summarized with the main methods such as the FEM, AMM, and lumped parameter (LPM) method (Zhang et al., 2018; Wu & Deng, 2011; Dwivedy & Eberhard, 2006).

The AMM is widely used in the literature. Studies presented by Gao et al. (2019); Kim (2015); He et al. (2017); Dwivedy & Eberhard (2006); Tso et al. (2003) are used the mathematical model of the FM using the Lagrange principle and AMM. Also, a similar study is reported utilizing the Newton-Euler Expression in the single-link FM by Rakhsha & Goldenberg (1985). To obtain the equations of motion, Krishnamurthy et al. (1990) which presented the FM produced from the composite materials studied an extended Hamilton's principle with unconstrained and constrained vibration modes. Also, using the same approach, the dynamic equation of the FM is studied with a prismatic joint as opposed to the revolute joint by Theodore & Ghosal (1997). The dynamic behavior of a single-link FM is presented by the deflection of tip position using the AMM (Sun et al., 2004). In the LPM, Zhang et al. (2018, 2017); He et al. (2017); Kim (2015); Wu & Deng (2011); Khalil & Gautier (2000) studied with a lumped model to model and analyze the displacement at the tip point of single-link FM. The lumped-parameter modeling is actually an easy method in order to separate the flexible elements in the FM (He et al., 2017; Kim, 2015). Therefore, it is often used to model FM in previous studies. With the same principle, using multi-layered PZT materials to control the vibrations of FM is developed by Tzou et al. (1990). Also, Tzou (1991) presented the distributed modal identification to find the mathematical model of FM for vibration control. Structural modeling of FM is developed with fiber-reinforced composite material by Saravanos & Lamancusa (1990). A computationally efficient modeling method is presented for the dynamics analysis of flexible manipulators (Zhang et al., 2018). Also, partial differential equation model is chosen to design the dynamic characteristics of flexible structure accurately (He et al., 2018; Zhang & Liu, 2013).

The FEM is widely used to analyze the dynamic behavior of different engineering structures for example beams, plates, manipulators (Zhang et al., 2018; Wu & Deng, 2011). Khulief developed an algorithm of system dynamics in combination with modal reduction schemes for a double span beam (Khulief, 2001). Korayem et al. introduced a comparative assumed mode analysis and the FEM with more suitable principles to computing the analysis of dynamic equations of the FM in Korayem et

al. (2010). Besides, the global mode method is presented to analyze the dynamic model of single-link FM in Wei et al. (2017). Gurses et al. (2009) considered the FEM to model a single-link FM with a rotating hub and PZT element in order to use as the actuator in vibration control. The dynamic equations for the rigid and flexible links are developed by Nagarajan & Turcic (1990). The Lagrange equation and FEM are used to advance the equation of motion of the FM. Mathematical models of a single-link FM using the finite element principle are developed by Martins et al. (2003); Martins et al. (2002); Tokhi et al. (2000) and then, compared the experimental natural frequencies with the FE results. Chung & Yoo (2002) studied the dynamic behavior of the revolving cantilever beam with FEM. Dubay et al. (2014) proposed a FE to reduce the vibration of smart FM using the model predictive control method. By Theodore & Ghosal (1995), a comparison of the AMM and FEM is presented to investigate the effect of modeling method in the FM.

The FM can be produced with composite materials because of the requests such as upward of payload/weight rate and lighter mass with more robot arms. Sun & Huang (2001) obtained an analytic formulation to model the laminated beams with the PZT sensors and actuators. A shear deformation theory is offered, formulated, and validated for a variety of numerical examples of laminated composite plates by Sahoo & Singh (2013). For the composite plates, the free vibration and static analyses are reported the use of a multiquadrics discretization and layerwise deformation method by Roque et al. (2013).

The composite structures such as shell, beam and plate can be modeled by using the FEM. Lam et al. studied the FEM based on the classical laminated theory for a composite plate using the PZT materials (Lam et al., 1997). Reddy (1999) presented Navier solution and FEM used the shear deformation and classical plate theories to analyze the composite beams placed the sensor and actuator. Different lay-ups for composite box structures are studied with the Extended Galerkin's Method and natural frequencies are presented by Qin & Librescu (2002). Mitra et al. (2004b) represented the composite thin wall beam element with open or close contour. Composite beam element is developed to formulate the effects of transverse shear

deformation, restrained warping, elastic coupling related with thin wall composite box structure, free vibration analyses for various beam configuration are performed. In Mitra et al. (2004a), Mitra et al. examined a smart composite box beam with the FEM using a first-order shear deformation principle. In Vo & Lee (2008), the vibration analysis of a thin-walled composite box structure is reported with numerical analyses of influences of boundary conditions, modulus ratio, and fiber angle onto the vibration mode shapes and frequencies.

For the rigid and flexible complex structures, the dynamic equations can be readily acquired from the FEM with inputs and outputs of the system in ANSYS. Firstly, the FE model of the system is created with a numerical approach such as signal analysis and FE Analysis (FEA). Then, the system matrices of structures can be extracted by using the inputs and outputs of the system. Khot & Yelve (2011) studied the extraction of a system model of the 2-DOF mechanical system by using ANSYS and MATLAB. The authors demonstrated the SS model in principle coordinate system obtained from modal analysis in ANSYS. With a similar method, the reduced and full order models of a cantilever beam are obtained from modal analysis results of its FE model in ANSYS (Khot et al., 2012). They offered the modal and transient analysis of reduced and full models with the matched results, closely and then, designed the PID controller with output feedback to reduce the vibrations in SIMULINK. Also, Gülbahçe & Çelik (2018) proposed positive position feedback and tuner-based PID controllers to control the smart structure by using the system matrices obtained from the FE model. Two different harmonic inputs are applied to the mathematical models to simulate the dynamic behavior of the proposed system.

1.1.2 Control Issues of Flexible Manipulators

Vibration control of the FM is generally classified into two categories as active or passive techniques. The PC techniques do not need additional hardware or equipment in practice. However, the AC methods require additional equipment in the closed-loop control. Active and passive vibration controls are important to suppress

the vibrations of the FM. Literature survey of single-link FM formed of composite and metal materials is summarized below.

In the PC, modified input shaping is a vibration and motion control techniques to suppress the undesired vibrations at the endpoint of the FM. Schemes of the input shaping method can be improved to change the motion input of the system without a feedback signal. The input shaping is not affected by vibration modes. Input shaping techniques are used to generate trajectory for the vibration suppression (Xu et al., 2019; Cao et al., 2018; Abe & Hashimoto, 2015; Cole & Wongratanaphisan, 2011; Hu et al., 2008). For input shaping method, Gürleyük & Cinal (2007) presented three-impulse sequence. Also, The optimization of the minimum settling time and energy consumption of a flexible manipulator is studied on the control method (Phan et al., 2012). Singhose (2009) presented the input shaping techniques to reduce the vibrations of flexible structures and command shaping application examples are reviewed. An appropriate trajectory planning can effectively reduce residual vibration in Meng et al. (2018); Yavuz et al. (2016). Singer & Seering (1990) studied the input shaping approach depend on postponing the command shaping using half of the period. Magee & Book (1994, 1995) reported that the input shaping principle is applied into a feedback loop with an effective pre-filter and then, proposed a filter design with the feedback of a joint PD controller. The control algorithm for the closed-loop can be used with an input shaping method to suppress the residual vibrations of the FM. The input shaping techniques are not affected by the disturbances due to be open-loop controller. This technique is applied to the FM with a feedback control by Kilian et al. (2013) and then, discussed the effect of open and closed-loop controllers.

The AC and PC techniques in metal structures have been studied in the literature. An AC strategy is designed in Malgaca et al. (2017); Gurses et al. (2009) to reduce the amplitudes of vibrations at the tip of a flexible smart manipulator. Gurses et al. (2009) studied a consubstantiated design of PD-based motion control with linear and angular velocity feedback and a pair of PZTs placed to the surface of a single-link FM to minimize the vibrations. The authors developed an original fiber-optic bending

sensor array named “ShapeTape”. The PID control algorithm is designed via the PZT patches in ANSYS. Tip position control is achieved by Lu et al. (2018) using the optimal placement criterion in Gurses et al. (2009). The composite controller consists of optimal placement and modal H2 norm is utilized for the AC of a FM. Choi et al. (2013) presented the position control of a robot arm with a PZT actuator experimentally. A quantitative feedback theory is designed to enhance vibration control. The output-based filter is proposed to reduce the residual vibrations of a manipulator. The effectiveness of combining the input shaping and feedback control methods are presented by Newman & Vaughan (2018) for energy-efficient point-to-point motion while simultaneously minimizing command-induced vibration by Tahir et al. (2017). Mohamed et al. (2006); Mohamed & Tokhi (2004) developed the strategies of feed-forward controller in order to reduce the vibrations using the input shaping method based on band-stop and low-pass filtering. In other works of the author, torque input to analyze the dynamic behavior is used as the bang-bang type and implemented to the command input shaping method to decrease the unwanted vibrations of the FM (Tokhi & Mohammed, 2001; Tokhi et al., 1999).

The feedback signals for the FM are measured with the displacement, acceleration and strain signals. In closed-loop controller, PZT material can be commonly used as an actuator. The strain feedback control is one of the most common methods to sense the deflection due to be handy and cheap (Bascetta & Rocco, 2006). Strain signal is utilized as feedback for the closed-loop control of the FM in order to estimate the vibrations. Strain results are verified with camera data using a Kalman Filter experimentally (Bascetta & Rocco, 2006). A Full-bridge strain gauge is used to gage the elastic deformation along the FM by Neto et al. (2010). Then, the amplified feedback signal is utilized to control the vibrations and output voltage applied to the servo motor through the power amplifier. A similar sensor is reported by Tumari et al. (2011) for its use as feedback for PID control in a FM. Direct strain feedback results are proved with experiments. Bascetta & Rocco (2006) presented two different controllers used with feedback signals acquired from both strain gauge and camera. For the repetitive control of the FM, Feliu et al. (2005) studied feedback signals

acquired from motor torque and angle calculated with a strain gauge. Also, a strain gauge is used to detect the position and force signals of the FM (Gu & Piedboeuf, 2003). The accuracy of the proposed technique is confirmed with two force sensors and a vision system.

Generally, PZT materials can be used as a sensor and an actuator in the closed-loop control. PZT material generates an electric field when distorted the form of its or receives the deformation when exposed to the electric field. Sun et al. (2004) explained an approximation for the usage of smart material to examine the stability and damping of vibration control, specifically, in control of a single-link FM. They studied a new method with an assembled scheme of PD feedback. The simulation and experimental vibration responses are confirmed with theoretical estimations. Lou et al. (2015) presented the hybrid control algorithm for rotating and vibration minimization of a FM with PZT materials. It contains a PD controller to carry out the motion control, and an effective multi-mode positive position feedback (EMPPF) control to reduce the multi-mode vibrations. Then, the control performance of the suggested strategy is represented by experiments and simulations. Elliott et al. (2014) aimed to provide the control actions to reduce residual vibrations by mounting PZT actuators in bimorph configurations. Molter et al. (2010) presented a control strategy for the FM using PZT actuators placed on prismatic links. The dynamic equation of manipulator can be presented in a closed-form due to the Lagrange principles. They used the PZT actuators and motor torque to control the residual vibrations. The efficiency of control design and advised optimization is verified with numerical tests.

The control method for the FM aims the keeping rotation angle at the desired position and to minimize the vibrations of the endpoint. Akyüz et al. (2011) designed state feedback and PID control schemes to reduce the vibrations at tip of the FM. Hassan et al. (2007) developed the Model Predictive Control (MPC) principle using PZT actuators to reduce the vibrations of a FM having multiple inputs and outputs. Then, they investigated the composition of strain gauge and PZT actuators according to controller efficiency in the vibration control. Using a similar method, the prediction ability of MPC can be very efficient for vibration control in a quick dynamic system.

Boscariol et al. (2009) offered an MPC design to control the vibration and position of a single-link FM. Kim et al. (2013) showed a non-collocated controller method with an accelerometer sensor and servo motor at the tip point of the FM. A nonlinear State-Dependent Riccati Equation (SDRE) is developed by Shawky et al. (2013) to control the endpoint position of a FM with payload at the tip. Robust nonlinear control is applied to the single-link FM to provide stability and to minimize the influences of vagueness with trajectory control by Khorasgani et al. (2011). Further, a sliding mode technique with two nonlinear controllers (Fareh et al., 2009), the pole placement procedure and linear quadratic regulator (LQR) controller (Saini et al., 2012) are established to decrease the vibrations at the tip and to increase the stability of a single-link FM. Singular perturbation theory is used to accomplish the tracking signal and vibration suppression (Xu, 2017).

The AC of composite structures is developed using PZT actuators and sensors. The suppression of vibrations of smart composite beams is performed with the analytical solution for the AC given by Sun & Huang (2001). Ma & Ghasemi-Nejhad (2008) investigated the vibration control of a composite manipulator with two PZT patches. The dynamic analysis of a composite manipulator is done. An independent modal space adaptive control and adaptive fuzzy logic control strategies for the composite manipulator are suggested for the active position and motion control. Phan et al. (2009) tuned the control method to optimal parameters based on the controlled initial maximum vibration responses with a genetic algorithm for the flexible composite manipulator. The optimal controller parameters on account of positive position feedback (PPF) used in a real-time control algorithm are determined with the objective functions. Single-link flexible composite manipulator with [0/90] and [45/-45] lay-ups are performed to reduce the end-point vibrations (Yavuz et al., 2016). Trapezoidal and triangular motion profiles are examined by creating cases for different motion times and stopping positions.

Yavuz analyzed a composite manipulator with the orientation of [0/90] lay-up using ANSYS and the amplitudes of residual vibrations are suppressed with a harmonic motion excitation (Yavuz, 2019). A composite controller (Xu, 2017), which

consists of a fuzzy sliding mode and linear quadratic regulator controllers, is designed by Lu et al. (2016). The linear and angular velocity feedback controllers, the positive position feedback and multi-mode controllers and the PD controller with the Lyapunov approach are the other examples of composite control method (Lou et al., 2017; Kumar, 2013). A neural network algorithm for faster residual vibration suppression comparing with the classical PD controller is proposed in Qiu & Zhang (2019).

The experimental and theoretical studies of vibration analysis for composite beams with rectangular cross-section is presented by utilizing distributed, surface fixtured PZT actuators by Mitra et al. (2004a). The dynamic behavior of laminated composite structure including the PZT actuators and sensors for the AC is conducted by the FEM given by Ramkumar et al. (2009).

Detail modeling and vibration control with the experiments on two and multi-link FM are reviewed by Qiu et al. (2019); Newman & Vaughan (2018). The studies for two and multi-link FM are categorized as the AMM, FEM, and LPM by Lochan et al. (2016). The researchers have studied different control methods such as the classical control technique by Choi et al. (2013); Kim (2015), robust control, intelligent control and artificial neural network by Castri & Messina (2012). Ge et al. (2001) presented an integral sliding mode controller with PZT actuators to accomplish the external forces and parametric changes to examine the angular displacement. The influence of the offered controller scheme is given by the simulations. Malzahn et al. (2011) planned a block diagram of robust controller united with the feed-forward input shaping and linear control for a 3-DOF FM expose to deflections due to gravity. The combined method demonstrated its capability to reduce the vibration and accomplish the end-effect positioning.

1.1.3 Contribution/Originality of the Thesis

The primary motivation in this thesis is to reduce residual vibrations by applying the active and passive control methods to single-link flexible smart manipulators.

One of the contributions of this thesis is to propose the modeling method for the vibration control of the smart manipulators as the FE simulations in transient analysis take a long time and a control system may not always be designed.

For this purpose, the SS models of the different smart manipulators made of metal and composite materials are obtained by using the FE models created in ANSYS. Then, mathematical models are used to design the control system specifically for the AC of smart manipulators. Active and passive vibration control analyses are performed in MATLAB by using the extracted SS models.

In addition, the other contribution to the thesis is the realization of the active and passive vibration control simulations in ANSYS. Closed-loop simulations in ANSYS are successfully performed with the integration of PD control actions into the FE transient solutions.

Another contribution of the thesis is the implementation of a hybrid vibration control using active and passive control methods.

As a result, comparison of the theoretical and experimental results is presented in this thesis.

1.2 Organization of the Thesis

This thesis consists of eight chapters and an appendix. Chapter 1 gives an introduction, literature survey, aims, and organization of the thesis. Vibration control methods of the single-link FM is given with the active and passive controls. For the smart manipulators, previous reference studies on vibration control are presented.

Chapter 2 presents vibration control of a two degree of freedom lumped parameter system with the analytical and SS analysis based on modal analysis in the FE method. The uncontrolled and controlled time responses are given with two different methods. The analysis of the AC is studied with the analytical solution (AS) by using the equation of motions of the mechanical system. For the SS analysis, the FE model of

the system is created using ANSYS. Inputs and outputs are defined in the FE model. Then, the SS matrices of the system based on modal analysis are extracted from ANSYS. The impulse and step responses are obtained in MATLAB. The SS analysis is verified with the analytical results.

In Chapter 3, the FE models of the smart manipulators are created for the simulation and analysis in ANSYS. For the simulations, active and passive control methods are incorporated into the FE model in ANSYS to conduct open and closed-loop control analyses. For the SS analysis, the inputs and outputs of the smart manipulators are defined in ANSYS to extract the system matrices based on modal analysis. The mathematical models are obtained using the model extraction method from ANSYS. The SS matrices are found for four different smart manipulators consisting of aluminum, composite and composite box links with the PZT actuators.

Chapter 4 examines the analysis of PC for the smart manipulators. Simulations are performed in ANSYS while the SS analysis is conducted in MATLAB. The mathematical models obtained in Chapter 3 are used for the SS analysis. The modal analysis of the smart manipulators is required to generate a motion profile. The modal analysis is done. The natural frequencies of reference studies are compared with the present works to confirm the FE models studied in this thesis. Two different motion profiles such as triangular and trapezoidal velocities are used to move the smart manipulators. The time parameters of the motion profiles based on the natural frequency of the smart manipulators are calculated. The amplitudes of residual vibrations can be reduced with selected deceleration times in the PC. For the PC, the PZT material is not used as an actuator. The simulation results are obtained to verify the SS responses. Root means square (RMS) values of the residual vibrations at the tip are evaluated. The simulation and SS results are presented.

Chapter 5 presents the closed-loop analyses in which the simulations are performed in ANSYS while the SS analysis is performed in MATLAB for the AC. The PD and FSF control methods are studied. In the closed-loop simulations, the PD control action is embedded into the FE solution in ANSYS. The smart manipulators moves with the

motion profiles used in Chapter 4. Then, the AC is applied to the smart manipulators when the motion stops. The AC results are presented for the PD control with the various controller gains. The simulation and SS results are compared in the AC. After compared the simulation and SS results, the PD and FSF controller gains are found using the design techniques. The optimized controller gains are obtained for the PD and FSF controls. Two design criteria are studied for the FSF control. The RMS values and reduction ratios for the residual vibrations are graphically shown to evaluate the effectiveness of implementing together with the AC on the PC.

In Chapter 6, the experiments are done to confirm the SS results. The experimental system is introduced for the smart manipulators. Firstly, the experimental modal analysis is studied by utilizing the chirp signals for the smart manipulators. The chirp signal is applied to the PZT actuators. Then, displacement signals at the tip point are measured with the laser displacement sensor and found the natural frequencies by taking Fast Fourier Transform (FFT) of the time responses in MATLAB. The experimental and theoretical natural frequencies are compared. The experimental vibration responses are obtained to verify the SS results for both the AC and PC.

Chapter 7 compares the experimental and SS analysis results for the PC and AC. Firstly, the PC results are presented for the triangular and trapezoidal motion profiles. The tables includes the RMS and error values according to the deceleration time. Then, the AC results are presented for the PD controller using the displacement feedback (DF). The tables give a comparison of the experimental and SS analysis results for the PD control.

In Chapter 8, the interpretation of main conclusions and suggestions for future works in active and passive vibration controls of the smart manipulators are explained.

Appendix gives the front panel and block diagram of LABVIEW code developed for the experimental studies in the thesis.

CHAPTER TWO

STATE-SPACE MODEL EXTRACTION FOR A LUMPED PARAMETER SYSTEM

2.1 Introduction

In vibration control, the dynamic modeling of smart structures is an important stage to compose a successful control method and to control the vibrations. The mathematical model of the smart structures can be obtained by using the FE method. The FE simulations can be applied to more complex structures while analytical methods are generally utilized to model simple structures. However, the dynamic analysis with the FE simulations takes more time. In this thesis, in order to decrease the time in the dynamic analysis with the FE simulations, the SS analysis, in which the system matrices are extracted from ANSYS is used. The times consumed in the dynamic analysis with the FE simulations are significantly reduced with this method. Then, the SS analyses of the smart manipulators are performed quickly for the vibration control in MATLAB. In order to test the effectiveness of the SS analysis, initially the analytical solution of a 2-DOF mass-spring-damper (MSD) is compared with the SS results in this chapter.

In this chapter, the 2-DOF MSD system is considered to analyze open-loop and closed-loop vibration responses and to verify the SS analysis with the AS. The 2-DOF MSD system is described as lumped parameter system (LPS) in this thesis. Firstly, the AS of the LPS is studied. The equation of motion of the system is established for the AS. The coupled second-order differential equations are obtained. Then, in order to compare the system matrices extracted from ANSYS, the uncoupled differential equations are found. In order to find the uncoupled differential equations, the eigenvectors and eigenvalues are determined with modal analysis. The dynamic equation is stated as a linear combination of normal vibration modes to normalize with respect to the mass matrix. Then, these equations, which give the AS of the LPS, are defined in nodal coordinates. Then, the SS analysis for the LPS is performed to

confirm the AS. The FE model of LPS is created using ANSYS. Modal analysis is carried out to obtain the undamped natural frequencies. The inputs and outputs are defined to extract the SS matrices based on modal analysis. The SS model requires modal analysis to extract the system matrices from ANSYS. So, the system matrices between inputs and outputs are found. The uncontrolled and controlled time responses of unit step and impulse inputs are simulated with the SS analysis in MATLAB.

Open and closed-loop vibration responses to the step and impulse inputs are successfully obtained in MATLAB. It is observed that the uncontrolled and controlled time responses are matched well with the AS and SS analysis.

2.2 Analytical Solution

Considering the LPS under this study is shown in Figure 2.1.

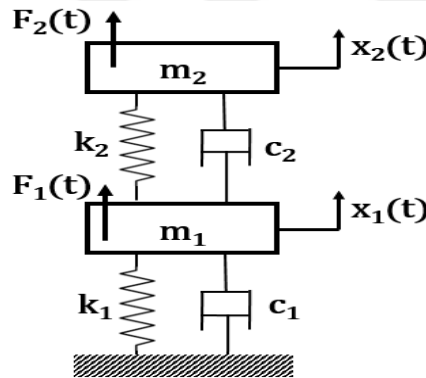


Figure 2.1 2-DOF mechanical system

where, the parameters m_1 , m_2 , k_1 , k_2 , c_1 and c_2 are the mass, stiffness and damping coefficients, respectively. The values of the system parameters are chosen as $m_1 = 1.2$ kg, $m_2 = 1$ kg, $k_1 = 350$ N/m, $k_2 = 300$ N/m, $c_1 = 4$ Ns/m and $c_2 = 2$ Ns/m. $x_1(t)$ and $x_2(t)$ are defined as the displacements of each mass. $F_1(t)$ and $F_2(t)$ are the forces. Therefore, the equations of motion for the LPS are given by

$$m_1 \ddot{x}_1 + (c_1 + c_2) \dot{x}_1 - c_2 \dot{x}_2 + (k_1 + k_2)x_1 - k_2 x_2 = f_1(t) \quad (2.1)$$

$$m_2 \ddot{x}_2 - c_2 \dot{x}_1 + c_2 \dot{x}_2 - k_2 x_1 + k_2 x_2 = f_2(t)$$

where, $x_1(t)$ and $x_2(t)$ are the general coordinates. The Rayleigh damping coefficients are used for the damping. The mass and stiffness damping coefficients are assumed to be $\alpha = 0.1364$ and $\beta = 0.01$. It is assumed that the damping coefficients are equal to zero for the AS. Therefore, the equation of motion written in matrix format is given in Equation 2.2.

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \begin{Bmatrix} \ddot{x}_1(t) \\ \ddot{x}_2(t) \end{Bmatrix} + \begin{bmatrix} c_1 + c_2 & -c_2 \\ -c_2 & c_2 \end{bmatrix} \begin{Bmatrix} \dot{x}_1(t) \\ \dot{x}_2(t) \end{Bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix} \begin{Bmatrix} x_1(t) \\ x_2(t) \end{Bmatrix} = \begin{Bmatrix} f_1(t) \\ f_2(t) \end{Bmatrix} \quad (2.2)$$

$$[M] \{\ddot{x}(t)\} + [C] \{\dot{x}(t)\} + [K] \{x(t)\} = \{F(t)\}$$

where, M, C, K and F(t) given in Equation 2.2 are the mass, damping, stiffness matrices and force vector, respectively. To find the natural frequencies (ω_i) of the system, the sum of all forces acting on the mass should be assumed to zero with $F(t) = 0$ an externally applied force given in Equation 2.2. The following eigenvalue equation is solved for the free-vibration eigenproblem associated with Equation 2.2.

$$|-\omega^2 M + K| = 0 \quad (2.3)$$

hence, the eigenvalue equation becomes as follow,

$$\begin{bmatrix} 650 - 1.2\omega^2 & -300 \\ -300 & 300 - \omega^2 \end{bmatrix} \begin{Bmatrix} x_1(t) \\ x_2(t) \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix} \quad (2.4)$$

and the characteristic polynomial equation is given by

$$\det \begin{bmatrix} 650 - 1.2\omega^2 & -300 \\ -300 & 300 - \omega^2 \end{bmatrix} = (650 - 1.2\omega^2)(300 - \omega^2) - 300 \times 300 = 0 \quad (2.5)$$

The characteristic polynomial equation is solved and the natural frequencies are found as

$$\omega_1 = 1.7543 \text{ Hz} \quad \omega_2 = 4.2711 \text{ Hz} \quad (2.6)$$

M , C , and K matrices are non-diagonal. As a result, the equation of motion is coupled. To uncouple this equation, the transformation matrix is used to transform the physical coordinates $\{x(t)\}$ into modal coordinates $\{\eta\}$. Firstly, the eigenvalue problem is worked out for the natural frequencies of the system. Then, the eigenvectors of the system, which are $X^{(1)}$ for first mode shape and $X^{(2)}$ for second mode shape, are solved to uncouple the equation of motion. Eigenvectors are given as follow

$$X^{(1)} = \begin{Bmatrix} X_1^{(1)} \\ X_2^{(1)} \end{Bmatrix} \quad X^{(2)} = \begin{Bmatrix} X_1^{(2)} \\ X_2^{(2)} \end{Bmatrix} \quad (2.7)$$

In order to find the eigenvectors of the system, motion of $x_1(t)$ and $x_2(t)$ general coordinates is taken as the harmonic motion of masses m_1 and m_2 at the same frequency ω and phase angle ϕ with the following equation

$$\begin{aligned} x_1(t) &= X_1 \cos(\omega t + \phi) \\ x_2(t) &= X_2 \cos(\omega t + \phi) \end{aligned} \quad (2.8)$$

where, X_1 , X_2 are maximum amplitudes of $x_1(t)$ and $x_2(t)$, and ϕ is the phase angle. Substituting Equation 2.8 into Equation 2.2. Then, the characteristic equation is solved. The roots of Equation 2.4 with undefined expressions are given by

$$\omega_{1,2}^2 = \frac{1}{2} \left\{ \frac{(k_1+k_2)m_2+k_2m_1}{m_1m_2} \right\} \pm \frac{1}{2} \left[\left\{ \frac{(k_1+k_2)m_2+k_2m_1}{m_1m_2} \right\}^2 - 4 \left\{ \frac{(k_1+k_2)(k_2)-k_2^2}{m_1m_2} \right\} \right]^{\frac{1}{2}} \quad (2.9)$$

In harmonic solution given in Equation 2.8, ω is equal to ω_1 and ω_2 given by Equation 2.9. It can be seen that the values of X_1 and X_2 based on the natural frequencies of the system. Also, the values of X_1 and X_2 are corresponded to $X_1^{(1)}$ and $X_2^{(1)}$ for ω_1 and $X_1^{(2)}$ and $X_2^{(2)}$ for ω_2 . Moreover, because the equation of motion is homogeneous, the values of $X_2^{(1)}$ and $X_2^{(2)}$ can be determined as $X_2^{(1)} = r_1 X_1^{(1)}$ and $X_2^{(2)} = r_2 X_1^{(2)}$ with the ratios r_1 and r_2 given in following equations

$$\begin{aligned}
r_1 &= \frac{X_2^{(1)}}{X_1^{(1)}} = \frac{-m_1\omega_1^2 + (k_1 + k_2)}{k_2} = \frac{k_2}{-m_2\omega_1^2 + (k_2)} \\
r_2 &= \frac{X_2^{(2)}}{X_1^{(2)}} = \frac{-m_1\omega_2^2 + (k_1 + k_2)}{k_2} = \frac{k_2}{-m_2\omega_2^2 + (k_2)}
\end{aligned} \tag{2.10}$$

where, the ratios r_1 and r_2 are identical as given in Equation 2.10. Therefore, the normal vibration modes corresponding to natural frequencies can be stated by

$$\begin{aligned}
X^{(1)} &= \begin{Bmatrix} X_1^{(1)} \\ X_2^{(1)} \end{Bmatrix} = \begin{Bmatrix} X_1^{(1)} \\ r_1 X_1^{(1)} \end{Bmatrix} = \begin{Bmatrix} -0.4984 \\ -0.8377 \end{Bmatrix} && \text{for first vibration mode} \\
X^{(2)} &= \begin{Bmatrix} X_1^{(2)} \\ X_2^{(2)} \end{Bmatrix} = \begin{Bmatrix} X_1^{(2)} \\ r_2 X_1^{(2)} \end{Bmatrix} = \begin{Bmatrix} -0.7647 \\ 0.5460 \end{Bmatrix} && \text{for second vibration mode}
\end{aligned} \tag{2.11}$$

where, $X_1^{(1)}$, $X_1^{(2)}$, $X_2^{(1)}$, and $X_2^{(2)}$ are calculated by using the initial conditions. The modal matrix for the system is determined as

$$[X_n] = \begin{bmatrix} X^{(1)} & X^{(2)} \end{bmatrix} \tag{2.12}$$

The eigenvectors $X^{(1)}$ and $X^{(2)}$ of the system are denoted the normal modes of vibration or modal vectors of the system. The normalized eigenvectors of the system are found as

$$[X_n] = \begin{bmatrix} -0.4984 & -0.7647 \\ -0.8377 & 0.5460 \end{bmatrix} \tag{2.13}$$

For the LPS, the solution of Equation 2.2 can be denoted as the linear combination of normal vibration modes as

$$x(t) = [X_n]\eta(t) \tag{2.14}$$

where, $\eta(t)$ is the displacement in general coordinates. The vibration modes are standardized as for the mass matrix. Therefore, the generalized mass and stiffness matrices associated with the normal modes are computed as

$$[M_p] = [X_n]^T [M] [X_n] \quad [M_p] = \begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix} \quad (2.15)$$

$$[C_p] = [X_n]^T [C] [X_n] \quad [C_p] = \begin{bmatrix} 1.3514 & 0 \\ 0 & 7.3381 \end{bmatrix} \quad (2.16)$$

$$[K_p] = [X_n]^T [K] [X_n] \quad [K_p] = \begin{bmatrix} 121.5 & 0 \\ 0 & 720.17 \end{bmatrix} \quad (2.17)$$

and force vector $F(t)$ for $f_1 = 0$ and $f_2 = 1$ are given by

$$[F_p] = [X_n]^T [F] \quad [F_p] = \begin{bmatrix} -0.8377 \\ 0.5460 \end{bmatrix} \quad (2.18)$$

where, $[F_p]$ is the force vector. Hence, the equation of motion in principal coordinates is written as

$$\begin{aligned} \ddot{x}_{p1} + 1.3514\dot{x}_{p1} + 121.5x_{p1} &= -0.8377 \\ \ddot{x}_{p2} + 7.3381\dot{x}_{p2} + 720.17x_{p2} &= 0.5460 \end{aligned} \quad (2.19)$$

The SS form of the system based on modal analysis may be constructed by using the eigenvectors and eigenvalues. The SS matrices of the system for two vibration modes are written in form of as

$$A = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -\omega_1^2 & 0 & -2\zeta_1\omega_1 & 0 \\ 0 & -\omega_2^2 & 0 & -2\zeta_2\omega_2 \end{bmatrix} \quad B = \begin{bmatrix} 0 \\ 0 \\ F_{p1} \\ F_{p2} \end{bmatrix} \quad (2.20)$$

$$C = \begin{bmatrix} [X_n]_{11} & [X_n]_{12} & 0 & 0 \\ [X_n]_{21} & [X_n]_{22} & 0 & 0 \end{bmatrix} \quad D = [0]$$

where, $[X_n]_{11}$, $[X_n]_{12}$, $[X_n]_{21}$ and $[X_n]_{22}$ are the parts of modal matrix regularized as for mass matrix $[M]$. These second-order uncoupled differential equations are modified

into the SS matrices of the LPS as the second-order differential equations. Therefore, the SS matrices for the AS are found as follow

$$A = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -121.5 & 0 & -1.3514 & 0 \\ 0 & -720.18 & 0 & -7.338 \end{bmatrix} \quad B = \begin{bmatrix} 0 \\ 0 \\ 0.8377 \\ 0.5460 \end{bmatrix} \quad (2.21)$$

$$C = \begin{bmatrix} -0.4984 & -0.7647 & 0 & 0 \\ -0.8377 & 0.5460 & 0 & 0 \end{bmatrix} \quad D = [0]$$

The system matrices are obtained with the AS. The SS analysis for the LPS is investigated with the model extraction method to verify the AS.

2.3 State-Space Model Extraction

The SS models of dynamic systems are commonly used in vibration control studies. It can be used to describe not only the linear system but also nonlinear systems. The SS model of a dynamic system is given with the following equations as

$$\begin{aligned} \dot{x} &= Ax + Bu(t) \\ y &= Cx \end{aligned} \quad (2.22)$$

where, $\dot{x}(t)$, rx1 (r rows by 1 column), is called the state vector, A (rxr) is the system matrix, B (rxc) (c columns by 1 row) is the input matrix, u (cx1) is the input vector, y (px1) is output, C (pxr) is the output matrix, respectively.

The mathematical model indicates the dynamic behavior of the system. The natural frequencies, damping ratios and vibration mode shapes of the system are basic parameters in a vibration system. The general approach for the SS model extraction used in this thesis is shown in Figure 2.2.

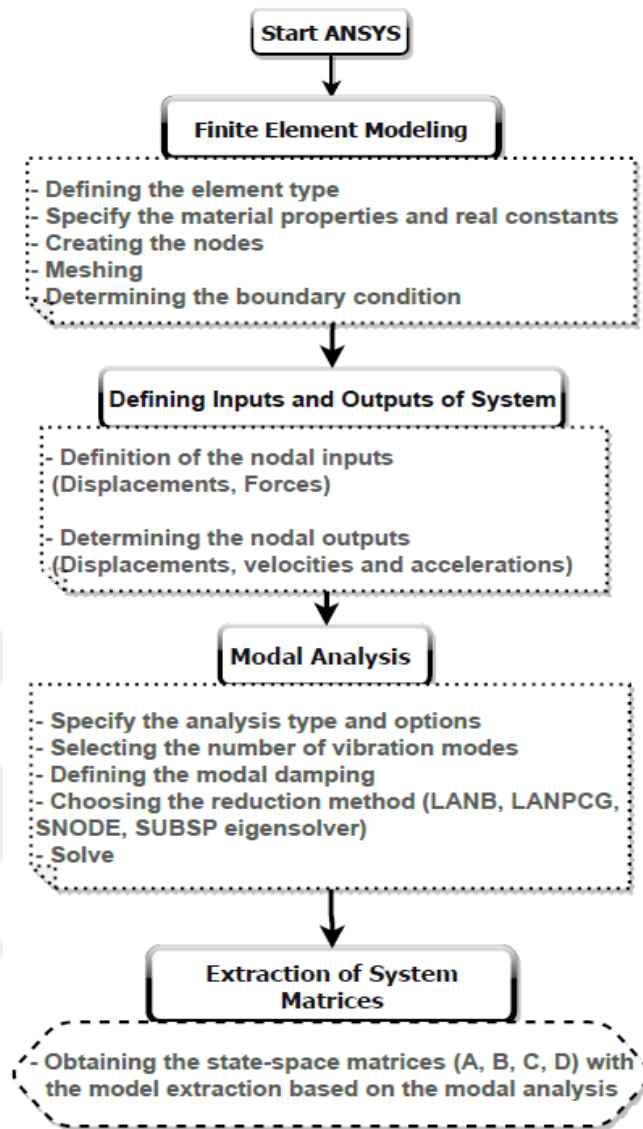


Figure 2.2 Principle procedure of SS model extraction

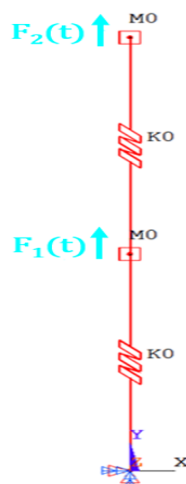


Figure 2.3 FE model

The first step for the model extraction is the modeling of the system in ANSYS. The input nodes are defined in the FE model. A similar technique is implemented to the output nodes of the system. The definition of inputs is all load vectors defined the structural DOF (1:ux, 2:uy, 3:uz, 4:rotx, 5:roty, 6:rotz) specifying the input nodes. As in the inputs, the outputs are determined in ANSYS. Additionally, the number of considered N vibration modes needs to be defined for the model reduction. Also, modal damping can be determined with Rayleigh's damping. Finally, the SS matrices are obtained with modal analysis after selecting the solver method.

The FE model of the LPS shown in Figure 2.3 is modeled with MASS21 elements and COMBIN14 element by using in ANSYS. MASS21 element is utilized for the masses m_1 and m_2 , and the COMBIN14 element is used for the stiffness and damping coefficients.

ANSYS macro code shown below are used in generating the FE model (ANSYS Inc., 2012).

Table 2.1 Ansys/APDL macro code for the modeling of the system

/prep7	real,2
et,1,mass21	e,2
et,2,combin14	type,2
r,1,,mb	real,3
r,2,,mt	e,2,3
r,3,k,c	real,4
r,4,kt,0	e,1,2
n,1,,,,	d,1,all,0
n,2,,0.5,	d,all,ux,0
n,3,,1,	d,all,uz,0
type,1	d,all,rotx,0
real,1	d,all,roty,0
e,3	d,all,rotz,0

The inputs are selected as the forces f_1 at node 1 and f_2 at node 2 while the outputs of the system are defined as the displacement $x_1(t)$ of mass m_1 at node 1 and displacement $x_2(t)$ of mass m_2 at node 2. The inputs and outputs define with the displacements and rotations at x, y, z- directions per node. The modal analysis is done to find the natural frequencies of the system. The natural frequencies of LPS is obtained as $f_{n1} = 1.7543$ Hz and $f_{n2} = 4.2711$ Hz. The time parameters for open-loop and closed-loop simulations are determined from the system frequencies. The time step Δt and steady-state time t_{ss} of the system adhere to the natural frequency. Therefore, the time step and steady-state time are calculated as $\Delta t = 1/f_{n1}/20 = 0.0285$ s, $t_{ss} = 8$ s, respectively.

The system matrices are obtained after performing modal analysis with the macro code in ANSYS below (ANSYS Inc., 2012).

Table 2.2 Ansys/APDL macro code to obtain State-Space Matrices

*dim,inputTab,,3,2	*dim,output,,1,2
*dim,inputLab,CHAR,3	output(1,1) = 3
inputTab(1,1) = 1	output(1,2) = 2
inputTab(1,2) = 2	*dim,outputLab,CHAR,1
INPUTLAB(1)='IN1'	outputLab(1) = 'Node3'
inputTab(2,1) = 2	
inputTab(2,2) = 2	
INPUTLAB(2)='IN2'	
inputTab(3,1) = 3	
inputTab(3,2) = 2	
INPUTLAB(3)='IN3'	

So, the SS model of the LPS is extracted. The SS matrices are obtained in principal coordinates for the first two vibration modes as follow

It can be seen that the system matrices are compatible with the AS and SS model extraction. The obtained SS matrices are used in open-loop and closed-loop analysis.

$$A = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -121.5 & 0 & -1.3514 & 0 \\ 0 & -720.18 & 0 & -7.338 \end{bmatrix} \quad B = \begin{bmatrix} 0 \\ 0 \\ 0.8377 \\ 0.5460 \end{bmatrix} \quad (2.23)$$

$$C = \begin{bmatrix} -0.4984 & -0.7647 & 0 & 0 \\ -0.8377 & 0.5460 & 0 & 0 \end{bmatrix}$$

2.4 Open-loop Analysis

Open-loop block diagram is given in Figure 2.4.

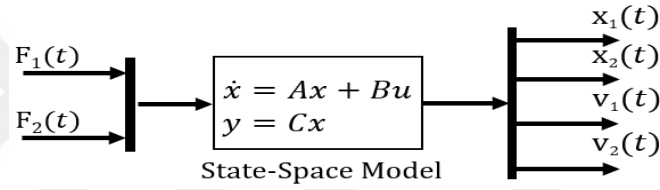


Figure 2.4 Open loop block diagram

The open-loop analysis are performed in MATLAB. The unit impulse and step inputs are used for the simulations. The forces $F_1(t)$ and $F_2(t)$ are applied to the system. Then, the uncontrolled vibration responses for $x_2(t)$ are shown in Figure 2.5 and Figure 2.6.

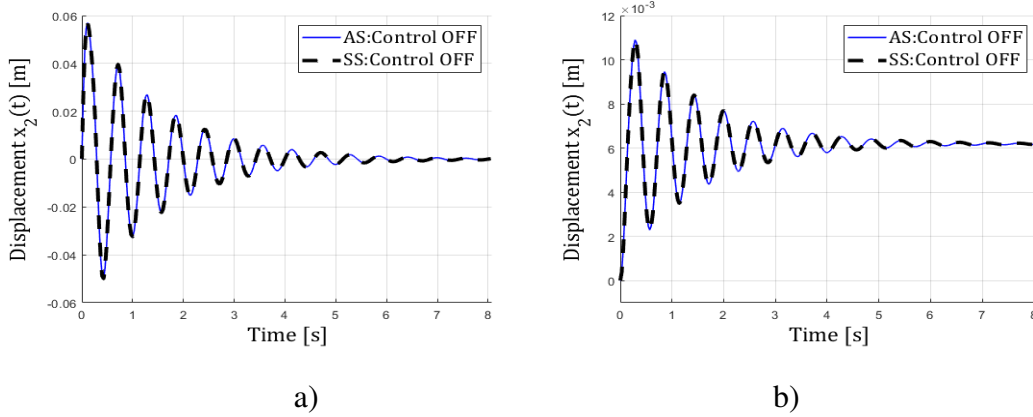


Figure 2.5 Open loop responses from $F_1(t)$ to $X_2(t)$, a) impulse, b) step

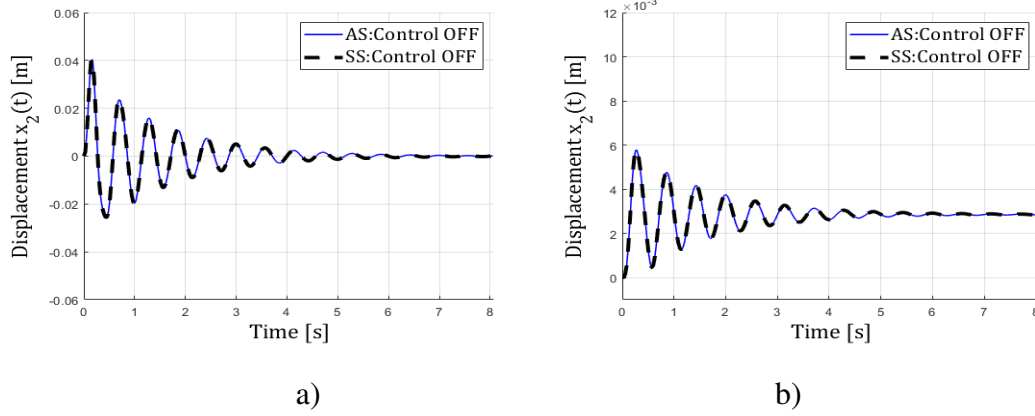


Figure 2.6 Open loop responses from $F_2(t)$ to $X_2(t)$, a) impulse, b) step

Figure 2.5 and Figure 2.6 compare the analytical and SS solutions. It can be seen that the results are in good agreement. Figure 2.5 shows open-loop responses for $X_2(t)$ to the input $F_1(t)$ while Figure 2.6 shows open-loop responses for $X_2(t)$ to the input $F_2(t)$.

2.5 Closed-loop Analysis

After obtaining the mathematical modeling of the system, the closed-loop control is studied in MATLAB. The closed-loop block diagram is shown in Figure 2.7.

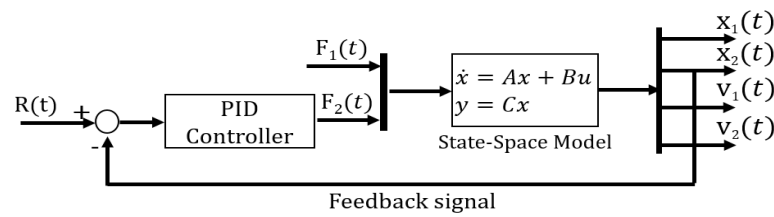


Figure 2.7 Closed-loop block diagram

where, $R(t)$ is the reference input taken as zero to eliminate the vibrations. $F_1(t)$ is selected as the disturbance input of 1 N. $F_2(t)$ is defined as the control force for the PID controller. K_p , K_i , and K_d are the proportional, integral, and derivative controller gains, respectively. $X_2(t)$ is selected as the feedback signal. The error value $e(t)$ is the input of PID controller. The actuating signal is applied to the SS model to reduce the

vibrations of $X_2(t)$. The controlled time responses for the inputs are obtained.

The closed-loop responses obtained with the AS and SS analysis for various controller gains are shown in Figure 2.8 and Figure 2.9.

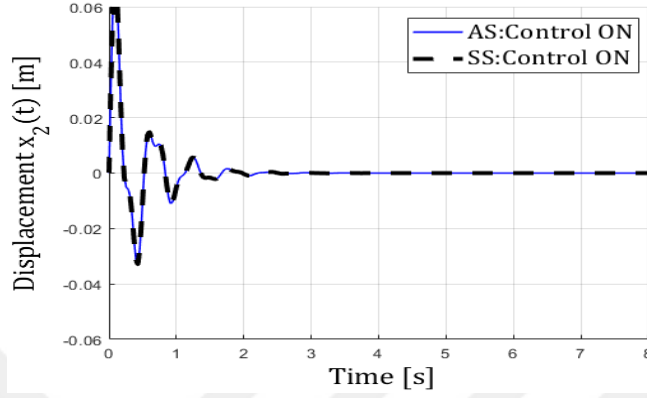


Figure 2.8 Impulse responses from $F_1(t)$ to $X_2(t)$ for controller gains, $K_p = 20$, $K_i = 0$, $K_d = 7$

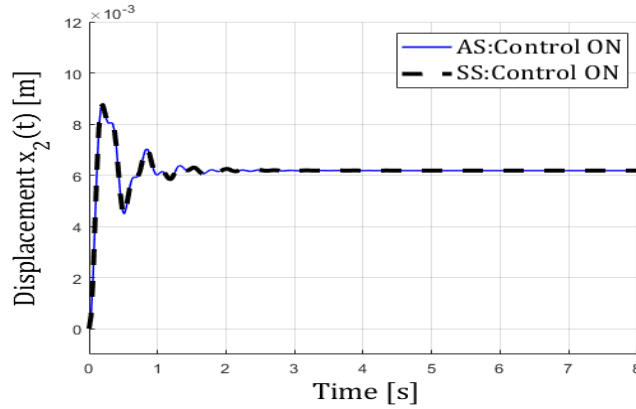


Figure 2.9 Step responses from $F_1(t)$ to $X_2(t)$ for controller gains, $K_p = 20$, $K_i = 4$, $K_d = 10$

It is observed from Figure 2.8 and Figure 2.9 that the AS and SS analysis results are matched well. To investigate the performance of PID controller, the uncontrolled and controlled vibration responses are given in Figure 2.10 and Figure 2.11.

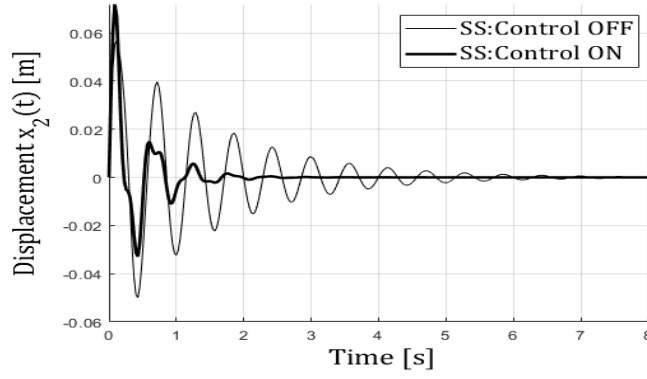


Figure 2.10 Impulse responses from $F_1(t)$ to $X_2(t)$ for controller gains, $K_p = 20$, $K_i = 0$, $K_d = 7$

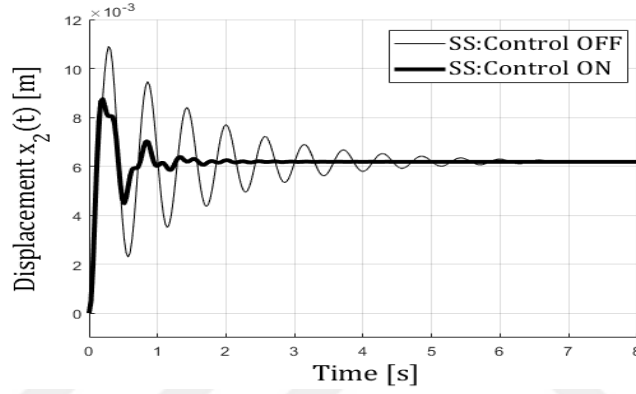


Figure 2.11 Step responses from $F_1(t)$ to $X_2(t)$ for controller gains, $K_p = 20$, $K_i = 4$, $K_d = 10$

Figure 2.10 and Figure 2.11 show open-loop and closed-loop vibration responses to the impulse and step inputs. Reductions in the vibration amplitudes and the settling time are an important criterion for the evaluation of vibration control. As seen from Figure 2.10 and Figure 2.11, it is observed that the amplitudes of controlled responses are successfully reduced in the closed-loop control. The settling times are decreased from 8 s to 2.5 s in the impulse response and from 8 s to 2.1 s in the step responses.

In this chapter, the SS model extraction is successfully achieved for a 2-DOF LPS. Then, the SS model extraction approach will be applied to the smart manipulators in the following chapters.

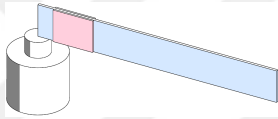
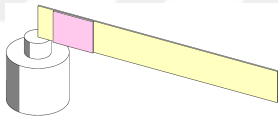
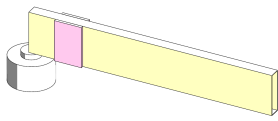
CHAPTER THREE

MODELING OF FLEXIBLE SMART MANIPULATORS

3.1 Introduction

In this chapter, the FE models of single-link flexible smart manipulators are created in ANSYS to obtain the SS models by using the model extraction approach. The configuration and dimension of the smart manipulators studied in this thesis are given in Table 3.1.

Table 3.1 Types of flexible smart manipulators

Manipulator	Sketch view	Orientation	Dimension, L x h x t
Smart manipulator (SM)		—	Aluminum link 450 x 20 x 1.5 mm ³ Two piezoelectric actuators 25 x 20 x 1 mm ³
Smart composite manipulators (1-SCM-[0/90]) (2-SCM-[45/-45])		[0/90] [45/-45]	Composite link 350 x 20 x 0.85 mm ³ Single piezoelectric actuator 25 x 20 x 1 mm ³
Smart composite box manipulator (SCBM)		[0/90/0/90]	Composite box link 955 x 80.1 x 1 mm ³ Two piezoelectric actuators 80.1 x 50 x 5 mm ³

As seen from Table 3.1, four smart manipulators made of an aluminum link, two composite epoxy-glass links, and a composite epoxy-glass box link with a rectangular cross-section are studied. The smart manipulators are modeled with the PZT materials used as an actuator to reduce the vibrations in the AC. The PZT actuators are placed on the surfaces of the smart manipulators at certain distances from the root.

For the FE analysis, the FE models are established in ANSYS. A revolute joint is created at the global axis to rotate the smart manipulators around the z-axis. For the SS analysis, in order to extract the SS models, the inputs are defined as the torque and control voltages while the outputs are determined as the acceleration, velocity and displacement at three directions (x, y, z) of the tip point. The SS models are obtained with the model extraction method from ANSYS.

After obtaining the mathematical models including the SS matrices, the AC and PC are performed to confirm the FE simulation results with the SS analysis in Chapter-4 and Chapter-5. In the following parts of the thesis, only the word "simulation" will be used for simulations performed in ANSYS.

3.2 Finite Element Models of Smart Manipulators

The FE models of the smart manipulators studied in the thesis are created by using ANSYS. Four different configurations with two different materials for the FE models are introduced in the following sections.

3.2.1 Smart Manipulator

The schematic view of a single-link flexible smart manipulator (SM) considered in this thesis is presented in Figure 3.1.

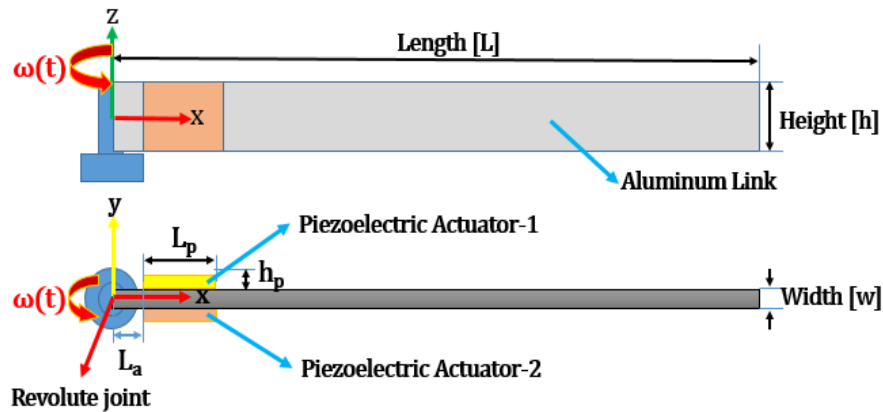


Figure 3.1 Schematic view

The SM consists of an aluminum link and two piezoelectric actuators. The actuators, which are made of PZT5-H, are located at 10 mm away from the root. The dimension of the aluminum link and actuators in the SM is given as $450 \times 20 \times 1.5 \text{ mm}^3$, $25 \times 20 \times 1 \text{ mm}^3$, respectively. Material properties are given in Table 3.2.

Table 3.2 Material properties of SM

Aluminum link	Piezoelectric material, PZT-5H
$\rho = 2700 \text{ kg/m}^3$	$\rho = 7500 \text{ kg/m}^3$
<i>Young's modulus (GPa)</i>	<i>Elastic stiffness matrix (N/m²)</i>
$E = 69$	$C_{11} = 126 \times 10^9$
	$C_{12} = 79.5 \times 10^9$
	$C_{13} = 84.1 \times 10^9$
	$C_{33} = 117 \times 10^9$
	$C_{44} = 23.30 \times 10^9$
<i>Poisson's ratio</i>	<i>Piezoelectric strain matrix (C/m²)</i>
$\nu = 0.3$	$E_{31} = -6.5$
	$E_{33} = 23.3$
	$E_{15} = 17$
<i>Shear modulus (GPa)</i>	<i>Dielectric matrix (F/m)</i>
—	$\epsilon_{11} = 15.03 \times 10^{-9}$
	$\epsilon_{22} = 15.03 \times 10^{-9}$
	$\epsilon_{33} = 13.00 \times 10^{-9}$

The global axis and a revolute joint are defined at the root of the smart manipulators. The SM rotates about the z-axis. Rotational spring constant K_m of the revolute joint is described as 16000 N.m/rad owing to the flexibility of gearbox of the servo motor to be taken into consideration in the experiments. The FE model is shown in Figure 3.2.

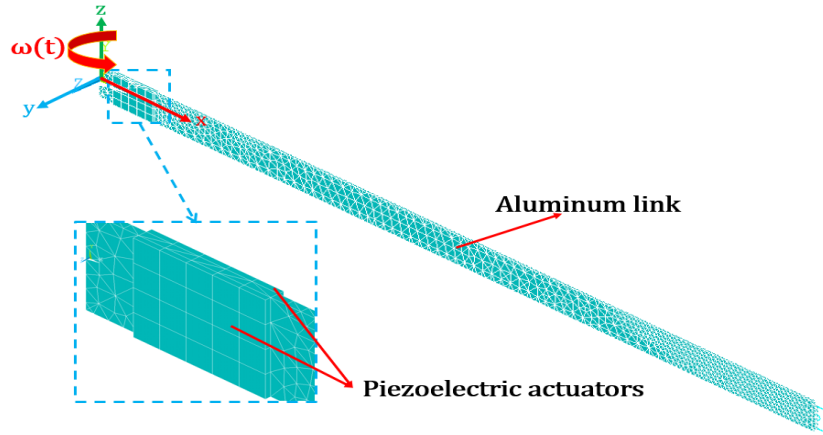


Figure 3.2 FE model of SM

The SM is modeled with SOLID226 element and SOLID92 element (ANSYS Inc., 2012). The aluminum part is modeled using SOLID92 element while the PZT patch is created using SOLID226 element. SOLID92 element has 5250 nodes with x, y, z displacement DOFs at per node. SOLID226 element has 398 nodes with x, y, z displacement and voltage DOFs at per node. The FE model has 2506 elements and 5648 nodes in total.

3.2.2 Smart Composite Manipulators

The FE model of a smart composite manipulator (SCM) with two different orientations is shown in Figure 3.3.

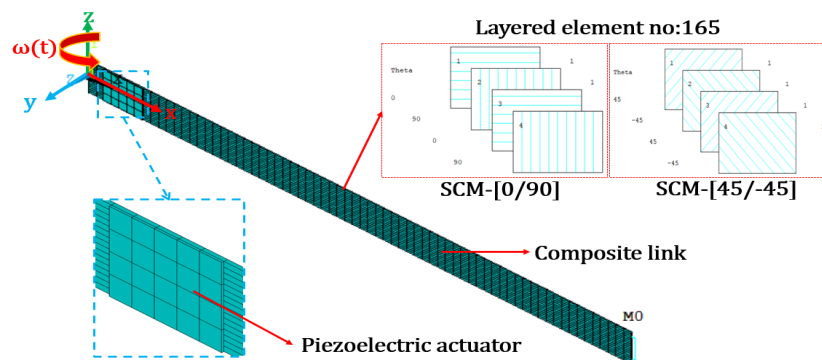


Figure 3.3 FE models of SCM

The SCM is produced from a symmetric laminated epoxy-glass composite link and

a PZT actuator upon the upper surface. The geometry, coordinate system and lay-ups are presented in Figure 3.3. The FE models of the SCM with $[0/90]_s$, and $[45/-45]_s$ orientations are specified for each layer. Two different models for the SCM is called as SCM-[0/90] and SCM-[45/-45]. Lay-ups of the SCM-[0/90] is created as $[0/90]_s$ while those of SCM-[45/-45] is modeled as $[45/-45]_s$. The global axis, revolute joint and K_m are defined at the root of the SCM. The SCM rotates about the z-axis. Material properties for the composite link is given in Table 3.3.

Table 3.3 Material properties of SCM

Density	Young's modulus	Poisson's ratio	Shear modulus
kg/m^3	GPa		GPa
$\rho = 1600$	$E_1 = 40.51$	$\nu_1 = 0.22$	$G_1 = 3.1$
	$E_2 = 13.96$	$\nu_2 = 0.22$	$G_2 = 3.1$
	$E_3 = 13.96$	$\nu_3 = 0.11$	$G_3 = 1.5$
Numerical values of SCM-[0/90] and SCM-[45/-45]			
Distance for actuator, La		10 mm	
Rayleigh damping, α, β		0.356728, 0.000049	for SCM-[0/90]
Rayleigh damping, α, β		0.450014, 0.000136	for SCM-[45/-45]

In the FE models, the composite link is modeled with SOLID186 layered solid element which has eight nodes and three DOFs (ux, uy, uz) at each node. The PZT actuator is created by using SOLID226 element.

3.2.3 Smart Composite Box Manipulator

The smart composite box manipulator (SCBM) is shown in Figure 3.4.

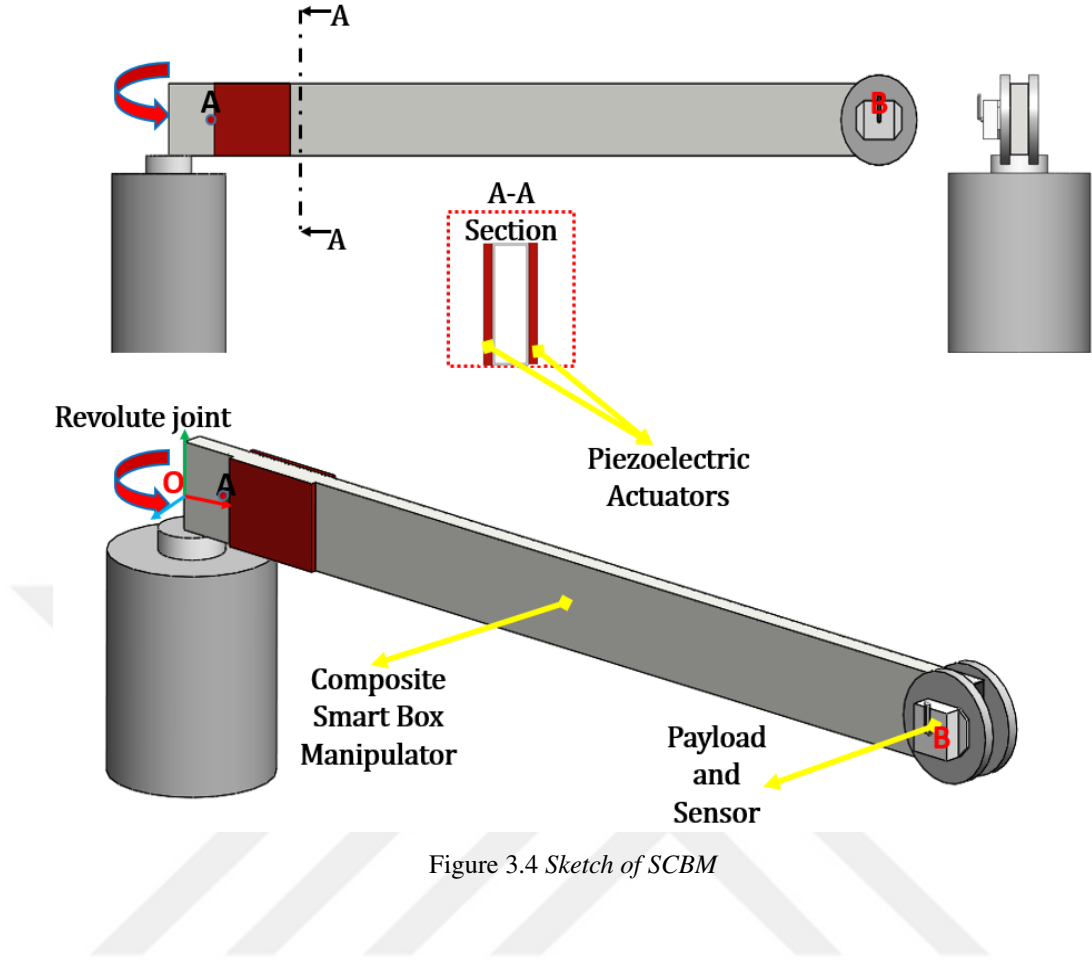


Figure 3.4 Sketch of SCBM

The SCBM is an OB beam with epoxy-glass material and rectangular cross-section. The global axis and a revolute joint are identified at O. The SCBM rotates around the z-axis and carries a load placed at distance d_p . The flexibility of geared servo motor to be taken into consideration in the experiments, K_m is defined at O. The mass of accelerometer sensor placed at distance d_p used in the experiments is considered in the model. Two PZT patches with the bimorph configuration to be used in the AC are attached to the upper and lower surfaces at a distance d_a from the axis of rotation of the smart manipulators. The properties of the SCBM and the values of K_m provided by the manufacturer, d_p , d_a are given in Table 3.4.

Table 3.4 Material properties of CBM

Composite structure	Density	Young's modulus	Poisson's ratio	Shear modulus
	kg/m^3	GPa		GPa
	$\rho = 1600$	$E_1 = 24$	$\nu_1 = 0.1$	$G_1 = 3$
		$E_1 = 24$	$\nu_2 = 0.3$	$G_2 = 5$
		$E_1 = 40$	$\nu_3 = 0.3$	$G_3 = 5$

Numerical values of SCM

Size of manipulator, $h \times w \times t$	20.5 mm x 80.1 mm x 1 mm
Distance for payload and sensor, dp	935 mm
Weight of sensor, mp	0.054 kg
Distance for actuators, da	60 mm
Rayleigh damping, α, β	1.53169, 0.000343
Motor rotational spring constant, K_m	16000 Nm/rad

The FE model is shown in Figure 3.5 and the geometry, lay-ups and coordinate systems are shown in Figure 3.6.

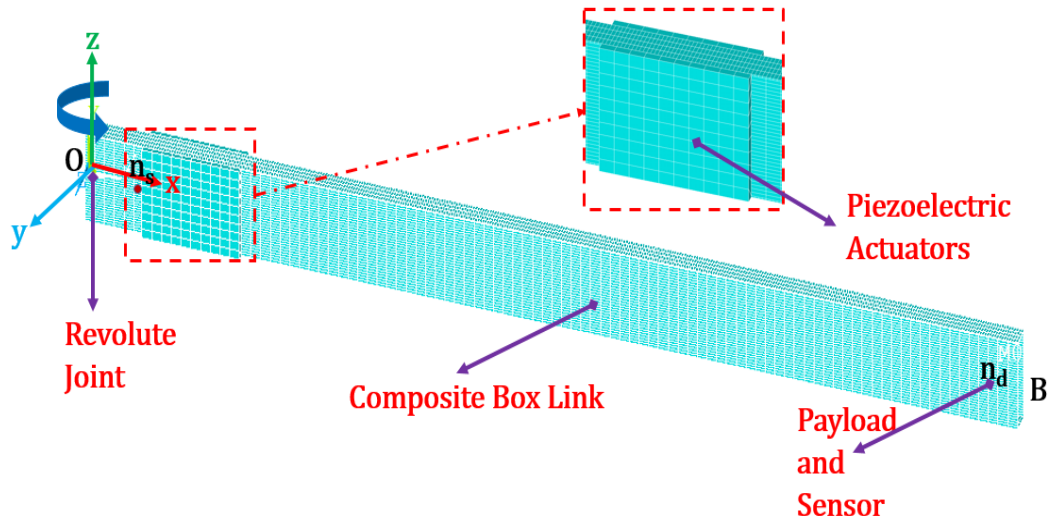


Figure 3.5 FE model of SCBM

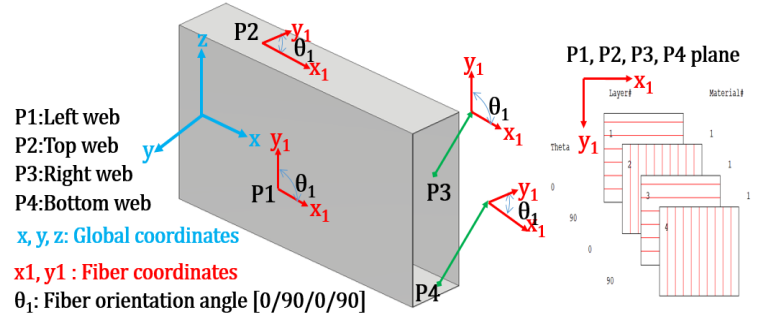


Figure 3.6 Geometry and lay-ups of SCBM

For the FE model shown in Figure 3.5, the SOLID186 layered solid element is used. The FE model includes four layers and the orientations are determined for each layer. In Figure 3.6, the lay-ups are defined on the left (P1), top (P2), right (P3), and bottom (P4) webs. Two coordinate systems are required to model the SCBM. x , y , and z axes define the general coordinate system while x_1 and y_1 axes define the local coordinate system. The local axis z is perpendicular to each plane. θ_1 is the orientation angle for each layer. The orientations on each layer are defined according to x_1 . The lay-up [0/90/0/90] is considered. So, the values of θ_1 are 0 and 90 degrees.

SOLID226 element is used to model the PZT actuators with the bimorph configuration. The mesh size is chosen as 10 mm. In total, the FE model has 2629 elements and 17219 nodes. Volt DOF is combined with the nodes at the upper and lower surfaces of the PZT actuators. For the bimorph actuation, the PZT actuators coupled sets are determined.

3.2.4 Finite Element Formulation of a Piezoelectric Patch

The constitutive equations for the FE modeling by Preumont (2011) to model the PZT materials are presented by

$$\begin{aligned} \{T\} &= [c] \{S\} - [e] \{E\} \\ \{D\} &= [e^T] \{S\} - [\epsilon] \{E\} \end{aligned} \quad (3.1)$$

here, mechanical variables are defined as strain S and stress T vectors. Electrical factors are the electric field E and electric displacement D vectors, separately. $[c]$, $[e]$ matrices and $[\epsilon]$ as given in Table 3.4 are the PZT properties, which $[e]$, $[c]$, $[\epsilon]$ are the piezoelectric, elasticity and permittivity matrices.

The Lagrangian function of a smart structure with a finite number of discrete piezoelectric material can be defined in common form by

$$L = T^* + W_c^* = \frac{1}{2}\dot{x}^T M \dot{x} - \frac{1}{2}x^T K_{xx}x + \frac{1}{2}\Phi^T K_{\Phi\Phi}\Phi + \Phi^T K_{\Phi x}x \quad (3.2)$$

where, L is the Lagrange parameter, W_c^* is the total energy and, T^* is the total kinetic energy, which includes mechanical energy stored in the elastic material, electrical energy stored in the dielectric material and piezoelectric energy.

The FE formulation can be derived from Hamilton's principle. For a smart manipulator, the equations of motion can be stated by

$$\begin{aligned} [M]\{\ddot{x}\} + [K_{xx}]\{x\} + [K_{x\phi}]\{\Phi\} &= \{F\} \\ [K_{x\phi}]^T\{x\} + [C_{\phi\phi}]\{\Phi\} &= \{Q\} \end{aligned} \quad (3.3)$$

where, $[M]$ is the mass matrix emerged out of volume and density, $[K_{xx}]$ is the stiffness matrix obtained from the elasticity including the mechanical part of piezoelectric with short-circuited electrical boundary conditions, $[K_{x\phi}]$ is the coupling matrix of piezoelectric properties relating the mechanical and electrical variables. x and Φ are the vectors of general coordinates and electric potential, respectively. F and Q are the force vectors appearing on the inertial loads and electric charges appearing on the electrodes.

The damping matrix $[C]$ is taken as a linear combination of mass $[M]$ and stiffness matrices $[K_{xx}]$ given in Eq. 3.4

$$[C] = \alpha[M] + \beta[K_{xx}] \quad (3.4)$$

where, α and β , which are Rayleigh damping coefficients, found from the vibration responses by experiments are given in Table 3.2 for the SM, 3.3 for the SCM-[0/90], SCM-[45/-45] and 3.4 for the SCBM.

3.2.5 Defining Motion Input

A revolute joint is defined to move the smart manipulators. The configuration of the revolute joint is shown in Figure 3.7.

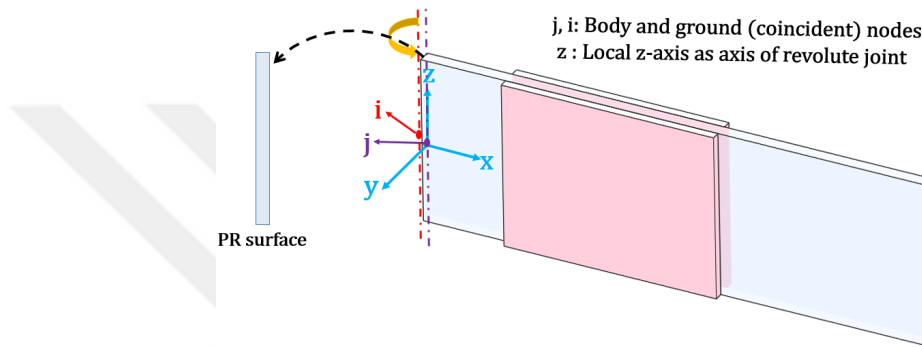


Figure 3.7 Revolute joint configuration

MPC184 element is used to create the revolute joint in ANSYS (ANSYS Inc., 2012). This element constraints “ux, uy, uz, rotx, roty” DOFs of revolute joint and releases angular rotation “rotz” degree of freedom in the z-axis. The settings must be done for the revolute joint as KEYOPT (1),6 and KEYOPT(4),1. These settings correspond to define the revolute joint between the rigid and flexible body and the local z-axis as the rotation axis.

As shown in Figure 3.7, in order to create a revolute joint, two coincident local coordinate systems are defined on the PR surface. The ground i and body j nodes are defined in the two local coordinate systems. A pilot node, which couples the nodes on the surface, is required to define a rotation. TARGE170 element is used to create the pilot node. As shown in Figure 3.7, the nodes on the PR surface are coupled with this pilot node. The defined pilot node is connected by j node. Then, the revolute joint element, which allows a single relative rotation around the z-axis is created between i and j nodes.

3.3 State-Space Model Extraction of Smart Manipulators

In the FE model, the SS matrices based on modal analysis can be obtained with the normal vibration modes (Khot & Yelve, 2011; Khot et al., 2012; Meirovitch, 2001). Modal analysis is done to find the mode shapes and undamped natural frequencies in ANSYS. The dynamic behavior of structures with fewer equations is obtained for the modeling and transient analysis with the model extraction method. The SS method as detail described in Chapter-2 is used to extract the SS models of the smart manipulators based on modal analysis in ANSYS. In order to obtain the system matrices, the number of extracted vibration modes, outputs and inputs of the system should be defined in modal analysis. The model extraction is performed using ANSYS. Then, the SS matrices are written to a text file with the nodal translation and rotational outputs.

The form of a system matrix A for the first three vibration modes is composed of the natural frequencies (ω_r) and damping ratios (ζ_r) as follows

$$A = \begin{bmatrix} 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ -\omega_1^2 & 0 & 0 & -2\zeta_1\omega_1 & 0 & 0 \\ 0 & -\omega_2^2 & 0 & 0 & -2\zeta_2\omega_2 & 0 \\ 0 & 0 & -\omega_3^2 & 0 & 0 & -2\zeta_3\omega_3 \end{bmatrix} \quad (3.5)$$

where, ω_1 , ω_2 , and ω_3 are the first, second and third natural frequencies while ζ_1 , ζ_2 , and ζ_3 are the damping ratios for first, second and third vibration modes, respectively. Input matrix B is determined using a normalized modal matrix and the input vector (u) with size nxn for the number of inputs. Output matrix C, which consists of modal displacements, velocities and accelerations of defined output DOFs, is formulated according to the number of outputs and selected vibration modes. The SS matrices of the smart manipulators are extracted from modal analysis using the outputs and inputs of the system. Inputs and outputs are chosen as given in Table 3.5.

Table 3.5 Inputs and outputs of smart manipulators

Manipulator	Inputs	Outputs
1- SM	Torque, $\tau(t)$	Displacement, $d_{x,y,z}(t)$
	Actuator voltage-1	Velocity, $v_{x,y,z}(t)$
2- SCBM	Actuator voltage-2	Acceleration, $a_{x,y,z}(t)$
3- SCM-[0/90]	Torque, $\tau(t)$	Displacement, $d_{x,y,z}(t)$
	Actuator voltage-1	Velocity, $v_{x,y,z}(t)$
4- SCM-[45/-45]		Acceleration, $a_{x,y,z}(t)$

Torque $\tau(t)$ of the revolute joint and the control voltages (VA) of the PZT actuators are defined as the inputs. In the SM and SCBM, the inputs are selected as $\tau(t)$ of the revolute joint, VA1 and VA2 for PZT actuator-1 and 2 while those is selected as $\tau(t)$ and VA1 for the PZT actuator-1 for the SCM-[0/90] and SCM-[45/-45] to suppress the residual vibration in the AC. The outputs of all smart manipulators under thesis are chosen as the displacement, velocity and acceleration in the bending direction (y-axis) at the tip. After defining the inputs and outputs of the system, the SS models are obtained for the three vibration modes as shown in Figure 3.8.

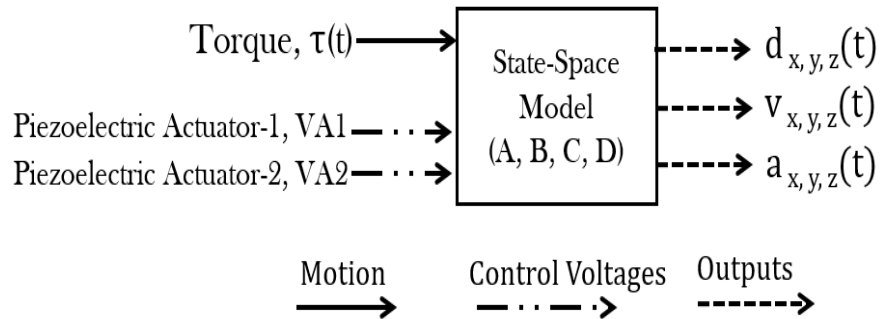


Figure 3.8 SS model of smart manipulators

$\tau(t)$ is utilized to move the smart manipulators for the desired angular position. Therefore, the angular velocity for the motion profiles is converted to the $\tau(t)$. Inertia of moments I_{zz} of the smart manipulators in the z-axis are found as $2.5115 \times 10^{-3} \text{ kg.m}^2$ for the SM, $3.8527 \times 10^{-4} \text{ kg.m}^2$ for the SCM-[0/90] and $3.1719 \times 10^{-4} \text{ kg.m}^2$ for the SCM-[45/-45] and 1.1104 kg.m^2 for the SCBM. The SS matrices between the inputs and outputs are obtained for the smart manipulators. The values of the SS matrices (A, B) of the smart manipulators are given for the first three vibration modes in equations below. Some values of the matrices are not given for the confidentiality of the thesis.

$$\begin{aligned}
 A_{SM} &= \begin{bmatrix} 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ \dots & 0 & 0 & \dots & 0 & 0 \\ 0 & -59.444 \times 10^3 & 0 & 0 & -1.7349 & 0 \\ 0 & 0 & \dots & 0 & 0 & -6.5683 \end{bmatrix} \\
 B_{SM} &= \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ \dots & \dots & -0.143 \times 10^{-3} \\ 153.02 & -0.866 \times 10^{-3} & \dots \\ \dots & -8.059 \times 10^{-7} & 2.201 \times 10^{-7} \end{bmatrix}
 \end{aligned} \tag{3.6}$$

$$A_{SCM-[0/90]} = \begin{bmatrix} 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ -1195.4 & 0 & 0 & \dots & 0 & 0 \\ 0 & \dots & 0 & 0 & -2.5527 & 0 \\ 0 & 0 & -263.930 \times 10^3 & 0 & 0 & \dots \end{bmatrix} \quad (3.7)$$

$$B_{SCM-[0/90]} = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ -50.166 & \dots \\ \dots & 1.475 \times 10^{-3} \\ -4.3437 & \dots \end{bmatrix}$$

$$A_{SCM-[45/-45]} = \begin{bmatrix} 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ \dots & 0 & 0 & -0.523 & 0 & 0 \\ 0 & -19.68 \times 10^3 & 0 & 0 & \dots & 0 \\ 0 & 0 & \dots & 0 & 0 & -20.066 \end{bmatrix} \quad (3.8)$$

$$B_{SCM-[45/-45]} = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ -52.305 & \dots \\ -375.43 & \dots \\ \dots & 1.206 \times 10^{-3} \end{bmatrix}$$

$$\begin{aligned}
 A_{SCBM} &= \begin{bmatrix} 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ -1905 & 0 & 0 & \dots & 0 & 0 \\ 0 & \dots & 0 & 0 & -6.994 & 0 \\ 0 & 0 & -609.87 \times 10^3 & 0 & 0 & \dots \end{bmatrix} \\
 B_{SCBM} &= \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0.747 & \dots & \dots \\ \dots & -5.933 \times 10^{-7} & \dots \\ 0.057 & \dots & -0.025 \times 10^{-3} \end{bmatrix}
 \end{aligned} \tag{3.9}$$

The SS models are found as given in Equation 3.6 for the SM, Equation 3.7 for the SCM-[0/90], Equation 3.8 for the SCM-[45/-45], and Equation 3.9 for the SCBM, respectively. After extracting the SS models of the smart manipulators, the SS analysis is studied in the next chapter for the AC and PC in MATLAB.

CHAPTER FOUR

PASSIVE CONTROL OF FLEXIBLE SMART MANIPULATORS

4.1 Introduction

In this chapter, the PC for the smart manipulators is theoretically investigated with the simulation and SS analysis. The simulation is implemented in ANSYS while the SS analysis is performed in MATLAB using the extracted SS models given in Chapter-3. Briefly, modal analysis is done to verify the accuracy of the FE models created in Chapter-3. The natural frequency results of the composite structures are presented for the reference studies with different dimensions and various layer orientations. For the PC analysis, the smart manipulators can be moved with a motion profile occurred random time parameters. If some of the time parameters are chosen suitably, the amplitudes of residual vibrations can be reduced with the motion profile. For the PC, the time parameters are found by using the natural frequencies obtained from modal analysis. The trapezoidal and triangular velocity profiles are determined. The PC analysis is conducted. The simulation and SS analysis results are compared. The effectiveness of the PC on the residual vibrations is examined.

4.2 Modal Analysis

Modal analysis is performed to find the natural frequencies and motion parameters for the PC analysis. For modal analysis, external loads such as $F = 0$ and $Q = 0$ are assume as equal to zero. Equation of motion for the smart manipulators are given in matrix form as

$$\begin{bmatrix} M & 0 \\ 0 & 0 \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{\Phi} \end{Bmatrix} + \begin{bmatrix} K_{xx} & K_{x\phi} \\ K_{x\phi}^T & C_{\phi\phi} \end{bmatrix} \begin{Bmatrix} x \\ \Phi \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix} \quad (4.1)$$

The eigenvalue equation is given as follows

$$\left[K^* - \omega_i^2 M \right] = 0 \quad (4.2)$$

and,

$$\left[K_m \right] = \left(\left[K_{xx} \right] + \left[K_{\phi x} \right]^T \left[C_{\phi\phi} \right]^{-1} \left[K_{\phi x} \right] \right) \quad (4.3)$$

There are n values of natural frequencies and ω_i is the i^{th} natural frequency of the system. Natural frequencies can be found as $f_{ni} = \frac{\omega_i}{2\pi}$ in Hz.

Different cantilever laminated composite box and smart structures with different configurations in the corresponding references (Yaman et al., 2002; Manning et al., 2000; Yousefi-Koma, 1997; Xu & Koko, 2004; Qin & Librescu, 2002; Mitra et al., 2004b,a; Vo & Lee, 2008) are studied to evaluate the correctness of present FE models in ANSYS. Reference studies taken into account, dimensions and layer orientations are given in Table 4.1 for the smart structures and smart composite box structures.

Table 4.1 Configurations in the corresponding references

Smart Structures			
Reference	Material	Structure (L x w x t), [mm ³]	Actuator (L x w x t), [mm ³]
Yaman et al. (2002)	Aluminum link	507x51x2	20x25x0.61
Manning et al. (2000)		350x24x1	72x24x0.5
Yousefi-Koma (1997)		508x25.4x0.8	76.2x25.4x0.305
Xu & Koko (2004)		226x25x0.965	39x25x0.75
Composite structure			
		Dimension (L x w x t) , [mm ³]	Layer orientation [°]
Qin & Librescu (2002)	Composite box link	762x52.3x13.6	[30] ₆ CAS
Mitra et al. (2004b)		254x33.5x16.7	[20/ − 70/20/ − 70/ − 70/20]
Mitra et al. (2004a)		655x57x19	[0 ₃ /90 ₂ /0 ₃]
Vo & Lee (2008)		8000x200x100	[45] ₆ CUS

*K is used as K for the SM while considered as K_m for the SCM-[0/90], SCM-[45/-45], SCBM.

A comparison of first three undamped natural frequencies is given in Table 4.2.

Table 4.2 Comparison of natural frequencies with references studies

Smart Structures									
Mode	Yaman et al. (2002)		Manning et al. (2000)		Yousefi-Koma (1997)		Xu & Koko (2004)		
	Reference	Present	Experiment	Reference	Present	Reference	Present	Reference	Present
	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz
1	6.67	6.60	3.41	3.11	3.04	19.85	19.94	9.00	9.004
2	41.85	41.24	16.90	16.07	15.40	104.78	96.37	-	52.43
3	117.21	115.17	-	-	40.46	259.22	230.82	-	133.27

Composite structure										
Mode	Qin & Librescu (2002)			Mitra et al. (2004b)		Mitra et al. (2004a)			Vo & Lee (2008)	
	Exp.	Analytical	Present	Analytical	Present	Exp.	Analytical	Present	Present	Analytical
	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz
1	20.96	19.92	19.89	3.09	3.03	34.6	34.46	34.90	10.88	10.71
2	128.36	124.73	121.79	5.59	5.64	-	81.32	78.53	18.39	18.81
3	-	-	189.49	19.36	18.98	206.00	206.59	206.23	73.57	66.96

As seen from the tables, the analytical and experimental frequency results are presented for the references given in Table 4.2. The smart structures and composite box structures with different configurations are studied by considering references in Table 4.1. Results of undamped natural frequencies are compared. It is observed that the current results are matched well with the natural frequency results given in the reference studies.

The natural frequencies should be determined before performing the transient analysis for the active and passive vibration controls. After evaluating the natural frequencies of the smart and different composite box structures, modal analysis is performed to find the natural frequencies of the smart manipulators. The undamped natural frequencies of SM, SCM-[0/90], SCM-[45/-45] and SCBM with various values of payloads are given in Table 4.3.

In order to examine the effect of thickness and stacking sequence on the dynamic characteristics of SCBM in terms of natural frequency, the simulations are extended. Table 4.4 presents the results without a payload for different values of thickness and stacking sequence.

Table 4.3 Natural frequencies of smart manipulators

Order of Natural Frequency (Hz)	SM	SCM-[0/90]	SCM-[45/-45]
f_{n1}	6.61	3.66	5.48
f_{n2}	39.20	22.24	33.27
f_{n3}	80.72	60.06	81.50
SCBM, Payloads			
	$M_p = 0$ kg	$M_p = 1.11$ kg	$M_p = 2.10$ kg
f_{n1}	20.41	6.58	4.87
f_{n2}	59.42	19.03	14.09
f_{n3}	102.10	90.95	90.20

Table 4.4 Effect of thickness and stacking sequence for no payload

Thickness (mm)	Natural Frequency (Hz)	[0/90/0/90]	[45/-45/-45/45]	[30/60/60/30]
1	f_{n1}	20.41	9.88	12.54
	f_{n2}	59.42	36.89	42.025
	f_{n3}	102.10	63.779	76.08
2	f_{n1}	21.91	12.51	15.09
	f_{n2}	70.82	51.85	56.05
	f_{n3}	112.98	73.85	84.66
		[0/90/0/90/0/90]	[45/-45/45/45/-45/45]	[30/60/30/30/60/30]
1	f_{n1}	20.41	9.88	14.61
	f_{n2}	59.43	36.92	45.72
	f_{n3}	102.11	63.80	83.77
2	f_{n1}	21.91	12.52	16.88
	f_{n2}	70.83	51.91	58.97
	f_{n3}	113.01	73.90	91.84

It can be seen from Table 4.4 that the thickness and stacking sequence affect the natural frequencies of SCBM significantly. The increase in the number of layers produces small increases in the natural frequency values. Among the stacking sequence considered for no payload condition, the most rigid layer orientation is [0/90/0/90], while the most flexible layer orientation is [45/-45/45/45]. The natural frequencies for all stacking sequences increase as the thickness increases. The example mode shapes corresponding to the first three natural frequencies for the SM and SCBM are shown in Figure 4.1.

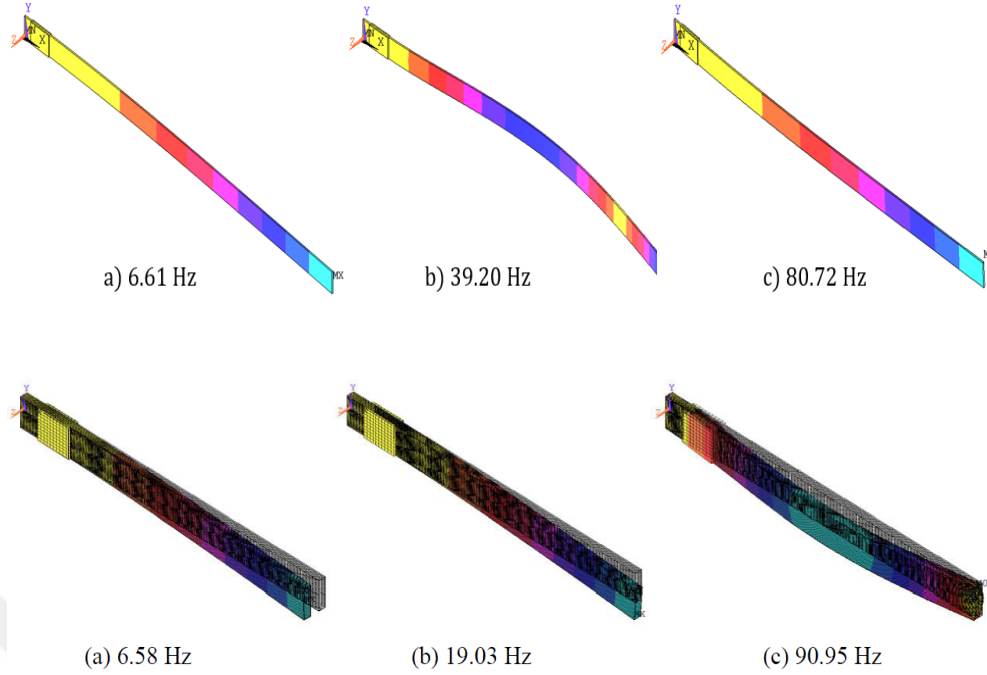


Figure 4.1 Mode shapes of SM and SCBM for $m_p = 1.11$ kg (a) 1st mode, (a) 2nd mode, (c) 3rd mode.

It is observed from Table 4.3 that payloads of smart manipulators affect the vibration modes. The natural frequencies decrease as the payload increases due to the effect of mass. For the SCBM, the SCM becomes flexible due to the payload and more flexible with higher payloads.

The time parameters for the transient analysis are calculated from the modal analysis results. The time step Δt and settling time t_{ss} , which depend on the undamped natural frequencies, are required to obtain the time responses of smart manipulators. The time step is calculated as $\delta t = 1/f_{n1}/20$, here, for the control of the first bending vibration mode, f_{n1} is the first undamped natural frequency. t_{ss} is selected as $t_{ss} = t_m + t_{rest}$, where t_{rest} is the residual time to obtain the residual vibrations after stopping the manipulator. Also, t_m is the motion time that takes to drive the smart manipulators.

Rayleigh damping is considered for the transient analyses. The Rayleigh coefficients α and β acquired from the experimental vibration responses are given in Table 3.2 for the SM, Table 3.3 for the SCM-[0/90] and SCM-[45/-45], and Table 3.4 for the SCBM.

4.3 Generating Motion

The motion profiles are defined to the revolute joint to move the smart manipulators. The motion profiles used in the PC are shown in Figure 4.2.

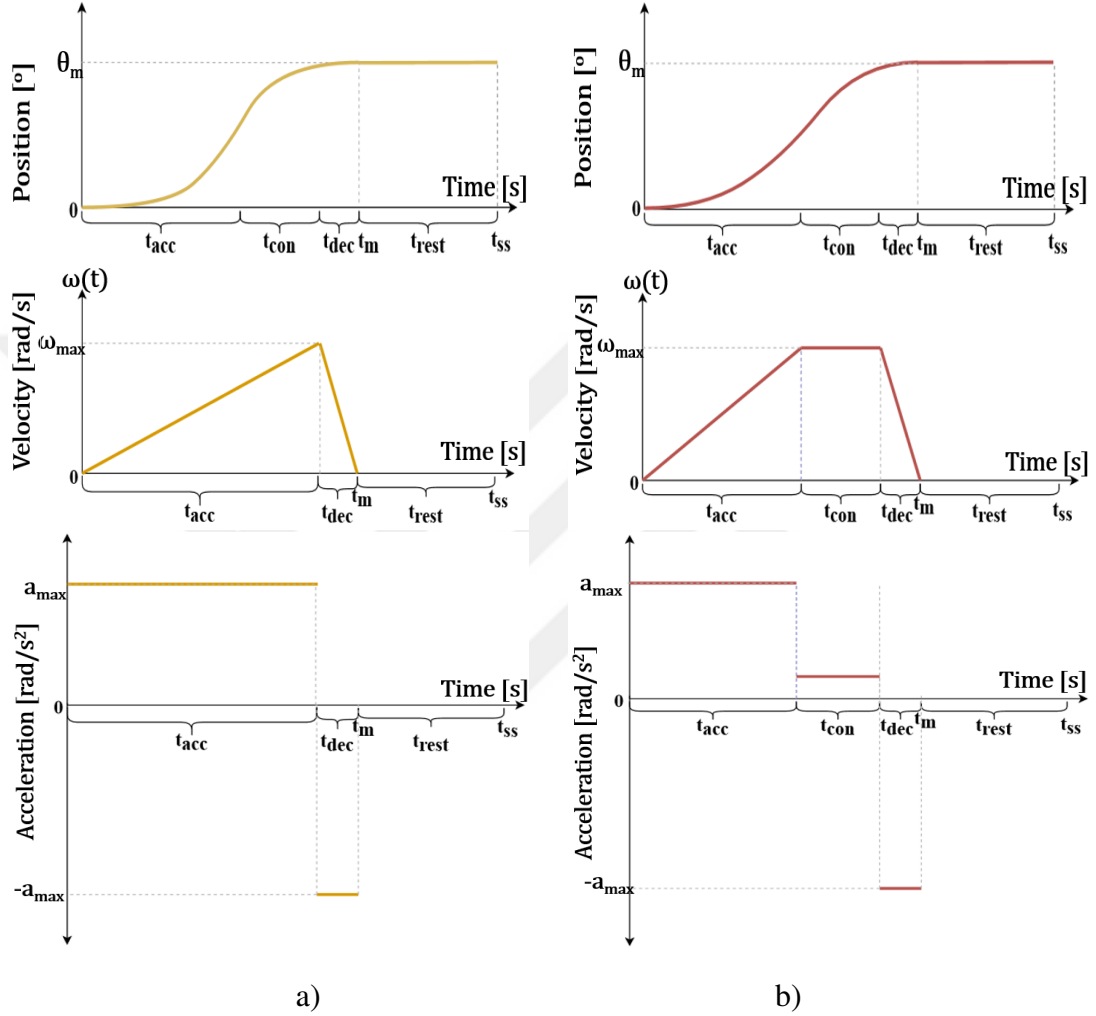


Figure 4.2 Position, velocity and acceleration profiles for a) triangular, and b) trapezoidal motion profiles

Two different motion profiles are studied to reduce the residual vibrations. Position, velocity and acceleration profiles for the motion profiles are shown in Figure 4.2. In the motion profiles, variables t_{acc} , t_{con} , t_{dec} , t_m , t_{rest} , and t_{ss} are defined as the acceleration, constant, deceleration, motion, residual vibration and settling times, respectively. Also, θ_m is determined as the angular position and ω_{max} is the maximum angular velocity while a_{max} is the maximum angular acceleration. t_{con} is not equal to zero for the trapezoidal velocity profiles while it is taken as zero for the triangular motion profiles.

In order to move the smart manipulators, the motion profiles are created with the t_{acc} , t_{con} , and t_{dec} . t_m is found by using the time parameters as follows

$$t_m = t_{acc} + t_{con} + t_{dec} \quad (4.4)$$

The smart manipulators move from a rest position at $t = 0$ to a stopping position θ_m at $t = t_m$. θ_m indicates an incremental value of angular position from the rest position. The units are in seconds for the time parameters and degree for angles unless otherwise stated. θ_m is found by integrating the angular velocity with respect to time. The motion time parameters are calculated as given in Equation 4.5.

$$\begin{aligned} t_{1h} &= 1/f_{n1}/2 \\ t_{con} &= 2t_{1h}, \quad \text{for trapezoidal velocity profile} \\ t_{con} &= 0, \quad \text{for triangular velocity profile} \\ t_{dec} &= rt_{1h}, \quad r = 1, 2, 3, 4, \dots \\ t_{acc} &= t_m - (t_{con} + t_{dec}) \end{aligned} \quad (4.5)$$

The maximum angular velocity ω_{max} is obtained using the following equation.

$$\omega_{max} = \frac{\theta_m}{0.5t_{acc} + t_{con} + 0.5t_{dec}} \quad (4.6)$$

where, θ_m is taken as radian given in Equation 4.6. For the motion profiles, the time parameters with the maximum angular velocity are calculated given in the equations above. The PC can be achieved by selecting suitable time parameters in a motion profile. The deceleration time in various motion profiles is effective to decrease the vibration levels of structures (Yavuz et al., 2016; Ankarali & Diken, 1997). Therefore, t_m and ω_{max} are defined as the input to move the manipulator. t_{dec} and t_{con} are calculated from Equation 4.5 and then, t_{acc} is found by determining t_{dec} and t_{con} . In the PC, half of period of first natural frequency of the smart manipulators is taken into account while determining t_{dec} , $t_{1h} = 2/f_{n1}$ is calculated at first. The deceleration time is designed as $t_{dec} = rt_{1h}$, where, r is an integer to investigate the effect of vibration

amplitudes in the motion control. The effect of t_{dec} is performed on the vibration control. t_{con} is chosen as equal to zero for the triangular motion profiles while selecting as $t_{con} = 2t_{1h}$ for the trapezoidal velocity profiles.

Example triangular and trapezoidal velocity profiles considering for t_{dec} are created as shown in Figure 4.3.

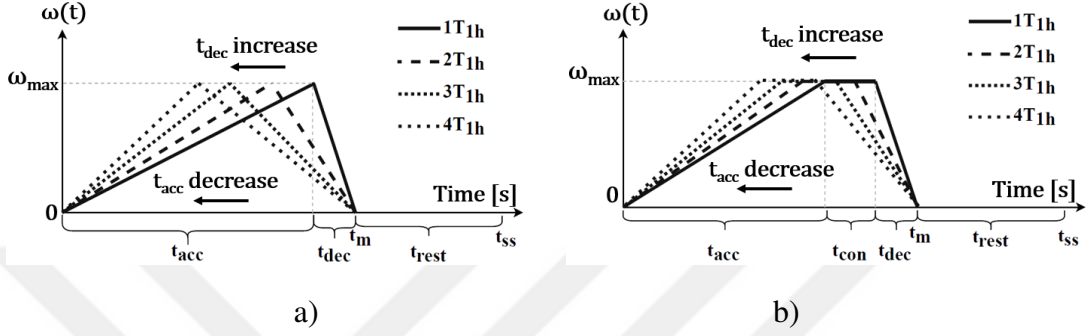


Figure 4.3 Example a) triangular, and b) trapezoidal velocity profiles

The example motion profiles are given for the different t_{acc} and t_{dec} shown in Figure 4.3. While t_{dec} is the minimum for the $1t_{1h}$, it is maximum for the $5t_{1h}$. Also, as t_{dec} increases, t_{acc} is decreases for all motion profiles while ω_{max} is constant. For the active and passive vibration controls, the motion profiles are created and, then, the time and velocity parameters are listed below.

For the SM, two different motion profiles given in Table 4.5 are defined as the triangular and trapezoidal velocity profiles.

For the SM, the motion time and angular displacement are chosen as $t_m = 1$ s and $\theta_m = 45^\circ$ for the motion profiles. In Table 4.5, SM-M1 is defined as the trapezoidal velocity profile while SM-M2 is the triangular motion profile. For two different motion profiles, the desired angular position is reached with different angular speeds at the same motion time. The maximum angular velocity is selected as two different values for the motion profiles.

For the SCM-[0/90] and SCM-[45/-45], four different motion profiles are created with the angular displacement $\theta_m = 45^\circ$ to move each SCM. t_m is chosen as 1 s. Motion

Table 4.5 Time parameters of motion profiles for the SM

Motion profile	f_{n1} [Hz]	θ_m [$^\circ$]	t_{acc} [s]	t_{con} [s]	t_{dec} [s]	t_m [s]	ω_{max} [rad/s]
SM-M1	6.59	45	0.7758	0.1495	0.0747	1	1.3665
			0.7010		0.1495		
			0.6263		0.2242		
			0.5516		0.2990		
SM-M2	6.59	45	0.9253	0	0.0747	1	1.5708
			0.8505		0.1495		
			0.7758		0.2242		
			0.7010		0.2990		

profiles for the SCM-[0/90] and SCM-[45/-45] are listed with the time and motion parameters given in Table 4.6.

Table 4.6 Time parameters of motion profiles for the SCM-[0/90] and SCM-[45/-45]

Motion profile	f_{n1} [Hz]	θ_m [$^\circ$]	t_{acc} [s]	t_{con} [s]	t_{dec} [s]	t_m [s]	ω_{max} [rad/s]
SCM-M1a	5.52	45	0.7283	0.1812	0.0906	1	1.3299
			0.6377		0.1812		
			0.5471		0.2717		
			0.4565		0.3623		
SCM-M2a	5.52	45	0.9094	0	0.0906	1	1.5708
			0.8188		0.1812		
			0.7283		0.2717		
			0.6377		0.3623		
SCM-M1b	3.67	45	0.5913	0.2725	0.1362	1	1.2344
			0.4550		0.2725		
			0.3188		0.4087		
			0.1826		0.5450		
SCM-M2b	3.67	45	0.8638	0	0.1362	1	1.5708
			0.7275		0.2725		
			0.5913		0.4087		
			0.4550		0.5450		

Table 4.7 Time parameters of motion profiles for the SCBM

Motion profile	Payloads m_p [kg]	f_{n1} [Hz]	θ_m [$^\circ$]	t_{acc} [s]	t_{con} [s]	t_{dec} [s]	t_m [s]	ω_{max} [rad/s]
SCBM-M1a	0	20.41	30	0.9265		0.0245		
				0.9020		0.0490		
				0.8775	0.0490	0.0735	1	0.9983
				0.8530		0.0980		
				0.8285		0.1225		
SCBM-M1b	1.11	6.58	30	0.7720		0.0760		
				0.6960		0.1520		
				0.6201	0.1520	0.2280	1	0.9090
				0.5441		0.3040		
				0.4681		0.3799		
SCBM-M1c	2.11	4.87	30	0.6920		0.1027		
				0.5893		0.2053		
				0.4867	0.2053	0.3080	1	0.8688
				0.3840		0.4107		
				0.2813		0.5133		
SCBM-M1d	1.11	6.58	45	0.7720		0.0760		
				0.6960		0.1520		
				0.6201	0.1520	0.2280	1	1.3636
				0.5441		0.3040		
				0.4681		0.3799		
SCBM-M1e	1.11	6.58	45	1.7720		0.0760		
				1.6960		0.1520		
				1.6201	0.1520	0.2280	2	0.7299
				1.5441		0.3040		
				1.4681		0.3799		

As seen from Table 4.6, the time and motion parameters for the SCM are given for $\theta_m = 45^\circ$ and $t_m = 1$ s. For the motion profiles, the four different velocity profiles are determined due to the different natural frequencies of the SCM-[0/90] and SCM-[45/-45]. SCM-M1a and SCM-M2a are defined for the SCM-[0/90]. SCM-M1b

and SCM-M2b for the SCM-[45/-45] are determined. SCM-M1a and SCM-M1b are the trapezoidal velocity profiles while SCM-M2a and SCM-M2b are defined as the triangular velocity profiles.

The motion profiles for the SCBM are created with $\theta_m = 30^\circ$ and $\theta_m = 45^\circ$ for the three different payloads. The effect of mass on the motion profiles is examined with payloads placed at the tip of the manipulators. Also, two different motion time for $\theta_m = 45^\circ$ is selected as $t_m = 1$ and $t_m = 2$ for payload $m_p = 1.11$ kg to compare the effect of motion time. For the SCBM, the time parameters are given in Table 4.7.

The time and motion parameters for the smart manipulators are calculated to move the manipulator in Table 4.5, Table 4.6 and Table 4.7. The effect of payloads on the motion profiles, different angular displacement, velocity, and motion time is investigated on the PC. The first natural frequency is considered.

4.4 Analysis of Passive Vibration Control

The passive vibration control for the smart manipulators is studied with the simulation in ANSYS and SS analysis in MATLAB.

4.4.1 Finite Element Simulation

For the PC analysis, a script based on macro code for the transient analysis is integrated into the FE models of the smart manipulators in ANSYS. During the transient analysis, the PC does not require an external force or effect, so the control action is carried out with the created motion profiles. For the PC, the motion profiles are applied to the FE model. The time parameters are t_{acc} , t_{con} , t_{dec} , t_m , and t_{ss} , while the velocity is selected as ω_{max} given in Table 4.5, Table 4.6 and Table 4.7.

The time and velocity parameters of motion profiles are tabulated as the time-dependent velocity profile to move the smart manipulators in ANSYS. A velocity table is parametrically prepared by using the values of motion profiles. The

intersection of time and angular velocity values determines the rows and columns of the velocity table. For example, the angular velocity value is defined as equal to ω_{max} in the second row in the velocity table while the time value is equal to t_{acc} .

After defining the velocity table, the motion profile by using “DJ” command is applied to the revolute joint element in the simulation. The command is used as “ $dj,nrev,omgz,wt$ ”. Here, “ $nrev$ ” is the element number of revolute joint. “ $omgz$ ” is the angular velocity in the z-axis. “ wt ” is the name of velocity table that including data for the time-dependent motion profiles.

The PC analysis is conducted with the motion profiles by the simulations. The PC results are obtained as the acceleration and displacement vibration responses at the tip of the smart manipulators.

4.4.2 State-Space Analysis

The PC of the smart manipulators is performed with the SS analysis in the MATLAB. The block diagram is shown in Figure 4.4.

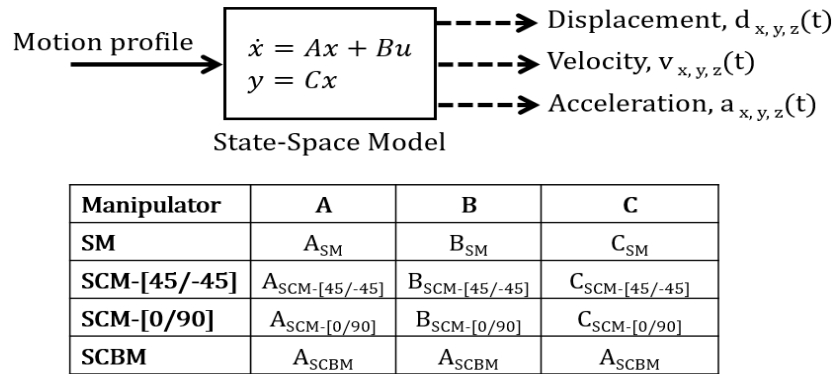


Figure 4.4 Block diagram of SS analysis in PC

The SS models given in Chapter-4 are used. In the block diagram of SS analysis, the motion profile is chosen as the input of the system while the outputs are the displacement, velocity and acceleration responses at the tip of the smart manipulators. The motion profile is converted to the torque profile by taking the second derivative of

a velocity profile and then, multiplying the moment of inertia of the smart manipulators before applied to the SS models. The torque profiles are applied to the SS models. The SS analysis are conducted using the block diagram given in Figure 4.4. The SS results for the PC are obtained in MATLAB.

4.5 Vibration Responses in Passive Vibration Control

The simulation and SS analysis results are compared for the PC. The PZT actuators are not available. In this section, different motion profiles for the SM, SCM, and SCBM mentioned above are used to move the smart manipulators. The simulation and SS results are presented.

The vibration responses are shown in Figure 4.5, Figure 4.6 and Figure 4.7.

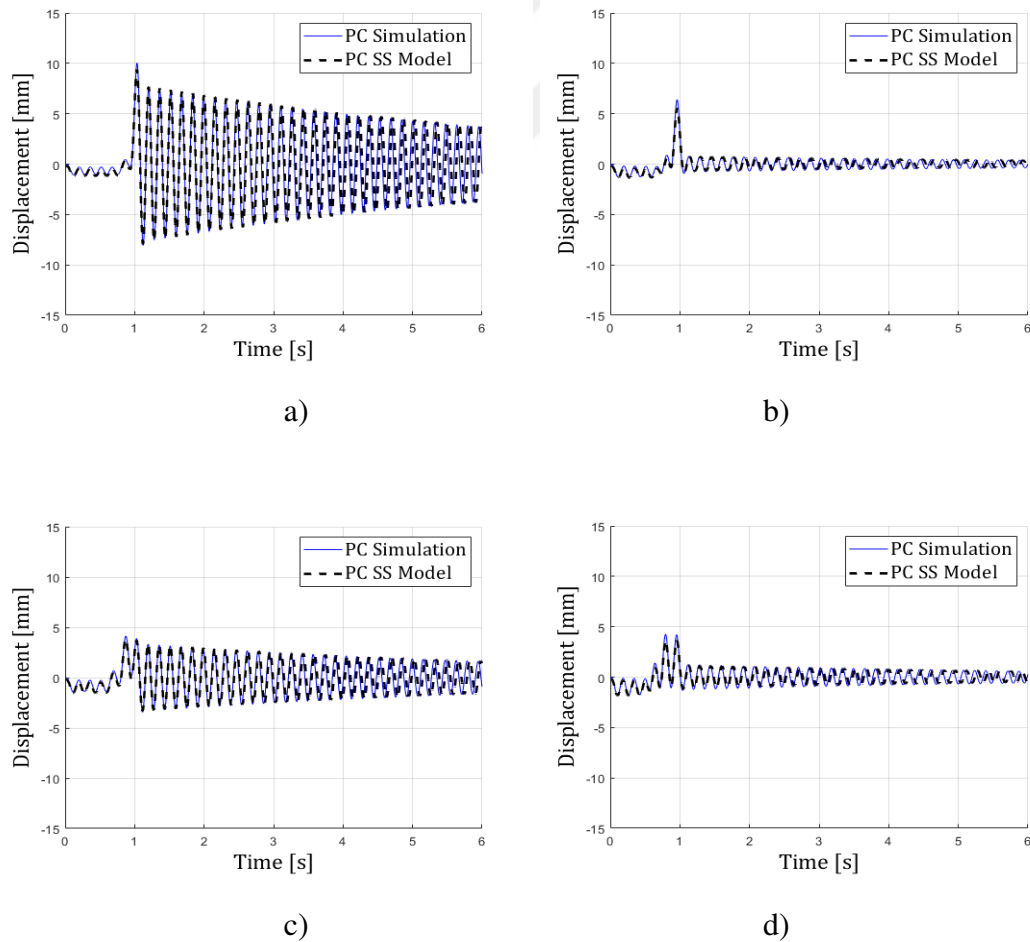
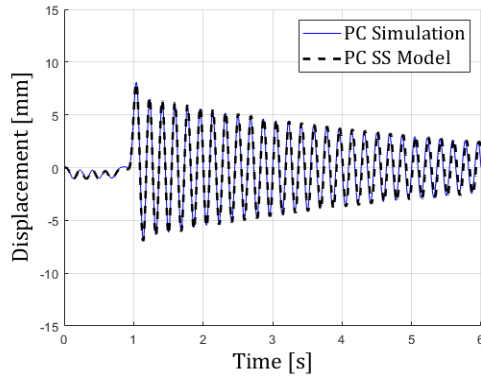
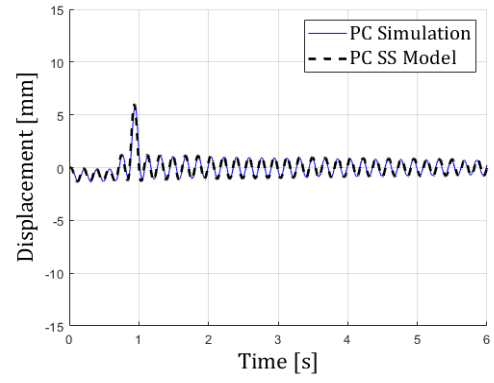


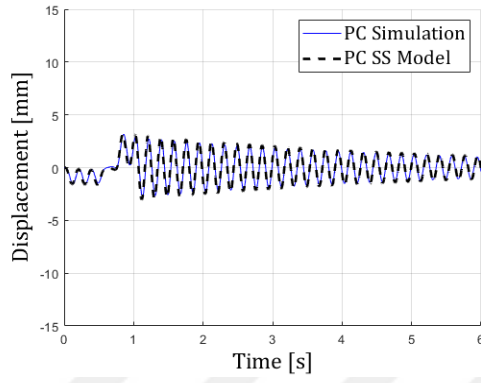
Figure 4.5 Vibration responses for SM with SM-M1, a) $1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, and d) $4t_{1h}$



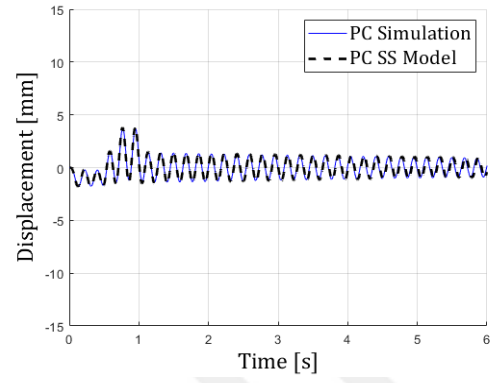
a)



b)

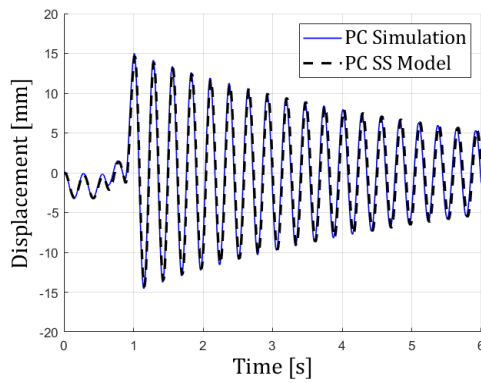


c)

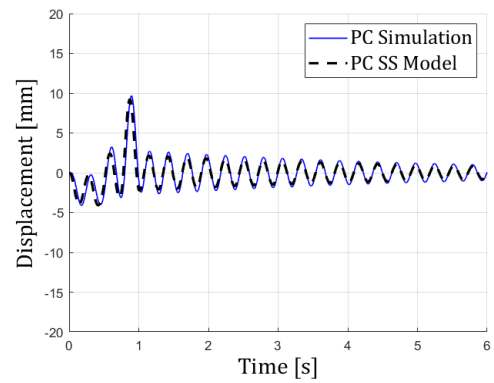


d)

Figure 4.6 Vibration responses for SCM-[0/90] with SCM-M1a, a) $1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, and d) $4t_{1h}$

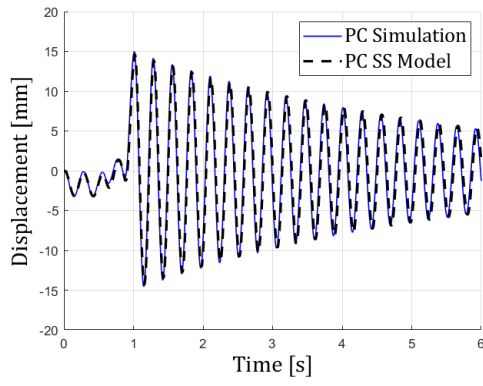


a)

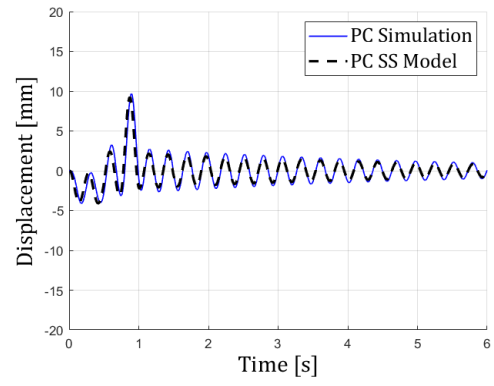


b)

Figure 4.7 Vibration responses for SCM-[45/-45] with SCM-M1b, a) $1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, and d) $4t_{1h}$



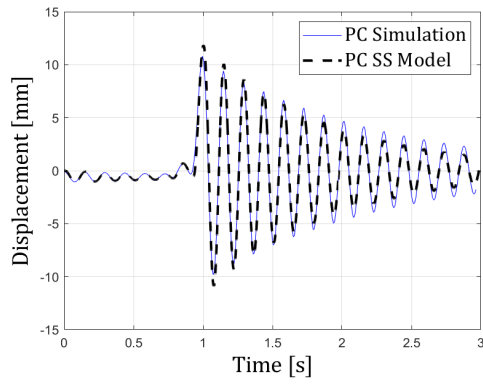
c)



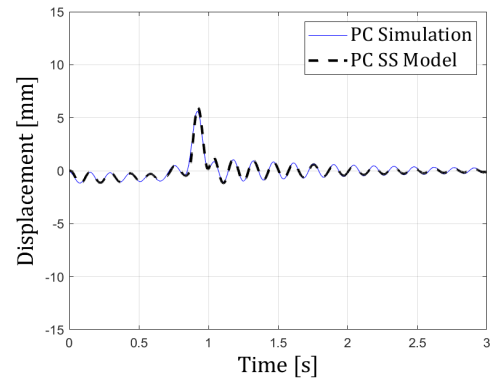
d)

Figure 4.7 continues

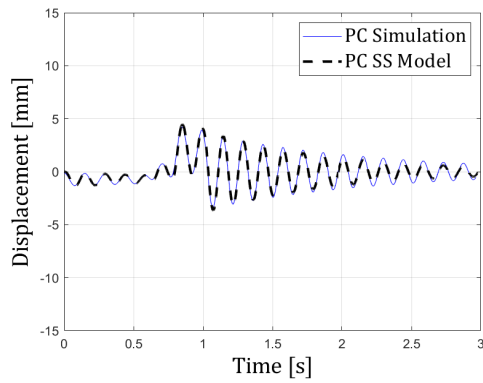
For the SCBM, the vibration responses are shown in Figure 4.8.



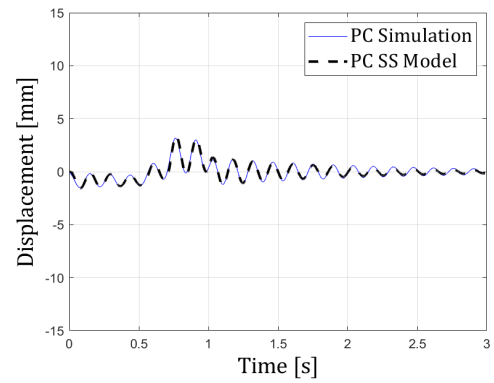
a)



b)



c)



d)

Figure 4.8 Vibration responses for SCBM for SCBM-M1b, a) $1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, and d) $4t_{1h}$

It is observed from the figures that the simulation and SS results are in very good agreement for the PC of the smart manipulators. For the PC, the SS results are verified with the simulations. Also, the residual vibration amplitudes are the lowest for $t_{dec} = 2t_{1h}$, while those are the highest for $t_{dec} = 1t_{1h}$.

The RMS values are determined for the residual vibration responses to evaluate the effectiveness of the PC. The RMS value, which is determined as the arithmetic means of squares of a set of n values in a vibration signal, is calculated as given in Equation 4.7.

$$\begin{aligned} y_{RMS} &= \sqrt{\frac{1}{n} \int_0^T y(t) dt} \\ &= \sqrt{\frac{1}{n} (y_1^2 + y_2^2 + \dots + y_n^2)} \end{aligned} \quad (4.7)$$

In order to calculate the RMS value of a vibration signal, the vibration signal in n numbers is selected and summed by squaring each value. Then, the arithmetic mean of these values is obtained and then, the RMS value is found by taking the square root.

Residual vibrations, which is the vibrations that occur after completing the movement of the smart manipulators, are considered with tabulation for evaluating the effectiveness of the PC. The smart manipulators complete the motion at $t = t_m$ due to the motion time. The window of residual vibrations is considered as $t_m < t_{rest} < t_{ss}$ in order to analyze the residual vibrations. The time of residual vibrations is chosen as $t_{rest} = 5$ s for the smart manipulators and $t_{rest} = 2$ s for the SCBM. Therefore, the RMS values is computed and listed in Table 4.8 and Table 4.9.

Table 4.8 and Table 4.9 give the RMS values of residual vibrations obtained with the simulation and SS analysis. Percent error values are calculated by using the RMS values in the PC. The RMS and error values are summarized in Table 4.8 and Table 4.9, respectively. As expected, the numerical errors are less than 5% for the smart manipulators in the PC. But, the error values of SCBM for $t_{dec} = 1t_{1h}$ remain relatively high within 10%. It is observed that the error values are within 10% to be accepted for

a good matching in the PC analysis.

Table 4.8 RMS values of SM, SCM-[0/90] and SCM-[45/-45] in the PC

Motion Profile	t_{dec}	Simulation	SS Model	Error
		RMS [mm]	RMS [mm]	[%]
SM-M1	$1T_{1h}$	4.0400	4.0912	-1.267
	$2T_{1h}$	0.4405	0.4327	0.193
	$3T_{1h}$	1.7444	1.7631	-0.462
	$4T_{1h}$	0.6321	0.6428	-0.264
SM-M2	$1T_{1h}$	4.4261	4.4492	-0.521
	$2T_{1h}$	0.4836	0.4644	0.433
	$3T_{1h}$	1.9014	1.8433	1.312
	$4T_{1h}$	0.4066	0.4235	-0.381
SCM-M1a	$1T_{1h}$	3.2128	3.2021	0.333
	$2T_{1h}$	0.6475	0.6343	0.410
	$3T_{1h}$	1.3202	1.3306	-0.323
	$4T_{1h}$	0.8313	0.8207	0.329
SCM-M2a	$1T_{1h}$	3.7962	3.7841	0.318
	$2T_{1h}$	0.6458	0.6559	-0.266
	$3T_{1h}$	1.5612	1.5045	1.493
	$4T_{1h}$	0.7819	0.7950	-0.345
SCM-M1b	$1T_{1h}$	4.8176	4.8771	-1.235
	$2T_{1h}$	1.1156	1.2065	-1.886
	$3T_{1h}$	2.2793	2.2819	-0.054
	$4T_{1h}$	2.5960	2.5600	0.747
SCM-M2b	$1T_{1h}$	5.8091	5.8414	-0.556
	$2T_{1h}$	1.0051	0.9920	0.225
	$3T_{1h}$	2.4453	2.4537	-0.144
	$4T_{1h}$	1.3871	1.4014	-0.246

The error values of t_{dec} are considered as reference values for comparison. For SM-M1, the errors are found as -1.267% , 0.193% , -0.462% , and -0.264% , for the $t_{dec} = 1t_{1h}$, $2t_{1h}$, $3t_{1h}$ and $4t_{1h}$, respectively. However, maximum error value for SM-M2 are found as 1.312% for $t_{dec} = 3t_{1h}$. For the PC, it is observed that the simulation and SS results were matched very well for the trapezoidal and triangular motion profiles.

Table 4.9 RMS values of SCBM in the PC

Motion Profile	Payloads m_p (kg)	t_{dec} [s]	Simulation	SS Model	Error
			RMS [mm]	RMS [mm]	[%]
SCBM-M1a	0.00	$1t_{1h}$	0.5936	0.5813	2.072
		$2t_{1h}$	0.1299	0.1014	4.801
		$3t_{1h}$	0.1716	0.1873	-2.644
		$4t_{1h}$	0.0741	0.0680	1.027
		$5t_{1h}$	0.0931	0.1047	-1.954
SCBM-M1b	1.10	$1t_{1h}$	4.7000	4.2076	10.476
		$2t_{1h}$	0.5282	0.5120	0.344
		$3t_{1h}$	1.6000	1.4961	2.210
		$4t_{1h}$	0.5858	0.5434	0.902
		$5t_{1h}$	1.0000	0.9606	0.838
SCBM-M1c	2.11	$1t_{1h}$	7.9000	7.1869	9.026
		$2t_{1h}$	0.6191	0.6997	-1.020
		$3t_{1h}$	3.2000	3.0370	2.063
		$4t_{1h}$	0.7792	0.8190	-0.503
		$5t_{1h}$	2.8000	2.5609	3.026

In order to evaluate the effectiveness of the PC, reduction ratios according to the reference values of $t_{dec} = 1t_{1h}$ are calculated to indicate the reduction ratio over the RMS values of the PC. For example, the reduction ratios are calculated as given in Equation 4.8.

$$R_{i*t_{1h}}^{PC} = \frac{RMS_{1t_{1h}}^{PC} - RMS_{i*t_{1h}}^{PC}}{RMS_{1t_{1h}}^{PC}} * 100 \quad i = 1, 2, 3, 4 \quad (4.8)$$

where, $R_{i*t_{1h}}^{PC}$ is the reduction ratios of i^{th} for the deceleration time $t_{dec} = i * t_{1h}$ in the PC. $RMS_{1t_{1h}}^{PC}$ is the RMS value for $t_{dec} = 1t_{1h}$. For the SM-M1, the example reduction ratio $R_{2t_{1h}}^{PC}$ for $t_{dec} = 2t_{1h}$ is computed for the simulation result given in Table 4.8 as follows

$$RMS_{1t_{1h}}^{PC} = 4.0400, \quad RMS_{2t_{1h}}^{PC} = 0.4405, \quad \text{read from Table 4.8}$$

(4.9)

$$R_{2*1t_{1h}}^{PC} = \frac{RMS_{1t_{1h}}^{PC} - RMS_{2t_{1h}}^{PC}}{RMS_{1t_{1h}}^{PC}} * 100 = \frac{4.0400 - 0.4405}{4.0400} * 100 = 89.09\%$$

As indicated in Equation 4.9, the reduction ratio of simulation result for $t_{dec} = 2t_{1h}$ are calculated as 89.09%. The reduction ratios according to the reference values of $t_{dec} = 1t_{1h}$ are calculated to indicate the reduction over the RMS values of the PC. Comparison of the reduction ratios for the smart manipulators are shown in Figure 4.9, Figure 4.10, and Figure 4.11.

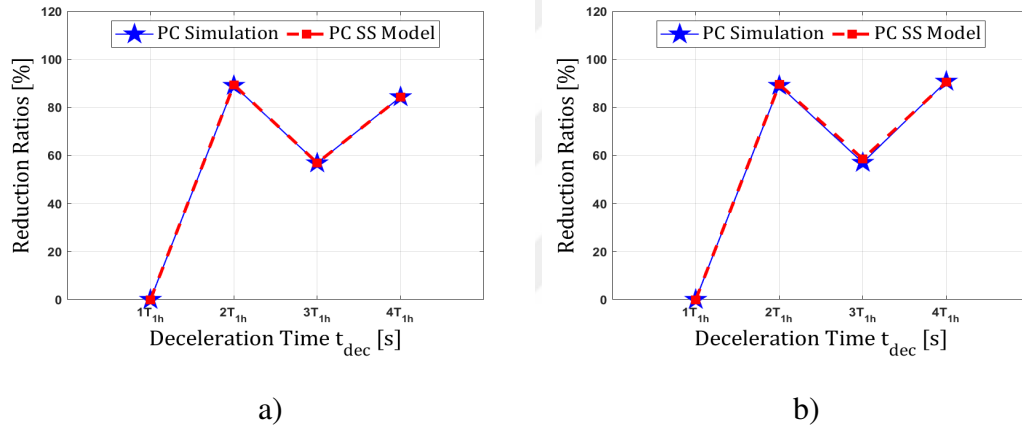


Figure 4.9 Change of reduction ratios of SM, a) SM-M1, b) SM-M2

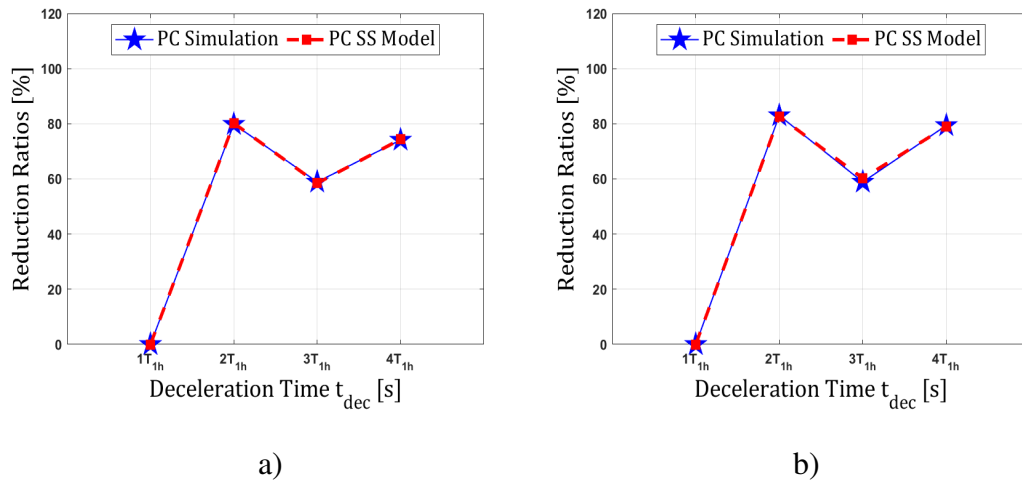
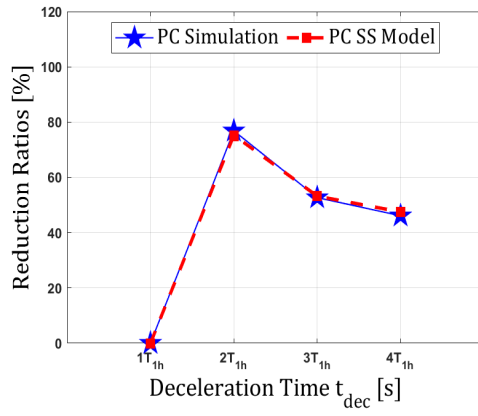
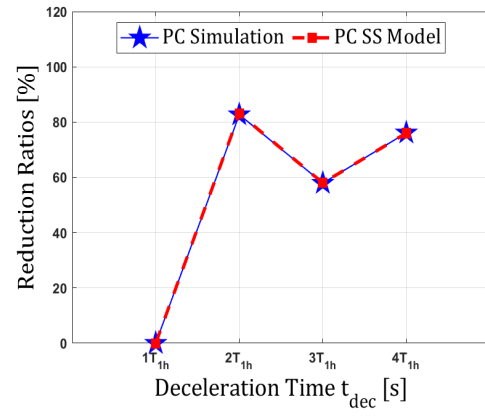


Figure 4.10 Change of reduction ratios of SCM-[0/90] and SCM-[45/-45], a) SCM-M1a, b) SCM-M2a, c) SCM-M1b and d) SCM-M2b

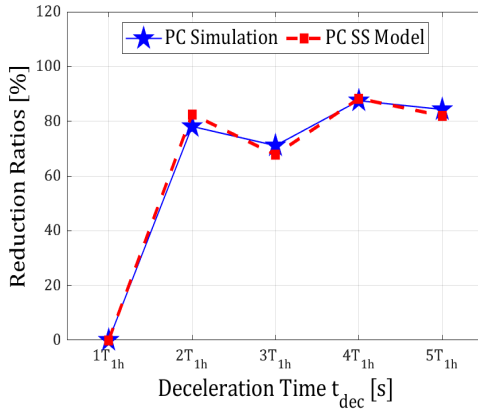


c)

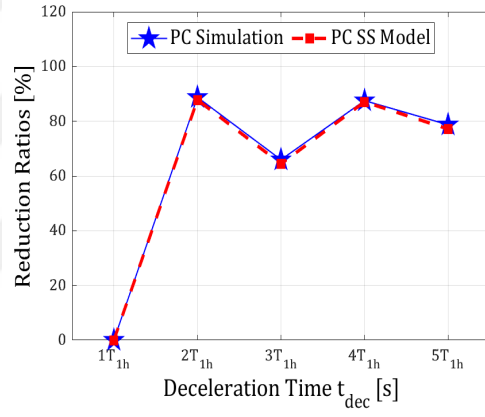


d)

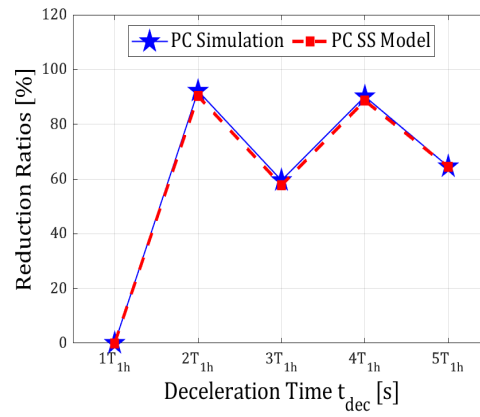
Figure 4.10 continues



a)



b)



c)

Figure 4.11 Change of reduction ratios of SCBM, a) SCBM-M1a, b) SCBM-M1b, and c) SCBM-M1c

The comparison of reduction ratios for the simulation and SS analysis are graphically shown in the figures above for the PC. The amplitudes of residual vibrations are the highest for the $t_{dec} = 1t_{1h}$, while those are the lowest for the $t_{dec} = 2t_{1h}$ and $4t_{1h}$. Also, for the $t_{dec} = 3t_{1h}$, are higher than $t_{dec} = 2t_{1h}$ and $4t_{1h}$.

The reduction ratios for $t_{dec} = 2t_{1h}$ are approximately 90% for SM-M1 and SM-M2 while those for SCM-M1a, SCM-M2a, SCM-M1b and SCM-M2b are found as 80%. For $t_{dec} = 4t_{1h}$, the reduction ratios for the SCBM are around 90% for SCBM-M1a, 88% for SCBM-M1b and 90% for SCBM-M1c, respectively.

As seen from the figures, it is observed that the simulation results are matched well with the SS results. After this verification, only the SS results will be compared with the experimental PC results of the smart manipulators in Chapter-7.

4.6 Comparison of Elapsed Time in Simulation and State-Space Analysis

The simulation of the smart manipulators can be conducted with commercial software such as ABAQUS, ANSYS, ADAMS, etc.. The elapsed time for the transient analysis may vary depending on the number of nodes and elements of FE model. Consequently, the analysis times can take days for some FE models. For the simulations performed in ANSYS and SS analysis realized in MATLAB, the example elapsed analysis times can be computed as given in Figure 4.12.

For the smart manipulators, the elapsed analysis times are compared for $t_{dec} = 1t_{1h}$ and $2t_{1h}$ as given in Figure 4.13. Also, the computer properties utilized for the transient analysis is given in Figure 4.13.

All transient analyses are conducted on the same computer. Four different smart manipulators are examined with the simulation and SS analysis for two different motion profiles. For $t_{dec} = 1t_{1h}$, it is observed that the analysis times with the SS analysis according to the simulation are reduced approximately from 475, 99, 52, and 234 minutes to 0.9, 0.8, 0.6, and 2.4 seconds for the SM, SCM-[0/90], SCM-[45/-45]

and SCBM, respectively. The transient analysis with the SS models will be performed in a short time.

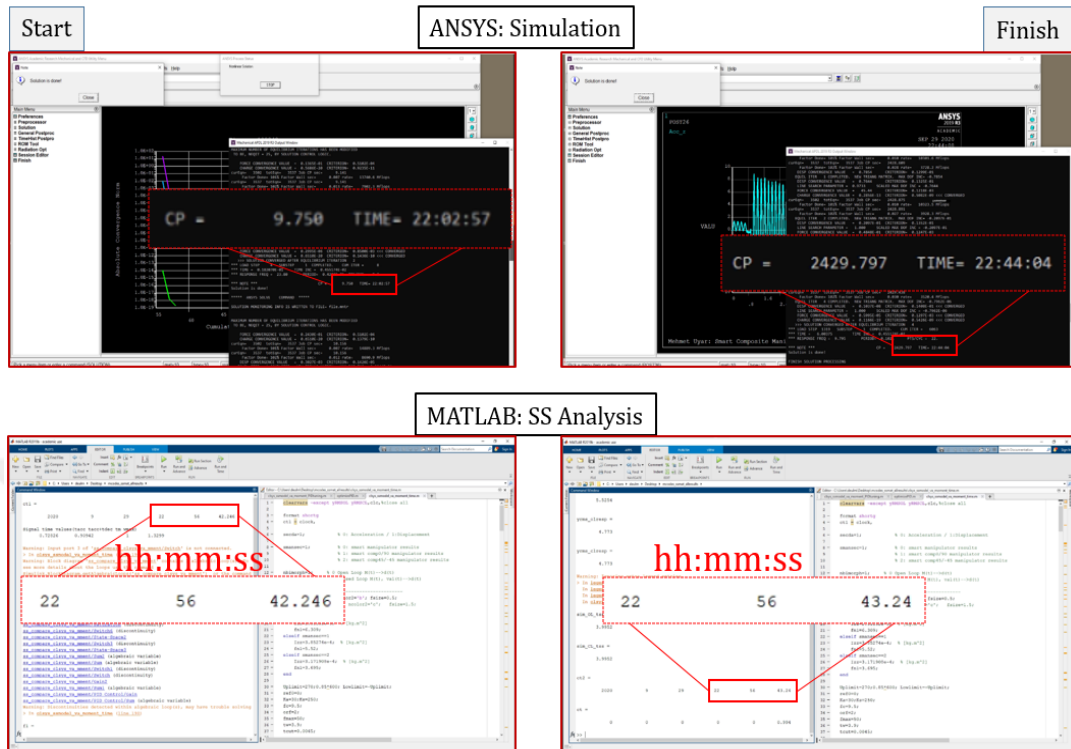


Figure 4.12 Example ANSYS and MATLAB pictures in the transient analysis

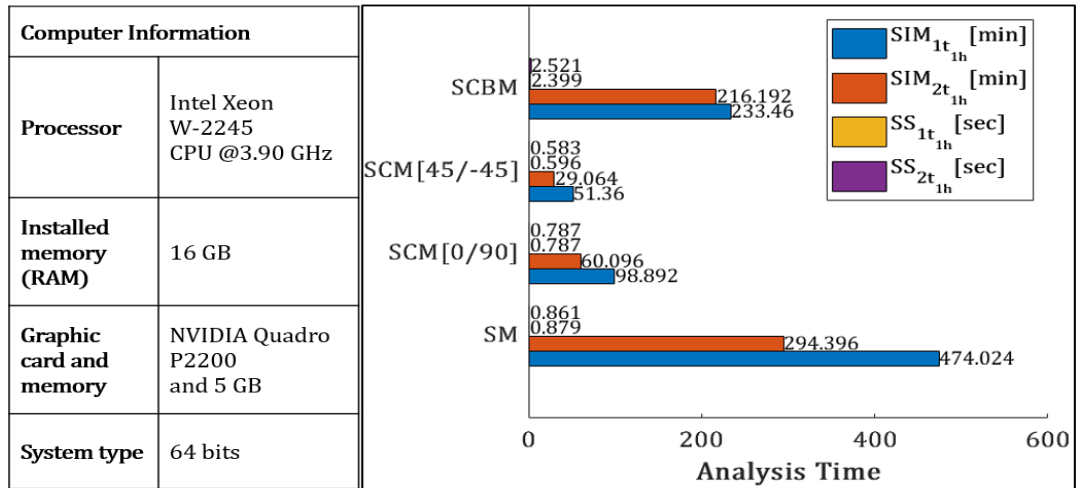


Figure 4.13 Comparison of elapsed times in PC analysis

CHAPTER FIVE

ACTIVE AND PASSIVE CONTROLS OF FLEXIBLE SMART MANIPULATORS

5.1 Introduction

In this chapter, the AC is performed with the simulation and SS analysis using the PD and FSF controls. The PD control with the displacement feedback (DF) is applied to the smart manipulators with the motion profiles considered in the PC for further suppressing residual vibrations at the tip point. The simulation and SS analysis results are compared using the PD controller in which the gains are found with the trial and error method. After comparison of the results, design techniques are studied using the SS models to optimize the PD and FSF controller gains. Ziegler-Nichols (Z-N), Integrator Plus Dead Time (IPDT), and Chien-Hrones-Reswick (CHR) tuning methods are utilized for the PD controller design while two state feedback design criteria are used for the FSF control. The PD parameters are settled with open-loop responses of the smart manipulators. Later, the PD gains are optimized by considering the maximum operation voltage and minimum error values of fitness functions which are Integral of Squared Error (ISE) and Integral of Absolute Magnitude of Error (IAE) functions. The optimized AC results together with the PC of the smart manipulators are presented.

5.2 Proportional-Integral-Derivative (PID) Control

The closed-loop simulations are performed in ANSYS, the closed-loop SS analyses are performed in MATLAB. The PD control is implemented with the DF for the AC.

5.2.1 Closed-loop Simulation

The closed-loop block diagram of AC is shown in Figure 5.1.

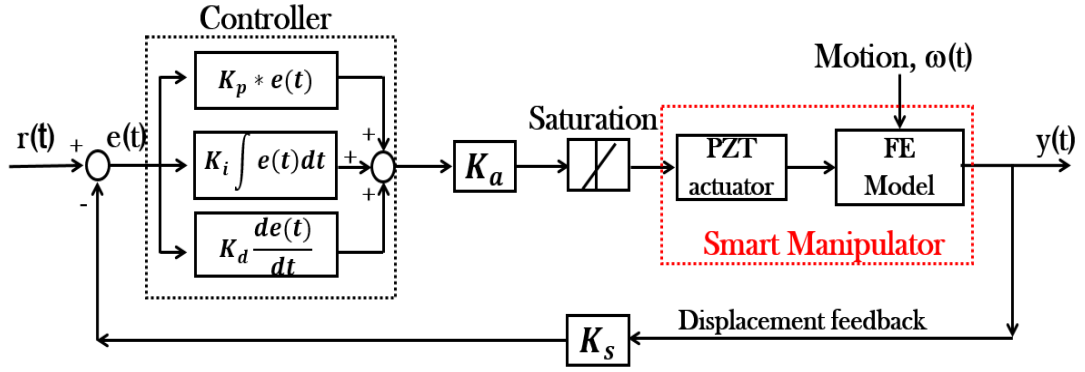


Figure 5.1 Closed-loop block diagram for the simulation

The closed-loop simulation is performed using the DF in ANSYS. For the closed-loop simulation, a script based on macro code for the transient analysis is developed and integrated into the FE models of the smart manipulators for the simulations. The motion profiles are applied to the smart manipulators. The control-loop simulation begins after the smart manipulators have completed their movement to reduce the residual vibrations. Knowing the input at $t = 0$, the feedback signal $v_f(i)$ at the time step i is calculated at the step $i = 1$. The feedback signal is selected as the displacement signal $y(t)$ in bending direction for the DF. For the DF, $y(t)$ is the tip displacement signal of the smart manipulators in the z -direction. $y(t)$ is read from node n_d at the tip point corresponding to the distance d_d . n_d and d_d is the node number and the distance of node n_d at the tip point of the smart manipulators, respectively. $v_f(i)$ is found by multiplying the sensor gain K_s by $y(i)$ for the DF. The error signal $e(t)$ at the step $i = 1$ can be calculated as $v_e(i) = v_r(i) - v_f(i)$. $v_e(t)$ is first multiplied by K_p , its derivative is multiplied by K_d and the control signal $u(i)$ is found by summing these three signals. The integral values are found by summing error signals at each time step. The derivative values are approximated by dividing the difference between the values in the i^{th} and the previous time steps by the time step. The time-dependent control signal $u(t)$, which is the output of PD controller shown in the block diagram is given in Equation 5.1.

$$u(t) = K_p e(t) + K_d \frac{de(t)}{dt} \quad (5.1)$$

The actuation signal is obtained as $v_a(i) = K_a * u(i)$. The value v_a is applied to the coupled nodes where the PZT actuators are located to reduce the residual vibration of the smart manipulators. The closed-loop simulation is continued until the steady-state response is reached.

Saturation is considered for $v_a(t)$ due to the limitation of piezoelectric actuators to verify the experiments. Set value for the saturation is selected. The saturation values depend on the thickness of the PZT actuators. The power amplification gain is chosen as $K_a = 30$. The sensor gain is taken as $K_s = 250$ for the DF. In their earlier work, the authors conducted experimental works on the AC of smart structures. The K_s and K_a values used in the simulation were based on these reference studies (Malgaca & Karagülle, 2009; Malgaca, 2010).

A macro code for the closed-loop control is developed to implement the AC in ANSYS. The example macro is given as follows

The macro code for the closed-loop simulation is embedded into the transient solution of the smart manipulators in the FE analysis. The AC is achieved with a loop. The loop starts with **do* and ends with **enddo* in the transient analysis. The variables *ym* and *err* are the feedback signal and error signal at a time step *dt*, respectively. *ym* is the tip displacement signal calculating with *dz* and *dx* at the node n_d for the DF for the smart manipulators. *thetam* is the angular displacement corresponding to θ_m . *ref* which is a reference signal, is taken as zero. By considering control gains (k_p , k_d) and taking the numerical integral and derivative of *err*, the control signal *u* is calculated. The integral values *sum* are found by summing *err* at each time step. The derivative values *dif* are found by dividing the difference between the values in the present and the previous time steps by the time step. Then, actuation signal *va* is obtained. The saturated actuation voltage is limited to **if – *endif* commands. The saturated *va* is applied to the nodes *nv1* and *nv2* of PZT actuators. The loop continues until reaching the number of samples *n* corresponding to the selected analysis time.

Table 5.1 Ansys/APDL macro code for the closed-loop control

```
vmax=270
*do,i,1,n
time,i*dt
solve
*get,dx,node,nr,u,x
*get,dz,node,nr,u,z
ym=dz*cos(thetam)-dx*sin(thetam)
ref=L*sin(thetam)
err=(ref-ym)*ks
sum=sum+err*dt
diff=(err-errp)/dt
u=kp*err+ki*sum+kd*diff
va=ka*u
*if,va,ge,vmax,then
va=vmax
*endif
*if,va,le,-vmax,then
va=-vmax
*endif
d,nv1,volt,va
d,nv2,volt,va
errp=err
*endif
*enddo
```

The closed-loop simulation is applied to the smart manipulators. For the AC, simulations are performed to confirm the SS analysis for the motion profiles.

5.2.2 Closed-loop State-Space Analysis

The closed-loop control is studied using the SS models to verify the simulation results. The closed-loop block diagram is shown in Figure 5.2.

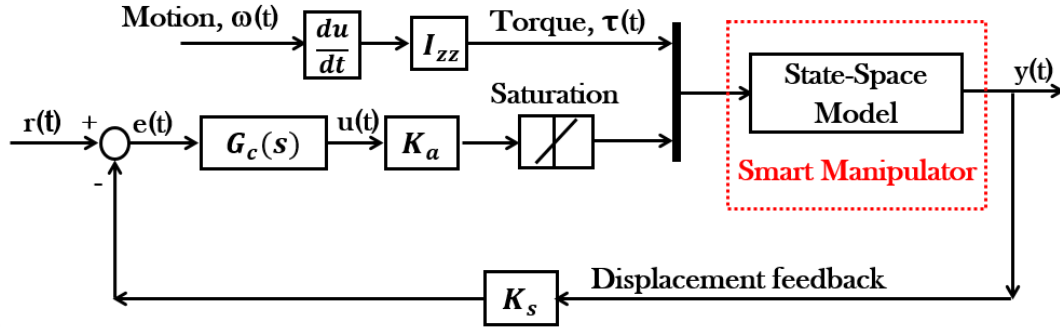


Figure 5.2 Closed-loop block diagram for the SS method

In the PD control, the DF is used as the feedback signal. The closed-loop block diagram is created in MATLAB. The equation of PID control is given by

$$G_c(s) = K_p \left(1 + \frac{1}{T_i s} + T_d s \right) \quad (5.2)$$

where, T_i and T_d are the integrator and derivative time constants, separately. The integral and derivative controller gains are found as $K_i = K_p/T_i$ and $K_d = K_p T_d$. K_i is equal to zero due to utilizing the PD controller.

In the closed-loop block diagram, the torque $\tau(t)$ of revolute joint and actuator voltages of the PZT actuators are chosen as the inputs of the system. $r(t)$ is the reference signal taken as zero to eliminate the residual vibrations of the tip point. As in the simulation, K_a and K_s are taken as 30, 250, respectively. For the AC, the PD controller is selected. $\tau(t)$ is applied to the revolute joint to drive the smart manipulators. The actuator voltage is used to minimize the residual vibrations. Since the motion profiles are defined as the velocity profiles, the motion profiles need to be modified to the torque curves. Therefore, the angular velocity $\omega(t)$ profiles are converted to $\tau(t)$ by multiplying the inertia of moment I_{zz} of the smart manipulators after taking the derivative of $\omega(t)$. After the movement of the smart manipulators is

completed, the closed-loop control with the DF begins in MATLAB. The feedback signal $v_f(t)$ is calculated as $v_f(t) = K_s y(t)$. The error signal $e(t)$ is found by subtracting the $v_f(t)$ from the reference signal $r(t)$. $e(t)$ is multiplied by K_p , its derivative is multiplied by K_d . Then, the control signal $u(t)$ is calculated by summing these three signals. The actuation signal is obtained by multiplying K_a with $u(t)$ and the actuation signal is applied to the PZT actuators after limited with the saturation. The closed-loop simulation continues until the steady-state response is reached to reduce the residual vibrations. Then, the closed-loop vibration responses are obtained.

5.2.3 Closed-loop Results

The closed-loop simulation and SS analysis results are presented with all motion profiles used in the PC for the AC. The AC results for SM-M1 are shown in Figure 5.3.

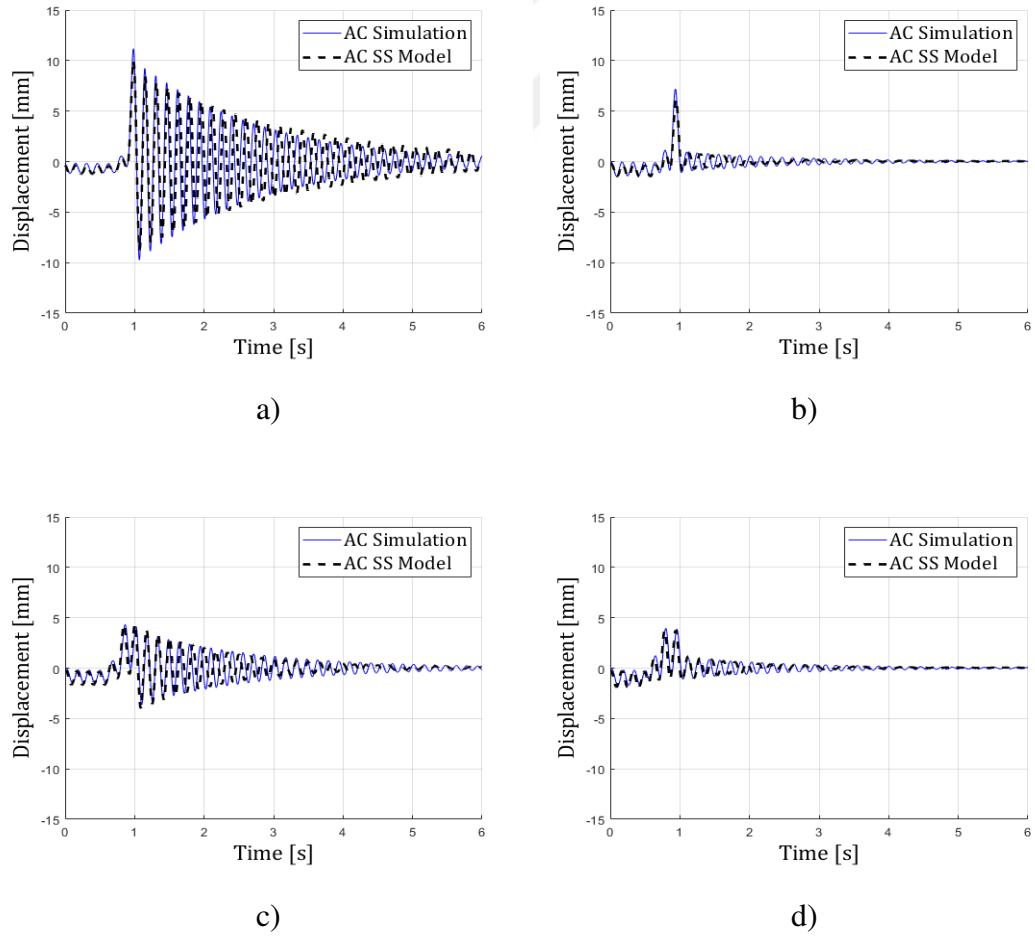


Figure 5.3 AC and PC vibration responses for SM-M1, a) $t_{dec} = 1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, and d) $4t_{1h}$

For the SCM-[0/90], SCM-[45/-45] and SCBM, the AC results are shown in Figure 5.4, Figure 5.5 and Figure 5.6 for the SCM-[0/90], SCM-[45/-45] and SCBM, respectively.

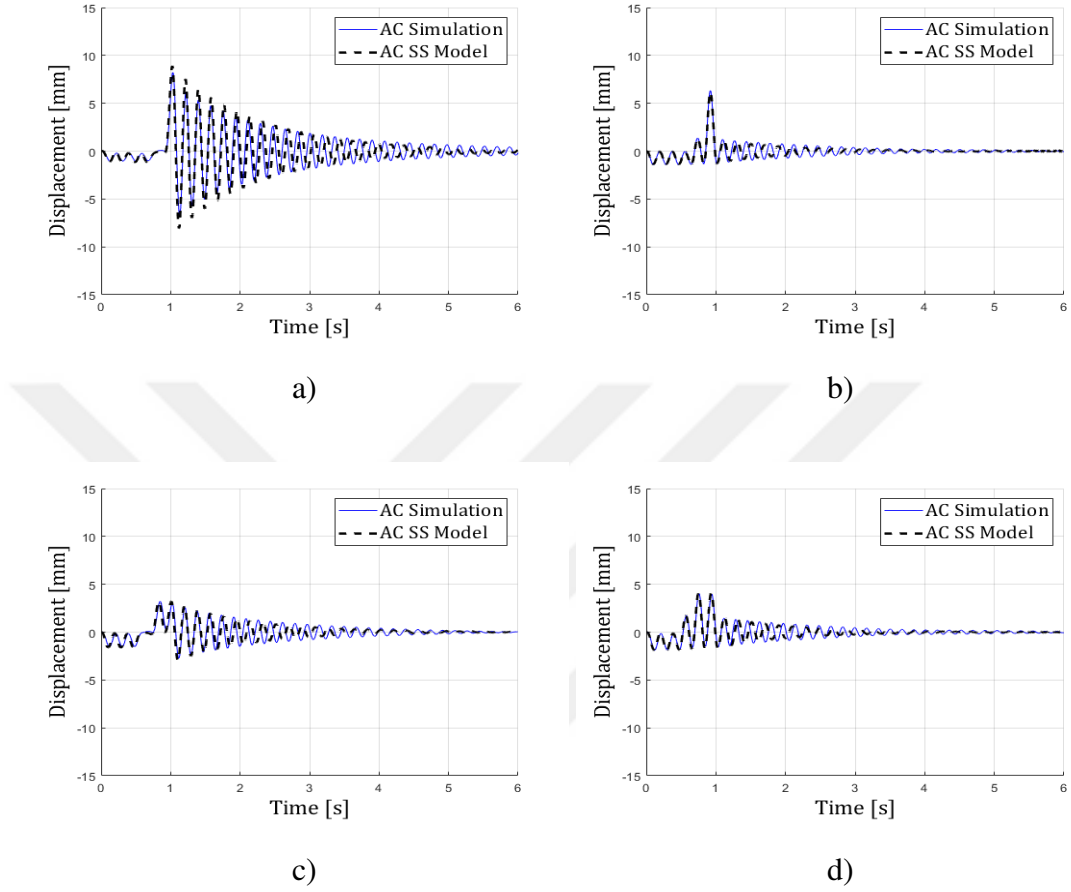


Figure 5.4 AC and PC vibration responses of SCM-[0/90] for SCM-M1a, a) $t_{dec} = 1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, and d) $4t_{1h}$

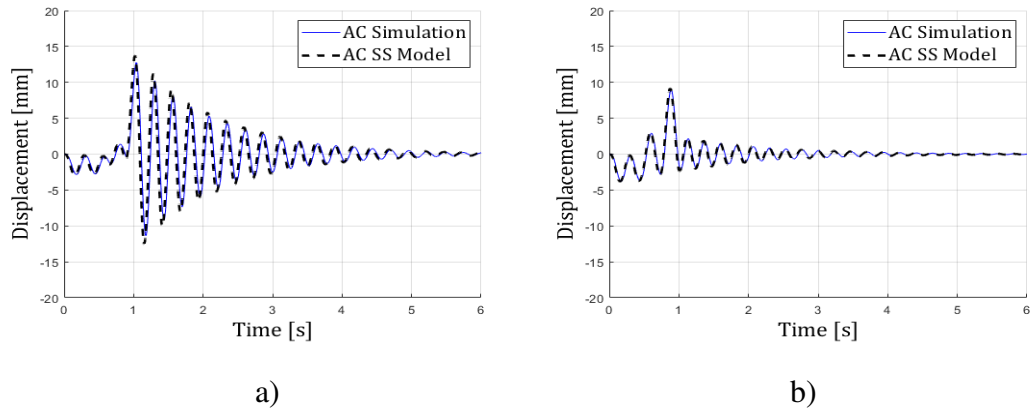
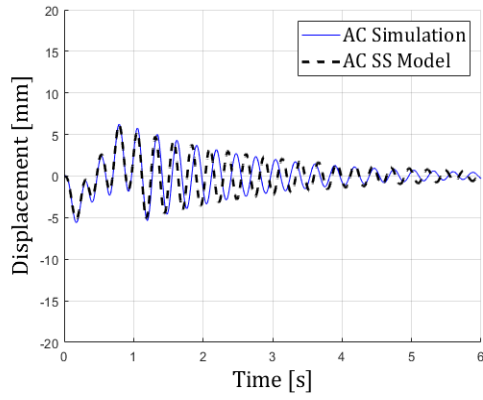
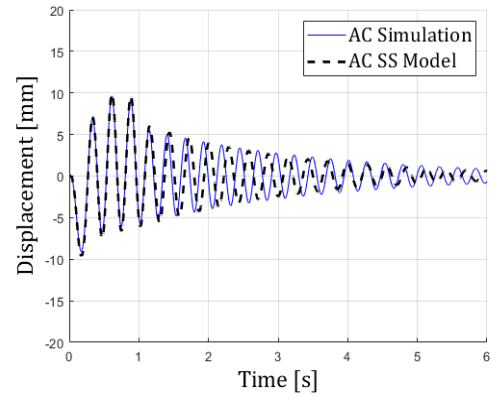


Figure 5.5 AC and PC vibration responses of SCM-[45/-45] for SCM-M1b, a) $t_{dec} = 1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, and d) $4t_{1h}$

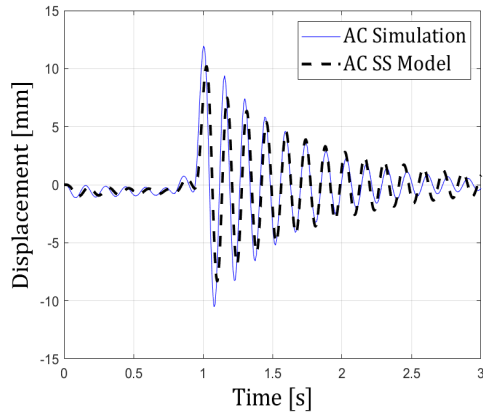


c)

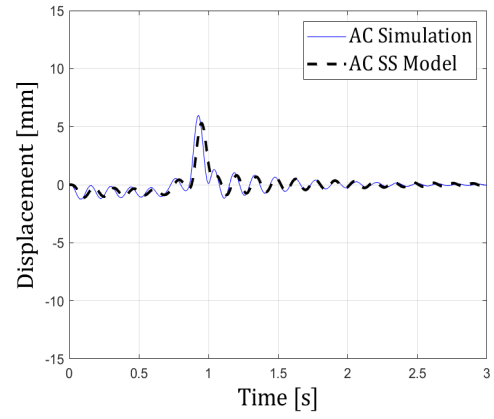


d)

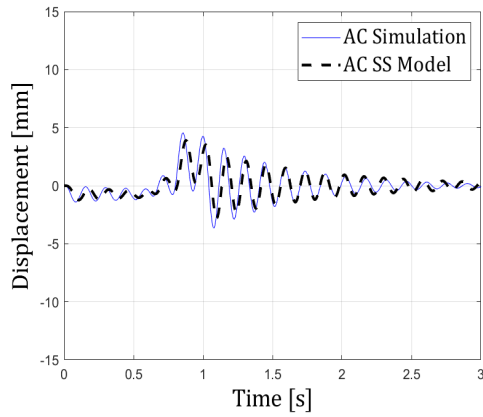
Figure 5.5 continues



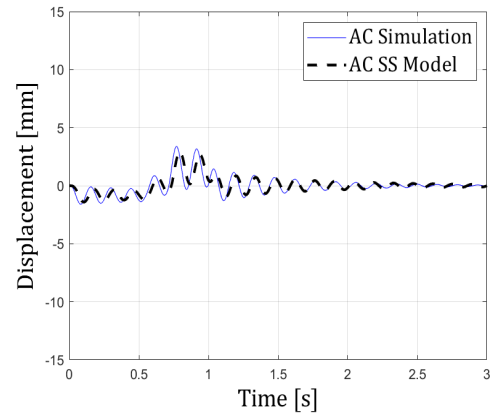
a)



b)



c)



d)

Figure 5.6 AC and PC vibration responses of SCBM for a) $t_{dec} = 1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, and d) $4t_{1h}$

As seen from the figures, the closed-loop simulation results are in good agreement with the closed-loop SS analysis results. The SS models obtained with the model extraction method are also verified for the AC. The RMS values of residual vibrations are computed to investigate the performance of AC implementing over the PC. The comparison of RMS values is listed in Table 5.2 for the SM, SCM-[0/90] and SCM-[45/-45], and Table 5.3 for the SCBM, respectively, with various PD controller gains.

Table 5.2 RMS values of SM, SCM-[0/90] and SCM-[45/-45] in the AC

Motion Profile	t_{dec} [s]	K_p	K_d	Simulation	SS Model	Error
				RMS [mm]	RMS [mm]	[%]
SM-M1	$1t_{1h}$	1	0.025	2.9613	3.0380	-2.590
	$2t_{1h}$	4	0.025	0.2953	0.2984	-0.104
	$3t_{1h}$	3	0.025	0.9519	0.9321	0.668
	$4t_{1h}$	4	0.025	0.3206	0.2575	2.130
SM-M2	$1t_{1h}$	1	0.025	3.4110	3.4334	-0.656
	$2t_{1h}$	4	0.025	0.2951	0.2404	1.603
	$3t_{1h}$	3	0.025	1.0442	1.1072	-1.847
	$4t_{1h}$	4	0.025	0.2823	0.2457	1.073
SCM-M1a	$1t_{1h}$	1	0.025	1.6696	1.6407	1.731
	$2t_{1h}$	4	0.025	0.1696	0.1559	0.820
	$3t_{1h}$	2	0.025	0.5074	0.4994	0.479
	$4t_{1h}$	4	0.025	0.2229	0.2183	0.275
SCM-M2a	$1t_{1h}$	1	0.025	1.8726	1.8904	-0.950
	$2t_{1h}$	4	0.025	0.1995	0.1997	-0.010
	$3t_{1h}$	2	0.025	0.5315	0.5467	-0.811
	$4t_{1h}$	4	0.025	0.2036	0.1831	1.094
SCM-M1b	$1t_{1h}$	1	0.025	2.8069	2.7626	1.578
	$2t_{1h}$	4	0.025	0.4690	0.4572	0.420
	$3t_{1h}$	3	0.025	0.9396	0.9860	-1.653
	$4t_{1h}$	3	0.025	0.9875	1.0419	-1.938
SCM-M2b	$1t_{1h}$	1	0.025	3.5989	3.5425	1.567
	$2t_{1h}$	4	0.025	0.3690	0.4033	-0.953
	$3t_{1h}$	3	0.025	1.0623	1.1125	-1.394
	$4t_{1h}$	3	0.025	0.6047	0.5908	0.386

Table 5.3 RMS values of SCBM in the AC

Motion Profile	t_{dec}	K_p	K_d	Simulation	SS Model	Error
				RMS [mm]	RMS [mm]	[%]
SCBM-M1a	$1t_{1h}$	13.05	0.21	0.5126	0.5043	1.619
	$2t_{1h}$			0.1240	0.1334	-1.833
	$3t_{1h}$			0.1505	0.1482	0.448
	$4t_{1h}$			0.0668	0.0681	-0.253
	$5t_{1h}$			0.0836	0.0826	0.195
SCBM-M1b	$1t_{1h}$	14.26	0.23	3.500	3.7377	-6.791
	$2t_{1h}$			0.3904	0.4244	-0.971
	$3t_{1h}$			1.200	1.2414	-1.182
	$4t_{1h}$			0.4343	0.4505	-0.462
	$5t_{1h}$			0.7358	0.7383	-0.071
SCBM-M1c	$1t_{1h}$	12.01	0.19	6.100	6.2250	-2.049
	$2t_{1h}$			0.4901	0.5174	-0.447
	$3t_{1h}$			2.500	2.6127	-1.847
	$4t_{1h}$			0.6172	0.6231	-0.096
	$5t_{1h}$			2.200	2.2852	-10.659

Table 5.2 and Table 5.3 give the comparison of the RMS values of residual vibrations. The error values are presented. When the error values are evaluated, it is observed that the results are in good agreement. The error values are within 10%, which is an acceptable value for the AC. For instance, the error values for SM-M1 given in Table 5.2 are found as approximately -2.6% , -0.1% , 0.7% and 2.2% for $t_{dec} = 1t_{1h}$, $2t_{1h}$, $3t_{1h}$, and $4t_{1h}$, respectively. The effectiveness of the PD control is tested for various controller gains in the AC.

Firstly, the controller gains are found with the trial and error method to confirm the simulation and SS results. Then, the controller gains are optimized by using the maximum actuation voltage. The PZT actuators may be damaged in practice when the maximum actuator voltage is exceeded. So, saturation is applied to limit the actuation voltage as given in closed-loop block diagrams.

After verifying the simulation results with the SS analysis in the AC, the design techniques are studied using the SS models to optimize the controller gains in the next section.

5.3 Controller Design

The controller design techniques are studied to optimize the gains of PD and FSF controls. In this section, the PD tuning methods and design criteria are used for the controller design.

5.3.1 *PID Controller Design Using SS Models*

The PID control gains can be found with the design techniques (Xue et al., 2007) and trial and error method. Also, the PID design methods are classified as closed-loop and open-loop methods. For the closed-loop method, the tuning techniques are designed automatically the controller gains during the closed-loop while open-loop methods are manually adjusted with the vibration response of plant in open-loop.

In this study, the PD parameters are found with Z-N, CHR and IPDT tuning methods (Xue et al., 2007) for the open-loop tuning techniques. Z-N method is used to tune the controller gains for the smart manipulators while CHR and IPDT methods are only applied to the SCBM. Also, IAE and ISE fitness functions are used for the SM, SCM-[0/90] and SCM-[45/-45] to optimize the values of PD gains. For the SCBM, the controller gains are obtained by optimizing the maximum actuation voltage of PZT actuators.

For the open-loop tuning method, Figure 5.7 gives the flowchart of tuning method to find the tuning coefficients.

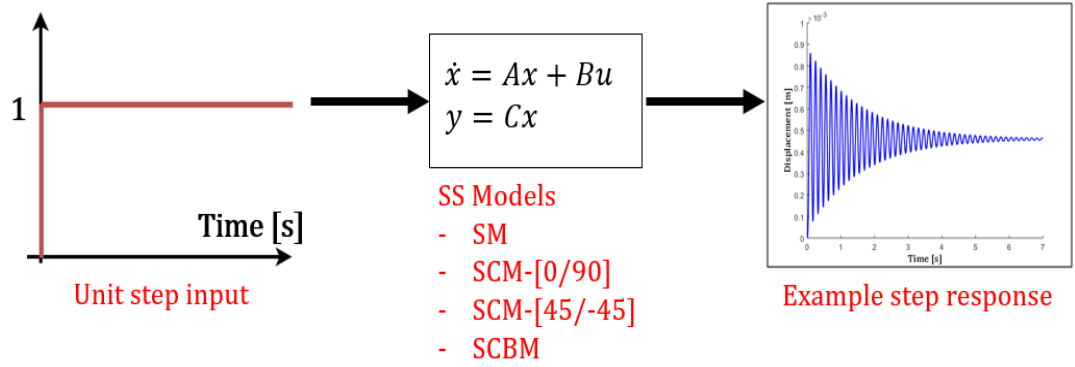


Figure 5.7 Flowchart for PD tuning coefficients

The step responses for the smart manipulators are analyzed to extract the tuning parameters K , L and T . In order to determine the tuning parameters, the step force is applied as unit step input of $u_{ss} = 1$ N to the node at the endpoint. The tip displacement is obtained from the node at the tip of the smart manipulators. The tuning parameters are found using the step response and its tangent. The obtained tuning parameters are given in Table 5.4.

Table 5.4 PID Tuning coefficients

Manipulators	K	L	T
SM	0.000970	0.014332	0.021841
SCM-[0/90]	0.004667	0.016441	0.026031
SCM-[45/-45]	0.003877	0.024414	0.043239
SCBM	0.000457	0.015700	0.029800

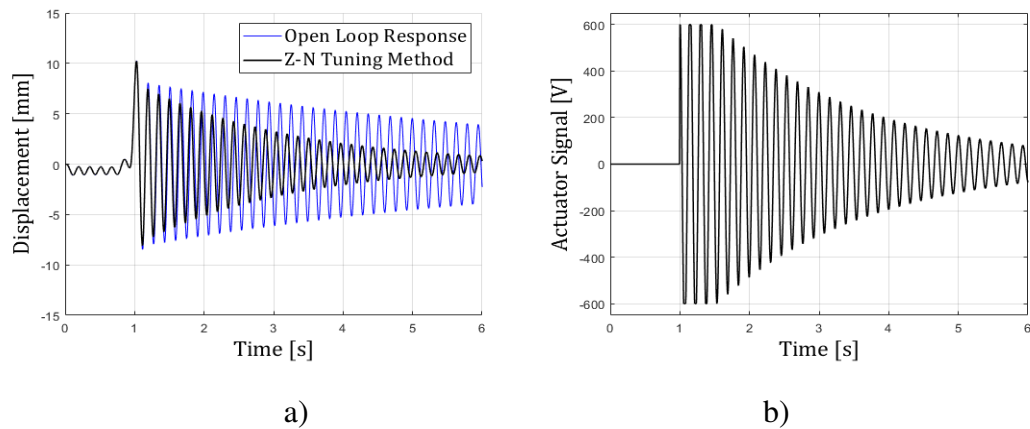
The PD controller gains are found with formulas given in Table 5.5 by using the tuning parameters listed in Table 5.4.

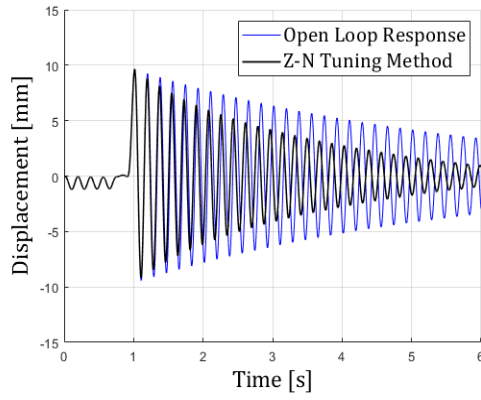
Table 5.5 Formulas of PID parameters and gains

Manipulators	Tuning Techniques	K_p	T_i	T_d	$m_p, (\text{Kg})$	K_p	K_d
SM	Z-N Method	$\frac{1.2}{a}$	$2L$	$\frac{L}{2}$	0	7.53	0.053
SCM-[0/90]	Z-N Method				0	1.62	0.013
SCM-[45/-45]	Z-N Method				0	2.19	0.026
SCBM	Z-N Method				0	0.602	0.0014
					1.11	0.667	0.0052
					2.10	0.426	0.0045
	CHR Method	0	0.602	0.0012			
		1.11	0.667	0.0043			
		2.10	0.426	0.0038			
	IPDT Method	0	13.05	0.2155			
		$\frac{1.37}{KL}$	1.49L	0.59L	1.11	14.26	0.2356
					2.10	12.01	0.1985

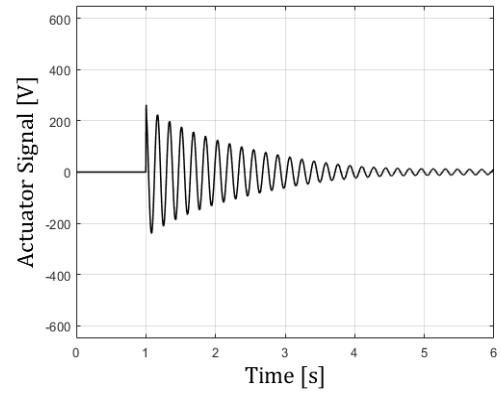
Where, $a = KL/T$. The parameter K is found from the steady-state values of response and input as $K = y_{ss}/u_{ss}$. The starting values of input and response are zero. Then, the controller gains for the smart manipulators with various payloads are calculated and listed in Table 5.5.

For the smart manipulators, the closed-loop vibration and actuation responses for $t_{dec} = 1t_{1h}$ to investigate the effectiveness of the PD tuning method are shown in Figure 5.8, Figure 5.9, Figure 5.10 and Figure 5.11.

Figure 5.8 Tuned responses of SM for $t_{dec} = 1t_{1h}$ a) vibration, b) actuation voltage

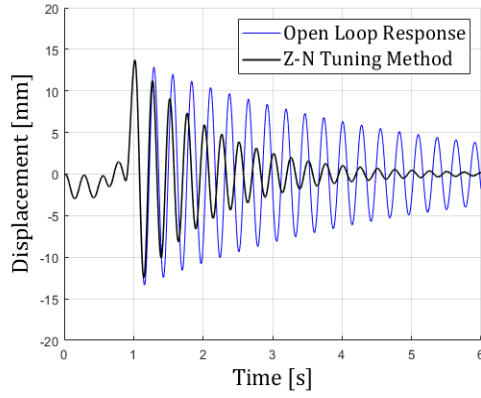


a)

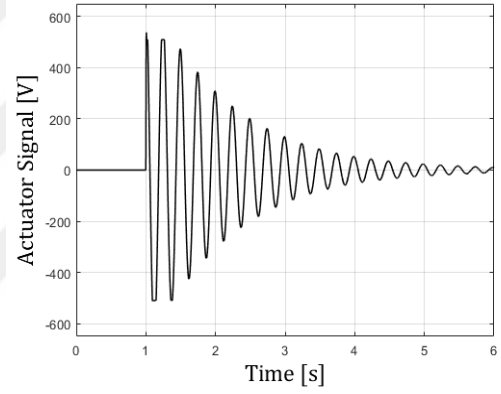


b)

Figure 5.9 Tuned responses of SCM-[0/90] for $t_{dec} = 1t_{1h}$ a) vibration, b) actuation voltage

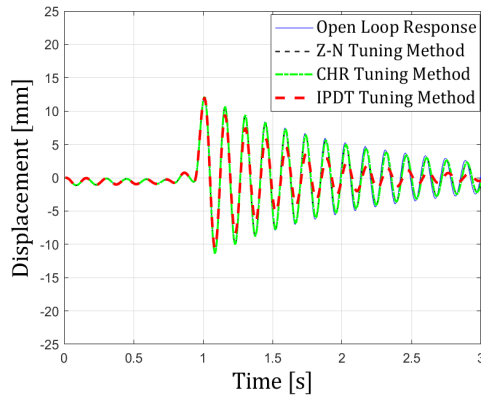


a)

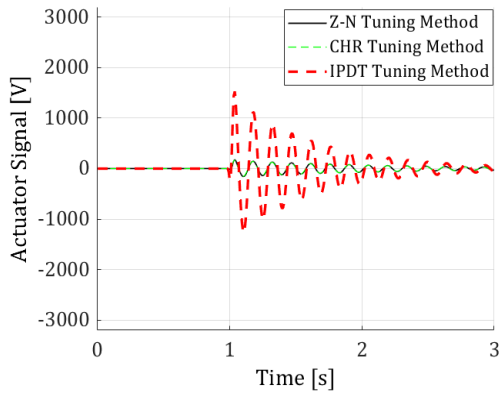


b)

Figure 5.10 Tuned responses of SCM-[45/-45] for $t_{dec} = 1t_{1h}$ a) vibration, b) actuation voltage



a)



b)

Figure 5.11 Tuned responses of SCBM for $t_{dec} = 1t_{1h}$ a) vibration, b) actuation voltage

The vibration and actuation responses for the tuning methods are shown in Figure 5.8, Figure 5.9, Figure 5.10 and Figure 5.11 for the SM, SCM-[0/90], SCM-[45/-45] and SCBM, respectively. The maximum actuation voltage can be applied at about 600 V/mm for PZT-5H to each PZT actuator (Preumont, 2011). For the PID controller design, it must ensure that the maximum actuator voltages do not exceed 600 V per mm thickness of PZT actuator, which restricts the actuator voltage to ± 600 V for the SM, SCM-[0/90], and SCM-[45/-45] of 1 mm thickness and ± 3000 V for the SCBM of 5 mm thickness, and then, those values are determined as an upper limit for the control gain.

As seen from the figures, the PD controller gains determined by the PID tuning method can be optimized by considering the stability and maximum operating voltage to increase the actuator efficiency.

For the SM, SCM-[0/90] and SCM-[45/-45], minimum error criteria, which are IAE and ISE fitness functions, are used to optimize the gains of PD controller. Here, the fitness functions aim to minimize the error value of closed-loop response. Also, the smallest value of error criteria indicates that the error is minimized. Fitness functions as stated in Equation 5.3 are utilized to optimize the PD gains which supply the design criteria.

$$\begin{aligned} IAE &= \int_0^{\infty} |r(t) - y_f(t)| dt = \int_0^{\infty} |e(t)| dt \\ ISE &= \int_0^{\infty} (r(t) - y_f(t))^2 dt = \int_0^{\infty} (e(t))^2 dt \end{aligned} \quad (5.3)$$

The main objective of design requirements is to reduce the settling time, achieve the minimum steady-state error and the optimum controller gains with minimal actuator energy to adjust the PD gains.

The PD control is simulated by using the fitness functions with the design criteria. A MATLAB code to establish the fitness function is created to minimize the error value of closed-loop. The starting values are selected as the tuning PD gains given in Table 5.5 to optimize the K_p and K_d gains. The controller gains are found by increasing the initial

values in the loop, taking into account the minimum settling time and error values. The optimized PD gains are determined by considering the stability and operating actuator voltage. The maximum and minimum actuator voltages are selected as ± 600 V to compare the performance of SS results with the PD controller gains. For that reason, the actuator voltage in the closed-loop response is limited with the saturation. The optimized PD gains for IAE and ISE functions are found for the SM, SCM-[0/90], and SCM-[45/-45]. In order to evaluate the performance of optimized PD gains, the PC and AC vibration responses are shown in Figure 5.12, Figure 5.13 and Figure 5.14.

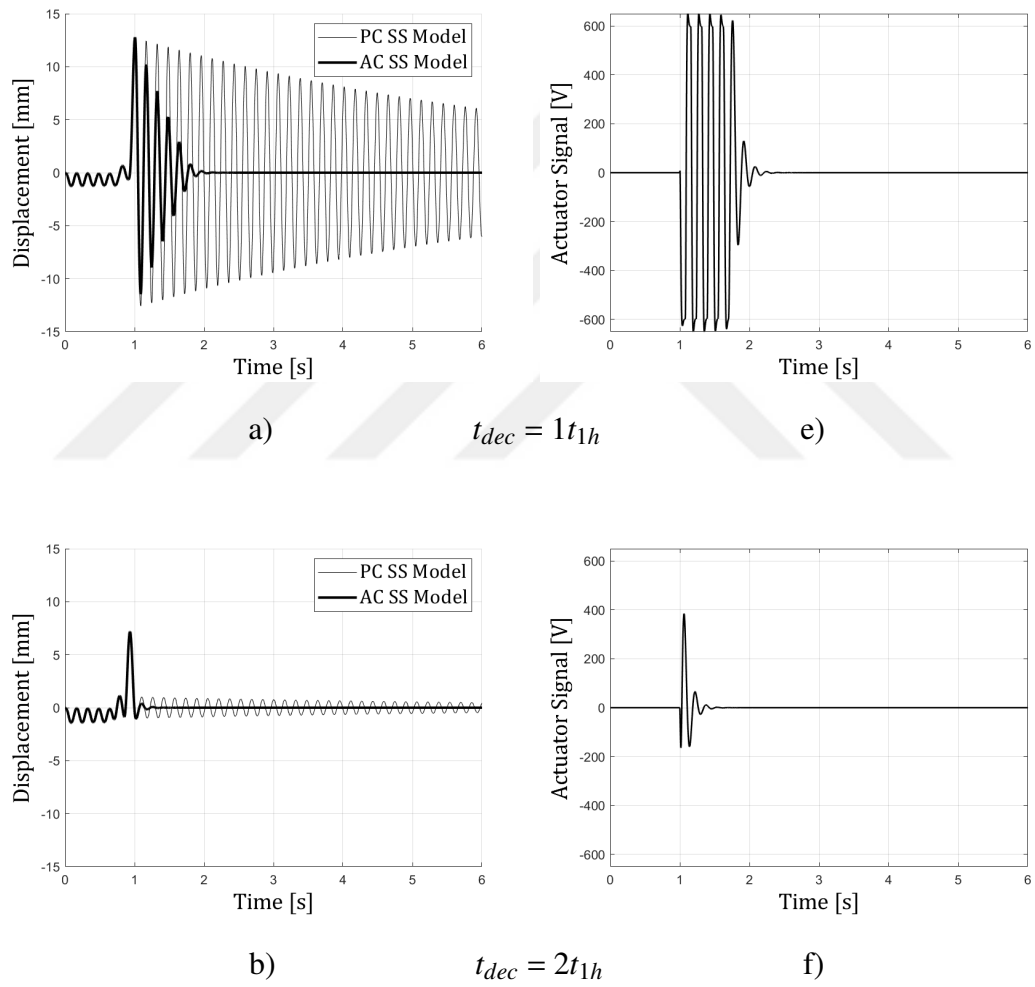
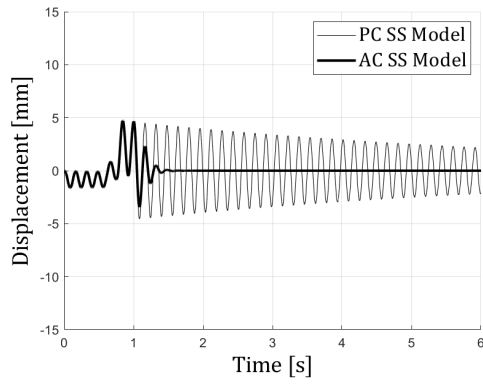
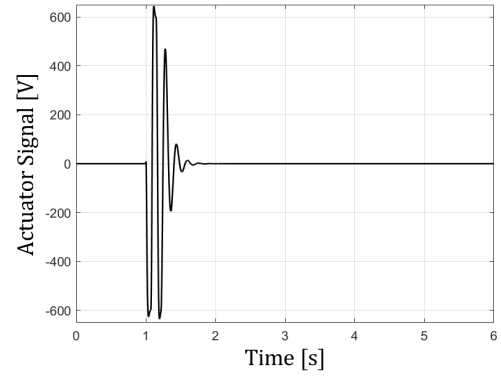


Figure 5.12 (a-d) Vibration and (e-h) actuation responses with optimized PD gains for SM-M1

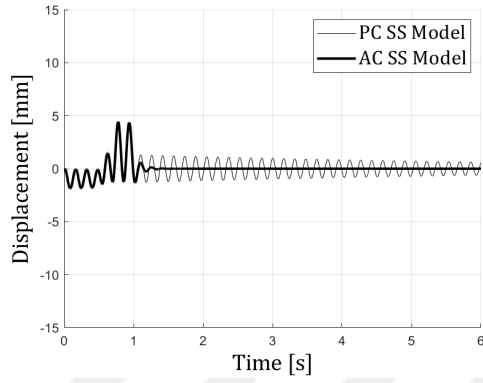


c)

$$t_{dec} = 3t_{1h}$$

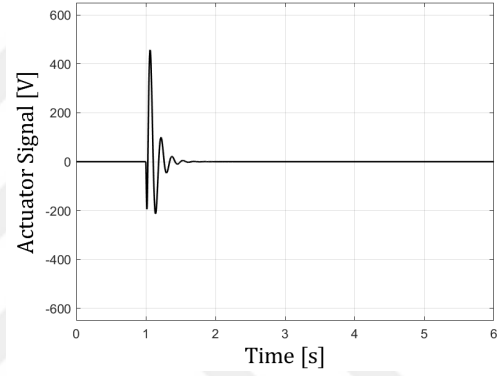


g)



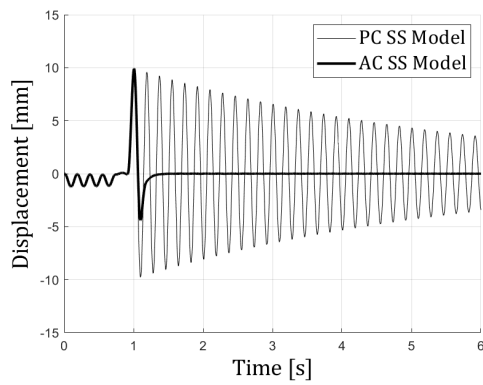
d)

$$t_{dec} = 4t_{1h}$$



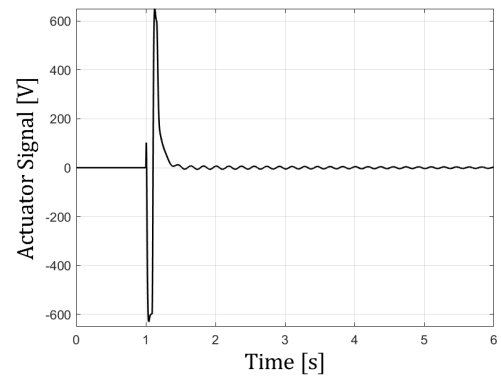
h)

Figure 5.12 continues



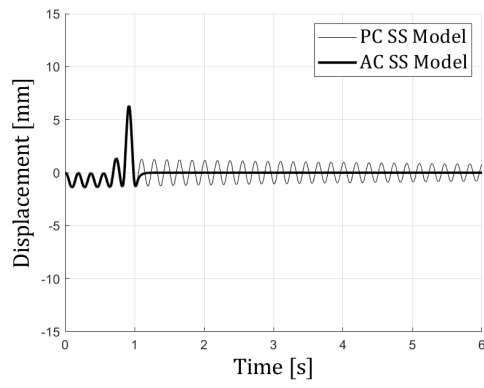
a)

$$t_{dec} = 1t_{1h}$$



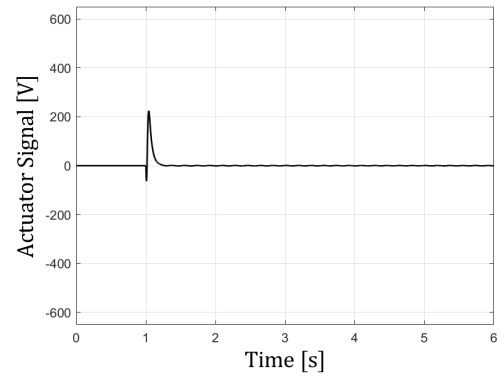
e)

Figure 5.13 (a-d) vibration and (e-h) actuation responses with optimized PD gains for SCM-M1a

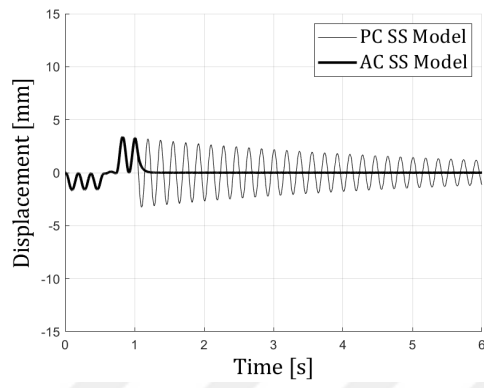


b)

$$t_{dec} = 2t_{1h}$$

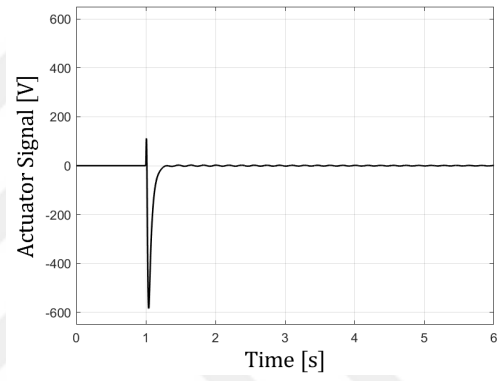


f)

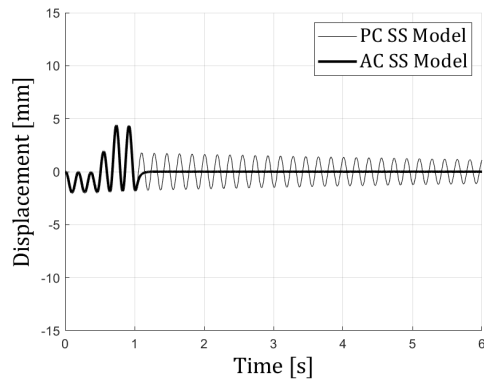


c)

$$t_{dec} = 3t_{1h}$$

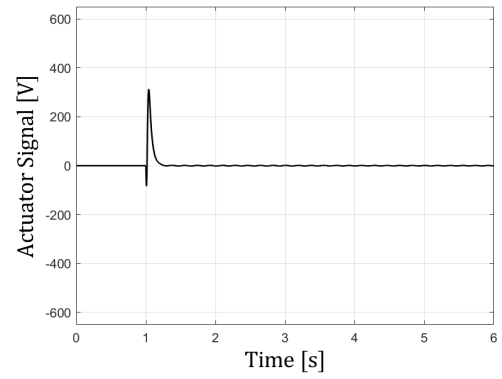


g)



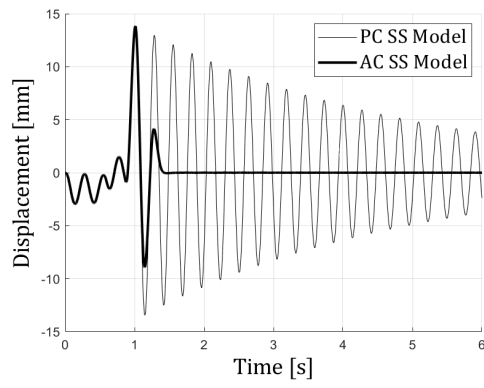
d)

$$t_{dec} = 4t_{1h}$$



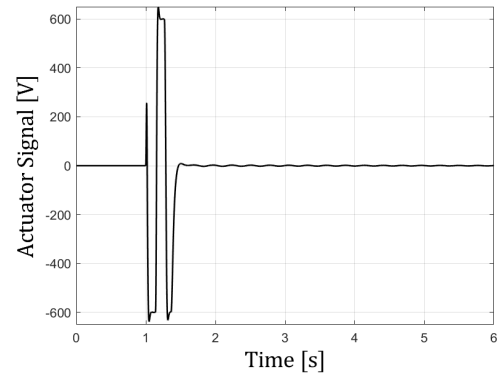
h)

Figure 5.13 continues

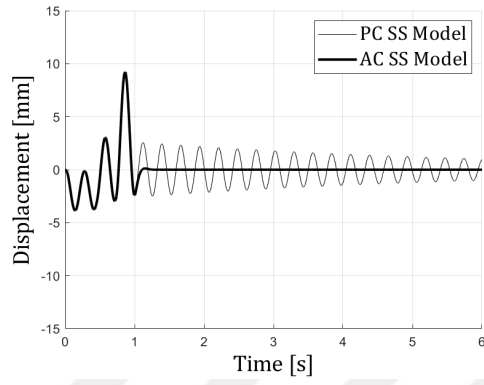


a)

$$t_{dec} = 1t_{1h}$$

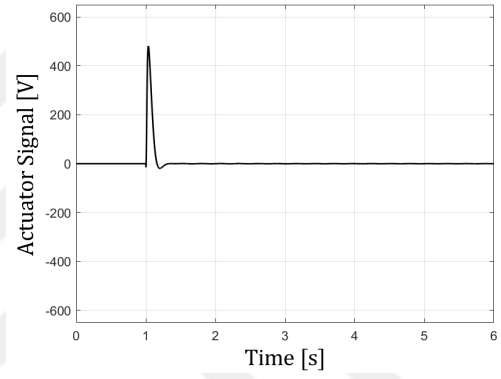


e)

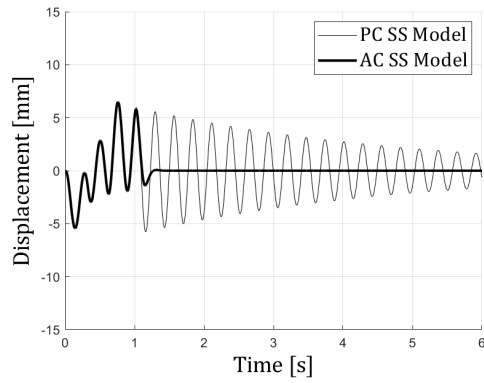


b)

$$t_{dec} = 2t_{1h}$$

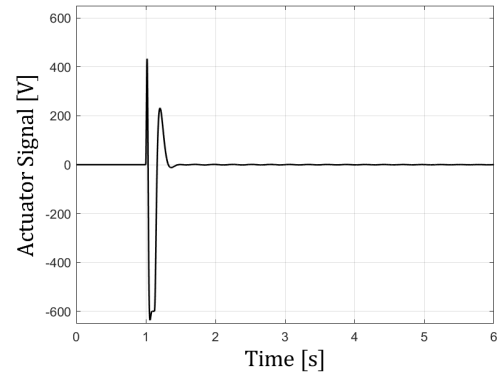


f)



c)

$$t_{dec} = 3t_{1h}$$



g)

Figure 5.14 (a-d) vibration and (e-h) actuation responses with optimized PD gains for SCM-M1b

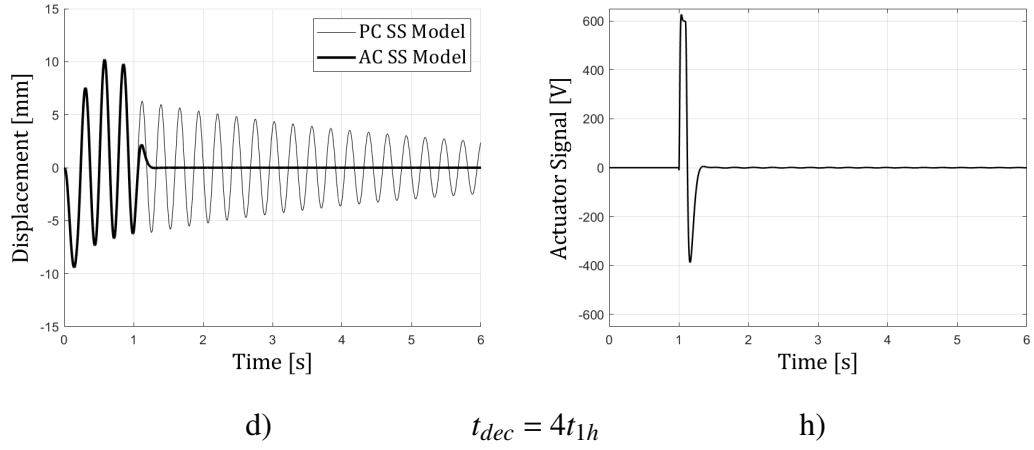


Figure 5.14 continues

Figures show the PC and AC results for SM-M1, SCM-M1a and SCM-M1b. The vibration and actuation responses are obtained for $t_{dec} = 1t_{1h}$, $2t_{1h}$, $3t_{1h}$ and $4t_{1h}$. For the SM, SCM-[0/90] and SCM-[45/-45], the residual vibrations reduced by the PC are also successfully further suppressed by the optimized PD controller gains in the AC.

For tabulation and evaluating the effectiveness of active vibration control, the residual vibration results for the optimized controller gains are considered. The RMS results and maximum actuation voltages are given in Table 5.6.

As seen from Table 5.6, the optimized PD gains are different according to the fitness functions. However, it is observed that the RMS values are the same approximately for IAE and ISE fitness functions. For an example, for SM-M1 and $t_{dec} = 2t_{1h}$, the references values are reduced from 0.4357 mm to 0.1214 mm and 0.1238 mm for IAE, and ISE fitness functions, respectively. When the controller gains are optimized, it is observed that the amplitudes of residual vibrations are further reduced in the AC.

Reduction ratios according to the reference values of $t_{dec} = 1t_{1h}$ are calculated to indicate the reduction ratio over the RMS values of implementing together both the AC and PC. For example, the reduction ratios for the pure AC, and the implementation together of active and passive vibration controls (APC) are calculated as given in Equation 5.4 and Equation 5.5, respectively.

Table 5.6 RMS values for the optimized PD gains

Motion Profile	Fitness Function	t_{dec}	K_p	K_d	AC	Actuation	
					$RMS [mm]$	$RMS [V]$	$[V_{max}]$
SM-M1	IAE	$1t_{1h}$	2.45	1.74	2.6091	138.0744	600
		$2t_{1h}$	9.99	1.90	0.1214	32.7510	384.32
		$3t_{1h}$	8.24	1.89	0.6447	82.5826	600
		$4t_{1h}$	9.99	1.66	0.1230	38.6752	457.67
	ISE	$1t_{1h}$	3.88	2.68	2.6086	139.858	600
		$2t_{1h}$	9.97	1.25	0.1238	29.0275	286.95
		$3t_{1h}$	9.99	2.29	0.6443	83.6002	600
		$4t_{1h}$	9.99	1.30	0.1251	36.3067	385.17
SCM-M1a	IAE	$1t_{1h}$	9.99	1.93	1.1966	62.6138	600
		$2t_{1h}$	11.94	1.35	0.1135	15.8047	225.29
		$3t_{1h}$	11.96	1.45	0.3186	32.1119	111.74
		$4t_{1h}$	11.86	1.33	0.1574	21.7055	313.48
	ISE	$1t_{1h}$	9.99	1.46	1.1934	61.7177	600
		$2t_{1h}$	11.95	0.98	0.1081	14.0887	211.06
		$3t_{1h}$	11.99	1.05	0.3147	31.0015	142.95
		$4t_{1h}$	11.89	0.96	0.1511	19.2691	293.11
SCM-M1b	IAE	$1t_{1h}$	9.37	2.99	2.4515	95.9476	600
		$2t_{1h}$	9.99	2.36	0.2471	36.3309	549.0
		$3t_{1h}$	9.99	2.26	0.8140	62.3402	600
		$4t_{1h}$	9.99	2.65	0.8046	63.8081	600
	ISE	$1t_{1h}$	9.90	2.99	2.4514	95.9278	600
		$2t_{1h}$	9.99	2.17	0.2470	36.0776	459.88
		$3t_{1h}$	9.99	1.83	0.8128	61.0593	600
		$4t_{1h}$	9.99	1.78	0.8024	62.1655	600

$$R_{i*t_{1h}}^{AC} = \frac{RMS_{i*t_{1h}}^{PC} - RMS_{i*t_{1h}}^{AC}}{RMS_{i*t_{1h}}^{PC}} * 100 \quad i = 1, 2, 3, 4 \quad (5.4)$$

$$R_{i*t_{1h}}^{APC} = \frac{RMS_{1t_{1h}}^{PC} - RMS_{i*t_{1h}}^{AC}}{RMS_{1t_{1h}}^{PC}} * 100 \quad i = 1, 2, 3, 4 \quad (5.5)$$

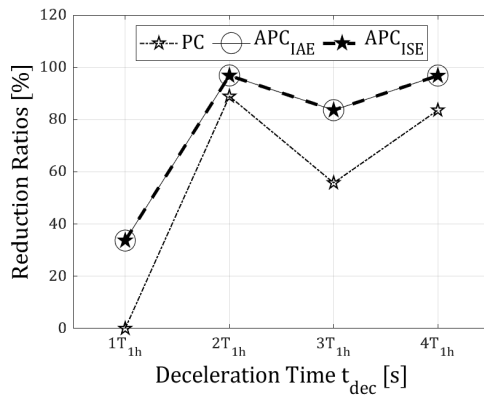
where, $R_{i*t_{1h}}^{AC}$ and $R_{i*t_{1h}}^{APC}$ is the reduction ratios of i^{th} for the deceleration time $t_{dec} = i * t_{1h}$

in the AC, and in the implementation together both the AC and PC, separately. The AC reduction ratios examine the influence of AC within each motion profile in the AC. On the other hand, the APC reduction ratios show the total reduction occurred by the implementation of AC and PC together. $RMS_{1t_{1h}}^{PC}$, $RMS_{1t_{1h}}^{AC}$ is the RMS values for $t_{dec} = 1t_{1h}$ in the PC and AC, respectively. For SM-M1, the example reduction ratio, $R_{i*1t_{1h}}^{AC}$ and $R_{i*1t_{1h}}^{APC}$ for SM-M1 of $t_{dec} = 2t_{1h}$ and IAE fitness function is computed for the simulation results using Table 5.6 and Table 4.8 with equation given by

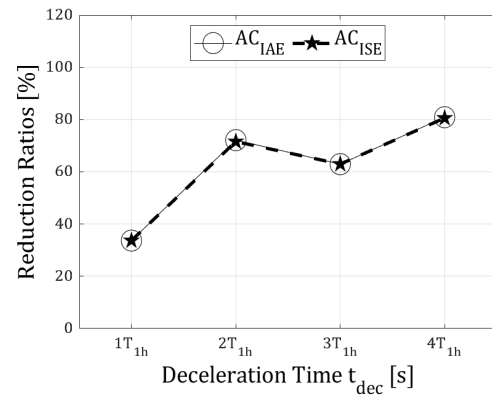
$$\begin{aligned} RMS_{1t_{1h}}^{PC} &= 4.0400 & RMS_{2t_{1h}}^{PC} &= 0.4405 & \text{read from Table 4.8} \\ RMS_{2t_{1h}}^{AC} &= 0.1214 & & \text{read from Table 5.6} \end{aligned} \quad (5.6)$$

$$\begin{aligned} R_{2t_{1h}}^{AC} &= \frac{RMS_{2t_{1h}}^{PC} - RMS_{2t_{1h}}^{AC}}{RMS_{2t_{1h}}^{PC}} * 100 = \frac{0.4405 - 0.1214}{0.4405} * 100 = 72.44\% \\ R_{2t_{1h}}^{APC} &= \frac{RMS_{1t_{1h}}^{PC} - RMS_{2t_{1h}}^{AC}}{RMS_{1t_{1h}}^{PC}} * 100 = \frac{4.0400 - 0.1214}{4.0400} * 100 = 96.99\% \end{aligned} \quad (5.7)$$

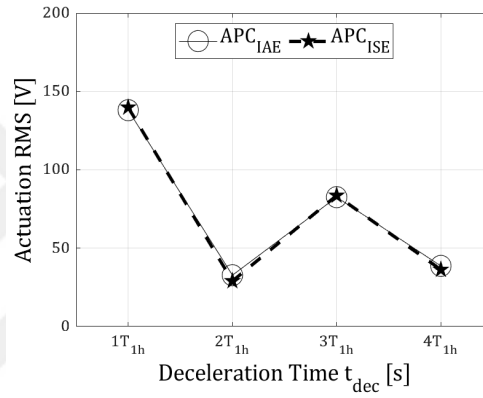
The example reduction ratios for $t_{dec} = 2t_{1h}$ are calculated with the equations above. The APC and AC reduction ratios for the optimized PD gains are calculated in order to evaluate the control performance. For SM-M1, SCM-M1a and SCM-M1b, the reduction ratios are found by using Equation 5.4 and Equation 5.5. Change of reduction ratios and actuator RMS values for the AC are shown in Figure 5.15, Figure 5.16 and Figure 5.17 for SM-M1, SCM-M1a, and SCM-M1b, respectively.



a)

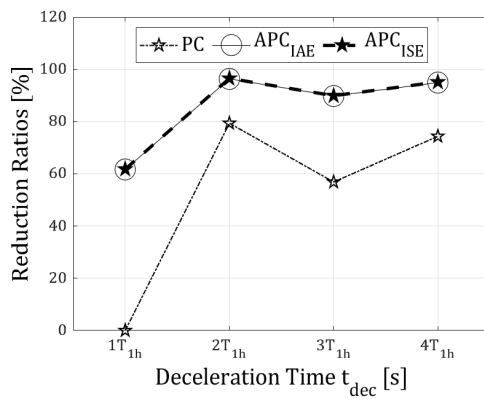


b)

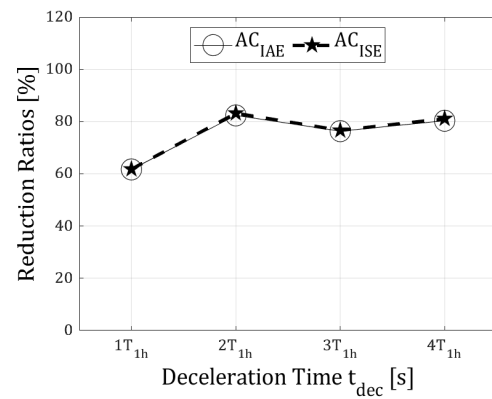


c)

Figure 5.15 Optimized PD control results of SM-M1, a) active and passive control, b) pure active control, c) actuation RMS

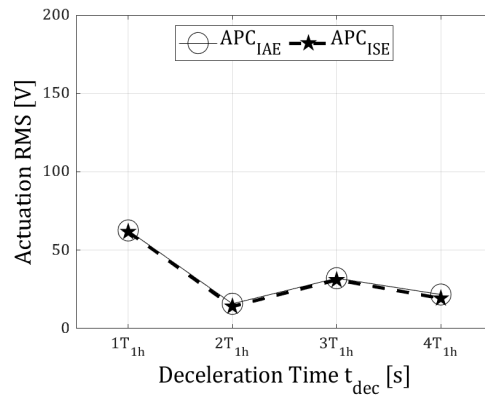


a)



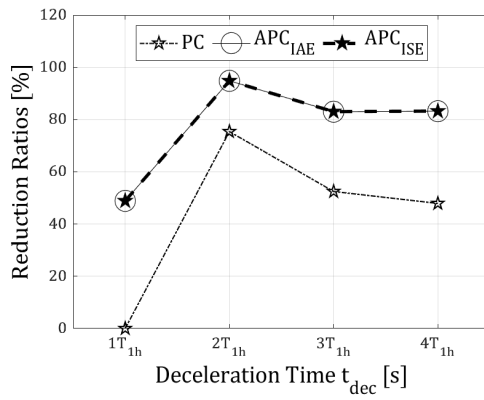
b)

Figure 5.16 Optimized PD control results of SCM-M1a, a) active and passive control, b) pure active control, c) actuation RMS

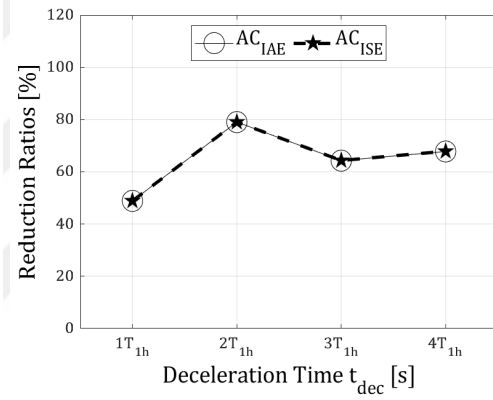


c)

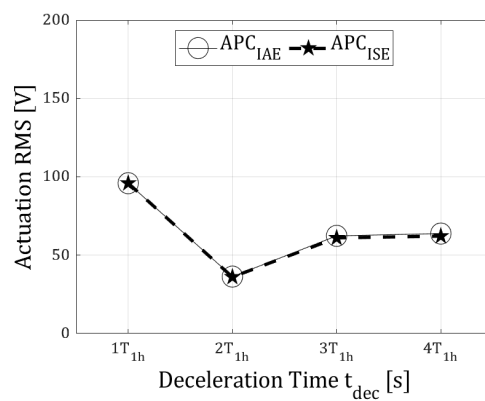
Figure 5.16 continues



a)



b)



c)

Figure 5.17 Optimized PD control results of SCM-M1b, a) active and passive control, b) pure active control, c) actuation RMS

As seen from the figures, the best results in the AC are obtained together with $t_{dec} = 2t_{1h}$ and $4t_{1h}$ as in the PC. It is observed that the AC reduces the residual vibration amplitudes for $t_{dec} = 1t_{1h}$ and $3t_{1h}$. For SM-M1, SCM-M1a and SCM-M1b and $t_{dec} = 2t_{1h}$, the APC reduction ratios are increased from 90%, 80%, 78% to 98%, 97%, 97%, respectively, while the RMS values of actuator responses are raised from 0 V for the PC to approximately 40 V, 20 V, 20 V for the AC.

As indicated in Figure 5.15(a), Figure 5.16(a), and Figure 5.17(a), for $t_{dec} = 4t_{1h}$, the APC reduction ratios are found as 99%, 98% and 82% for SM-M1, SCM-M1a and SCM-M1b. In the case of $t_{dec} = 2t_{1h}$ for the SM, when the optimized PD control is primarily considered, this result means that the residual vibration amplitudes are reduced by 80% with the AC after 78% reduction with the PC. The total reduction ratios for $t_{dec} = 2t_{1h}$ are obtained as 98% reduction with the APC.

The AC reduction ratios are also shown in Figure 5.15(b), Figure 5.16(b), and Figure 5.17(b). For $t_{dec} = 2t_{1h}$, the reduction ratios are found as 73%, 82% and 79% for the SM, SCM-[0/90] and SCM-[45/-45], respectively. As seen from the figures, it seems that the optimized AC results for the IAE and ISE fitness functions is close to each other. It is observed that the best results in the AC are achieved with $t_{dec} = 2t_{1h}$ and $4t_{1h}$ as in the PC.

As indicated in Figure 5.15(c), Figure 5.16(c), and Figure 5.17(c), the RMS values of actuation responses are grown from 0 V to 40 V, 44 V while the reduction ratios are further increased by implementing together of AC and PC from 89%, 82% to 99%, 98% for the $t_{dec} = 2t_{1h}$ and $4t_{1h}$, respectively.

For the SCBM, as seen from Figure 5.11(a), IPDT method among the tuning methods gives a better response to suppress the vibrations. As seen from Figure 5.11(b), the maximum voltages are below ± 3000 V. Since the actuator has a thickness of 5 mm for the SCBM, the maximum operating voltage can be increased to approximately 3000 V. The controller gain is optimized by considering the stability and maximum operating voltage. Saturation is applied to the actuator voltage.

In order to improve the performance of PD control in the AC, the controller gains are optimized by keeping the K_p values constant and increasing the K_d values according to maximum operation voltage of the PZT actuators. The integral gain is also ignored for better performance. For SCBM-M1b, the vibration and actuation responses are shown for the optimized PD gains in Figure 5.18.

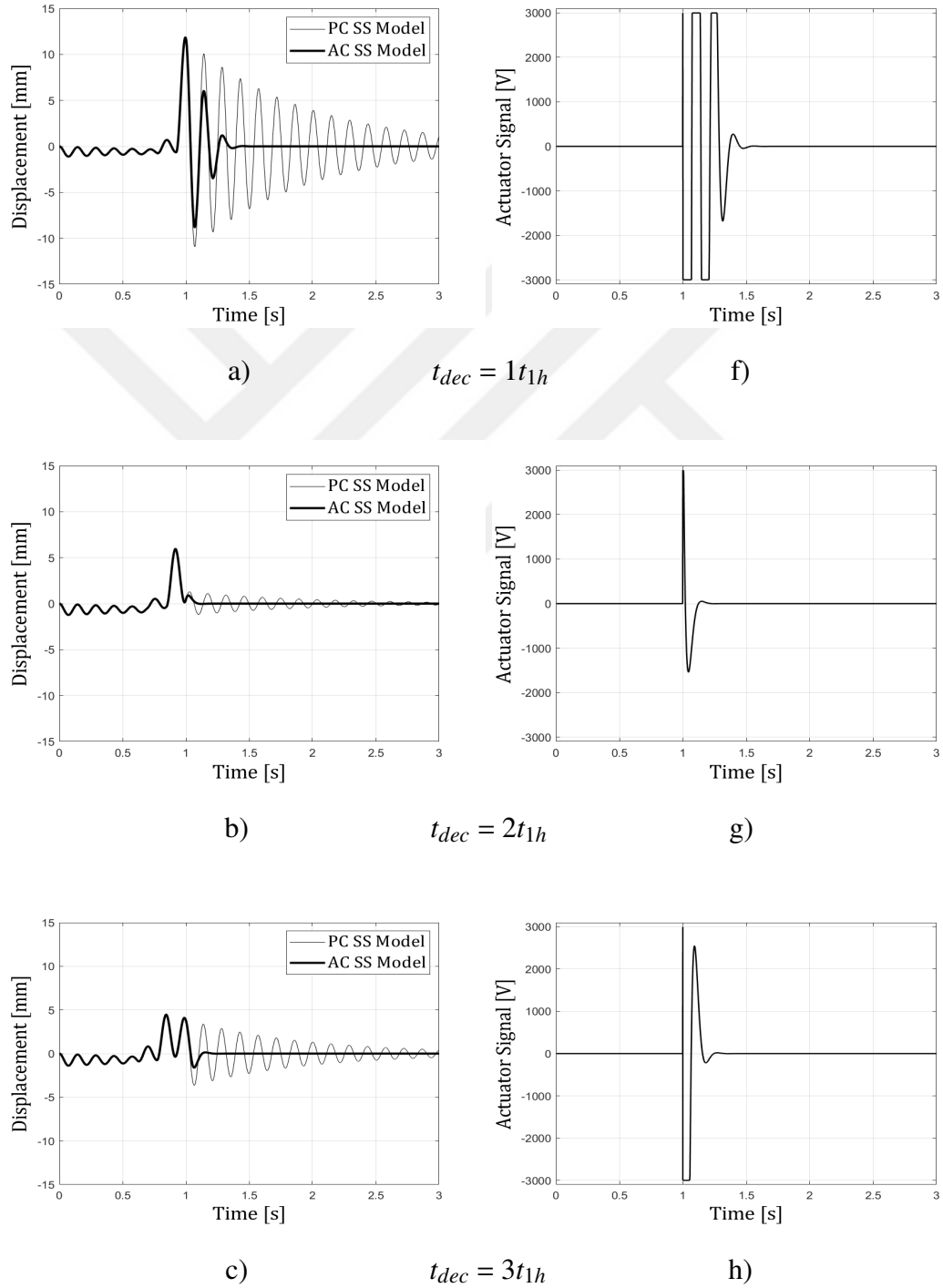


Figure 5.18 (a-e) vibration and (f-j) actuation responses with optimized PD gains for the SCBM-M1b

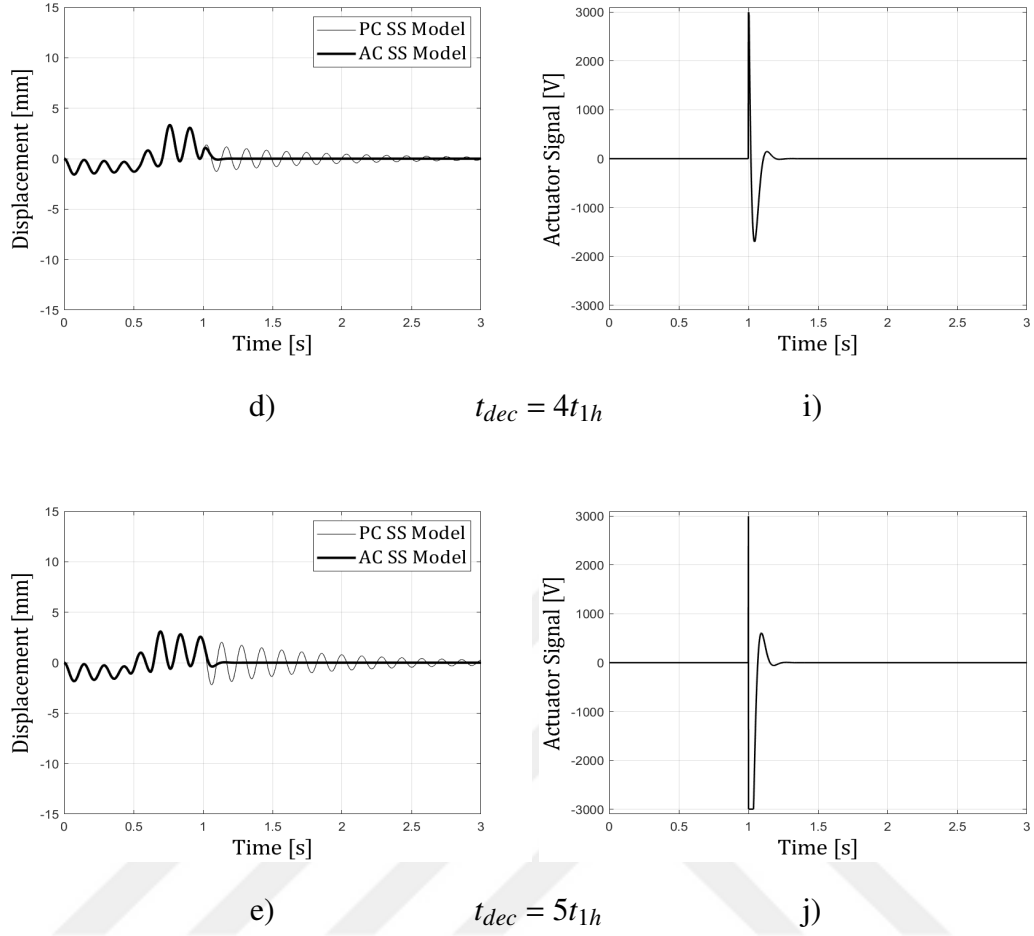


Figure 5.18 continues

As seen from Figure 5.18, the residual vibrations reduced by the PC are also effectively further suppressed by implementing together both the AC and PC. The RMS values of residual vibrations are calculated for the optimized PD controller gains. The comparison of RMS values for the optimized PD gains according to the actuation is listed in Table 5.7.

As seen from Table 5.7, the optimized PD controller gains are different for the AC. For SCBM-M1a and $t_{dec} = 4t_{1h}$, the reference value is reduced from 0.0741 mm to 0.0537 mm. For SCBM-M1c and $t_{dec} = 2t_{1h}$, the reference value is reduced from 0.6191 mm to 0.1445 mm. There is saturation for the respective actuator voltages. When the PD controller gains are optimized according to the actuator performance, it is observed that the reduction values are further reduced with the AC. The reduction ratios and actuation RMS values calculated from Table 5.7 are shown in Figure 5.19, Figure

Table 5.7 RMS results for the optimized PD controller gains according to the actuation

Motion Profile	t_{dec}	K_p	K_d	AC	Actuation	
				RMS [mm]	RMS [V]	[V_{max}]
SCBM-M1a	$1t_{1h}$		4.5	0.4415	174.86	3000
	$2t_{1h}$		4.25	0.1141	32.68	3000
	$3t_{1h}$	13.05	4.35	0.1290	94.80	3000
	$4t_{1h}$		4.25	0.0537	28.07	3000
	$5t_{1h}$		4	0.0713	46.36	3000
SCBM-M1b	$1t_{1h}$		8	2.400	736.94	3000
	$2t_{1h}$		12	0.1332	312.55	3000
	$3t_{1h}$	14.26	10	0.5716	516.80	3000
	$4t_{1h}$		10	0.1381	302.08	3000
	$5t_{1h}$		9.75	0.2992	395.63	3000
SCBM-M1c	$1t_{1h}$		22	4.100	1229.10	3000
	$2t_{1h}$		22	0.1445	419.42	3000
	$3t_{1h}$	12.01	18	1.100	798.61	3000
	$4t_{1h}$		27.5	0.1665	440.73	3000
	$5t_{1h}$		22	0.9392	779.65	3000

5.20, and Figure 5.21 for SCBM-M1a, SCBM-M1b, and SCBM-M1c, respectively.

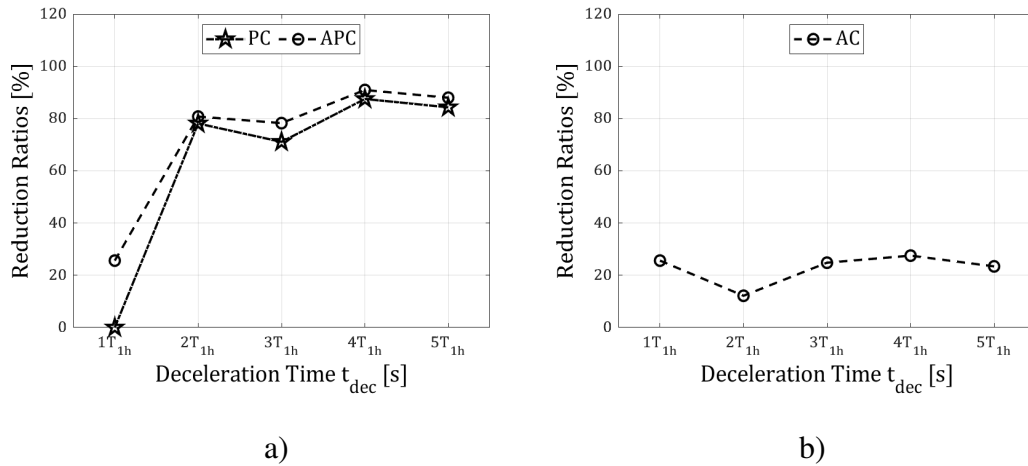
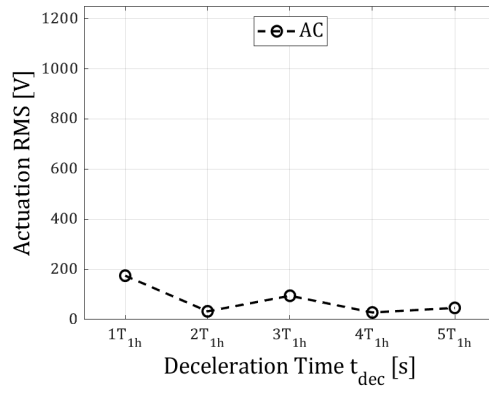
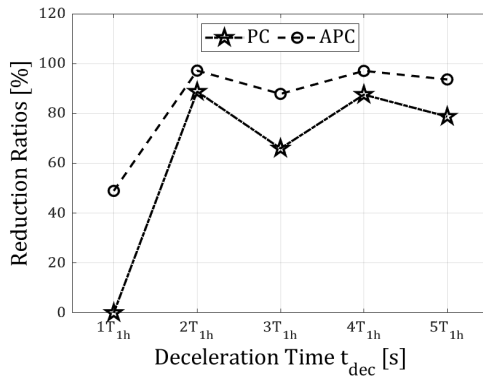


Figure 5.19 Results for the optimized PD control according to the actuator performance of SCBM-M1a, a) active and passive control, b) pure active control, c) actuation RMS

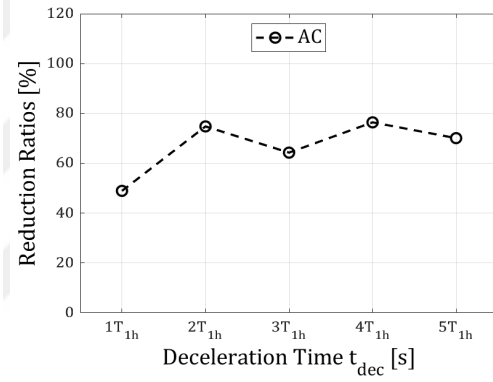


c)

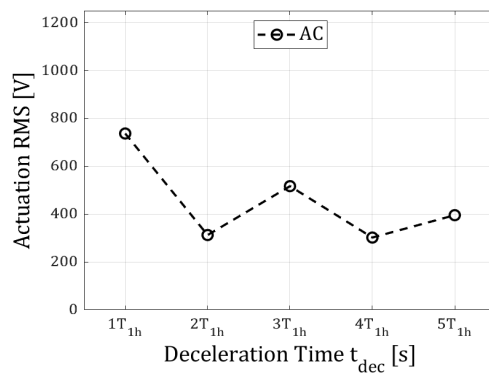
Figure 5.19 continues



a)



b)



c)

Figure 5.20 Results for the optimized PD control according to the actuator performance of SCBM-M1b

a) active and passive control, b) pure active control, c) actuation RMS

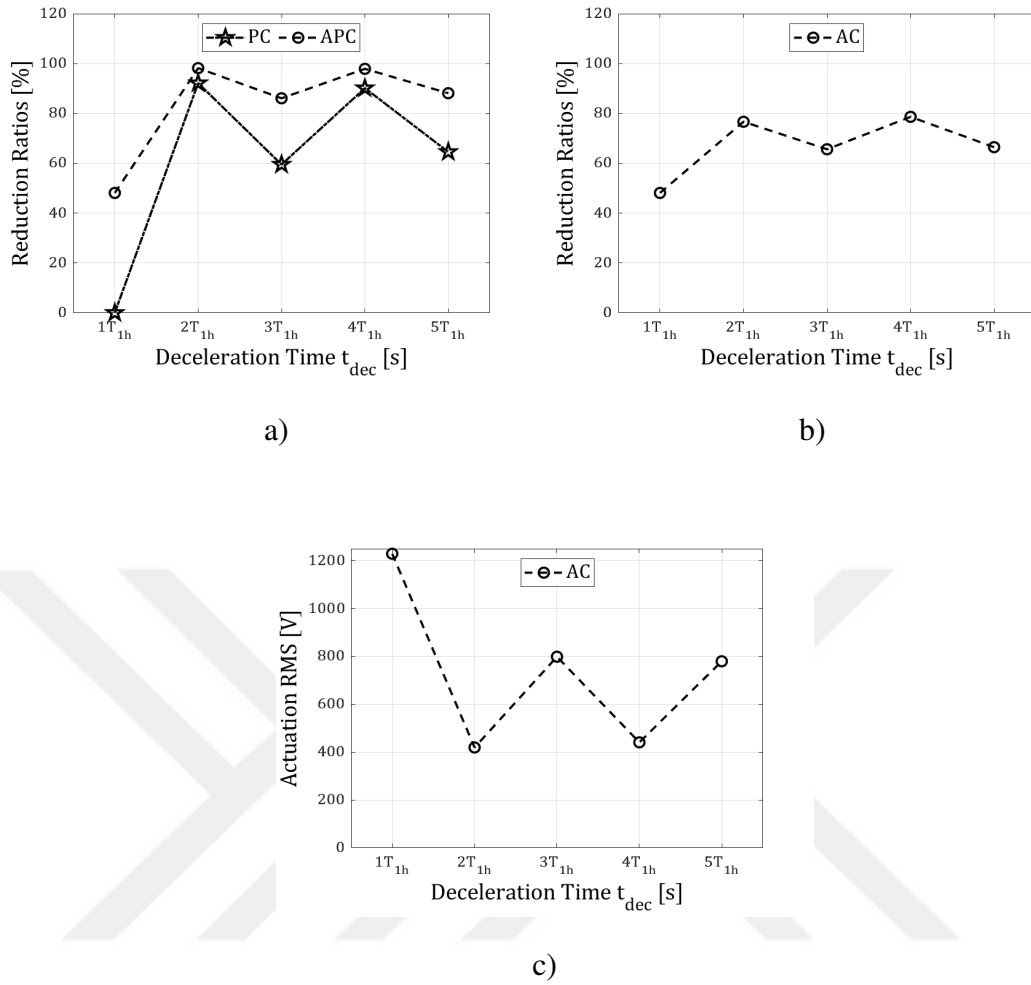


Figure 5.21 Results for the optimized PD control according to the actuator performance of SCBM-M1c, a) active and passive control, b) pure active control, c) actuation RMS

As seen from the figures, the best result for SCBM-M1a in the AC is obtained together with $t_{dec} = 4t_{1h}$ in the PC. For SCBM-M1b and SCBM-M1c, however, the best results in the AC are obtained together with $t_{dec} = 2t_{1h}$ and $4t_{1h}$ in the PC. It is observed that the AC reduces the amplitudes of residual vibrations for $t_{dec} = 1t_{1h}$, $3t_{1h}$, and $5t_{1h}$.

As seen from Figure 5.19(a), the reduction ratios for SCBM-M1a are close to $t_{dec} = 2t_{1h}$, $3t_{1h}$, $4t_{1h}$ and $5t_{1h}$ for the PC, and the optimized AC results. For SCBM-M1a, the reduction ratios are around 80%, 80%, 90%, and 87% for $t_{dec} = 2t_{1h}$, $3t_{1h}$, $4t_{1h}$ and $5t_{1h}$, respectively. When the effect of AC is evaluated for SCBM-M1a shown in Figure 5.19(b), the AC reduction ratios for the optimized controller gains are found as

approximately 28%, 17%, 24%, 26%, and 22% for $t_{dec} = 1t_{1h}$, $2t_{1h}$, $3t_{1h}$, $4t_{1h}$ and $5t_{1h}$, respectively.

As seen from Figure 5.20(a), the best results for SCBM-M1b are obtained together with $t_{dec} = 2t_{1h}$ and $4t_{1h}$. The reduction ratios for $t_{dec} = 2t_{1h}$ and $4t_{1h}$ are increased from 90% and 90% to 99% and 98%. The reduction ratios for the AC on the vibration control are obtained as 78% and 79% for $t_{dec} = 2t_{1h}$ and $4t_{1h}$. For SCBM-M1c, the maximum reduction ratios for $t_{dec} = 2t_{1h}$ are obtained as 90% for the PC, and 99% for the APC with the optimized AC result.

For the optimized AC results shown in Figure 5.19(c), Figure 5.20(c) and Figure 5.21(c), the RMS values of actuator signal for $t_{dec} = 2t_{1h}$ and $4t_{1h}$ are acquired as about 75 V and 65 V, 350 V and 345 V, 440 V and 445 V for SCBM-M1a, SCBM-M1b, SCBM-M1c, respectively. As indicated in the figures, when the optimized AC results are primarily considered, it is observed that the residual vibration amplitudes are further raised with the optimized AC results after increasing the reduction ratios with the PC.

As expected, if the optimized AC results are compared with the PC, it is observed that the optimized AC is more effective than the PC on the vibration control. Also, the level of vibration control is improved by the optimized PD controller gains. As the controller gain increases according to the maximum operating voltage, the success of vibration control increases.

5.3.2 Full-State Feedback Control

The FSF control is conducted for the SCBM in order to investigate the effect of state feedback design. SCBM-M1b, SCBM-M1d and SCBM-M1e are used for the FSF control in MATLAB. The SS model for the SCBM is given in Equation 3.9. For the FSF control, the feedback gains are designed. The state feedback gains are found to place the closed-loop poles at desired locations. Closed-loop block diagram for the FSF control is shown in Figure 5.22.

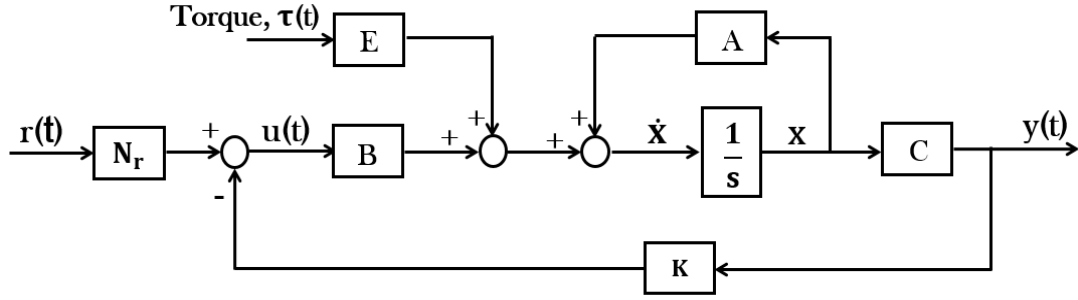


Figure 5.22 Block diagram of FSF control

where, the torque $\tau(t)$ and control voltage $u(t)$ in the block diagram are chosen as the inputs. A, B, C, E are the system, input, output and disturbance matrices, separately. K is the feedback gains and N_r is the tracking gains. $r(t)$ and $y(t)$ are the reference signal and the output of the system, separately. $\tau(t)$ is applied to the revolute joint to move the SCBM. After the movement of SCBM is completed, the FSF control with the DF begins in MATLAB by determining the gains K and N_r . The control goes on up to the steady-state response is reached to reduce the residual vibrations.

For the regulation problem in the state feedback design, the states required to approach zero from the initial state, the design problem is known as the regulation problem for the stability of the system with the desired poles. Therefore, control input $u(t)$ is calculated follow,

$$u(t) = -Kx(t) \quad (5.8)$$

In the block diagram, since the output is required to trace a reference input, the design problem is called as the tracking. So, $u(t)$ with the tracking gain is given by

$$u(t) = -Kx(t) + Nr(t) \quad (5.9)$$

where, $x(t)$ is the state vectors. Control law is applied to the input signal $u(t) = Nr(t) - Kx(t)$ and obtain the closed-loop state equation given as follows

$$\begin{aligned}
\dot{x}(t) &= Ax(t) + B(Nr(t) - Kx(t)) + E\tau(t) \\
&= (A - BK)x(t) + BNr(t) + E\tau(t) \\
&= A_{CL}x(t) + BNr(t) + E\tau(t) \\
y(t) &= Cx(t) + Du(t)
\end{aligned} \tag{5.10}$$

$A_{CL} = A - BK$: Closed-loop matrix

If the closed-loop system is controllable with the designed feedback gains K properly, we can locate the eigenvalues of closed-loop matrix A_{CL} at the desired poles. In order to design the K values, the design criteria are utilized for the FSF control.

5.3.2.1 State feedback design criteria

The location of eigenvalues can be designed with the desired settling time t_{ss} , overshoot M_p , and rise time t_r of the system. A prototype second-order system is considered for the desired criteria. t_{ss} , M_p , and t_r are calculated by using equations given by

$$\begin{aligned}
\text{Settling time}(2\%), t_{ss} &= \frac{4.2}{\zeta w_n} \\
\text{Overshoot}, M_p &= e^{-\pi\zeta / \sqrt{1-\zeta^2}} \\
\text{Rise time}, t_r &= \frac{1.8}{w_n}
\end{aligned} \tag{5.11}$$

Poles-zeros desired with the design criteria are selected at the stable area in root locus analysis. Therefore, two design criteria are chosen as Case-1 and Case-2 given by

- **Case 1:** Settling time, $t_{ss} = 0.05$ s and overshoot, $M_p = 10$ %,

- **Case 2:** Settling time, $t_{ss} = 0.05$ s and rise time, $t_r = 0.005$ s,

The design criteria for the FSF control are determined with Case-1 and Case-2. The desired poles are calculated with equations given in Equation 5.11. The natural frequency ω_n and damping ratio ζ for the desired poles corresponding to poles are calculated. Therefore, ω_n and ζ values for the desired poles are found as given in Equation 5.12 for Case-1, and Equation 5.13 for Case-2.

$$\zeta_1 = \sqrt{\frac{\log(0.10)^2}{\pi^2 + \log(0.10)^2}} = 0.5912$$

$$w_{n1} = \frac{4.2}{\zeta_1 t_{ss}} = \frac{4.2}{0.5912 * 0.05} = 142.095$$
(5.12)

$$w_{n2} = \frac{1.8}{t_r} = \frac{1.8}{0.005} = 360$$

$$\zeta_2 = \frac{4.2}{t_{ss} w_{n2}} = \frac{4.2}{360 * 0.005} = 2.333$$
(5.13)

The desired poles are found as follows

$$p_{1,2} = -\zeta_1 w_{n1} \pm w_{n1} \sqrt{1 - \zeta_1^2} i \quad \text{for Case 1}$$

$$p_{1,2} = -84.0 \pm 114.61 i$$
(5.14)

$$p_{1,2} = -\zeta_2 w_{n2} \pm w_{n2} \sqrt{1 - \zeta_2^2} i \quad \text{for Case 2}$$

$$p_{1,2} = -84.0 \pm 350.06 i$$

After calculating the desired poles, the feedback gains can be found. Feedback gain matrix K are designed by Ackermann's Formula given as follows.

$$K_a = \begin{bmatrix} 0 & \dots & 0 & 1 \end{bmatrix} M_c^{-1} q(A)$$

$$M_c = \begin{bmatrix} B & AB & \dots & A^{n-1} B \end{bmatrix}$$

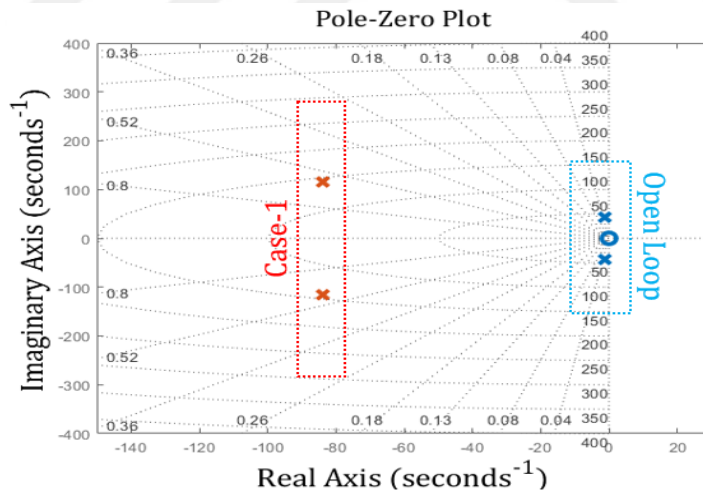
$$q(A) = A^n + \alpha_1 A^{n-1} + \dots + \alpha_n I$$
(5.15)

where, $q(A)$ is the characteristic equation for the closed-loop poles evaluated at matrix A . Then, the tracking problem of FSF control can be solved with the tracking gains. N_r values are determined with the equation given by

$$N_r = \left[C(sI - (A - BK))B \right]^{-1} \quad (5.16)$$

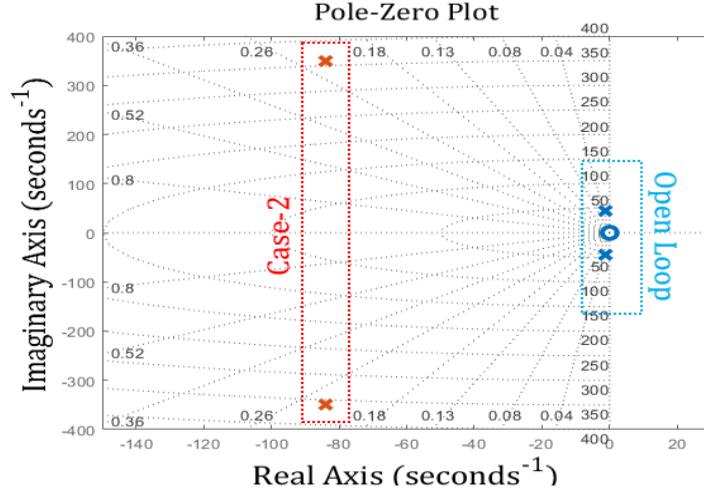
The values of K and N_r are calculated with Ackermann's Formula and Equation 5.16.

In the stability analysis for the control system, the root locus analysis, which is a graphical process, is explained how the roots of system modify with the change of control parameters. The root locus schemes the desired poles of closed-loop system in the complex s-plane. Briefly, the selection of feedback gains is crucial for the system stability in the FSF control. Therefore, the control parameters are chosen in the stability area on the root locus plot coincided with the desired frequency and damping ratio. The root locus plots are shown in Figure 5.23 for the first vibration mode.



a)

Figure 5.23 Root locus plot for a) Case-1, and b) Case-2

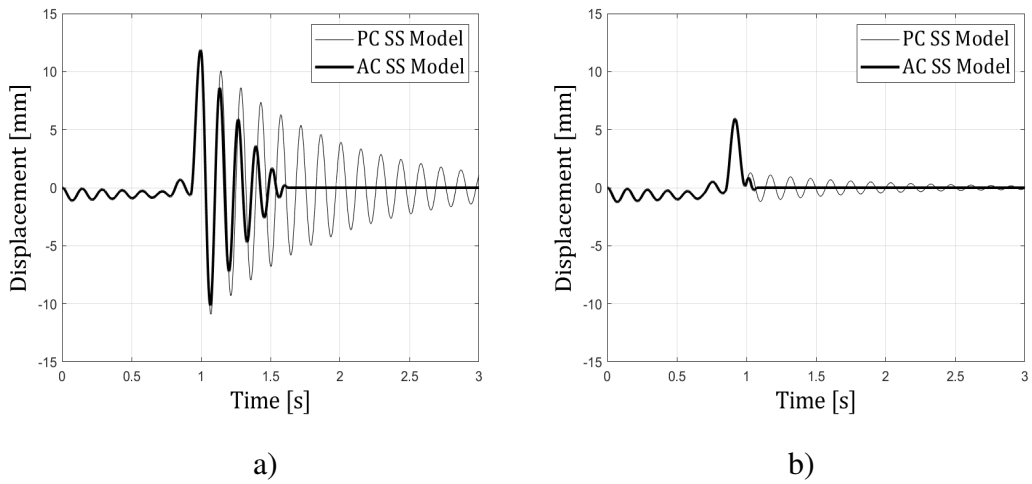


b)

Figure 5.23 continues

It is observed from Figure 5.23 that open-loop system of SCBM is stable and damping ratio is found as 0.025. The damping of closed-loop system is calculated as 0.591 for Case-1, and 0.233 for Case-2, respectively. The overshoot of root locus plots is found as 10% for Case-1, and 12.8% for Case-2 while these are found as 92.4 for open-loop system.

The FSF control is performed using SCBM-M1b, SCBM-M1d and SCBM-M1e. The SS results for SCBM-M1b are shown in Figure 5.24.



a)

b)

Figure 5.24 FSF control results for SCBM-M1b and Case-1, t_{dec} a) $1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$ and d) $4t_{1h}$

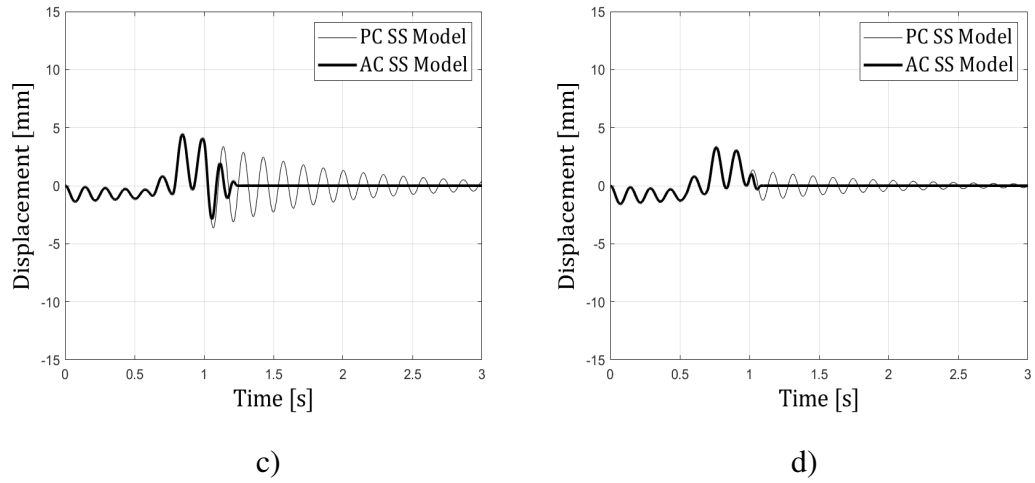


Figure 5.24 continues

Figure 5.24 indicates the example vibration results for the SCBM-M1b with Case-1. The RMS values of vibration responses are calculated using SCBM-M1b, SCBM-M1d and SCBM-M1e for Case-1 and Case-2. The FSF control results are given in Table 5.8.

Table 5.8 RMS values and settling times of FSF control

Motion Profile	t_m [s]	t_{dec} [s]	PC	Case-1		Case-2	
			RMS [mm]	RMS [mm]	t_{ss} [s]	RMS [mm]	t_{ss} [s]
SCBM-M1b	1	$1t_{1h}$	2.8346	1.7808	0.6272	2.5004	1.6943
		$2t_{1h}$	0.3207	0.0652	0.0887	0.1006	0.2994
		$3t_{1h}$	0.9400	0.3812	0.2569	0.6900	0.9440
		$4t_{1h}$	0.3399	0.0810	0.1012	0.1304	0.3542
		$5t_{1h}$	0.5582	0.1665	0.1548	0.3467	0.6420
SCBM-M1d	1	$1t_{1h}$	4.2519	2.9889	0.8352	3.8928	2.0104
		$2t_{1h}$	0.4811	0.1242	0.1224	0.2029	0.4554
		$3t_{1h}$	1.4101	0.6817	0.3566	1.1193	1.1935
		$4t_{1h}$	0.5099	0.1490	0.1424	0.2444	0.5155
		$5t_{1h}$	0.8374	0.3071	0.2311	0.5775	0.8453
SCBM-M1e	2	$1t_{1h}$	2.2024	1.2798	0.5219	1.8834	1.5036
		$2t_{1h}$	0.2183	0.0395	0.0645	0.0576	0.2155
		$3t_{1h}$	0.6754	0.2369	0.1951	0.4516	0.7471
		$4t_{1h}$	0.2064	0.0421	0.0657	0.0629	0.2294
		$5t_{1h}$	0.3611	0.0888	0.1184	0.1923	0.4622

As indicated in Table 5.8, the results of SCBM are given for two different motion times as $t_m = 1$ s for SCBM-M1d and $t_m = 2$ s for SCBM-M1e. The RMS values are decreased using the FSF control. In order to evaluate the effectiveness of active vibration control, the change of APC and AC reduction ratios and settling times are graphically shown in Figure 5.25, Figure 5.26 and Figure 5.27.

As seen from the figures, it is also observed that the deceleration time for $t_{dec} = 2t_{1h}$ and $4t_{1h}$ is the most effective performance of vibration control to reduce the residual vibration amplitudes.

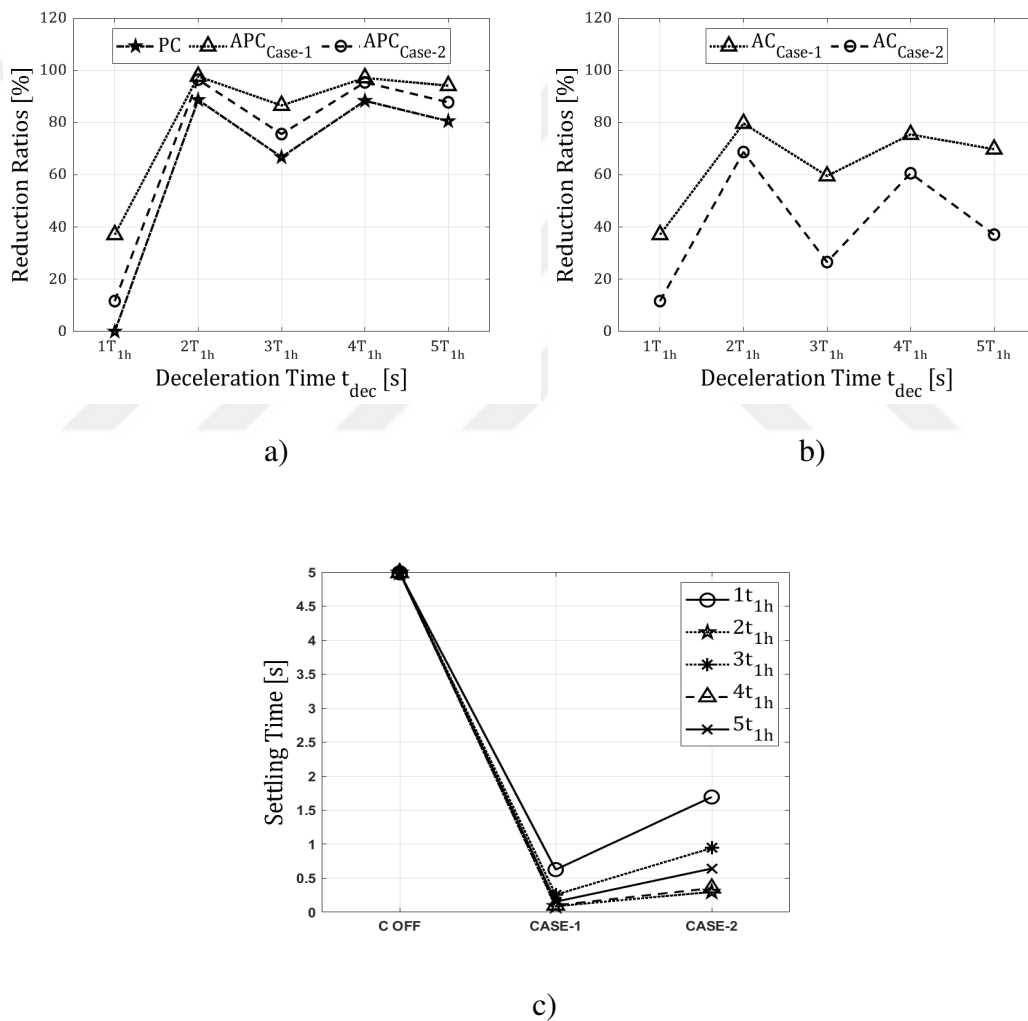
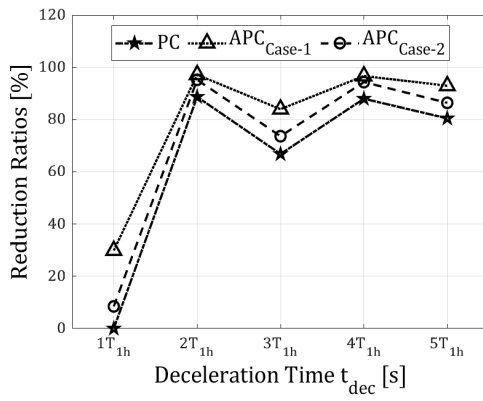
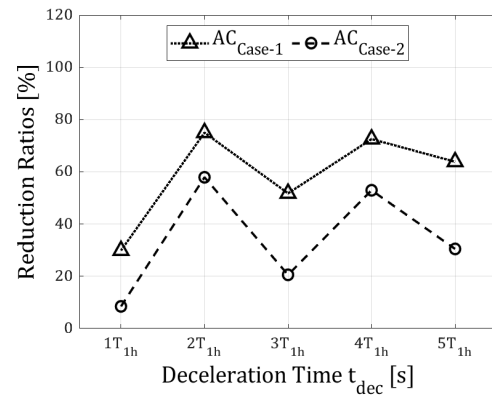


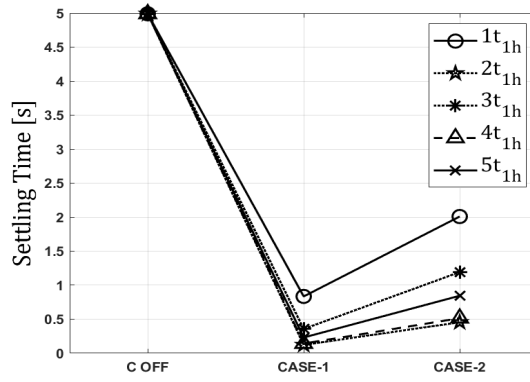
Figure 5.25 FSF control results of SCBM-M1b for Case-1 and Case-2, a) APC reduction ratios, b) AC reduction ratios and c) settling times



a)

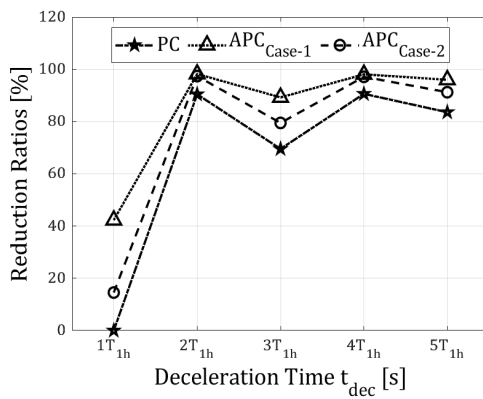


b)

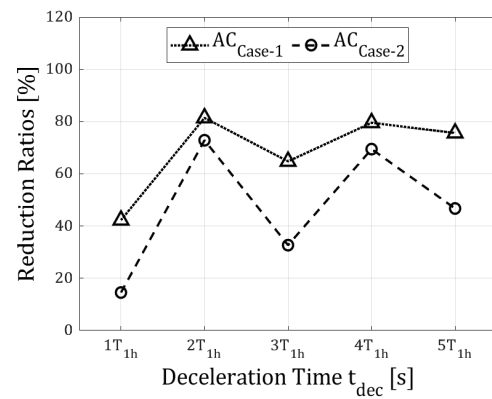


c)

Figure 5.26 FSF control results of SCBM-M1d for Case-1 and Case-2, a) APC reduction ratios, b) AC reduction ratios and c) settling times

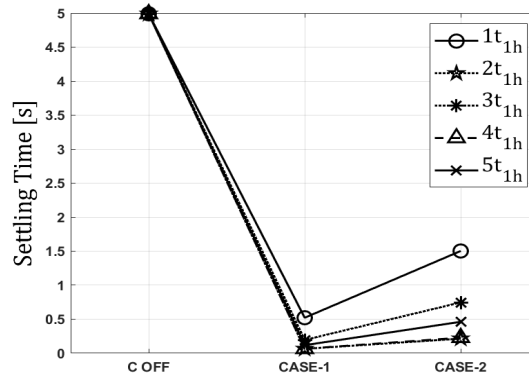


a)



b)

Figure 5.27 FSF control results of SCBM-M1e for Case-1 and Case-2, a) APC reduction ratios, b) AC reduction ratios and c) settling times



c)

Figure 5.27 continues

Table 5.8 shows the FSF control results with saturation due to the maximum operating voltage of the PZT actuators. In order to compare the saturation and non-saturation results for SCBM-M1b, the non-saturation results are given in Table 5.9.

Table 5.9 RMS values and settling times of non-saturation results

Motion Profile	t_{dec} [s]	Case-1		Case-2	
		RMS [mm]	t_{ss} [s]	RMS [mm]	t_{ss} [s]
SCBM-M1b	$1t_{1h}$	0.4644	0.0505	0.2968	0.0485
	$2t_{1h}$	0.0205	0.0416	0.0112	0.0350
	$3t_{1h}$	0.1487	0.0499	0.0960	0.0482
	$4t_{1h}$	0.0295	0.0462	0.0169	0.0412
	$5t_{1h}$	0.0829	0.0490	0.0541	0.0478

As seen from Table 5.9, it is observed that the reduction values are slightly decreased for the saturation and non-saturation results. The reduction ratios of saturation and non-saturation results are graphically shown in Figure 5.28.

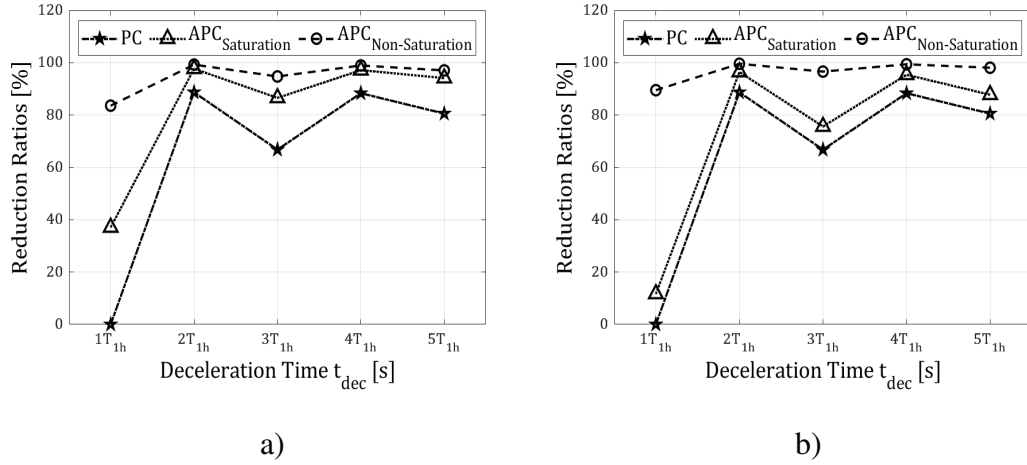


Figure 5.28 Reduction ratios of saturation and non-saturation results for a) Case-1 and b) Case-2

When the saturation and non-saturation results are compared, it is observed that better results for the settling time are obtained for the non-saturation results. For the non-saturation results, it is observed that the minimum settling time for Case-2 is found as approximately 0.035 s while these is calculated as 0.041 s for Case-1. The settling times for the design criteria are determined as $t_{ss} = 0.05$ s. Therefore, it seems that t_{ss} are in good agreement with the non-saturation results. However, for the saturation results and $t_{dec} = 2t_{1h}$, t_{ss} is increased from 0.0416 s, 0.0350 s to 0.0887 s, 0.2994 s for Case-1 and Case-2, respectively.

In the non-saturation results, the maximum reduction ratios are approximately calculated as 99% for $t_{dec} = 2t_{1h}$. Also, the reduction ratios of results with saturation are observed to be better results with the reduction of approximately 90% in the residual vibrations according to the PC. Higher reduction ratios are achieved for the FSF control without saturation. It seems that the results without saturation are more useful to control the vibration amplitudes.

CHAPTER SIX

EXPERIMENTS

6.1 Introduction

In this chapter, the experiments of the smart manipulators are conducted for the comparison of SS analysis results presented in Chapter-4 and Chapter-5. The experimental system is introduced. In the experiments, a displacement sensor is used for the smart manipulators while an acceleration sensor is utilized for the CBM at the tip point. The CBM is a composite box manipulator without piezoelectric actuators as SCBM cannot be produced due to the absence of 5 mm piezoelectric actuators. Experimental modal analysis is performed to find the natural frequencies for two vibration modes by applying a chirp signal to the PZT actuators. Experimental natural frequencies are found by taking the Fast Fourier Transform (FFT) and then, the experimental results are compared with the simulation results given in Chapter-4. After comparing the natural frequency results, the experiments for the AC and PC are performed by using the developed LABVIEW code. Experimental displacement and acceleration vibration responses are presented in this chapter.

6.2 Measurement and Control System

Experimental results are obtained using the motion profiles considering in the PC given in Chapter-4. The experimental setup for the AC and PC is shown in Figure 6.1.

The experimental block diagram consists of four-part as creating a drive signal, obtaining the actuator signal, acquiring the displacement and acceleration signals. The drive signal is used to move the smart manipulators while the actuator signal is obtained to implement the PZT actuators to reduce the amplitudes of residual vibrations in the AC. The displacement and acceleration signals are used to measure the vibrations at the tip point. In the experimental setup, a 200 W Mitsubishi Electric servo motor and drive for the drive signal, a power amplifier (SENORTECH SA-10) for the actuator signal,

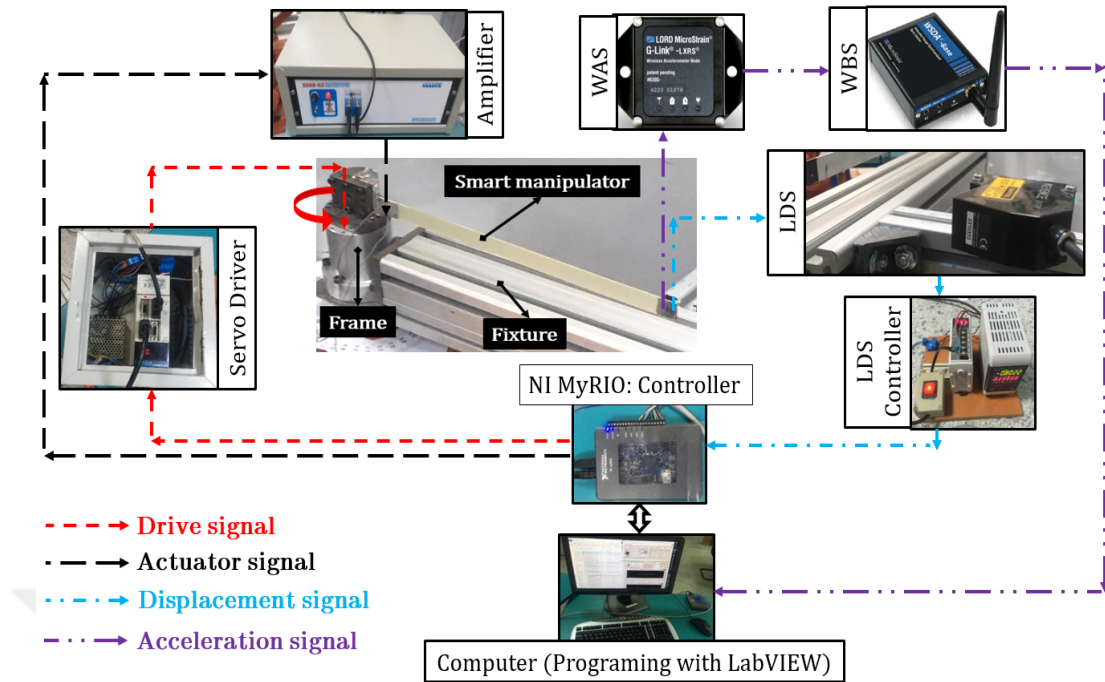


Figure 6.1 Block diagram of experimental system (Personal archive, 2020)

a KEYENCE LK-G157 laser displacement sensor (LDS) and LDS controller for the displacement signal, and a Microstrain Wireless Base Station (WBS) and acceleration sensor (WAS) for the acceleration signal are utilized. Also, NI MyRIO 1900 DAQ card of National Instruments which has analog inputs and outputs, digital input and output terminals is utilized to read the sensor data, generate and transmit the control signal, and to communicate with the computer for programming in LABVIEW software.

Generally, MyRIO 1900 is a tool which uses to implement the multiple design concepts with one reconfigurable Input/Output (I/O) device (MyRIO1900, 2016). Servo motor is used with a Harmonic Drive gearbox with a ratio of 100. The servo driver has the position control with a pulse command, the speed and torque control with the pulse and analog command, and execute precision positioning of 131072 pulses/rev resolution. For the analog control mode, an external analog speed command is used to control the speed with 0 to ± 10 VDC, torque with 0 to ± 8 VDC and direction of servo motor smoothly. A power amplifier is used to amplify the control voltage for the AC. A digital low pass filter of 30 Hz is used during the excitation in the power amplifier.

A LABVIEW code is developed to move the smart manipulators and perform the AC and PC given in Appendix-A. The LABVIEW code has a comprehensive set of tools for analyzing, acquiring, storing data and displaying (MyRIO1900, 2016). The uncontrolled and controlled vibration responses are obtained by using the AI and the drive signal driven the smart manipulators is sent to the servo drive via the AO in the LABVIEW code. The experimental results are obtained repeatedly for various control gains.

The drive signal is created to move the smart manipulators with using the analog output of control card. The motion commands are forward to the servo driver through MyRIO 1900. MyRIO 1900 is connected to the computer via a USB port.

The displacement signal is acquired with the analog input of MyRIO 1900 through the LDS via the LDS controller. The LDS can measure the displacement amplitudes from +40 mm to -40 mm. The output of LDS controller is taken the analog voltages between +10 and -10 V. The LDS is connected to the controller which enables to display of measurement values and to obtain the voltage output simultaneously. The LDS controller has the voltage output in the range of ± 10 V to acquire the sensor data using the LABVIEW code in MyRIO 1900. For the LDS controller, the measurement range of ± 40 mm and ± 5 mm corresponds to the voltage output of ± 10 V in the controller. The displacement data is recorded to the computer by using LABVIEW code.

The acceleration signal is obtained by using Node Commander software from the WAS that communicates with the WBS. The acceleration data is measured the vibrations in three directions with their embedded accelerometer. The measured vibration signals are transmitted from the WAS to the WBS. Node Commander software of Microstrain Company is used to transfer the acceleration data to the computer. The sampling rate for the acceleration signal is set as 617 Hz during the experiments. The displacement signal is used to verify the AC and PC results of smart manipulators while the acceleration data is only utilized to confirm the PC results of the CBM.

The experimental studies for the PC are realized with the drive signal, displacement and acceleration signals. In addition to the PC, the actuator signal is used in the AC.

The actuator signal is created to eliminate the residual vibrations of the smart manipulators in the experiments. The actuation voltage is transmitted to the power amplifier via the analog output of MyRIO 1900. The control voltage applied for the PZT actuators is transferred via the power amplifier and limited in the control card by ± 9 V due to the maximum operating voltage of PZT actuators. Briefly, the actuator signal is simultaneously carried off with the analog output in order to actuate the PZT actuators through the high voltage amplifier.

6.2.1 Experimental Smart Manipulators

The SM consists of an aluminum link and a PZT actuator while the SCM-[0/90] and SCM-[45/-45] have a composite link and a PZT actuator. PZT actuators are bonded on the surface of the manipulators by using ELECOLIT 325-A/B conductive epoxy adhesive. The type of PZT actuators are the BM532 (PZT5-H) of SENSORTECH. The experimental smart manipulators are shown in Figure 6.2 and Figure 6.3. The SCM consists of two different manipulators as the SCM-[0/90] and SCM-[45/-45].

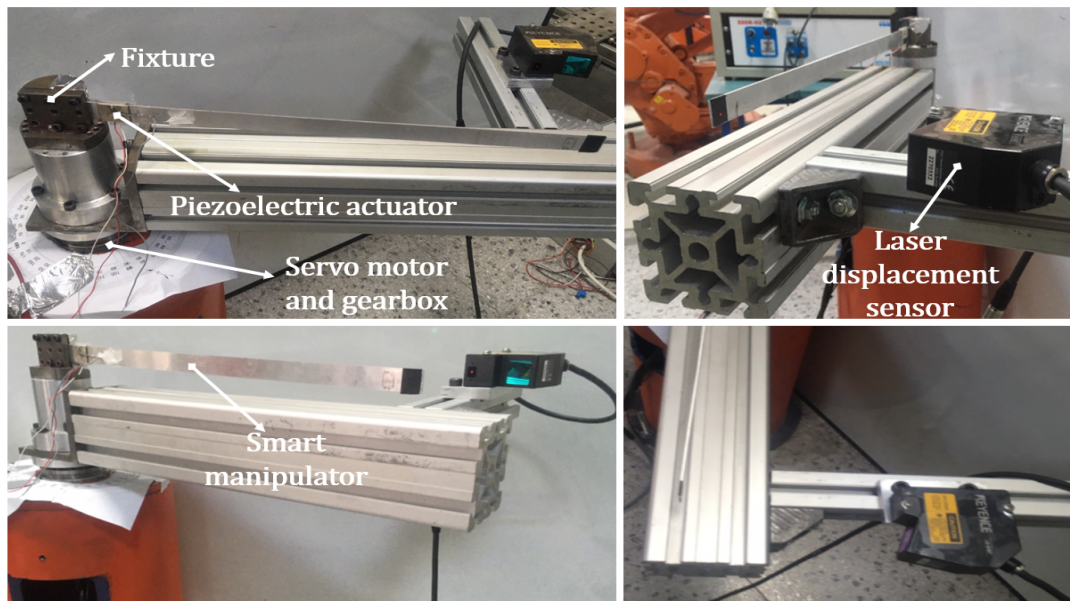


Figure 6.2 Experimental system with the SM (Personal archive, 2020)

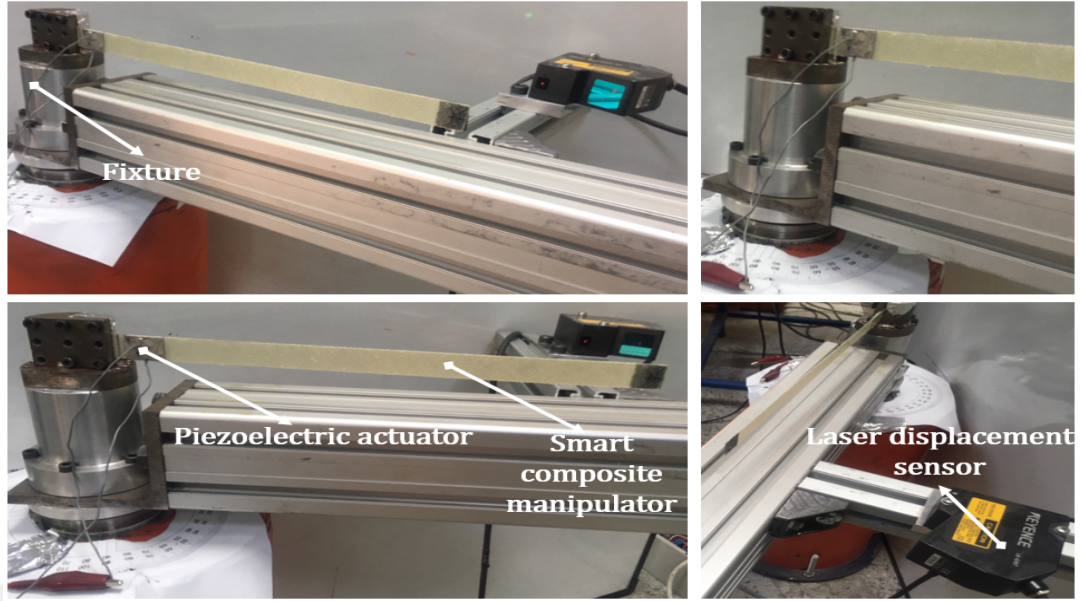


Figure 6.3 Experimental system with the SCM-[0/90] and SCM-[45/-45] (Personal archive, 2020)

6.2.2 Composite Box Manipulator

The CBM is defined as the SCBM without the PZT actuators. In this section, the experimental system for the CBM is setup to verify the PC results. The SCBM whose SS analysis results given in Chapter-4 and 5 have been verified by simulations consists of 5 mm thickness of PZT actuators. Besides, the actuation voltage of PZT actuators has been limited to ± 3000 V due to the maximum operating voltage of about 600 V/mm. In order to verify the theoretical results, the experimental system requires two 5 mm thickness of PZT actuators and a power amplifier with an operating voltage of ± 3000 V. The SCBM cannot be produced due to the absence of PZT actuators of 5 mm and a power amplifier of 3000 V.

So, the experiments for the CBM are conducted to verify the SS results in the PC. The experimental system of CBM considered in this study is shown in Figure 6.4.

The experimental CBM includes a Microstrain wireless acceleration sensor (WAS) that communicates with a wireless base station (WBS). The mass effect of WAS is considered in simulations.

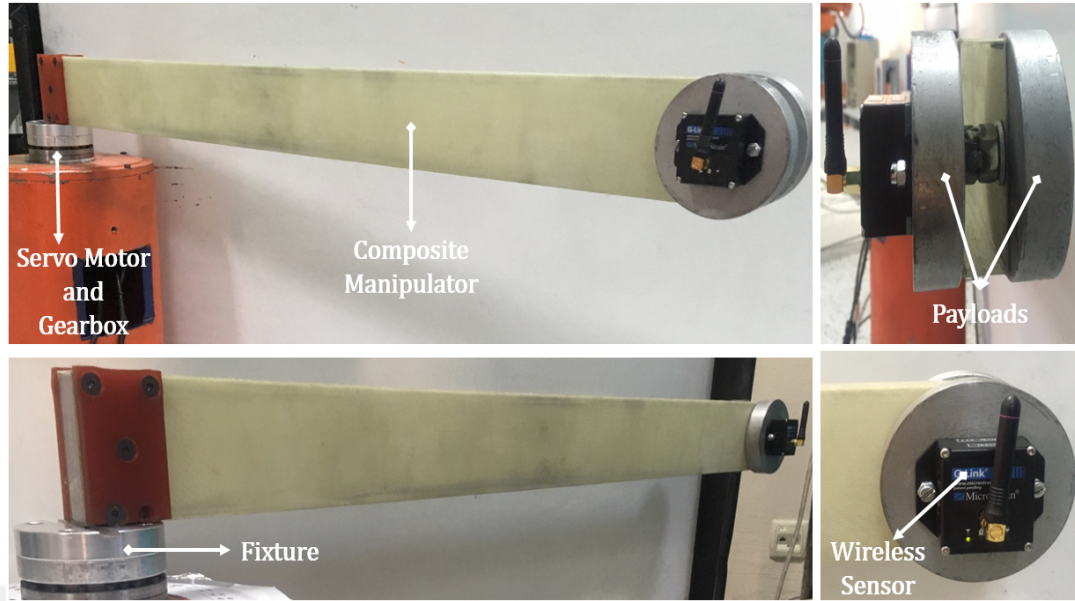


Figure 6.4 Experimental system for the CBM (Personal archive, 2020)

Also, the CBM is made of four-layer epoxy-glass material with a layer orientation of $[0/90/0/90]$. The wrapping method is used. Composite layers are wrapped around a 20 mm x 80 mm rectangular steel bar utilized as a mold. A compression is exerted by dint of a vacuum at 120 in degree. After curing, the CBM is fabricated by removing the steel rod.

6.3 Experimental Modal Analysis

The experimental modal analysis is performed by using a chirp signal to found the natural frequencies. A chirp signal is created with its amplitude of ± 9 V in the LABVIEW code. In order to determine the first two natural frequencies, the initial and target frequencies of chirp signal are defined from 0.01 Hz to 60 Hz. The duration of experimental modal analysis is selected as 50 s. The chirp signal is applied to the PZT actuators. For the vibration responses, the displacement signal at the tip of smart manipulators is simultaneously measured by using the LDS and its controller.

The frequency response is obtained by applying the Fast Fourier Transform (FFT) in MATLAB. The experimental modal analysis results are shown in the figures.

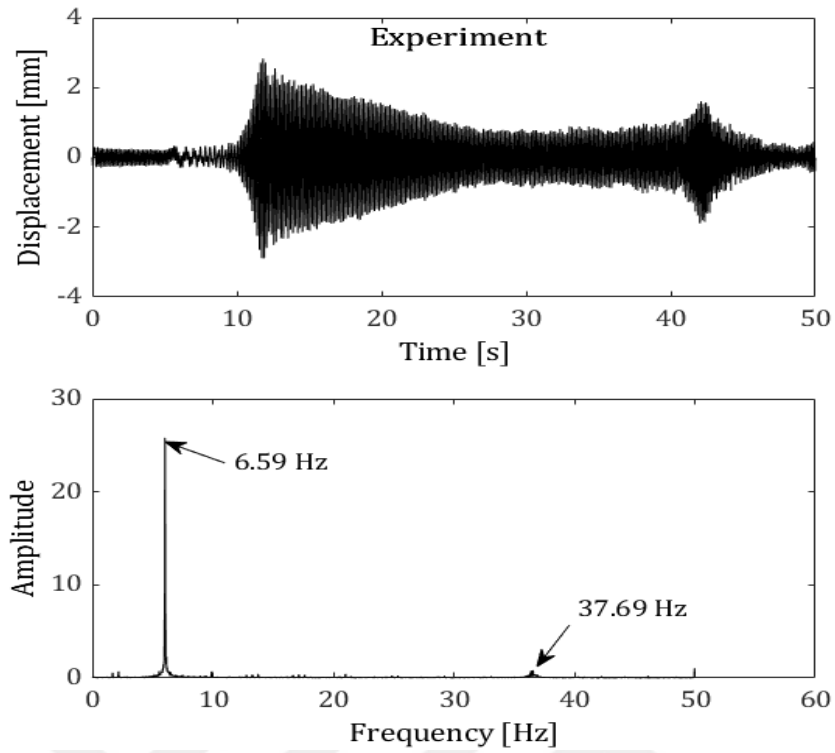


Figure 6.5 Experimental modal analysis results of SM

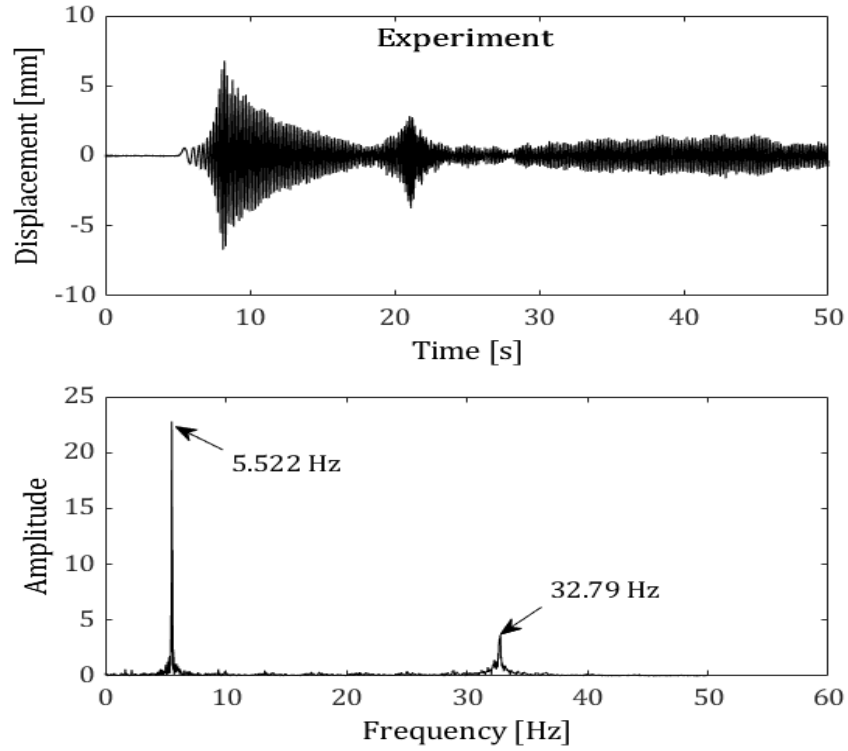


Figure 6.6 Experimental modal analysis results of SCM-[0/90]

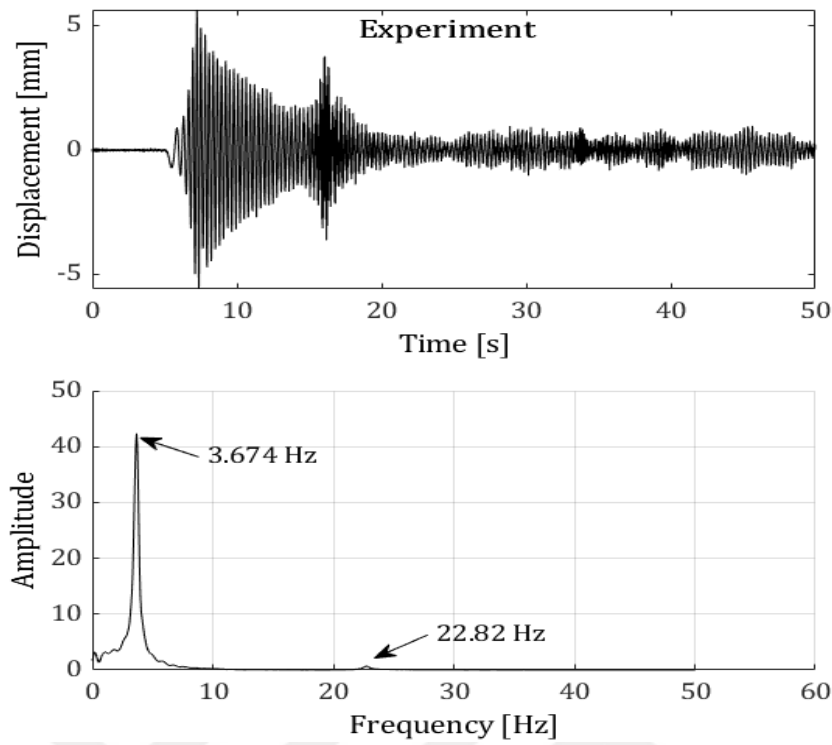


Figure 6.7 Experimental modal analysis results of SCM-[45/-45]

As shown from the figures, the experimental natural frequencies for the first two vibration modes are found as 6.59 Hz and 37.69 Hz for the SM, 5.52 Hz and 32.79 Hz for the SCM-[0/90] and 3.67 Hz and 22.82 Hz for the SCM-[45/-45]. The comparison of natural frequencies is given in Table 6.1.

Table 6.1 Undamped natural frequencies of smart manipulators

Order of frequency	SM		SCM-[45/-45]		SCM-[0/90]	
	Simulation (Hz)	Experiment (Hz)	Simulation (Hz)	Experiment (Hz)	Simulation (Hz)	Experiment (Hz)
f_{n1}	6.610	6.59	3.667	3.674	5.485	5.522
f_{n2}	39.203	37.69	22.242	22.820	33.275	32.79

Also, for the natural frequencies of CBM for various payloads are given in Table 6.2.

Table 6.2 Undamped natural frequencies of CBM

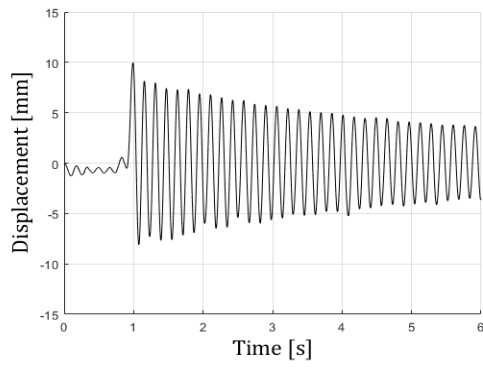
Natural Frequency (Hz)	Payloads					
	$m_p = 0$ kg		$m_p = 1.11$ kg		$m_p = 2.10$ kg	
	Simulation	Experimental	Simulation	Experimental	Simulation	Experimental
f_{n1}	18.297	18.105	6.025	6.326	4.423	4.677
f_{n2}	55.991	50.966	17.647	17.942	13.556	13.274
f_{n3}	104.59	111.99	87.423	92.497	90.152	91.564

It is observed from Table 6.1 and Table 6.2 that the experimental natural frequency results matched well with the simulation results for the smart manipulators and CBM.

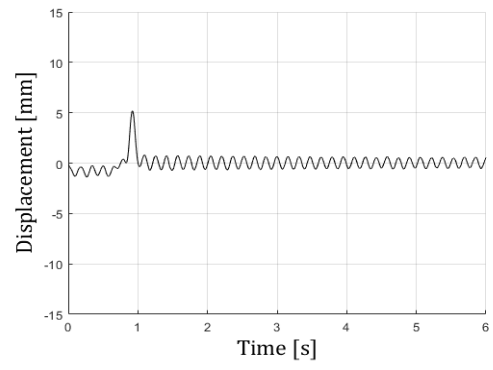
As seen from Table 6.2, since the payload at the endpoint causes flexibility, the natural frequencies of CBM decrease as the payload increases. For example, the first natural frequency of CBM decreases from 18.297 Hz for $m_p = 0$ kg to 4.423 Hz for $m_p = 2.11$ kg. When tables 4.3 and 6.2 for the CBM and SCBM are compared, the PZT actuators increase the stiffness, especially in bending vibration modes and less increment are observed at lower frequencies than at higher frequencies.

6.4 Experiments for Passive Vibration Control

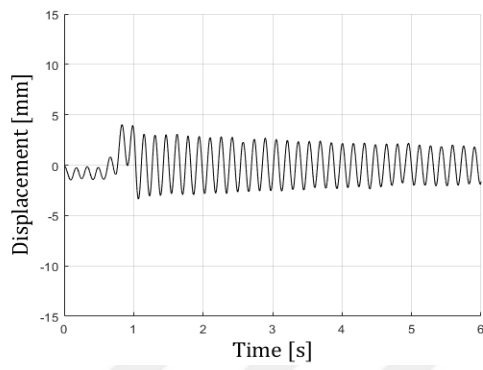
The PC is accomplished by considering the fundamental natural frequency of the smart manipulators given in Table 6.1 and Table 6.2. In the experiments, the motion profiles are considered in Chapter-4. For SM-M1, SCM-M1a, SCM-M1b and $t_{dec} = 1t_{1h}$, $2t_{1h}$, $3t_{1h}$ and $4t_{1h}$, the experimental results are shown in Figure 6.8, Figure 6.9 and Figure 6.10.



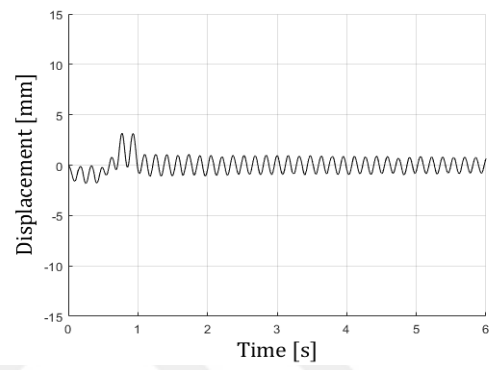
a)



b)

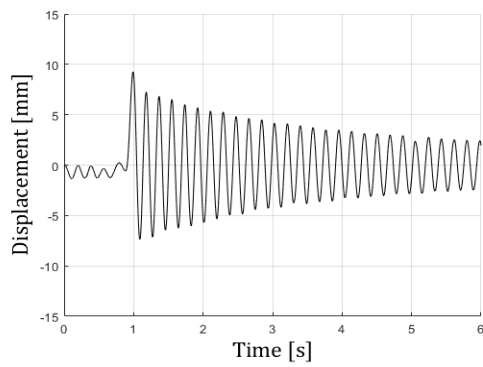


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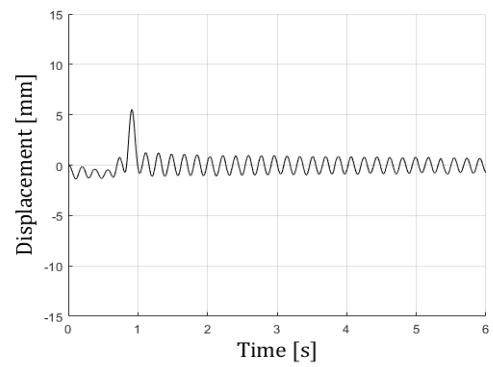


d)

Figure 6.8 PC experimental results for the SM, a) $1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, d) $4t_{1h}$

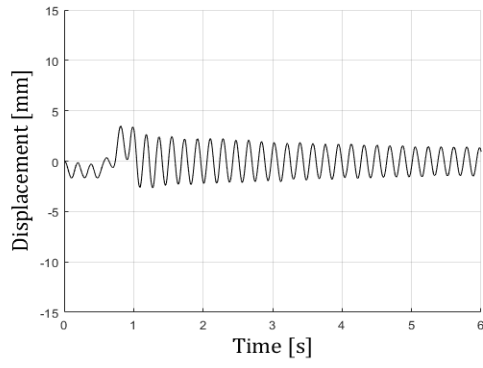


a)

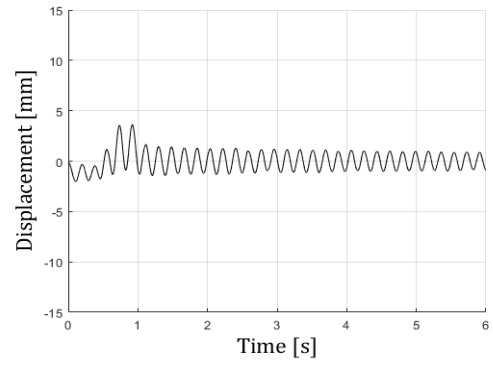


b)

Figure 6.9 PC experimental results for the SCM-[0/90], a) $1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, d) $4t_{1h}$

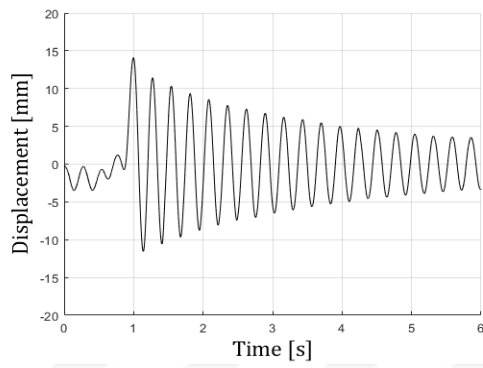


c)

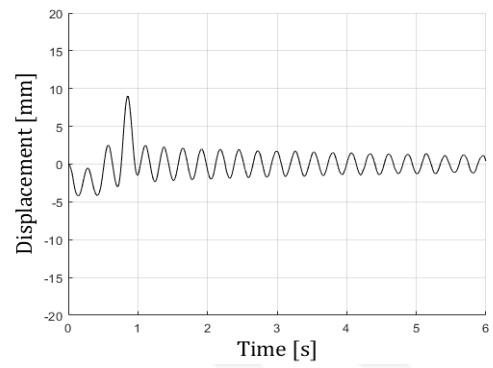


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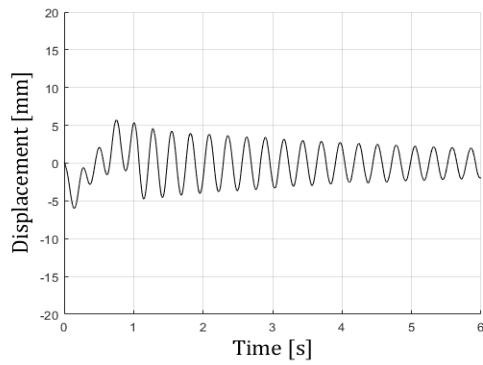
Figure 6.9 continues



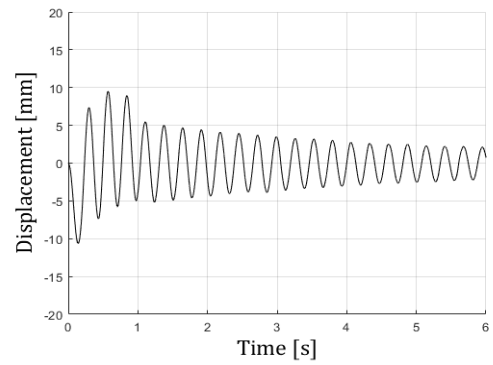
a)



b)



c)



d)

Figure 6.10 PC experimental results for the SCM-[45/-45], a) $1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, d) $4t_{1h}$

In this section, the experimental results are presented as mm for the smart manipulators due to the displacement sensor and m/s^2 for the CBM owing to the

accelerometer sensor used in the experiments. For the CBM, the motion time and angular displacement are chosen as $t_m = 1$ s and $\theta_m = 30$ deg. For $m_p = 1.11$ kg, example PC vibration responses for $t_{dec} = 1t_{1h}$, $2t_{1h}$, $3t_{1h}$, $4t_{1h}$ and $5t_{1h}$ are shown in Figure 6.11.

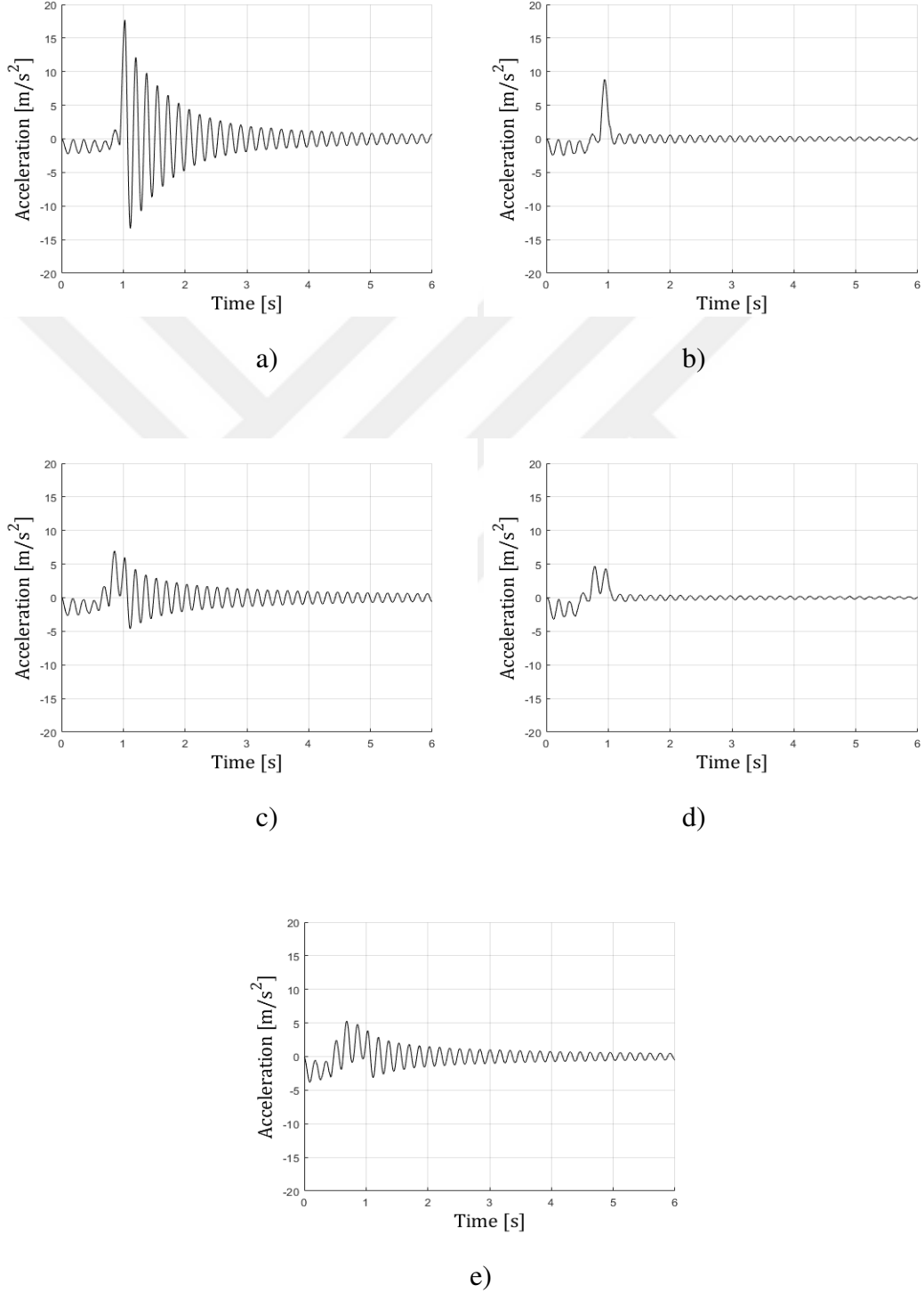


Figure 6.11 PC experimental results for the CBM, a) $1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, d) $4t_{1h}$ and e) $5t_{1h}$

It is observed from the figures that the residual vibration amplitudes for the PC are the highest for $t_{dec} = 1t_{1h}$, while those are the lowest for $t_{dec} = 2t_{1h}$. The detailed results of the PC will be explained for all motion profiles in Chapter-7.

6.5 Experiments for Active Vibration Control

After obtaining the experimental results for the PC, the AC is studied for the smart manipulators. The actuator signal for the experiments is limited to ± 270 V with the saturation considering to the maximum operating voltage of power amplifier. The PD control with the DF is used. The experimental results are obtained with the motion profiles given in Chapter-4. The AC and PC experimental results are shown in Figure 6.12, Figure 6.13 and Figure 6.14.

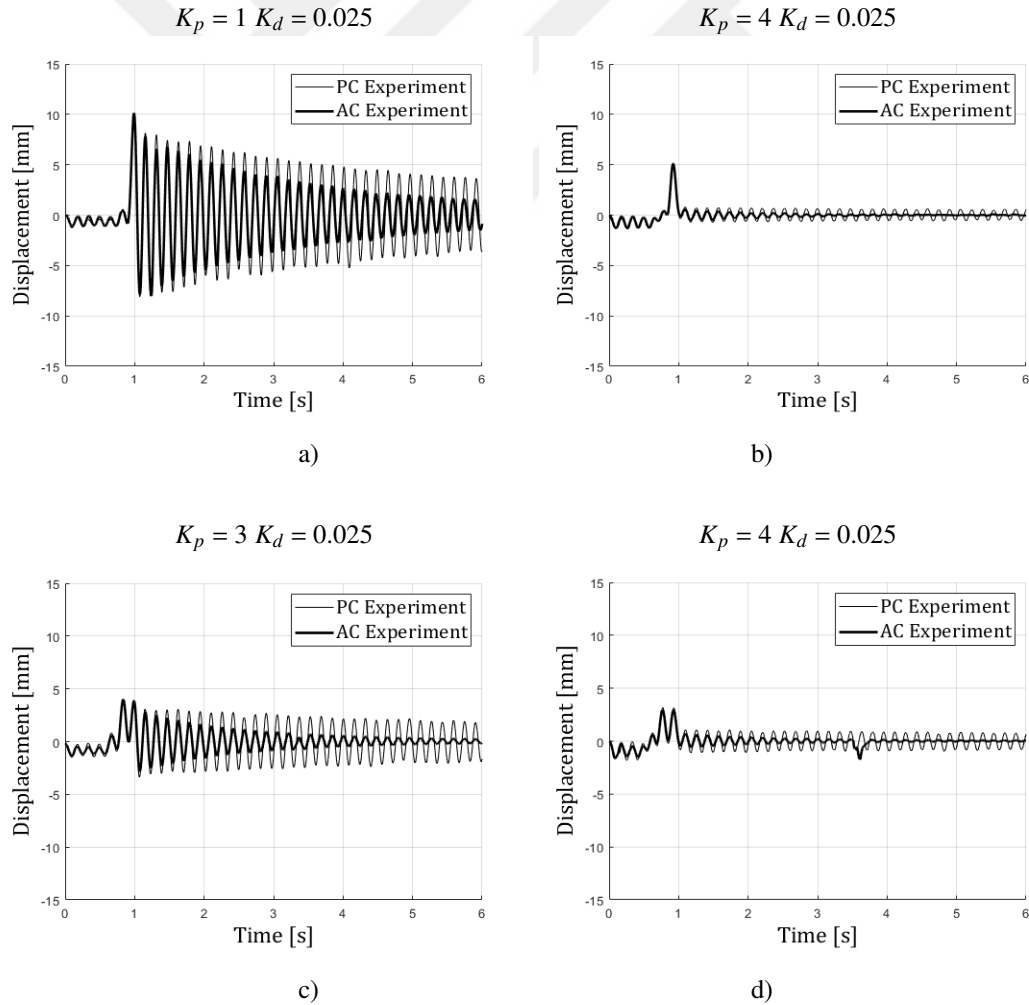


Figure 6.12 Change of PC and AC experimental results of SM for, a) $t_{dec} = 1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, d) $4t_{1h}$

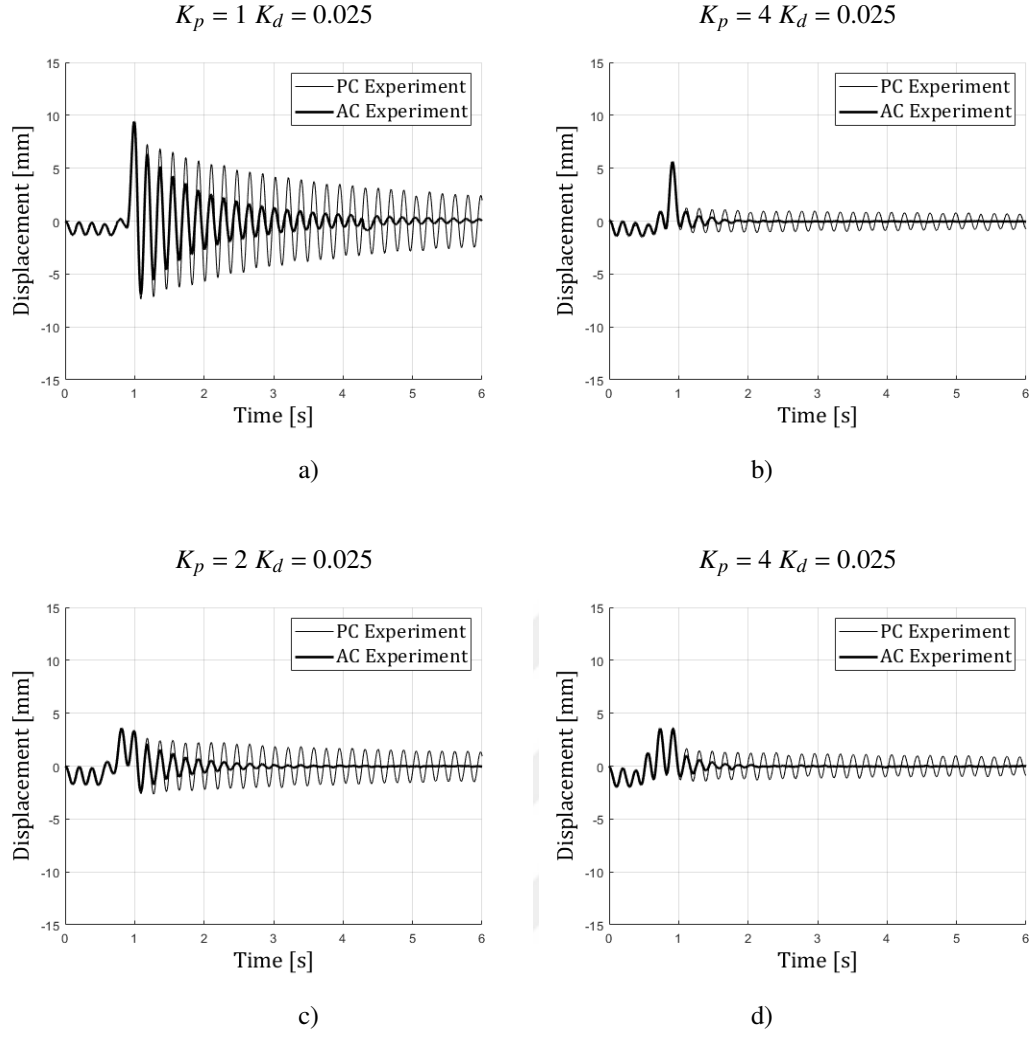


Figure 6.13 Change of PC and AC experimental results of SCM-[0/90] for, a) $t_{dec} = 1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, d) $4t_{1h}$

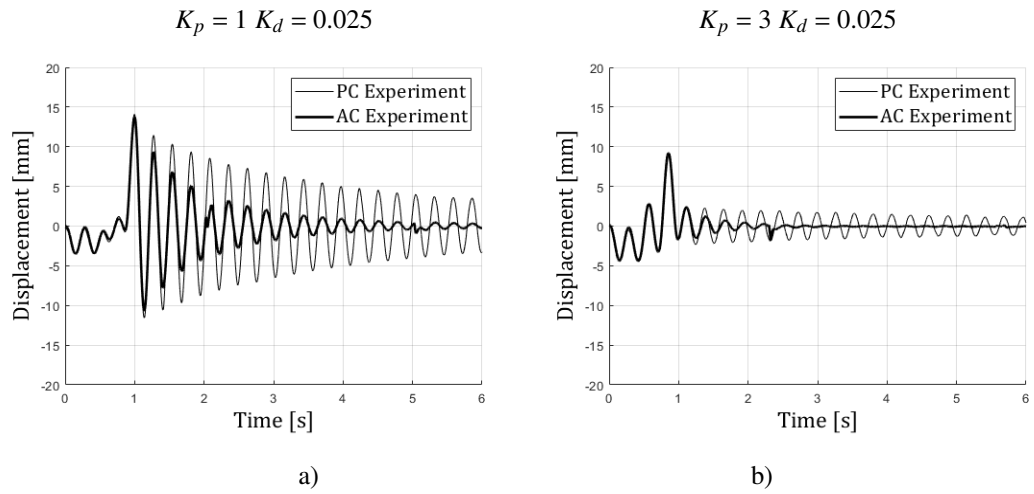


Figure 6.14 Change of PC and AC experimental results of SCM-[45/-45] for, a) $t_{dec} = 1t_{1h}$, b) $2t_{1h}$, c) $3t_{1h}$, d) $4t_{1h}$

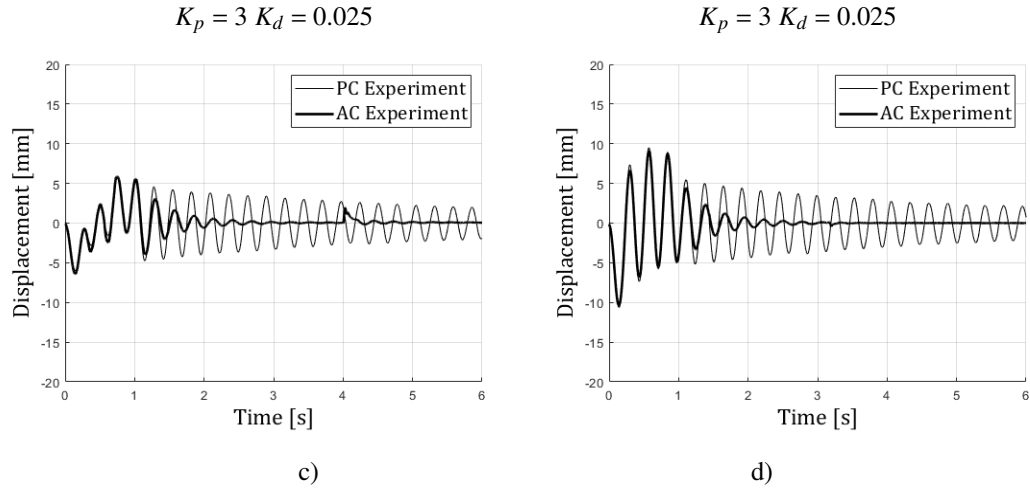


Figure 6.14 continues

The figures show the change of PC and AC experimental results with SM-M1, SCM-M1a, and SCM-M1b. The AC results for the motion profiles in the PC will be presented in Chapter-7. The SS results is compared with the experiments in Chapter-7 in detail.

CHAPTER SEVEN

COMPARISON OF EXPERIMENTAL AND STATE-SPACE ANALYSIS RESULTS

7.1 Introduction

In this chapter, the experimental AC and PC results for all motion profiles are presented to compare with the SS results given in Chapter-4 and Chapter-5. For the AC, the results are presented for the PD control with the DF. The RMS values of the residual vibrations and actuation responses are calculated and listed to compare the vibration responses in the experiment and SS analysis. For each motion profile, the error values are calculated by considering the RMS values of experimental results as reference. Also, the reduction ratios of active and passive vibration controls (APC) are presented for the residual vibrations to evaluate the performance of the AC and PC.

7.2 Passive Vibration Control Results

The motion profiles defined in Chapter-4 are used to drive the smart manipulators to reduce the residual vibrations at the tip point. The results for the passive vibration control is presented.

7.2.1 Results of Smart Manipulators

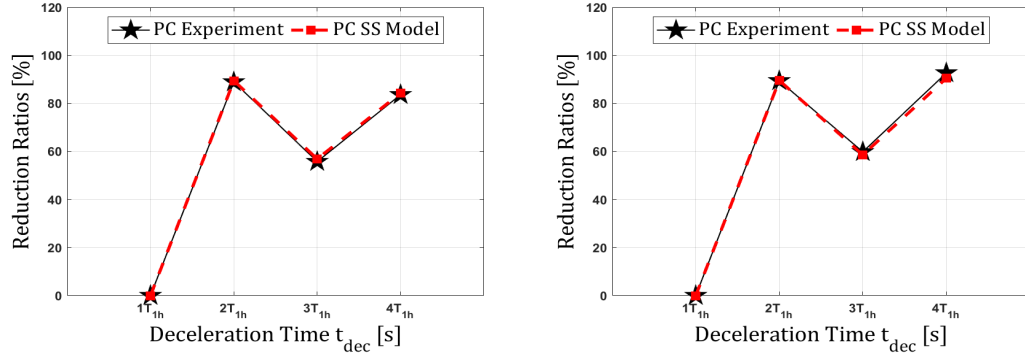
For the PC, the experimental and SS vibration responses are obtained with SM-M1, SM-M2, SCM-M1a, SCM-M2a, SCM-M1b, and SCM-M2b. Then, the RMS and error values of the residual vibrations are computed and given in Table 7.1.

Table 7.1 RMS values of smart manipulators

Motion Profile	t_{dec}	Experiment	SS Model	Error
		RMS [mm]	RMS [mm]	[%]
SM-M1	$1t_{1h}$	3.9344	4.0912	-3.985
	$2t_{1h}$	0.4357	0.4327	0.076
	$3t_{1h}$	1.7399	1.7631	-0.589
	$4t_{1h}$	0.6447	0.6428	0.048
SM-M2	$1t_{1h}$	4.4877	4.4492	0.857
	$2t_{1h}$	0.4758	0.4644	0.254
	$3t_{1h}$	1.8022	1.8433	-0.915
	$4t_{1h}$	0.3281	0.4235	-2.125
SCM-M1a	$1t_{1h}$	3.1231	3.2021	-2.529
	$2t_{1h}$	0.6439	0.6343	0.307
	$3t_{1h}$	1.3470	1.3306	0.525
	$4t_{1h}$	0.8005	0.8207	-0.646
SCM-M2a	$1t_{1h}$	3.6044	3.7841	-4.985
	$2t_{1h}$	0.6900	0.6559	0.946
	$3t_{1h}$	1.4440	1.5045	-1.678
	$4t_{1h}$	0.7732	0.7950	-0.604
SCM-M1b	$1t_{1h}$	4.7962	4.8771	-1.686
	$2t_{1h}$	1.1793	1.2065	-0.567
	$3t_{1h}$	2.2798	2.2819	-0.043
	$4t_{1h}$	2.4986	2.5600	-1.280
SCM-M2b	$1t_{1h}$	5.7681	5.8414	-1.270
	$2t_{1h}$	1.1142	0.9920	2.118
	$3t_{1h}$	2.4284	2.4537	-0.438
	$4t_{1h}$	1.4306	1.4014	0.506

The RMS values are given to determine the reductions in vibration control. The error values are calculated using the RMS values. For SM-M1, the error values are found as -3.9 , 0.07 , -0.5 , and 0.04 for $t_{dec} = 1t_{1h}$, $2t_{1h}$, $3t_{1h}$, and $4t_{1h}$, respectively. It is observed that the error values are less than 5 percent for all motion profiles. Also, it seems that the results are matched well in the PC.

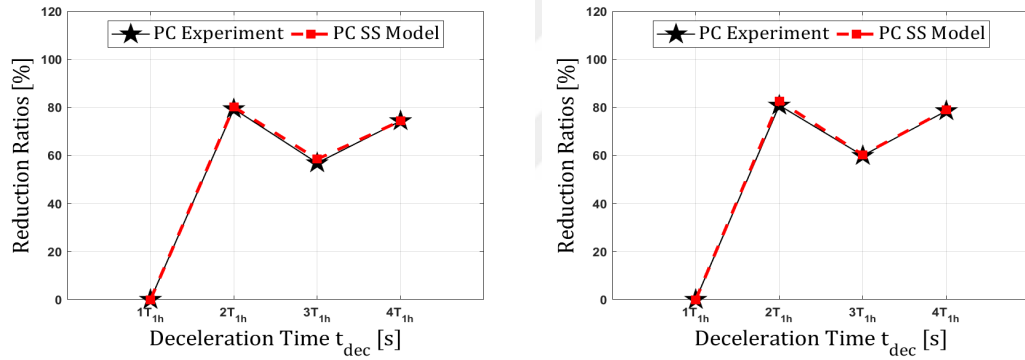
The reduction ratios according to the reference values of $t_{dec} = 1t_{1h}$ are calculated to investigate the reduction in the PC. The change of PC reduction ratios are shown in Figure 7.1 and Figure 7.2.



a)

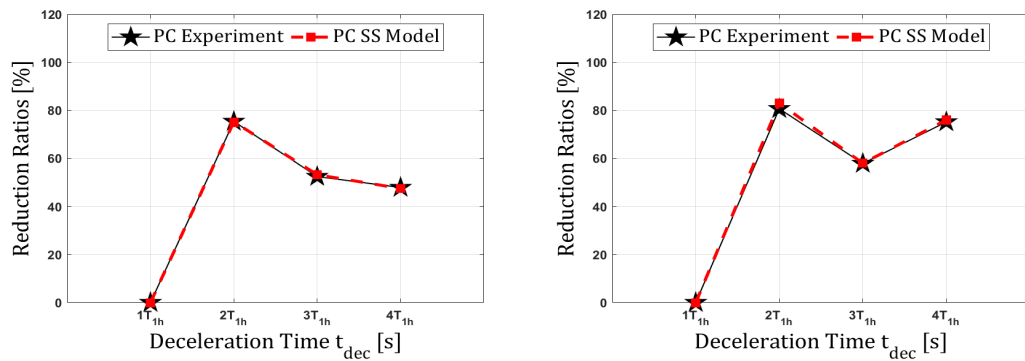
b)

Figure 7.1 Change of reduction ratios in PC for a) SM-M1, b) SM-M2



a)

b)



c)

d)

Figure 7.2 Change of reduction ratios in PC for a) SCM-M1a, b) SCM-M2a, c) SCM-M1b, and d) SCM-M2b

As seen from Figure 7.1 and Figure 7.2, the amplitudes of residual vibrations are the highest for the $t_{dec} = 1t_{1h}$, while those are the lowest for the $t_{dec} = 2t_{1h}$ and $4t_{1h}$ as expected. Also, the results for the $t_{dec} = 3t_{1h}$ are higher than $t_{dec} = 2t_{1h}$ and $4t_{1h}$.

As seen from Figure 7.1, the reduction ratios for $t_{dec} = 2t_{1h}$ are approximately 90% for SM-M1 and SM-M2 in the experimental and SS results. For $t_{dec} = 4t_{1h}$, the reduction ratios are 85% for SM-M1 and 92% for SM-M2. For the SCM-[0/90] and SCM-M1a, the reduction ratios shown in Figure 7.2a) are about 80%, 60%, and 77% for $t_{dec} = 2t_{1h}$, $3t_{1h}$ and $4t_{1h}$, respectively. For the $t_{dec} = 2t_{1h}$, the reduction ratios for the PC are approximately 80% for SCM-M1a, SCM-M2a, SCM-M2b and 77% for SCM-M1b.

7.2.2 Results of Composite Box Manipulator (CBM)

The CBM is considered for the comparison of experimental and SS vibration responses. For $t_m = 1$ s, $\theta_m = 30^\circ$ and $\theta_m = 45^\circ$, the results are obtained and the RMS values are tabulated in Table 7.2.

Table 7.2 RMS values of CBM

Motion Profile	t_{dec}	Experiment	SS Model	Error
		RMS [m/s^2]	RMS [m/s^2]	[%]
SCBM-M1b	$1t_{1h}$	5.1982	5.2466	-4.840
	$2t_{1h}$	0.4947	0.5670	-7.230
	$3t_{1h}$	1.7442	1.6376	10.66
	$4t_{1h}$	0.4801	0.5431	-6.300
	$5t_{1h}$	1.0040	0.9266	7.740
SCBM-M1d	$1t_{1h}$	8.0172	7.8700	14.72
	$2t_{1h}$	0.9739	0.8741	9.980
	$3t_{1h}$	2.6013	2.4564	14.49
	$4t_{1h}$	0.8585	0.8147	4.380
	$5t_{1h}$	1.4417	1.3898	5.190

For $t_m = 1$ s and $\theta_m = 30^\circ$, the error values are found as -4.8, -7.2, 10.6, -6.3 and

7.7 for $t_{dec} = 1t_{1h}$, $2t_{1h}$, $3t_{1h}$, $4t_{1h}$ and $5t_{1h}$, respectively. For $t_m = 1$ s and $\theta_m = 45^\circ$, the error values are the highest for $t_{dec} = 1t_{1h}$ and $3t_{1h}$, while those are the lowest for $t_{dec} = 4t_{1h}$ and $5t_{1h}$. It is observed that the experimental results of the CBM matched well with the SS results according to the error values. The SS model of CBM is verified with the experiments in the PC.

7.3 Active Vibration Control Results

To verify the SS results with the experiments in the AC, the PD control is applied to the smart manipulators to further minimize the residual vibrations at the endpoint. All cases considered in the PC are studied for the AC.

7.3.1 PID Control Results

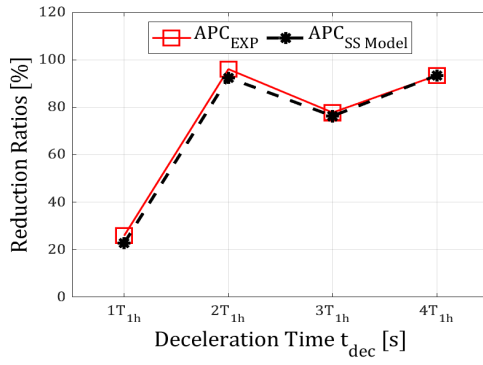
The PD control is considered with the DF for the active vibration suppression. The vibration responses are obtained with the experiments and SS analysis. For the smart manipulators, the RMS values of residual vibrations and actuation results, error values are calculated for the experimental and SS results as listed in Table 7.3.

It is observed that the amplitudes of residual vibrations are reduced experimentally from 3.9344 mm to 2.9154 mm for $t_{dec} = 1t_{1h}$ and 0.1553 mm for $t_{dec} = 2t_{1h}$ by implementing the AC over the PC in SM-M1. For SCM-M1a, the error values are 0.917, -0.573, -3.181, and -2.753 for $t_{dec} = 1t_{1h}$, $2t_{1h}$, $3t_{1h}$ and $4t_{1h}$, respectively. It is observed that the results are matched well with the experiment and SS analysis.

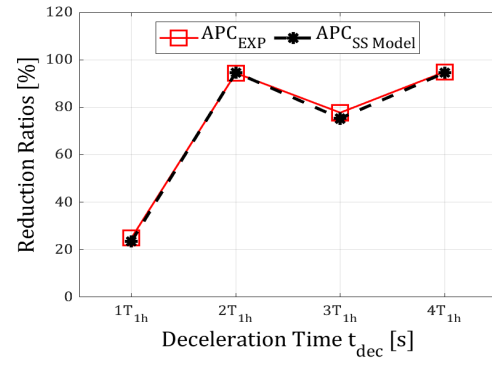
Table 7.3 RMS results of smart manipulators

Motion Profile	t_{dec} [s]	K_p	K_d	Experiment	SS Model	Error	Actuation
				<i>RMS</i> [mm]	<i>RMS</i> [mm]	[%]	<i>RMS</i> [V]
SM-M1	$1t_{1h}$	1	0.025	2.9154	3.0380	-4.205	50.2817
	$2t_{1h}$	4	0.025	0.1553	0.2984	-4.908	12.0074
	$3t_{1h}$	3	0.025	0.8746	0.9321	-1.972	40.6701
	$4t_{1h}$	4	0.025	0.2603	0.2575	0.096	15.5849
SM-M2	$1t_{1h}$	1	0.025	3.3698	3.4334	-1.887	57.9397
	$2t_{1h}$	4	0.025	0.2511	0.2404	0.317	11.2115
	$3t_{1h}$	3	0.025	1.0000	1.1072	-3.181	46.0259
	$4t_{1h}$	4	0.025	0.2292	0.2457	-0.489	13.8865
SCM-M1a	$1t_{1h}$	1	0.025	1.6559	1.6407	0.917	14.5043
	$2t_{1h}$	4	0.025	0.1464	0.1559	-0.573	6.1950
	$3t_{1h}$	2	0.025	0.4736	0.4994	-1.558	8.9112
	$4t_{1h}$	4	0.025	0.1727	0.2183	-2.753	8.6493
SCM-M2a	$1t_{1h}$	1	0.025	1.8766	1.8904	-0.735	16.7883
	$2t_{1h}$	4	0.025	0.1984	0.1997	-0.069	5.6988
	$3t_{1h}$	2	0.025	0.5301	0.5467	-0.884	10.2884
	$4t_{1h}$	4	0.025	0.1593	0.1831	-1.268	7.3165
SCM-M1b	$1t_{1h}$	1	0.025	2.7602	2.7626	-0.087	20.1665
	$2t_{1h}$	3	0.025	0.4394	0.4572	-0.644	10.9216
	$3t_{1h}$	3	0.025	0.9384	0.9860	-1.724	23.1237
	$4t_{1h}$	3	0.025	0.8130	1.0419	-8.292	26.2849
SCM-M2b	$1t_{1h}$	1	0.025	3.5417	3.5425	-0.022	25.1386
	$2t_{1h}$	3	0.025	0.3865	0.4033	-0.474	8.8143
	$3t_{1h}$	3	0.025	1.0034	1.1125	-3.080	25.3798
	$4t_{1h}$	3	0.025	0.5057	0.5908	-2.402	13.7889

Reduction ratios according to the reference values of $t_{dec} = 1t_{1h}$ are calculated to indicate the reduction ratio over the RMS values of AC. The reduction ratios for the APC are calculated. The change of reduction ratios for the APC are shown in Figure 7.3 and Figure 7.4.

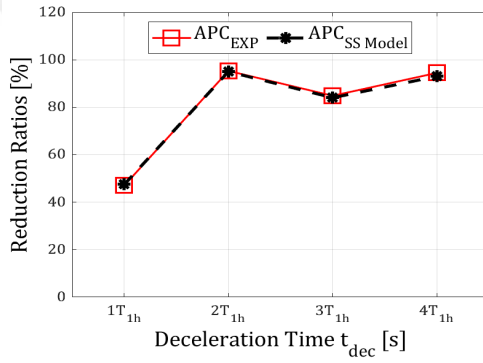


a)

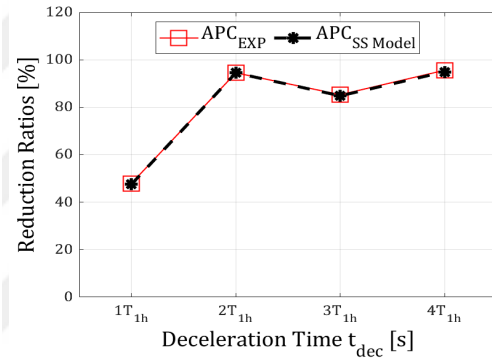


b)

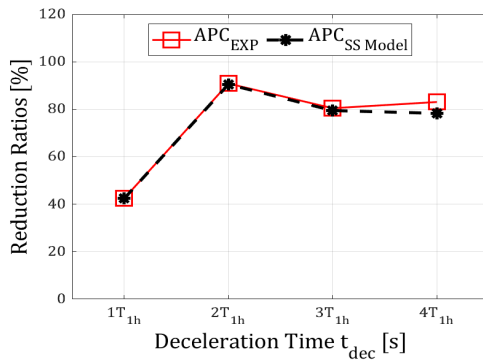
Figure 7.3 Change of reduction ratios for a) SM-M1, b) SM-M2



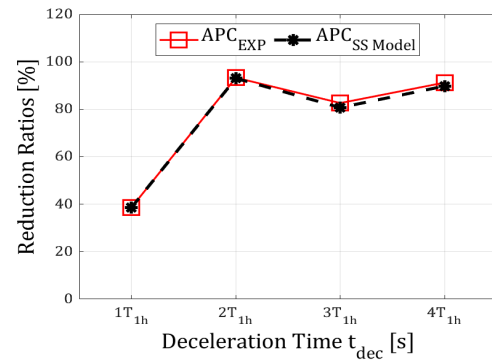
a)



b)



c)



d)

Figure 7.4 Change of reduction ratios for a) SCM-M1a, b) SCM-M2a, c) SCM-M1b, d) SCM-M2b

As seen from Figure 7.3 and Figure 7.4, the APC reduction ratios are approximately 25%, 97%, 79% and 95% for $t_{dec} = 1t_{1h}$, $2t_{1h}$, $3t_{1h}$ and $4t_{1h}$ in the

experiments, respectively. The reduction ratios of experimental and SS results for $t_{dec} = 2t_{1h}$ are found as 97% and 96% for the SM, 96% and 96% for the SCM-[0/90] and 95% and 95% for the SCM-[45/-45]. The best results in the AC are obtained for $t_{dec} = 2t_{1h}$ and $4t_{1h}$. Also, the residual vibrations reduced by the PC are also effectively suppressed by the AC.

The AC aims to further improve the reduction rates in the PC. When compared with the AC and PC results, it is observed that the amplitudes of residual vibrations for the PC further reduce the vibrations with the AC. As seen from Figure 7.3 and Figure 7.4, it is observed that the experimental results for the AC are matched well with the SS results.

CHAPTER EIGHT

CONCLUSIONS

In this thesis, the active and passive vibration control of different smart manipulators with payloads are studied experimentally and theoretically using the triangular and trapezoidal motion profiles. The PC and AC are carried out the simulation and SS analysis. The simulations are performed in ANSYS while the SS analysis is performed in MATLAB. The SS models are extracted from the FE models created in ANSYS. The FE simulations in transient analysis take a long time and a control system may not always be designed. In order to overcome these problems, different control methods are studied with the SS analysis that can be carried out effectively in a shorter time in this thesis.

Firstly, the SS analysis is studied for the 2-DOF mechanical system. Open and closed-loop vibration results are validated for the impulse and step inputs. It is observed that the analytical solutions are matched well with the SS results. Then, the AC and PC techniques are applied to the smart manipulators to reduce the amplitudes of the residual vibrations. Four different configurations with the SM, SCM and SCBM made of aluminum and composite materials are considered.

The PC is conducted using the triangular and trapezoidal motion profiles.. The PD and FSF controls with the DF are performed for the AC. The design techniques are used to optimize the gains of the PD control while the feedback gains of the FSF control are designed with different design criteria. The results of the simulation and SS analysis are also compared. It is observed the simulation results are good agreement with the SS results.

Experiments are performed to verify the SS analysis results in the AC and PC. It is observed that the results matched well the vibration responses with the SS analysis and experiments. The results show that the motion profiles affect directly the performance of the PC.

In order to evaluate the performance of the PC, the deceleration time t_{dec} is

considered for the comparison. For the deceleration time $t_{dec} = 2t_{1h}$ and $4t_{1h}$, reduction ratios are found between 80 – 90% for the smart manipulators while these for $t_{dec} = 3t_{1h}$ is obtained between 52 – 70%. So, the residual vibration amplitudes are the lowest for $t_{dec} = 2t_{1h}$ and $4t_{1h}$, while those are the highest for $t_{dec} = 1t_{1h}$. The amplitudes of residual vibrations for $t_{dec} = 3t_{1h}$ and $5t_{1h}$ are higher than those for $t_{dec} = 2t_{1h}$ and $4t_{1h}$. As a result, it is observed that double values of the deceleration time t_{dec} are more effective to control the residual vibrations at the endpoint in the PC. The reduction ratios of the PC are valid for the SM, SCM, and SCBM.

The AC is performed with the motion profiles used by the PC to move the smart manipulators. The AC and PC are applied together for vibration suppression. The maximum APC reduction ratio is found as 90% for the PC and increased about 99% with the optimized controller gains in the AC. The effect of the AC is increased by optimizing the controller gains. The best results for the APC are obtained with $t_{dec} = 2t_{1h}$ and $4t_{1h}$.

The AC can be purely applied to the smart manipulators without considering the PC. If the AC is only considered, any values for the deceleration time t_{dec} in the motion profiles can be used to move the manipulators. However, The AC together with the PC is considered for vibration suppression.

If the AC is a primary concern, each value of the deceleration time can be used to reduce the residual vibration amplitudes much more. Also, after increasing the reduction ratios with the PC, it is observed that the residual vibration amplitudes are further raised with the optimized AC results. The smart manipulators become flexible due to the payload and more flexible with higher payloads. Therefore, the natural frequencies decrease because of the payload increases. The reduction ratios for the deceleration time $t_{dec} = 2t_{1h}$ and $4t_{1h}$ are calculated as 80% and 84% for no payload, 88% and 87% for 1.1 kg, 90% and 89% for 2.1 kg. It is observed that the reduction ratios increase when the payload increases.

The effectiveness of pure AC on the vibration control is investigated. For the deceleration time $t_{dec} = 2t_{1h}$, the reduction ratios are obtained between 76 – 80%. The

reduction ratios are the highest for $t_{dec} = 2t_{1h}$ and $4t_{1h}$, while those are the lowest for $t_{dec} = 1t_{1h}$, $3t_{1h}$ and $5t_{1h}$. If the actuation RMS values are compared, the RMS values of actuation responses are the highest for $t_{dec} = 1t_{1h}$, while those for $3t_{1h}$ and $5t_{1h}$ are highest than $t_{dec} = 2t_{1h}$ and $4t_{1h}$. When the reduction ratios and actuation responses are compared, it is observed that higher reduction ratios can be achieved with fewer actuation RMS in the AC.

The following issues for vibration control of flexible manipulators can be investigated in the future:

- Different motion profiles can be studied.
- Hybrid controllers can be investigated.
- Different configurations such as bimorph and different actuators such as stack, SMA and Magnetostrictive material can be examined.
- Multi-link manipulators with different geometries can be studied.
- Nonlinear control methods can be investigated.

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APPENDICES

Appendix A: List of Abbreviations

Abbreviation	Meaning
<i>FM</i>	Flexible manipulator
<i>SM</i>	Smart manipulator
<i>SCM</i>	Smart composite manipulator
<i>SCBM</i>	Smart composite box manipulator
<i>CBM</i>	Composite box manipulator
<i>AC</i>	Active control
<i>PC</i>	Passive control
<i>APC</i>	Active and passive vibration control
<i>FE</i>	Finite element
<i>SS</i>	State-space
<i>AS</i>	Analytical solution
<i>2 – DOF</i>	Two degree of freedom
<i>MSD</i>	Mass-spring-damper
<i>LPS</i>	Lumped parameter system
<i>PID</i>	Proportional-integral-derivative
<i>PD</i>	Proportional-derivative
<i>FSF</i>	Full-state feedback
<i>DF</i>	Displacement feedback
<i>PZT</i>	Piezoelectric
<i>PVDF</i>	Polyvinylidene fluoride sensors
<i>SMA</i>	Shape memory alloys
<i>FEM</i>	Finite element method
<i>AMM</i>	Assumed modes method
<i>LPM</i>	Lumped parameter method
<i>E</i>	Elastic modulus

continued on next page

Abbreviation	Meaning
ρ	Density
ω	Natural frequency in Hz
K_m	Rotational spring constant
VA	Control voltage
M1	Trapezoidal velocity profile
SM – M1	Trapezoidal velocity profile for smart manipulator
M2	Triangular velocity profile
RMS	Root means square
FFT	Fast fourier transform
R	Reduction ratio
Z – N	Ziegler-Nichols
IPDT	Integrator Plus Dead Time
CHR	Chien-Hrones-Reswick
ISE	Integral of squared error
ISA	Integral of absolute magnitude of error
LDS	Laser displacement sensor
WBS	Wireless base station
WAS	Wireless acceleration sensor
AI	Analog input
AO	Analog output
PPF	Positive position feedback
EMPPF	Effective multi-mode positive position feedback
MPC	Model predictive control
LQR	Linear quadratic regulator
SDRE	Nonlinear state-dependent ricati equation

Appendix B: Some LABVIEW Codes for Experiments

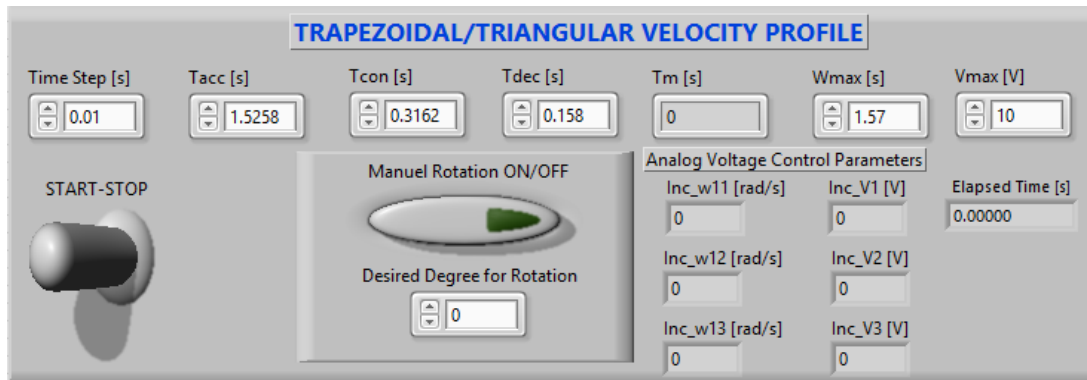


Figure B.1 Servo driving part of front panel in LABVIEW code

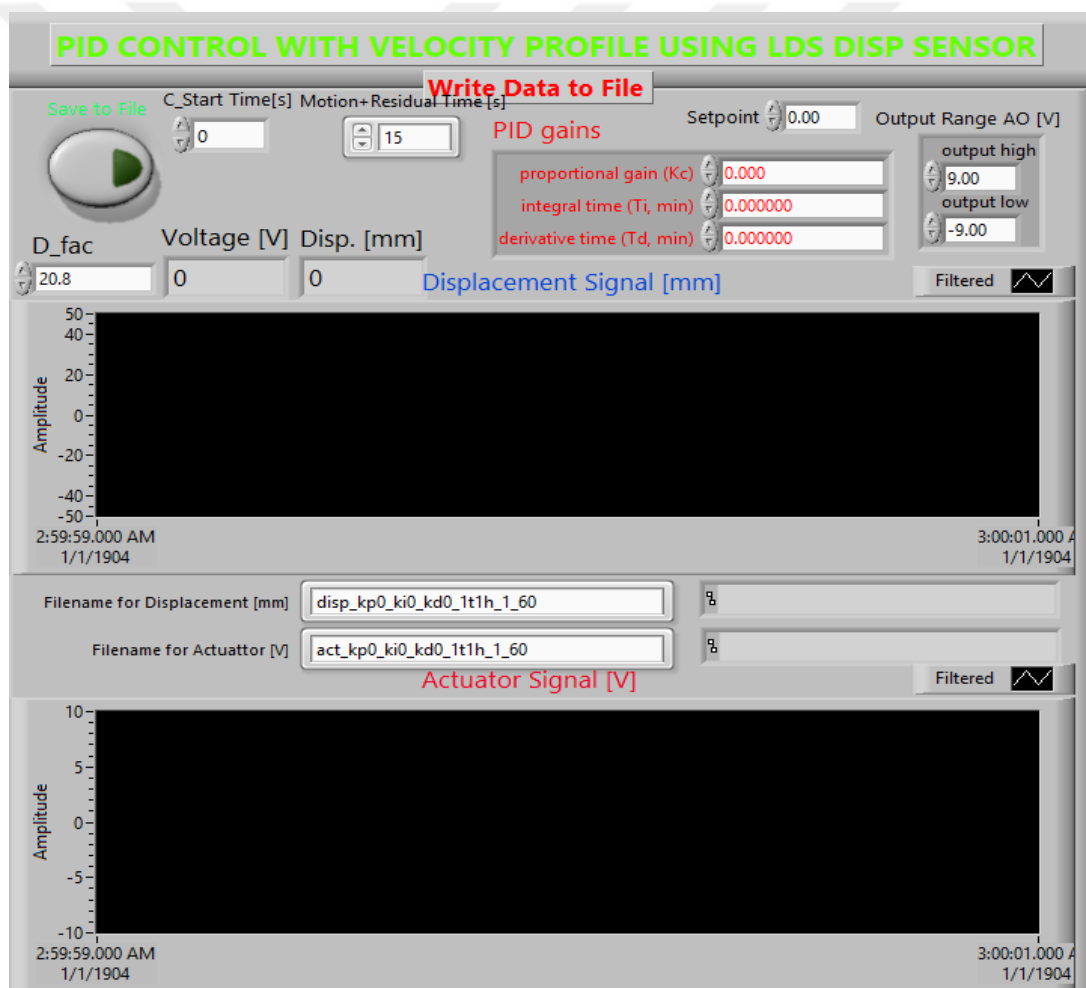


Figure B.2 PID Control part of front panel in LABVIEW code

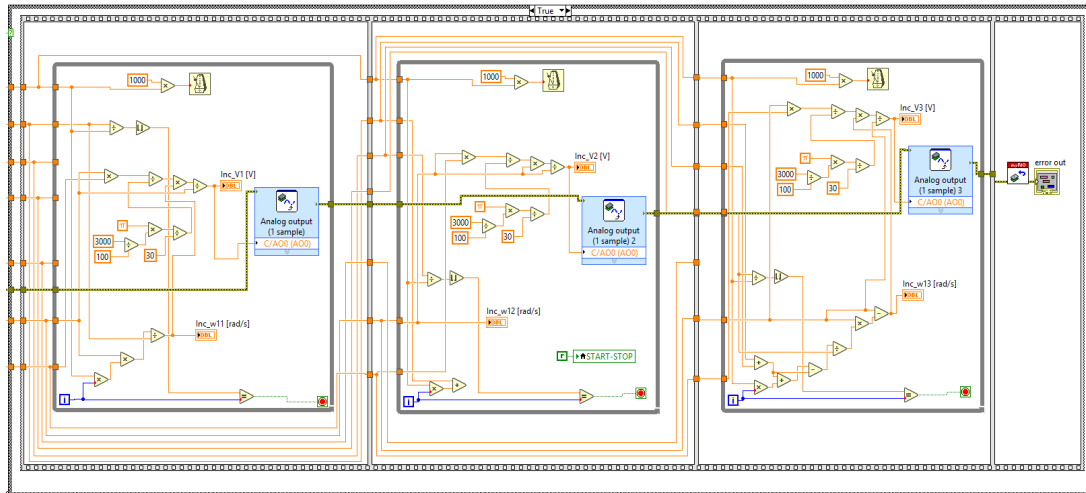


Figure B.3 Servo driving part of block diagram in LABVIEW code

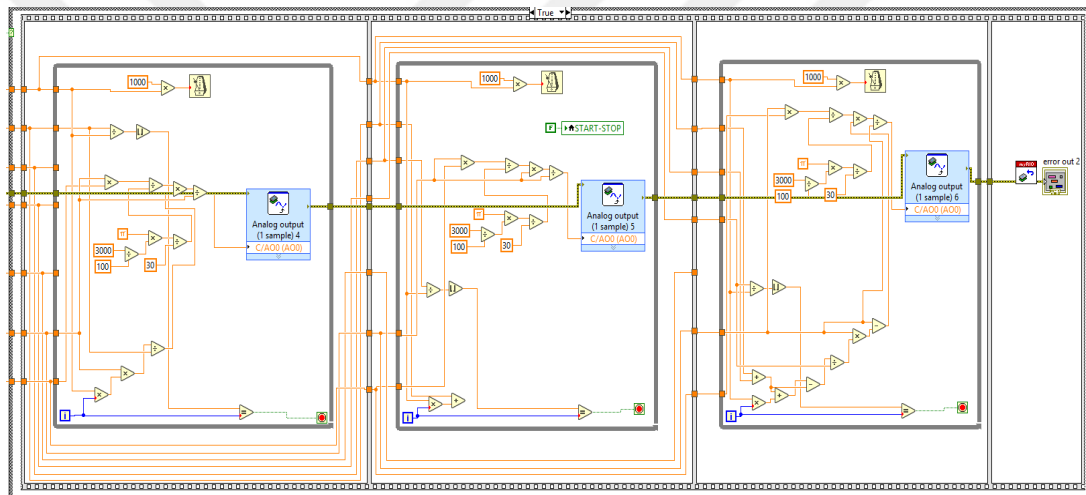


Figure B.4 Servo driving part for desired angular position in LABVIEW code

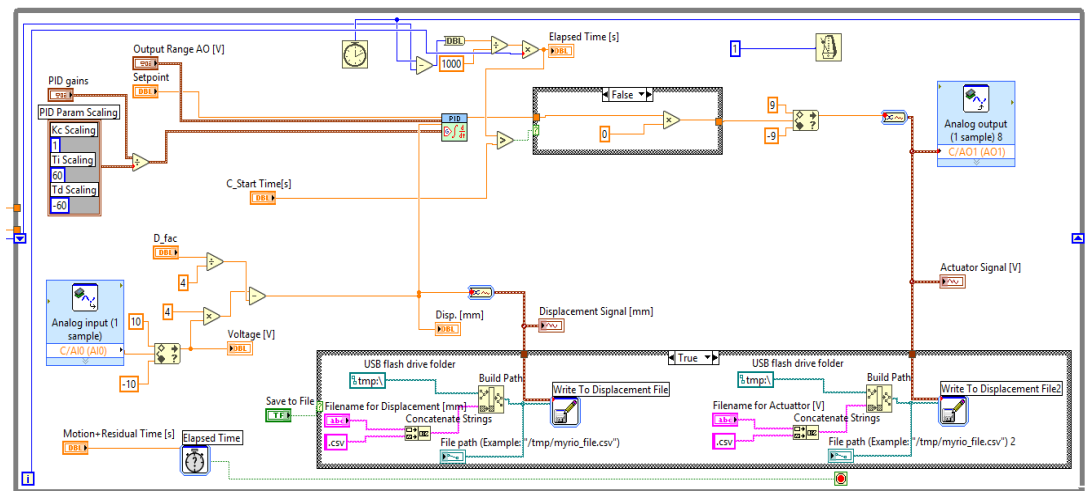


Figure B.5 PID control part of block diagram in LABVIEW code