Sway Reduction of Tower Crane

by

Muhamad Zulhairie bin Moideen

Dissertation submitted in partial fulfilment
the requirements for the
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(Mechanical Engineering)

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CERTIFICATION OF APPROVAL

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Approved by,

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(Mr. Azman bin Zainuddin)

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

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CERTIFICATE OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified source of persons.

Muhamad Zulhairie bin Moideen

ABSTRACT

Tower crane is a common fixture at any major construction site. They often rise hundreds of feet into the air, and can reach out just as far. Tower crane has a problem that a fast transfer of the load causes sway of the load. It takes a long time sometime to lift up and unload the load due to the wind and other disturbances. The objective of this project is to minimize the sway of the load suspended on the tower crane by developing a control method by using a concept of spherical pendulum and delayed position feedback. Develop equation for controller and solve using numerical simulation. From the result gained, the best method can be used to minimize the sway of the tower crane.

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Last but not least, I would like to thank all others that have contributed in completing this project. I really hope this project is beneficial to the society.

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CHAPTER 1
INTRODUCTION

1.1 Background of Study

Tower crane is a common fixture at any major construction site. They often rise hundreds of feet into the air, and can reach out just as far. The construction crew uses the tower crane to lift steel, concrete, large tools like acetylene torches and generators, and a wide variety of other building materials.

Tower cranes are typically used in a construction area to transport all kinds of equipment and materials. Tower crane has a base which is bolted to a large concrete pad that supports the crane. The base connects to the mast (or tower), which gives the tower crane its height. Attached to the top of the mast is the slewing unit which is consisting of gear and a motor that allows the crane to rotate as shown in Figure 1.1. On top of the slewing unit are two parts which are the jib and machinery arm. The long horizontal jib (or working arm), is the portion of the crane that carries the load. A trolley runs along the jib to move the load in and out from the crane's center. The horizontal machinery arm contains the crane motors and electronics as well as the large concrete counter weights (Figure 1.2).

Most of the cranes use their own mechanism to control the activities during the operation. The control system and the operator command are the most important things to make the operation run smoothly with the small value of sway angle or pendulation of payload.

The major task of this project is to develop a controller to reduce the sway of tower crane load by using a delayed position feedback controller. Several methods to reduce sway have been developed by King [1], Maarten De Munck[2] and Kenneth N[3] are presented in the literature review.

Figure 1.1: Slewing unit

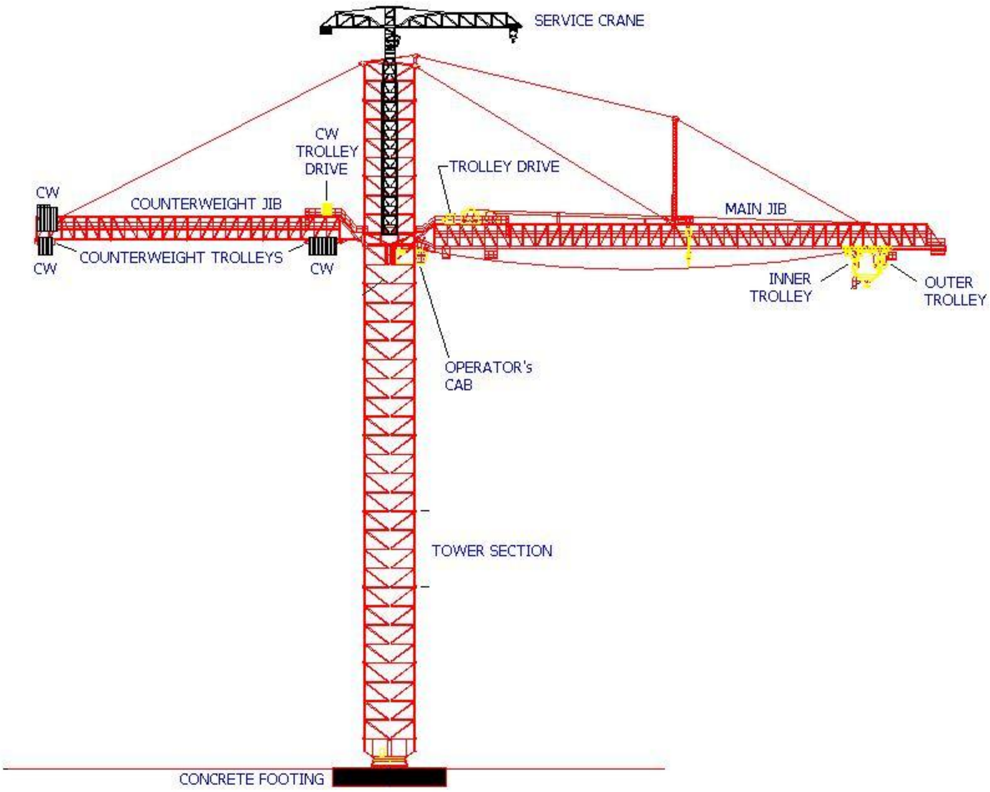


Figure 1.2: Tower Crane model

1.2 Problem Statement

1.2.1 Problem Identification

Tower crane has a problem that a fast transfer of the load causes sway of the load and the vibration of the flexible structure.

The difficulty with a tower crane is the fact that the movement of the load consists of a superposition of a rotation and a translation which involve motion at x, y, and z axis.

There are many factors and disturbances that can contribute to load sway such as wind and speed or velocity of the trolley controlled by the crane operator. It is also because of the control system used is not efficient enough in controlling the load sway.

The load sway gives a greater impact to the operation and the structure itself. The sway can slow down the operation hence will lead to project delay. It also can lead to structural vibration of the tower crane which can affect the stability of the tower crane when handling the load.

1.3 Objectives of the Project

In order to complete the project within the time limit and the scope given, several objectives for this project have been identified and listed such as below:

- Develop a control method to minimize the sway of the tower crane load.
- Develop a mathematical model based on the dynamic of hoisted payload and spherical pendulum concept using MATLAB.
- Simulate and analyze the control method through experiment.

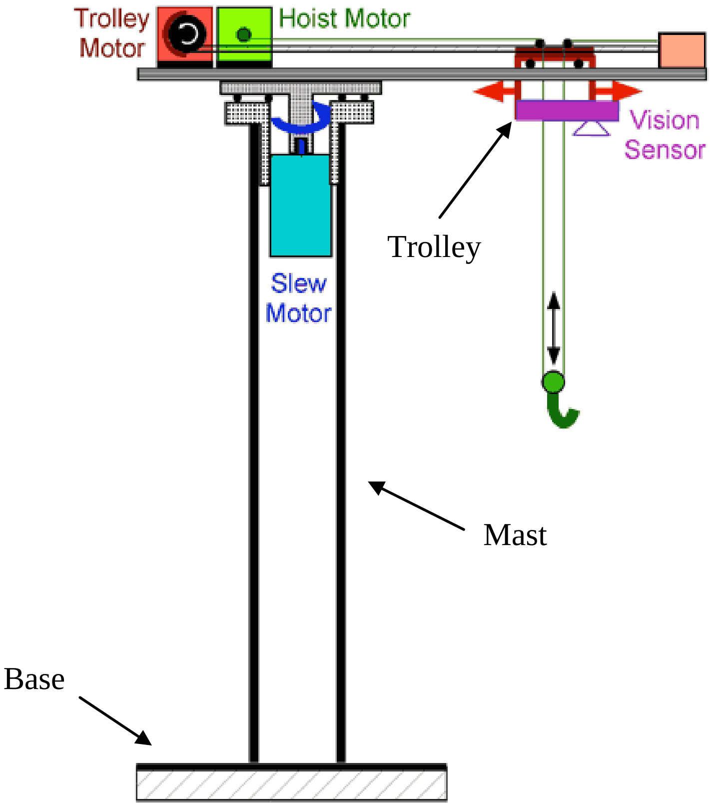


Figure 1.3: Tower crane main components

CHAPTER 2

LITERATURE REVIEW

Tower cranes are widely used in the construction of high-rise buildings, where these cranes may rise hundreds of meters into the air with the required reach to cover the working area. As compared to mobile cranes, tower cranes are normally designed and operated in a relatively severe environmental condition with the wind speed up to about to 36 m/s. the analysis of the dynamic characteristics is therefore significant for both the design and operation of such cranes.

Considering the fact that there is almost no damping in the load suspension system, it will take some time before the sway of the load would stop. A lot of time and money can be saved with an adequate anti-sway control mechanism. There a three dimensional nonlinear model of the tower crane motion is derived. The model consists of rigid body so all structural dynamics are ignored. Secondly an adequate control strategy is computed based on a linearized version of the model with adaptive parameters.

2.1 Concept of Spherical Pendulum

A method that is going to be utilized is by using the concept of spherical pendulum with moving support. It represents the crane systems which have a pendulum as payload and moving support as the crane boom.

Dynamical systems contain nonlinearities that play an important role in modeling and simulating crane systems. They create unpredictable behavior that is often hidden or neglected in traditional solutions. A simple dynamical system, the spherical pendulum, is introduced to illustrate issues, principles, and effects of chaos in dynamics. The spherical pendulum is two degree of freedom nonlinear system with a pivot point in space. The equations of motion for the pendulum are derived, simulated, and animated. A periodical perturbation is applied to the pivot point producing radically different behavior.

The crane configuration consist of a payload mass that swings like a spherical pendulum on the end of a lift-line which is attached to a boom capable of hub rotation (slewing). Positioning of a payload is accomplished through the hub and the load-line length. Since the configuration of the crane effects the excitation and response of the payload, the swing control scheme must account for the varying geometry of the system. Adaptive forward path command is employed to remove component of the command signal which induce payload swing.

2.2 Modeling of the Payload-Pendulation Problem

The nature of the payload-pendulation problem was explored in a number of studies. Most of these studies concluded that large load motions necessitate nonlinear analysis of the crane dynamics. One of the first to examine the nonlinearities involved in the dynamic response of a boom crane were Elling and McClinton [8]. They modeled the cable-payload assembly as a spherical pendulum subject to base excitations applied at the boom tip. Using numerical simulation, they solved the equations of motion of the pendulum subject to harmonic base excitations. Their results show a resonant response when the excitation frequency is near the natural frequency (primary resonance) or one-half the natural frequency (secondary resonance) of the assembly. Miles [9] examined the weakly nonlinear response of a lightly damped, spherical pendulum to a simple harmonic, planar displacement of the point of suspension. He found that nonplanar motions could be excited due to the nonlinear interaction between the two modes of oscillation.

A planar model of the in-plane motion of the crane was reduced to a Mathieu equation, thus showing that the load can be parametrically excited due to the vertical motion of the boom tip. Nojiri and Sasaki [10] found that payload pendulations due to an excitation frequency near the resonance frequency of the cable-payload assembly have a pronounced effect on the roll and pitch motions of the crane vessel under the influence of both regular and irregular waves.

Posiadala et al. [11] modeled the cable-payload assembly in a truck-mounted boom crane as a spherical pendulum. Base excitations due to the boom slew, luff, and telescopic (extension) motion and forcing due to cable reeling/unreeling were introduced into the pendulum's equations of motion. Numerical simulations were used to solve for the forced payload response under various motion combinations for 10 s and then its free response for the subsequent 10 s. They found that, except in the absence of slew motion, the payload response is three-dimensional and cannot be considered as a planar phenomenon. Posiadala et al. [11] extended the model to account for the flexibility of the cable as a Kelvin-Voigt body. They derived equations of motion describing the position of the payload and the dynamic stretching of the cable. The results showed a fast frequency component in the tension force in the cable, representing the oscillations due to the cable's dynamic stretching. Posiadala [11] to model a truck crane on an elastic support. He modeled the crane as a rigid body and the supports as elastic springs. He derived equations of motion describing the position of the payload and the position and orientation of the crane. Numerical simulations showed that the free response of the payload is quasi-periodic. The slow frequency in the (in-plane and out-of-plane pendulations) was due to the natural frequency of the cable-payload assembly, while the fast frequency was due to the support response and the resulting base excitations of the system at the boom tip. Towarek [12] derived a model of a truck-mounted boom crane standing on and interacting with a flexible soil.

The crane platform was modeled as a rigid body undergoing small oscillations, the boom as a flexible beam, the cables an elastic string, the cable-payload assembly as a spherical pendulum, and the soil as a viscoelastic Kelvin-Voigt body. He solved the equations of motion of the system for a complete revolution of boom slew at two different speeds. The system response showed that the crane was oscillating with a narrow band of frequencies, thus producing base excitations of the cable-payload assembly at its point of suspension (boom tip).

2.3 Control of Payload Pendulation

Many researches investigated control of payload pendulations for fixed-base cranes. The majority of control techniques were developed for gantry cranes. While some controllers were originally designed for boom cranes, others were modifications of earlier work on gantry cranes. Two main approaches can be identified among these researches: one targets pendulation suppression through out the whole transport maneuver, while the other is more concerned with pendulation suppression at the end-of-maneuver, so-called\elimination of residual pendulation". In both approaches, limited research included the operator as a part of the model plant. Lewis et al. [13] and Parker et al. [14] presented a three-dimensional linear model of a boom crane. A controller applies quasi-static filters to the operator's input commands to avoid exciting the natural frequency of the cable-payload assembly. Experimental results showed a significant reduction in both the in-plane and out-of-plane payload pendulations. However, the counter-intuitive delay of all operator inputs continues to exist.

Gustafsson [15] presented a three-dimensional nonlinear model of a boom crane. Two independent, in-plane and out-of-plane, linear position feedback controllers were designed based on a linearization of the model. Computer simulations were conducted. The results showed stable responses for commanded slewing rates away from the natural frequency of the cable-payload assembly. As soon as the ratio of the slewing rate to the natural frequency of the cable-payload assemblyapproached1=2, the controller failed due the impact of nonlinearities in the system. Burg et al. [16] reported that the neglected nonlinearities in a linearized state-space model may significantly impact the performance of the linear controller. Computer simulations showed that a linear controller applied to the system provide an acceptable performance only within a fixed operating range of small pendulation angles around the equilibrium point of the payload.

Hara et al. [17] presented a linear model of a planar telescopic boom crane. To control pendulations due to the telescopic motion of the boom, they designed an LQR controller using the boom telescopic motion as control input. A saturation condition was applied to the controller input to keep it within available control authority. In computer simulations and actual testing, the control strategy was successful in pendulation suppression.

Souissi and Koivo [18] modeled a boom crane as a spherical pendulum. They designed a two-step control scheme. A PID controller to track a reference trajectory using the slew and luff of the boom and the reeling/unreeling of the cable and a PD controller to dampen payload pendulations. Numerical simulation of the boom performing a luffing-slewing-
luffing maneuver at constant cable length showed significant payload pendulations, as high as 15 indicating that the linear controller is not effective in damping pendulations.

Kimiaghalam et al. [19] applied a fuzzy control approach to the level of Coulomb friction in the Maryland rigging. They also proposed a fuzzy controller to change the damping constant in an active friction adaptation of the Maryland rigging. The performance of the first approach was inferior to the original Maryland rigging setup, while the second was comparable to that of the original Maryland rigging. Kimiaghalam et al. [20] proposed a fuzzy logic controller to change the boom angle and the length of the cable on which the pulley rolls in the Maryland rigging. The performance of the proposed controller was inferior to that of the original Maryland rigging. Kimiaghalam et al. [21] proposed a feedforward controller and applied it to the boom of a Maryland rigged crane to reduce the equilibrium point displacement of the pulley due to the ship rolling. Another feedback controller was added to dampen payload pendulations by changing the length of the pulley cable. The combination was effective in suppressing load pendulations due to ship rolling and initial disturbances. However, the feedback controller assumed full authority on the lengths of the two segments of the pulley cable independently and hence the pulley

2.4 Anti-Sway Control Tower Crane

Controllers are designed to transfer the loads from point to point, as fast as possible, and at the same time the load swing is kept small during the transfer process. Moreover, variations of the system parameters such as cable length are taken into consideration by Maarten De Munck [4]

Controlling the sway of the load of a tower crane increases safety, simplifies its operation, and increases the speed of load positioning, thereby saving time and money. The development of this anti-sway controller consists of several parts: modeling the dynamic crane behavior, selection of an appropriate control structure, tuning of the control parameters and validation based on the developed crane model, implementation of the controller on a crane, fine tuning and experimental validation of the performance.

The development of a non-linear dynamic crane model was completed in 2002. Based on that model, an appropriate control structure was selected, tested and improved. The selected control structure consists of two independent adaptive controllers (one for the tower, and one for the trolley) consisting of model-based feed forward and 2dof sway angle measurement feedback. The control parameters are independent of the load, but vary with the length of the cable. Due to the limited availability of a crane, only the feed forward controller could be implemented successfully. The feed forward controller is able to reduce the sway angle by a factor of 3 or more, which is significant. The main advantages of this feedforward controller are its simplicity and robustness, and the fact it requires no sensors. Its main disadvantage is that it can only avoid load sway induced by the motion of the crane, and cannot react on disturbances, such as wind. This requires a feedback controller that will be further developed and implemented in 2004.

A system is disclosed for eliminating sway of a load in a crane or crane-like system subject to operator's command. The load is suspended by a cable from a horizontally movable trolley and can be hoisted vertically. The system uses the principle of cancellation to eliminate sway even when the crane has simultaneous trolley and hoisting motions. The system takes into account the full dynamical effect in computing cancellation signals. The use of a family of ordinary differential equations for the computation of the cancellation controls is a key component of the invention. In computing these controls, the differential equations are solved in real time using sensory measurement of the cable length and its time derivative. The cancellation controls handle the sway induced by the operator's command. Sway can also be induced by other factors, like wind load and external disturbances. This system also includes a feedback mechanism for eliminating sway due to such factors. The system ensures saturation limits, corresponding to the velocity and acceleration limits of the drive system of the trolley are not exceeded for proper functioning of the system.

2.5 System Modeling Assumption.

From the research, there are a few assumptions made by others to simplify the process of system modeling. Below are the assumptions made by King [1]:

a) Ignored the trolley friction.
b) The trolley and the payload can be considered as point of masses.
c) Ignored the tension force that may cause the hoisting rope elongate.
d) The trolley and the payload are assumed to move in two dimensional only which is x-y plane.

Claudio [6] et al. assumptions are:

a) The trolley can move only in one direction i.e. only a planar swinging of the load is considered.
b) The load can be regarded as a point mass.
c) The cable has negligible mass and it is rigid.

All these assumptions are important in designing and develop the mathematical model and the controller. Assumption is necessary and important because it simplifies the further work. All assumptions stated above must be taken into consideration in this project.

2.6 Control Strategies (Kriss Smolders) [7]

Two control strategies have been examined so far: proportional feedback of the sway angle and position feedback of the load position. The first one leads to a simple controller designed with the root-locus method. The second one is a triple lead compensator to secure stability combined with an extra PI-controller to improve the tracking behaviour.

To deal with the parameter dependency of the models, a gain-scheduling algoritme will be used to insure stability at all times.

CHAPTER 3

PROJECT METHODOLOGY

3.1 Procedure Identification

This project will be based on the following project design. This methodology was design to have a basic view of the project and as preparation for this project. Figure 3.1 below shows the design that will be followed in carrying out this project. The details of every stage will be identified after this stage. Researches will be done from time to time so that the best methods and formula could be determined in this project. It was a self study to detail understand and to get as much as knowledge and information about this project. Best methods used will contribute to the best result.

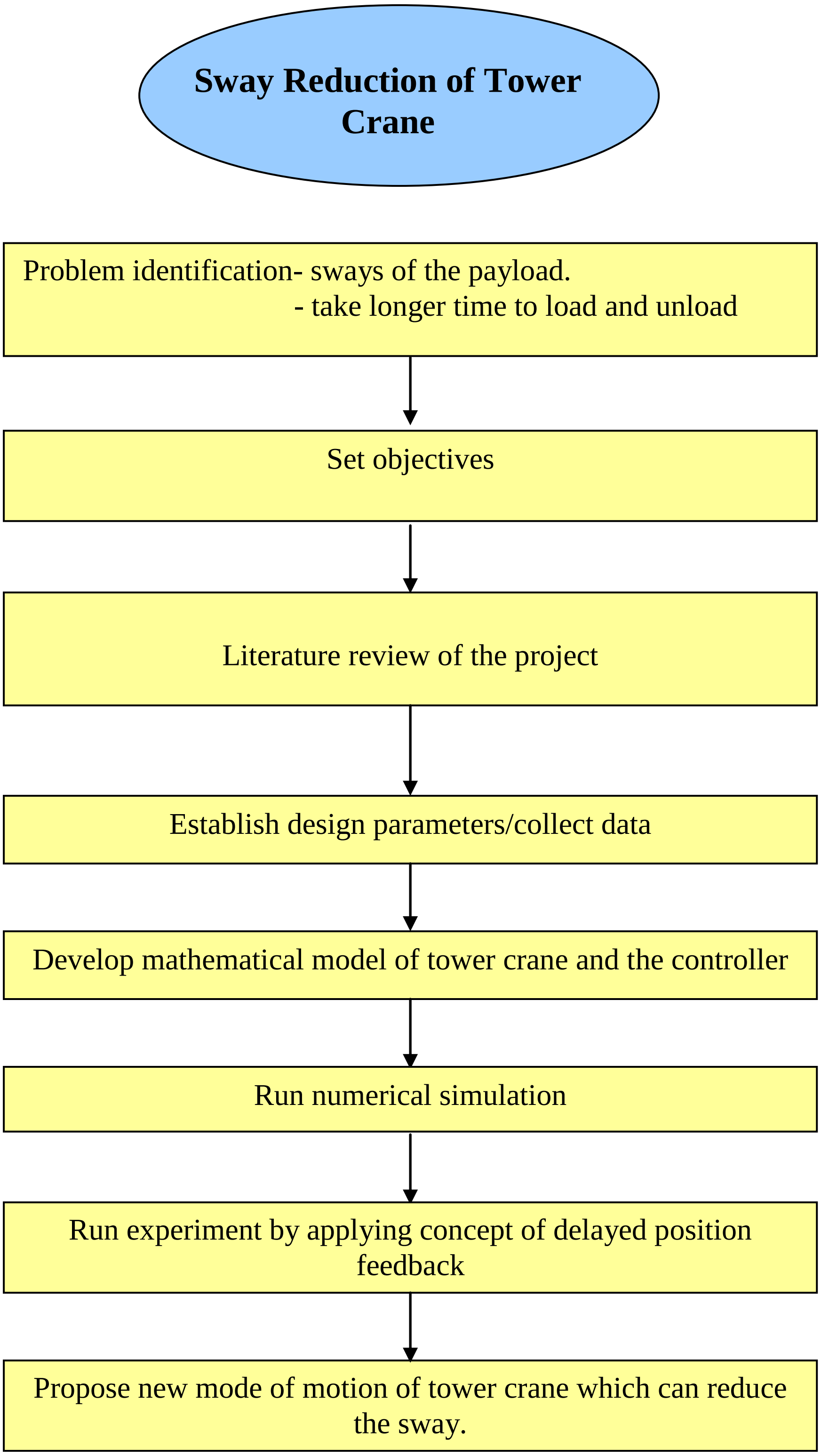


Figure 3.1: Project Design Flow Chart

3.2 Tools Required

Numerical simulation in MATLAB software is used to compute and solve complex mathematical equation. It also used to simulate equation in getting the result such as in a form of graph. A mathematical model including geometric and kinetic nonlinearities can be developed. The mathematical model was then solved numerically to analyze its nonlinearities.

CHAPTER 4

RESULT AND DISCUSSION

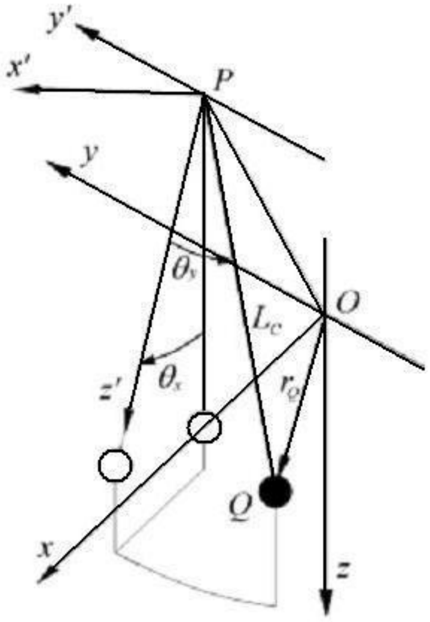
In theory, delayed-position feedback produces damping in the system; consequently, there is the expectation that the oscillations amplitudes will be significantly reduced

Furthermore, the control strategy was applied to, and tested on, an experimental scale model of a land-based tower crane. The control input was introduced through the rotary and gantry degrees of freedom of the crane. The controlled crane was tested in both rotary and gantry modes of operation.

4.1 Dynamics of Hoisted Payload

4.1.1 Mathematical Model by Todd [22]

Tower crane exhibits the typical dynamic behavior of a forced spherical pendulum, including chaotic and/or nonplanar responses to strictly planar excitations at frequencies near the natural frequency of the payload pendulation. In this work, the model used to develop the controller was a spherical pendulum with an inextensible massless cable and a massive point load, as shown in Fig 4.1. Points P and Q represent the boom tip and the load, respectively. The cable length is Lc



$\theta_x =$ Pendulation angle in x
$\theta_y =$ Pendulation angle in y

Lc = Length of the cable

Figure 4.1: A schematic diagram of the hoisting cable model.

To describe the orientation of the cable with respect to the inertial frame (x;y;z), it uses a sequence of two angles, represented by θ_x and θ_y in Fig 4.1. These two angles will be referred to as the in-plane and out-of-plane pendulation angles of the pendulum, respectively. We begin with the cable aligned parallel to the z-axis and then rotate it

Through the angle θ_x around an axis through P that is parallel to the inertial y-axis. This step forms the (x; y; z) coordinate system. Finally, we rotate the cable about the newly formed x-axis through the angle θ_y. The position of point P in the inertial frame is given by (x (t);y (t);z (t)). It follows that the inertial position r_Q of Q is given by

$$r_Q = \lfloor x_p(t) + \sin(\theta_x(t))\cos(\theta_y(t))L_c \rfloor i + \lfloor y_p(t) i \sin(\theta_y(t))L_c \rfloor j \\ + \left[z_p(t) + \cos(\theta_x(t))\cos(\theta_y(t))L_c \right] k \tag{4.1}$$

4.1.2 Analysis of Spherical-Pendulum Behavior.

To analyze the behavior of a payload suspended from a crane, dynamic equations of motion (4.1) of the spherical pendulum has been solved. Due to the complex nonlinearity of these equations, numerical techniques are used to solve for and analyze their behavior.

To find the most critical excitation scenario, we rescale the variables in the equations of motion as follows:

$$\theta_x(t) = {}^2\theta_x(t) \tag{4.2}$$

$$\theta_y(t) = {}^2\theta_y(t) \tag{4.3}$$

$$x_p(t) = {}^2x_p(t) \tag{4.4}$$

$$y_p(t) = {}^2y_p(t) \tag{4.5}$$

$$z_p(t) = {}^2z_p(t) \tag{4.6}$$

Where ² is a small nondimensional quantity, which is a measure of the amplitude of the motion. The rescaled variables are then substituted into the nonlinear equations of motion (4.1) and the equations are then expanded. Keeping terms up to order [22] we find that the equations of motion become

$$\ddot{\theta}_x(t)+2^1\dot{\theta}_x(t)+\frac{g}{Lc}\theta_x(t)+\frac{1}{Lc}\left[\ddot{x}_p(t)+2^1\dot{x}_p(t)\right]i\frac{2}{Lc}\theta_x(t)\left[\ddot{z}_p(t)+2^1\dot{z}_p(t)\right]=0 \tag{4.7}$$

$$\ddot{\theta}_y(t)+2^1\dot{\theta}_y(t)+\frac{g}{Lc}\theta_y(t)i+\frac{1}{Lc}\left[y_p(t)+2^1 y_p(t)\right]i\frac{2}{Lc}\theta_y(t)\left[\ddot{z}_p(t)+2^1\dot{z}_p(t)\right]=0 \tag{4.8}$$

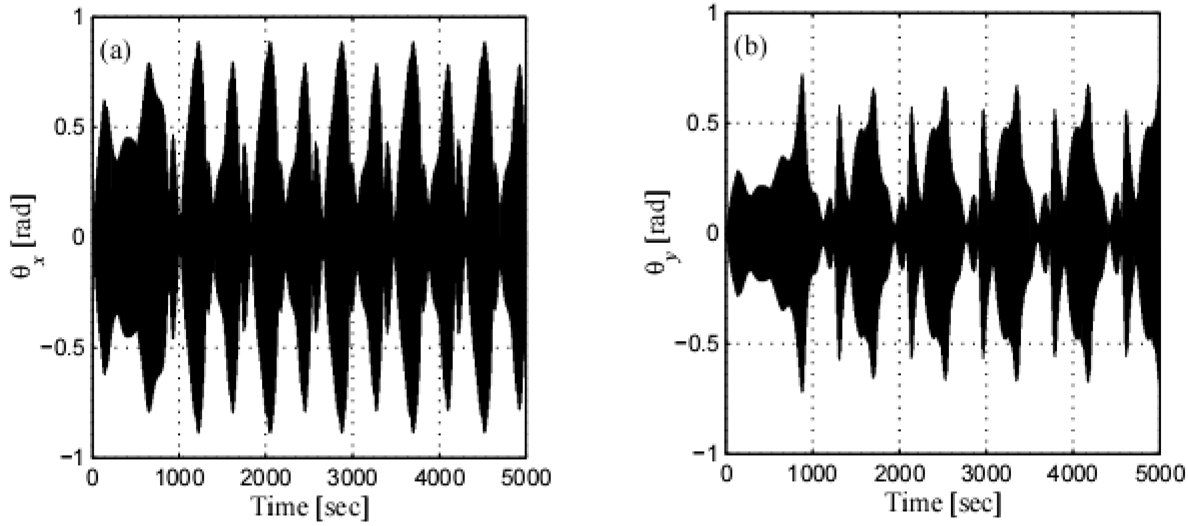
Because the terms involving x_p (t) and y_p (t) appear as additive terms in the rescaled equations of motion (4.9) and (4.10), the most critical conditions occur when their frequencies are approximately equal to the natural frequency ωn of the spherical pendulum. And because the term involving $z_p(t)$ appear as multiplicative terms in the rescaled equations of motion, the most critical conditions occur when the frequency of the $z_p(t)$ is approximately equal to twice the natural frequency of the spherical pendulum. Hence, to simulate the behavior of the spherical pendulum under the most critical conditions, I excited the suspension point of the pendulum ($x_p(t)$; $y_p(t)$;$z_p(t)$) sinusoidally in the x-and y-directions at the natural frequency of the pendulum and sinusoidally in the z-direction at twice the natural frequency of the pendulum; that is we let;

$$x_p(t)=A_x\sin(\omega_n t) \tag{4.9}$$

$$y_p(t)=A_y\sin(\omega_n t) \tag{4.10}$$

$$z_p(t)=A_z\sin(\omega_n t) \tag{4.11}$$

These excitations are substituted into the nonlinear equations of motion (4.1), and the resulting equations are then solved numerically for a long period of time. For the purpose of numerically solving these equations, we choose a spherical pendulum with a linear period of 10 seconds (Lc =24:849 m) and damping factor of 0:007. The amplitudes Ax, Ay, and Az of the base excitations are chosen to be 0:5 m, 0:25 m, and 0:125 m respectively. Results of the numerical solution are shown in Fig 4.2.



(a) (b)

Figure 4.2: Numerical simulation results of the (a) in-plane and the (b) out-of-plane pendulation angles of a spherical pendulum with base excitation at the natural pendulation frequency in the x-and y-directions and at twice the natural pendulation frequency in the z-direction.

Studying Fig 4.2, we notice that the system takes relatively very long time to reach steady state, from which we conclude that a controller design process should not be based on the steady-state response of the spherical pendulum. It also observe that, although the excitation magnitude in the x-direction is twice as large as that in the y-direction, both in-plane and out-of-plane motions of the hoisted payload grow to the same order of magnitude due to the nonlinear interaction between the in-plane and out-of-plane modes of oscillation. This means that smaller excitation amplitude or an excitation that is away from the natural frequency of the pendulum in one plane does not necessarily mean that the amplitude of the response in that direction would not grow to higher amplitude.

Another observation in Fig 4.2 is that the nonlinear interaction between the in-plane and the out-of-plane modes is relatively slow. This means that a controller designed to absorb the oscillation energy from one mode would not be efficient in absorbing energy from the other mode due to the slower rate of energy transfer between the two modes.

4.2 Controller Design and Analysis

4.2.1 Delayed-Position Feedback

In theory, delayed feedback in a controlled system produces damping in the system response; consequently, there is the expectation that the oscillation amplitude of a crane payload, represented by a spherical pendulum, will be significantly suppressed by forcing the suspension point of the payload hoisting cable to track inertial reference coordinates (x ref(t), y ref (t)) These reference coordinates consist of a percentage of the delayed motion of the payload in the inertial horizontal plane, relative to that suspension point, superimposed on fixed or slowly varying inertial input coordinates (x i(t), y i(t)) The
(x i(t), y i(t)) coordinates are defined by the crane operator. A tracking controller is used to ensure proper tracking of the desired (x ref(t), y ref(t)) coordinates of the suspension point.

This control concept applies to all types of cranes that use a cable for the purpose of hoisting and transferring load. To apply control concept to model of the payload hoisting cable assembly, represented by a spherical pendulum, actuate the suspension point of the hoisting cable in the x-and y-directions. These two degrees of freedom already exist in most crane types. The operator commands are transformed into the desired (x i(t), y i(t)) coordinates of the suspension point of the hoisting cable. Based on measurements of the angles of the payload hoisting cable, as shown in Fig 4.1, the delay control law takes the following form:

$$x_{ref}(t) = x_i(t) + k_x L_c \cos(\theta_y(t_i t_{dx}))\sin(\theta_x(t_i t_{dx})) \qquad (4.21)$$

$$y_{ref}(t) = y_i(t)_i + k_y L_c \sin(\theta_y(t_i t_{dy})) \qquad (4.22)$$

where k_x and k_y are the controller gains and t_{dx} and t_{dy} are the time delays. The time delay in the feedback loop of the controller creates the required damping effect in the system. A set of two PD tracking controllers is used to apply this control algorithm to ensure that the suspension point of the payload follows the prescribed reference position.

4.2.2 Controller Design for a Tower Crane

In tower cranes, the suspension point of the payload pendulum has the freedom to move to any prescribed horizontal position within the reach of the crane using the crane's rotational and translational degrees of freedom. Applying the delayed-position feedback controller to these motions can reduce the payload pendulation in- and out-of-the plane formed by the jib and crane tower.

To apply the delayed-position feedback control algorithm, we use two PD tracking controllers to drive the actuators of the rotating jib and the moving trolley. The operator input commands are routed through the delayed-position feedback controller to the PD controllers, thereby functioning transparently to the operator. The crane actuators are assumed to be strong enough to move the jib and the trolley rapidly to satisfy the reference rotation and translation signals at the end of each sampling period. Again, the limits on the crane actuators' speeds are taken into consideration in the process of designing the delayed-position feedback controller.

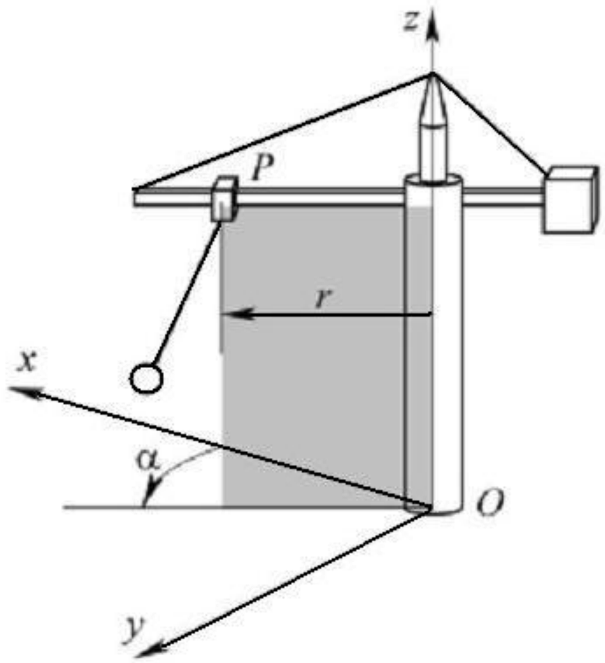


Figure 4.3: A model of tower crane.

For a land-based tower crane Fig above, point O is a reference point at the base of the crane. To describe the inertial position of the suspension point of the payload hoisting cable (trolley, P), both rotational and translational positions of the jib and trolley are measured. Using these measurements, we can give the inertial horizontal coordinates of the trolley as follows:

$$x_p(t) = r(t)\cos(\theta(t)) \qquad (4.23)$$

$$y_p(t) = r(t)\sin(\theta(t)) \qquad (4.24)$$

where r(t) is the trolley radial position and θ (t) is the rotational position of the jib.
inertial reference target position (x i(t), y i(t)) of the trolley using Eqs. (4.23) and (4.24).
And by using Eqs. (4.25) and (4.26);

$$\begin{aligned} x_{ref}(t) &= x_i(t) + k_{in}L_c \sin(\theta_{in}(t_i t_{d.in}))\cos(\theta_{out}(t_i t_{d.in}))\cos(\theta(t)) \\ &+ k_{out}L_c \sin(\theta_{out}(t_i t_{d.out}))\sin(\theta(t)) \end{aligned} \qquad (4.25)$$

$$\begin{aligned} y_{ref}(t) &= y_i(t) + (k_{in}L_c \sin(\theta_{in}(t_i t_{d.in}))\cos(\theta_{out}(t_i t_{d.in})))\sin(\theta(t))_i \\ &(k_{out}L_c \sin(\theta_{out}(t_i t_{d.out})))\cos(\theta(t)) \end{aligned} \qquad (4.26)$$

we superimpose a percentage of the time-delayed payload motion in the xy-plane derived from the time-delayed in-plane and out-of-plane pendulation angles of the payload on the x i(t)and y i(t) inputs of the operator to form the commanded trolley position (x ref(t), y ref(t)) in the inertial reference system. Using these references inertial coordinate system and equation (3.26) and (3.27), reference rotation and translation signal (r ref(t), θ ref(t)) can be obtained. The final part of the controller consist of two tracking PD controllers, which rapidly drive the jib and trolley of the crane to track the reference rotation and translation signals (r ref(t), θ ref(t)).

4.3 Experimental Validation

4.3.1 Experimental Setup for Tower Cranes

An experimental model of a tower crane was designed and built to validate the performance of the delayed-feedback control strategy on land-based cranes. The model crane has two degrees of freedom, a rotating jib on which a trolley can extend up to 1 m off the center of rotation of the jib, Fig 4.4. A rotary platform with a 1 : 45 gear ratio was used to give the crane its slewing degree of freedom. A ball screw with a pitch of 1: 2 in was used to drive the trolley along the jib.

Both the rotational degree of freedom of the jib and the translational degree of freedom of the trolley are actuated using two brushless servo-motors. Optical encoders embedded within the servo-motors were used to obtain information about the translational and rotational motion of the crane, Fig 4.4. Another set of two optical encoders placed at the suspension point of the hoisting cable were used to measure the in-plane and out-of-plane pendulation angles of the payload. A 1: 24 scale model of an 8 ft by 8 ft by 20 ft container weighing 20 tons was used as a payload. The center of gravity of the payload was located 1 m below the boom tip. This length yields a pendulation frequency of 0:498 Hz.

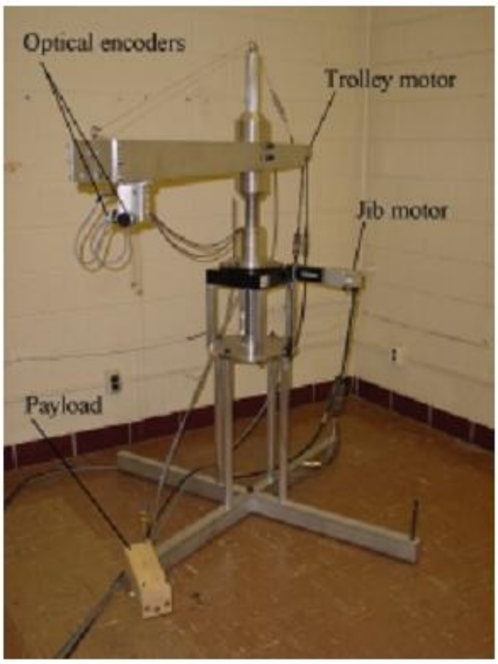


Figure 4.4: An experimental scale model of a tower crane.

The board was used to sample the optical encoders of the crane and the payload hoisting cable. A C++ code was developed, which included the delayed-position feedback control algorithm, and was used to sample the data acquisition board registers and drive the amplifiers of the jib and the trolley actuators. The code is capable of turning the controller on and off at any time during the experiment.

4.3.2 Experimental Results for Tower Cranes

To obtain the maximum damping, the controller parameters used were a gain of 0:4 for both the in-plane and out-of-plane parts of the controller, and a time delay of 0:56 seconds for the in-plane and out-of-plane angles of the payload hoisting cable, which was about 0:28 of the natural pendulation period of the model payload.

Two sets of experiments were conducted. In both sets, controlled and uncontrolled cases were run. The first set of experiments was conducted for the rotary mode of operation.
The trolley was placed at a radius of 40 in. The jib was then commanded to perform a rotation of 90±. The commanded motion consisted of a constant acceleration phase followed by a constant velocity phase and was terminated by a constant deceleration phase, Fig 4.5. In the uncontrolled case, as shown in Fig 4.6, the commanded motion caused the amplitude of

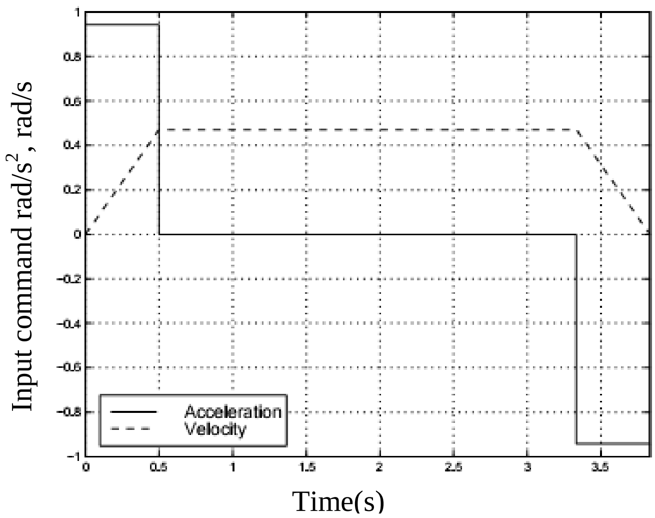


Figure 4.5: Commanded Rotary Motion

the pendulation angles to grow rapidly in-plane and out-of- plane. These pendulations continued to exist during the commanded motion and after it was concluded. The same experiment was then repeated with the controller turned on. The commanded motion caused the out-of-plane pendulation angle to grow to nearly the same amplitude at the beginning of the acceleration and the deceleration phases.

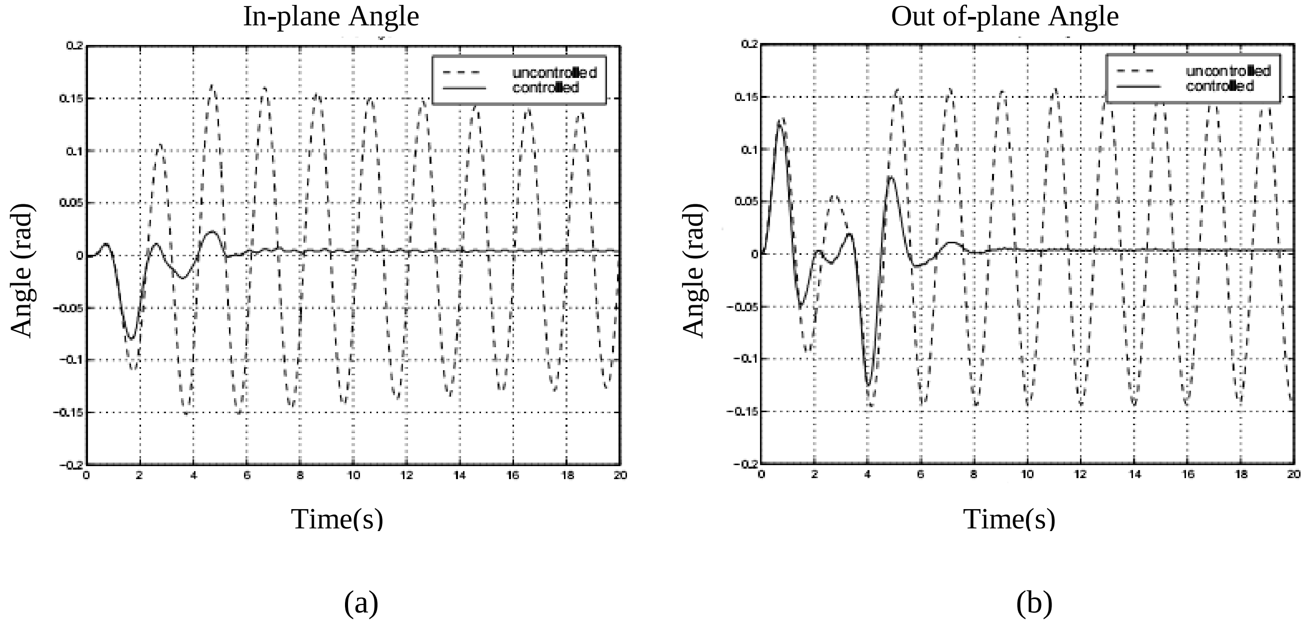
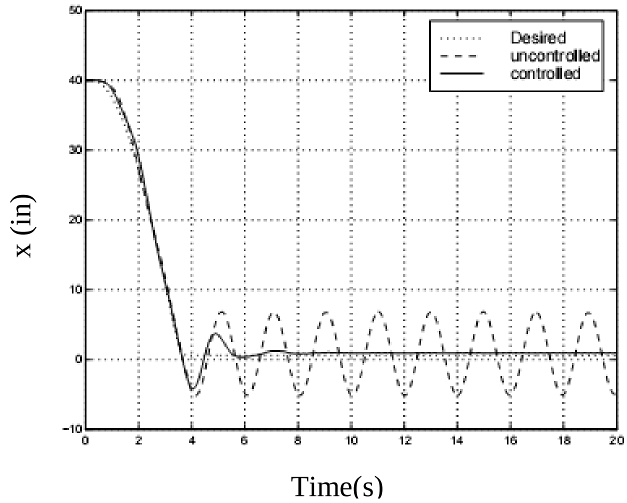
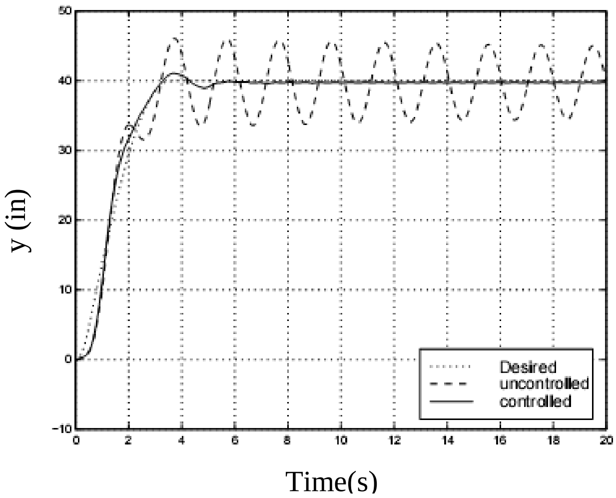


Figure 4.6: Experimental results for the rotary mode of operation: (a) in-plane and (b) out-of-plane angles of the payload cable as functions of time.

However, it dropped rapidly during the constant velocity phase as well as at the end of the commanded motion. On the other hand, the in-plane pendulation angle grew to a smaller amplitude at the beginning of the motion as a result of the centrifugal force on the payload, but was rapidly damped to a significantly smaller amplitude during the motion and virtually to zero shortly after the end of the commanded motion. Figure 4.7 shows the controlled and uncontrolled motions of the payload in the x-and y-coordinates. Although the controlled pendulation angles of the payload seamed to grow to amplitude close to that of the uncontrolled case, the controlled motion of the payload had a smaller deviation from the desired motion than the uncontrolled motion.



(a)



(b)

Figure 4.7: Experimental results for the rotary mode of operation: (a) x-motion and (b) y-motion of the payload as functions of time.

In the second set of experiments, the crane was operated in the gantry mode; that is, the only commanded motion was a translational motion of the trolley. The trolley was placed at a radius of 10 in. The trolley was then commanded to move to a radius of 40 in. As in the rotary mode, the commanded motion consisted of a constant acceleration phase followed by a constant velocity phase and was ended by a constant deceleration phase, Fig 4.8. In the uncontrolled case, as shown in Fig 4.9, the commanded motion caused the amplitude of the pendulation angles to grow. These pendulations continued to exist during the commanded motion and after it was concluded. The same experiment was then repeated with the controller turned on. The commanded motion caused the pendulation angle to grow to nearly the same amplitude at the beginning of the acceleration and deceleration phases, while in it dropped rapidly during the constant velocity phase a well as at the end of the commanded motion. Figure 5.10 also shows controlled and uncontrolled motions of the payload in the x-coordinate. The figure shows less deviation from the desired path of the payload and a faster drop in the pendulation at the end of the commanded motion.

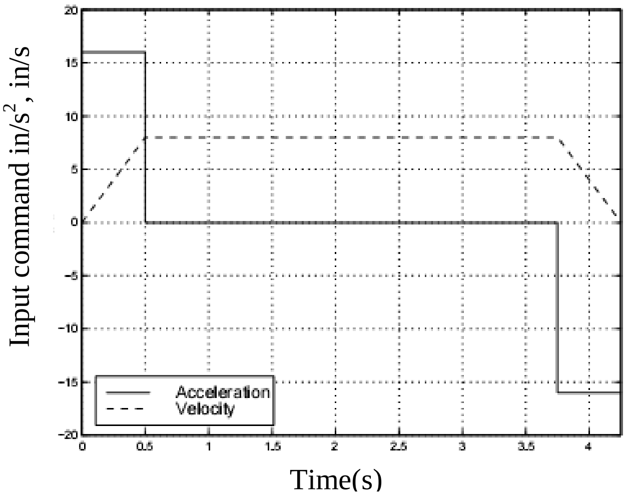
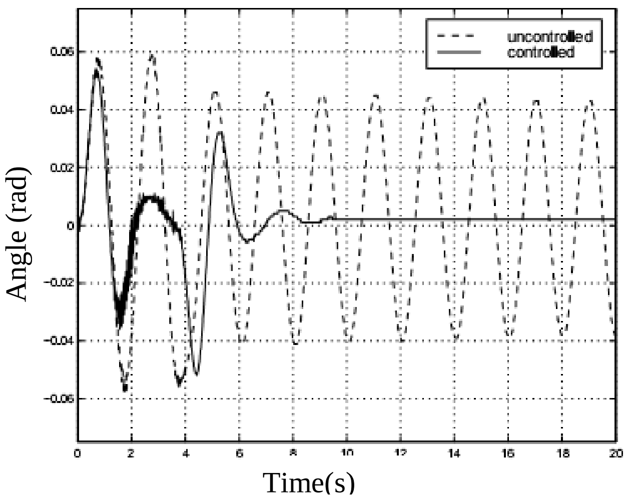
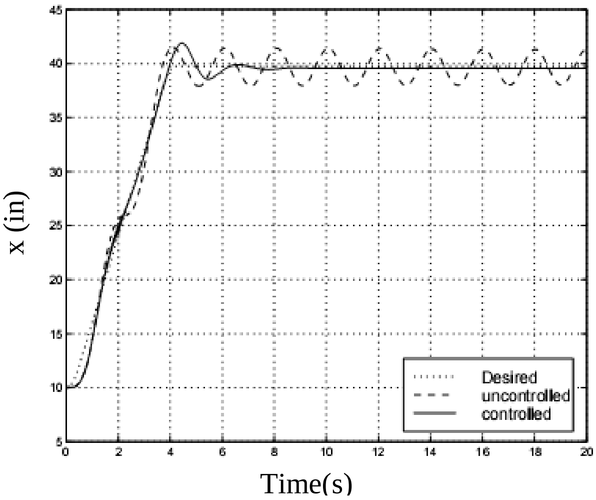


Figure 4.8: Commanded gantry motion.



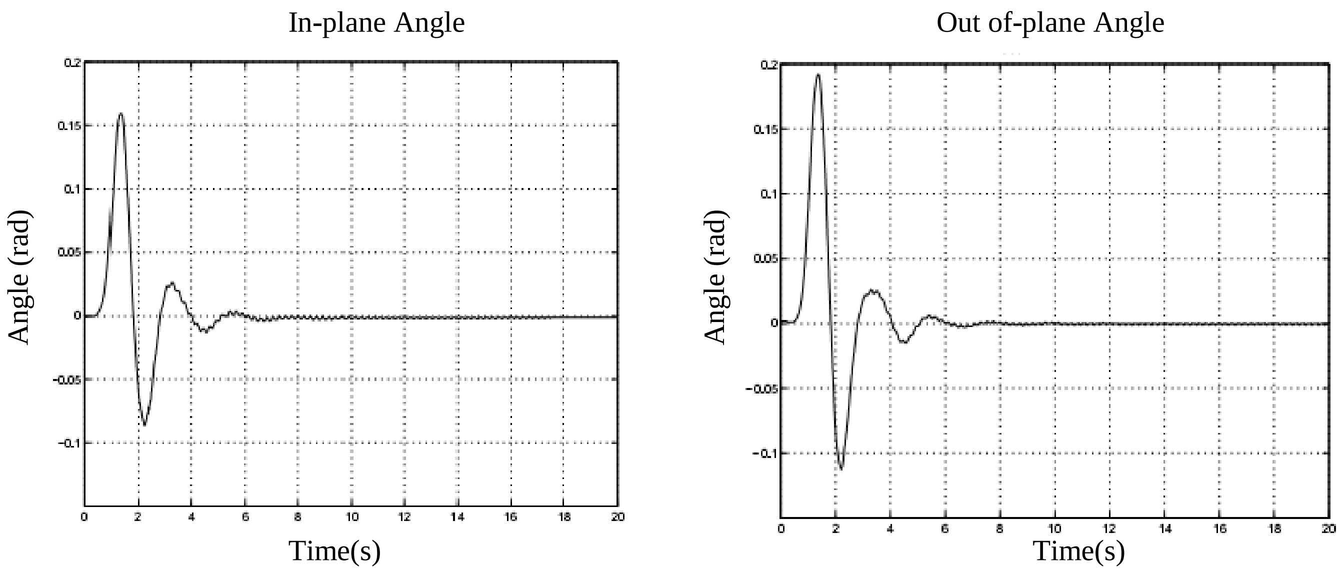
(a)



(b)

Figure 4.9: Experimental results for the gantry mode of operation: (a) pendulation angle of the payload cable and (b) payload motion as functions of time.

An additional experiment was conducted, Fig 4.91. After the experiment was started, the payload was given in-plane and out-of-plane disturbances. The pendulation angles of the payload dropped in 5 seconds to less than 0:5± . Figure 5.12 shows the corresponding x-and y-motions of the payload as a result of the disturbances.



(a) (b)

Figure 4.10: Experimental results: (a) in-plane and (b) out-of-plane angles of the payload cable as functions of time. The payload was given in-plane and out-of-plane initial disturbances.

CHAPTER 5
CONCLUSION AND RECOMMENDATION

5.1 Conclusion

This project is based on the study of behavior of tower crane in handling the load. Delayed-position feedback together with luff-and-slew angle actuation is an effective method for controlling pendulations of tower cranes. Dramatic reductions in the pendulation angles of the payload in the controlled system were achieved in the numerical simulations as well as the experimental tests. The simulations and experimental results demonstrated the effectiveness and robustness of the proposed strategy against disturbances in the system.

The model was tested in both the rotary and gantry modes of operation. Although sizable pendulations were observed at the beginning and at the end of the commanded motions of the crane, the controller was clearly effective in absorbing these pendulations during and at the end of the commanded motions at high rates.

5.2 Recommendation

This project can be more improved by applying more control system method by covering more type controllers such as PID controller. Each controller function differently to the responses applied hence more data can be obtained and more accuracy can be achieved.

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APPENDICES

Week Number / Activities/ milestones	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Project Work Continue -Modeling(Catia)								MID-SEM BREAK							
Submission of Progress Report 1															
Project Work Continue -Prototype fabrication															
Submission of Progress Report 2															
Seminar (compulsory)															
Project work continue -Testing & Analysis															
Poster Exhibition															
Submission of Dissertation (soft bound)															
Oral Presentation															
Submission of Project Dissertation (Hard Bound)															