OPTIMAL CONTROL OF SWAY IN HYDRAULIC BOOM CRANES OF RECOVERY VEHICLES

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DEDICATION

All glory rests for Allah Almighty for ranting an opportunity to me for carrying out this research and bestowing upon me with the ability to seek knowledge on the subject. I dedicate the completion of this research thesis to my beloved parents, my beloved wife and adorable kids Inayah and Mustafa whose whole-hearted support and encouragement inspired me to not only pursue my course work but also complete my research amicably and timely.

I am highly grateful to my Thesis Supervisor Dr. Mohsin Jamil who, for the entire period of my research work, has been guiding me on my novel research area. I am thankful to him for his dedicated, generous, and rightful guidance in perusing this research. His timely counselling and assistance encourage me to remain focus on my research work and complete it amicably. I am also very grateful to my Thesis Co-Supervisor Dr Syed Omer Gilani and other faculty of SMME who have guided me rightfully during the course of this thesis. Moreover, I would acknowledge the best possible research facilities and infrastructure provided by my School of Mechanical and Manufacturing Engineering for facilitating me with an encouraging and research oriented environment which enabled me to complete my Masters' degree.

The last but not the least, I am grateful to my friends, colleagues and well-wishers for their prayers and support without which; it would not have been possible to accomplish this milestone.

DECLERATION

We hereby declare that no portion of the work referred to in this project Thesis has been submitted in support of an application for another degree or qualification of this of any other university or other institute of learning. In any act of plagiarism found, we are fully responsible for every disciplinary action taken against us depending upon the seriousness of the proven offence, even the cancellation of our degree.

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ABSTRACT

Due to inertia, undesirable sway in the payload is observed during a recovery operation; in which a vehicle or equipment casualty is evacuated with the help of a recovery vehicle for its further extrication to a repair site. This recovery operation is usually carried out by military or commercial recovery resources with the assistance of a rotary crane vehicle 765 evacuate men and material from debris after a catastrophe like earthquake, accidents etc. Generally, the payload oscillations are manually controlled by detailing few personnel of the recovery crew to hold the payload which violates safety of recovery crane operation. Manual control of the payload oscillations also results in the commitment of skilled recovery crew thus retarding speed and efficiency of the process. Therefore, in this research work, a robust controller has been designed for intelligent control of these payload oscillations in a mobile recovery crane.

An EFI based Light Recovery Vehicle (LRV); IVECO Euro-Cargo has been selected for designing a robust controller to regulate its payload oscillations. The Rotary boom crane and its luffing motion were duly modeled while catering for the effects of varying environmental parameters like wind speed, disturbances etc. Various control schemes based on even intelligent and adaptive controllers have now been matured enough for overhead, fixed and tower types of cranes due to their wider applications on industrial scale. However, for mobile recovery vehicles operating in an open / unstructured field environment, a little research has been done to find a suitable anti-sway control scheme. Therefore, as an initial step, anti-sway control scheme based on classical PID controller has been suggested with the feedback available from newly installed sensors on the boom crane mechanism in addition to the already installed sensors in the EFI vehicle. The simulations results have been compared with an optimal LQR controller which showed an improvement in the settling time by 4.5 seconds as compared to the existing crane mechanism where no control scheme is available to control the payload oscillations. Therefore, the application of the suggested control scheme has resulted in reducing the sway present in vehicle / equipment casualty thereby improving the safety, efficiency and speed of the recovery process.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF ABBREVIATIONS

Rec	Recovery
3 D	Three Dimensional
3DOF	Three Degree of Freedom
PID	Proportional Derivative Integral
SP	Set Point
IC	Internal Combustion
Cas	Casualty
EFI	Electronic Fuel Injection
FLC	Fuzzy Logic Controller
DC	Direct Current
Eq	Equation
PV	Pre-set Value
MV	Manipulated Variable
LIC	Low Intensity Conflict

CHAPTER - 1

INTRODUCTION

1.1 Introduction

Mobile rotary crane vehicles are used both in military and civil environment during recovery operations[1]. Recovery is defined as the process of extricating an equipment casualty from the place where it has become disabled or defective and evacuating it to the first place where repairs can be affected or from where it can be back-loaded as shown in figure 1. Due to inertia of casualty, undesirable oscillations in the payload (equipment casualty) are observed while recovery operation is underway. This phenomenon of uncontrolled oscillations also termed as sway[2]; becomes paramount while a fallen casualty is being lifted form a deep ridge. These payload oscillations are usually controlled manually by detailing 2-3 personnel of the recovery crane crew. However, this practice violates the basic safety precautions of recovery operation which dictates ensuring a minimum safe distance form hoist cables under tension. Moreover, detailing of recovery crane personnel to hold the payload oscillations of casualty also leads to unnecessary commitment of recovery crew. The speed of the recovery operation is also adversely affected as crane operator has to wait for settling down of oscillations in the hanged payload as shown in figure 1 [3].

Figure 1 - Recovery Crane Operation in Field

Introducing automation in mobile recovery crane vehicles is a challenging task in unstructured field environment. These mobile recovery cranes are typically employed for transportation of load, lifting of vehicle / equipment casualties, and moving them from one location to other. The cranes mounted on these recovery vehicles are hydraulically operated i.e., the mechanical power of the engine is transmitted to hydraulic fluid which ultimately extend or retract the boom crane for hoisting of payload in the desired direction. It has however; always remain challenging to continuously track the amplitude and frequency of payload oscillations. The employment of sophisticated sensors is an resource intensive task which is not only costly but the integration of these sensors in any control model increases the complications due to sensitivity of these sensors and their compatibility with the hardware of the overall system. In addition, the need to provide a user friendly interaction of the crane system with the crane operator and prevailing safe and efficient working conditions are always on the priority. Due consideration of the field environment in which a typical recovery operation will be performed needs to be incorporated while designing a control scheme for a mobile recovery vehicle. The controlling of the payload oscillation in the mobile crane vehicle are also affected by the speed of recovery operation, the type of ground / terrain on which this operation is being carried out. The intricacies developed due to non-linear effects hydraulic operated boom retracting and extending cylinders, friction forces in the spool valves and joint axes of the hydraulic mechanism are also faced while designing a suitable controller for payload oscillations in a mobile recovery crane[4]. These non-linear effects, if not duly catered for during the modeling of the system affects the linearity of the model. The size and geometry of the payload during a field recovery operation as shown in figure 2 [5] also varies as per the size and geometry of the casualty to be recovered and thus the speed, capacity and duration of the resultant recovery process varies accordingly and thus needs to be catered for in the design stage of a suitable control scheme.

Figure 2 - Manual Regulation of Payload Oscillations

It is noteworthy that while carrying out a field recovery operation with the help of a mobile recovery crane, there is a need to continuously observe the states of the system for instance the angle of the payload oscillations, the frequency and amplitude of the resultant sway produced, and the varying environmental parameters like speed of wind, level of ground[6] and relative height of the terrain, vehicle speed in case the recovery vehicle is on the move. In the existing mobile cranes, all these varying parameters are manually controlled by the crane operator. Thus the overall safety, efficiency and speed of the recovery process depends on the experience and skills of the recovery operator[7]. By utilizing his experience, a skilled crane operator negotiates the irregularities posed by these variable field recovery process operating parameters. Thus, Recovery vehicles in field are usually preferred to be carried out manually. However, the need to regulate the sway intelligently with the help of some robust controller is enhanced by the requirement of improving the operating conditions for the manual crane operators and speed / efficiency of the process [8]. However, a good and user friendly interface is essentially required for the smoothness of the recovery operation. The aim of this thesis is to therefore reach a shared control between the crane operators and semi-autonomous control in field recovery operation. Conventional or classical approach is one way of controlling this undesirable swing. For better tuning of the observed results, PID gains can be calculated through Linear Quadratic Regulator (LQR) controller to achieve desirable output[9]. Performance measures in this case are settling time, steady state error, overshoot and undershoot. Command shaping method shows more efficacy than the traditional input phasing for reduction in vibration[10].

The paper is organized as follows. Chapter 1 explains the overall background of the study. Chapter 2 covers the literature review of the control schemes applied previously on different types of cranes to dampen the payload oscillations. Different types of cranes and basis safety design parameters in their design is covered in Chapter 3. In addition, it discusses the system description of the rotary boom crane mechanism by taking the case study of IVECO Euro Cargo Light Recovery Vehicle. Chapter 4 presents the sway phenomenon and methods applied on various cranes to reduce the payload oscillations. Chapter 5 entails the proposed control scheme duly modeled along with the Simulink model description in the following Chapter. The simulation and experimental results are mentioned in Chapter 6. The conclusion and future recommendations are briefed in chapter 7.

1.2 Thesis Objectives

The main objective of this thesis is to design an optimal control system to dampen payload oscillations in hydraulic rotary cranes of recovery vehicle. Therefore, following will summarize the objective of this research work:-

a. Analyze and evaluate the pattern of payload oscillations in the payload of the recovery crane vehicle as a result of sway phenomenon in the open loop crane system (in the absence of any control scheme).

b. Evaluate the closed loop response of the system after employment of PID classical controller and optimal LQR (Linear Quadratic Regulator) controller for controlling the sway in the payload.

c. Compare the results of open loop system of Recovery Crane with the closed loop response of Recovery Crane Vehicle.

d. Compare the results of classical PID controller with optimal LQR controller with the help of Simulink models.

1.3 Scope of Thesis

Developing a suitable controller for damping the sway in cranes is an important area of research. It is an obvious thing because of the following applications of hydraulic operated mobile rotary cranes:-

a. Mobile hydraulic cranes are frequently employed for lifting of fallen or overturned casualty for its further extrication to a repair site / echelon.

b. This operation is usually carried out by military or commercial recovery resources in the form of mobile / wheeled cranes.

c. These recovery tasks are also carried out in the aftermath of a catastrophe like earthquake, accidents etc. to evacuate men and material from debris.

d. In addition, these cranes are mainly used for carrying and transporting of heavier material all across the country [11].

e. Mechanical advantage available through hydraulic crane mechanism is utilized which moves the payload beyond the normal capability of a human.

f. Cranes are also used for carrying freight in the industry of transportation[12].

g. The versatile kinds of cranes are also used for assembly of heavier equipment in the manufacturing industry[13].

h. These are also used in the car industry for loading and unloading of the cars from one place to another[14].

i. The cranes are mainly used for lifting heavy things and transporting them to other places all over the country.

j. In construction industry cranes are used for movement of heavy raw materials.

k. Cranes are used in the manufacturing industry for assembling and disassembling of heavy equipment and machines[15].

l. In Integrated Circuits (IC) manufacturing industries, it is used to bring electrical components from one place to another place.

m. It is also possible to extend this research work to other applications which involve automation for example:-

(1) The controlling of artificial human hand motion.

(2) Smooth movement of automatic cars.

(3) Plane motion of belts that are controlled by motors for transportation of goods in shopping malls.

(4) The performance of electrical Elevators, which are often used in many buildings, can also be improved.

1.4 **Advantages**

The following advantages will be obtained due to robust control of sway produced in the recovery process:-

a. Optimizing the efficiency of recovery process by elimination of the manual effort to control sway displacement.

b. Increasing the speed of crane operation and recovery process possible due to swift control of the sway produced before the hanged mass can be transported from one to another location.

c. Enhancing the safety of crane operation due to elimination of need for employment of manpower to control the sway that used to be present in the closed proximity to the crane and hoisting cables under stress. The safety of the process is also improved since the chances of collision of the hanged mass will be minimized in the autonomous process.

d. In the manual crane operation, uncontrolled sway of the hung mass results in the generation of structural stresses in the extended boom of the crane[16]. These uncontrolled stresses may increase the wear and rear of the crane equipment in the long run. Thus by intelligent control of the sway displacement in the proposed method will enhance the serviceable life of the crane equipment.

CHAPTER - 2

LITERATURE REVIEW

In merchant trade, cranes were a source of great assistance for transporting containers on harbors[17]. It is noteworthy that except for few cranes which were fixed on factors for restricting their movement, most of the cranes, before 1870s used to be fixed to a position. The idea of steam powered cranes was introduced by Appleby brothers in 1873. In 1922, Henry Coles, came with the idea of truck mounted cranes. The internal combustion (IC) engines were adopted for cranes in 1922 along with the introduction invention of telescopic jibs. Additional booms were carried in cranes before 1960s to increase height at the cost of increasing cost, however, in 1959 crane and hydraulic experts; R.H.Neal, F.Taylor and Bob Lester integrated modernized cranes at that time. In 1966, Wafurther adapted with ten ton fully telescopic hydraulic boom. In 1968 thirty ton military versions with 4 wheel drive were introduced. Cosmos team created a twenty five tonnage crane in 1976 which marked new developments in the hydraulic cranes. The boom cranes were then utilized at sea ports for lifting and carrying of heavier containers from the ships[18]. Boom cranes which are used in small and midsize harbors with mixed freight types. In general, level of process automation was low as transporting and trans-loading of containers was carried out manually by crane operators. Nevertheless, over the years, designers realized that for automation of logistics in harbors, increasing container handling rates was prerequisite[19], only possible by increased progress on the automation of the overall process.

Spreaders were introduced for mounting of the crane hook in case of boom cranes[20]. Spreader is adjusted to the container position by controlling the rotational sway carried out manually or via automatic anti-sway system. Auto skew control on boom cranes was introduced with the help of a gyroscope supported on a hook. In addition, a position sensor for the purpose of evaluation of states, in conjunction with a position sensor was used.. A controller design was suggested for binary (on/off) rotary actuators optimizing the trajectories. A simulation model of the skew dynamics was also presented on gantry cranes. Further evaluation of the simulation model was carried out[21]. This was followed by explanation of damping in skew motion by using additional moveable frames. This was followed by a dynamic model on a four rope suspension based crane. Stiffness of rope was supposed to be finite while in this case. Adaptable linkage was catered for the skew movement to make mechanism of the spreaders compatible with the suspended load to reduce the vibrations[22].

The technique of numerical optimization was also used with the help of additional ropes as actuators. In addition for determination of feedback gains, a control approach based on state feedback was used[23]. The simultaneous skew and swing were also integrated and their results were examined. Similar set-ups with additional cables for skew control were suggested. [24]. Control cables were applied to control a suspended load from a boom crane in terms of six degrees of freedom of [25]. The modification resulted in the improvement of system but made it heavier as compared to the previous simple hook designs which resulted in reduction of loading capacity of the cranes.

Effects of skew control on electronic drives were analyzed [26]. Sensors for measuring tension in ropes were used to measure the skew motion. This innovative method proved to be fruitful yet the tuning of sensors became problematic and resulted in garbage values further adding to the sway. Inertia rotor was used as actuator for suppressing the rotational vibration and a fiber-optical gyroscope as a skew sensor, however, the high cost made the system less applicable. The idea of employing optical load tracking and a linear controller design for skew control system was also investigated in rotary cranes. The results were also presented on results of an automatic control system for overhead cranes including skew control[27]. Their findings investigated use of a skew observer to determine time slots during which even without skew control, a hung mass can be placed down. Recognition of optical position. The idea of optical position recognition was discussed in gantry cranes to develop an anti-skew system. Input of the velocity for sway control was emphasized in cranes. Problem arose in the time taken to lengthen the cables for skew control affecting the productivity of the system. Rope lengths were varied individually in a four-rope suspension and a hydraulic actuation system in a gantry crane[28]. PID control was employed for skew stabilization. Eigen frequencies of skew dynamics were quantified and proposed application of shaping control and used actuators for modification of rope lengths. Later, a mechanism was developed for tracking spreader consisting of one dimensional laser distance measurements. A camera was used for measurement of the skew angle[29].

During recent decades, the focus of the skew control efforts remained on gantry cranes. Boom cranes which required additional challenges thus remained out of focus of the researchers. For instance, long rope length and the small rope distance yielded lower torsional stiffness compared to gantry cranes. Thus in case of boom cranes, for producing sufficient reactive torque, greater angles of misalignment angles were required. Misalignment of hook angle became a prominent limitation in boom cranes. While skew angles of a few degrees are attainable in gantry cranes, there is always a possibility of arbitrary skew angles in boom cranes. Thus, in boom cranes, a careful planning of the transitional behavior is required. It is noteworthy that the technique of load tracking using cameras and markers which is well matured in case of gantry cranes cannot be applied to boom cranes. This limitation thus again highlights the state estimated issue[29].

Skew dynamics of a boom crane were also modeled by a non-linear model to design a suitable control system for the skew dynamics[30]. A two degree freedom of freedom structure was utilized for the controller to stabilize skew angle, decouple slewing gear motions and simplify operational control. Luenberger-type state observer was employed for the system state from a skew rate measurement. Estimation of trajectory of the controller system was carried out from real time operational input using a simulation of the model based on suitable control the reference trajectory tracks the operating signal and maintains constraints of the system.

A multivariable control scheme was also suggested to reduce the load swing while the load is hoisted and transferred[31]. The issue of controlling the linear time-varying system was simplified in terms of the optimal control of a linear time invariant system. The transformed linear model was decoupled. It became symmetric with respect to the crane motion (both traveling and traversing). The three dimensional overhead crane was linearized into a two dimensional overhead cranes in terms of control. Thus, by using the loop shaping, root locus, and gain planning methods, this investigation enabled in designing of a new decoupled control law for a 3D overhead crane. This makes application of the lagrangian equation easier as compared to the 3D case[32].

The variation of accuracy in respect of the boom tip position and its dependency on the type of the path and velocity was also investigated. Significant errors were caused by higher velocities. The path similar to a coil around a toroid was found to be harder. It was revealed that higher velocities were found as the boom tip started cutting corners which may be thought as a limitation of the current controller design for path tracking. The effect of the anti-sway control was studied as such; the cost in the object function for the sway angles and sway velocities was set to zero. The results of these experiments showed that the anti-sway control can decrease the swaying in sideways up to 64% and in the forward backward directly up to 76%. Accuracy of the path tracking or the velocity of the tip of the boom was not affected by activation of anti-sway element. The issue can be minimized by restricting crane motion in undesired paths. This can be possible by introduction of virtual obstacles in crane motion[33].

A nonlinear feedback control design was introduced for an overhead crane to be dealt by a singular perturbation time-scale separation as there was ill-defined relative degree in the gantry crane [34]. Consequent results showed a crane control comprising of two parts; a slow portion so that load follows a desired trajectory, and a superimposed fast portion for removing oscillations and sway. The result was found more accurate load control than the usual Jacobian linearization techniques in the literature.

A nonlinear controller was designed for a rotational crane to control its sway using optimum reference pattern based on linear approximation[35]. This method is not of significant value for the disturbances except control at a stop point. In order to simplify the problem, Control issues on the rotating action were addressed. The results of simulations for the anti-sway control produced satisfactory results.

Efforts were also made to develop techniques for ATC (Automated Transfer Crane)[36]. An algorithm was designed that builds the container path in order to prevent collision during movement. It is aimed at finding the anti-collision path for transferring the container in the least time. The simulation result showed superior performance about trolley position, sway angle and settling time. Techniques for unmanned automation of ATC were introduced. As a result, application of PID controller was evaluated for disturbance wind pattern in the yard.

A control system was proposed to minimize the payload oscillations at the termination of payload motion. For achieving this purpose, a mathematical model of the control system was developed for displacement of the payload to a selected point. A partial feedback type robust control system was suggested with integrator for a rotary crane. The information on natural frequencies was utilized on the crane rope. In order to reduce the accuracy in positional error of the rotary type crane boom, he included an integrator in the proposed control system. The results were successfully verified experimentally. A control system was studied for swing free transportation of payload by mobile boom cranes employed in harbors. A linearized approach was used to model the control system while which translated crane operator commands into swing free payload transportation. They used a trajectory generator to taking into account the input and output constraints while using a predictive control theory.

Anti-sway techniques of a gantry crane system were also inquired with disturbances effect using the DFS and PD controller and results were investigated in form of suppression of sway angle[37]. It was concluded that employment of DFS and PD controller can successfully negotiate the disturbances. Three modes of swaying frequencies were significantly reduced with both controllers. PD-type Fuzzy Logic Control's development with different polarities was investigated into input for anti-sway control of a gantry crane system[38]. In order to formulate dynamic model of the system, Euler Lagrange system was used. The designing of PD-type Fuzzy Logic Control for control of cart position in a gantry crane. It then extends to include input shaper control for controlling sway of the mass. The time as well as frequency domains were used for presenting the response of the suggested model in the form of different parameters like input tracking capability, swing angle, response time period and system rigidity to variation in surroundings parameters. Subsequently, amplitude polarities of the input shapers were evaluated which demonstrated an improvement in speed of the response, decrement in the level of vibration and robustness of input with the error in natural frequencies.

Sway feedback control approaches was also investigated to use a rotary crane system with disturbance effect in the dynamic system with Fuzzy type logic controllers[39]. Sway angle suppression and disturbances cancellation was investigated to evaluate performance of the controllers. The findings showed successful handling of disturbances in the system by PD and PD-type fuzzy logic controller. Sway in the system was significantly decreased with the PD controller as compared to the PD type fuzzy logic controller. PD-type Fuzzy logic controller showed a better performance in comparison to PD controller In addition in relation to the response speed.

The use of accelerometer for estimating swing of the crane was also investigated. The findings showed that the approach can be adapted to other crane areas that use symmetric wires[40]. An observer was proposed to estimate the swing angle of the crane and its validity was determined by simulation and experiment. Swing angle was determined with distributed values of the accelerometer against various values of input on x axis and z axis. In addition, a controller was designed to stabilize the swing of the crane with sea-induced base movement. The proposed method was also verified with a real, miniature crane.

Control schemes for anti-swaying and input tracking of a gantry crane system were also developed [41]. The relative performance of the controllers in gantry crane was gauged in the start with the help of a PD control for cart position control. A non-collocated PID was included with an input shaper based control schemes for controlling sway of the system. Time and frequency domains were utilized to present the results of simulation on the response of the gantry crane with the controllers. Evaluation of performances of the control schemes was carried out in terms of level of capability of gantry crane, decrease in the sway angle and system response in time as compared to the PD control.

The control schemes based on LQR by incorporating IIR low-pass and band-stop filter were studied aimed at tracking of input and reduction of sway in a rotary crane[42]. The study was carried out on a prototype of a rotary crane scaled to size. It was found that better sway suppression was achievable in LQR control with band-stop filter in comparison to the low-pass filter for each studied filter’s order. Reduction of sway was also observed when number of filter’s order was increased. But, in contrast, the higher response in tracking speed was found in a higher speed of tracking response. Thus it was concluded that the suggested hybrid controllers were able to reduce system sway in a rotary crane while ensuring the capability of input tracking.

The control of the rotary crane for the change of rope length was also investigated. A straight transfer transformation method was employed[43]. The position of load and its sway control was developed using the gain-scheduled controller. The reference governor with the maximum constrained positively invariant set was assumed to produce the reference input. This arrangement fulfilled the input constraints and suppression of sway. The method was the successfully demonstrated by the simulation and experiments.

The hybrid control schemes for sway suppression and rotational angle tracking of a rotary crane system as also studied. A lab-scaled rotary crane was considered for the arrangement[44]. The dynamic model of the system was derived using the Euler-LaGrange formulation. Initially a classical controller which is collocated proportional-derivative (PD) controller was developed to study the effectiveness of the controllers and to control rotary motion. The rotary crane system response was presented in time and frequency domains. The work formed the basis of design and development of experimental work for trajectory tracking and sway suppression of others crane system.

An anti-swing control scheme with fuzzy uncertainty compensation was also proposed[45]. The arrangement catered for the positioning control as well as overall closed-loop system stability dynamics as there was no requirement of the system parameters for control design in a priority. The error in position can be restricted to a bounded area in the suggested control schemes, while the swing angle of the load can be damped readily for suppression of sway in a crane. The results were supported by simulation results and stability analysis. Simulations were carried out to predict the behavior of a mobile crane used in docks. They used a software based mechanical model to investigate the reduction in the swing angle under the application of an adaptive controller.

From the view of available literature, it can be concluded that sophisticated controllers have been developed for overhead, fixed, gantry, harbor and tower cranes etc. which are successfully employed for various industrial processes of loading / unloading and transportation. However, for the mobile recovery cranes which are used in unstructured environment with varying operating parameters, practical control schemes have achieved a little success. Therefore, there is a need to explore this uncharted avenue of sway reduction for enhancing efficiency and speed of these mobile crane or recovery vehicles.

CHAPTER 3

TYPES OF CRANE AND IVECO RECOVERY CRANE VEHICLE

3.1 General

This chapter will discuss various types of cranes and the differences involved in their operating mechanism. This is essential to develop an understanding of the rotary boom cranes which are equipped on mobile recovery vehicles and their difference from the overhead / fixed / tower cranes, used on static role for various industrial applications. This chapter will also cover the description of the hydraulic operated boom crane found in recovery vehicles to assimilate the basic functioning and operating characteristics of a mobile rotary crane for which a control scheme will be presented in the subsequent chapters. As a case study, IVECO Euro-Cargo Light Recovery Vehicle (LRV), will be discussed along with its modulus of operandi.

3.2 Crane

A crane is a kind of equipment or machinery which uses principle of Mechanical Advantage (MA) to lift and lower the hoisted payload from one location to another. It is usually composed of a combination of hoisting cables, hoist pulleys and sheave blocks, wire ropes and chains, a rotating platform hydraulic actuating mechanism etc. Cranes make the transportation swift replacing the manual laborious tasks of carrying payloads consuming valuable man-hours. Cranes find applications in freight transport, construction and manufacturing industries.

A crane is designed keeping in view its loading capability without toppling or rupturing. Cranes are modified to meet the specific requirement of the pertinent industry where they are going to be employed. For instance, cranes vary in sizes thus we find small jib cranes in mini-workshops, tall tower cranes for construction of huge meg structures or even floating cranes for constructing oil rigs, aerial cranes for easy of air lifting during speedy / emergent situations etc.

3.3 Classification of Cranes

Different types of cranes are characterized by different names indicating the difference in either their operating mechanism or their configuration[46]. For example:-

a. Bridge or Overhead Cranes are employed in long factories. They work in functionality with the girder or overhead cranes. Its advantage is its small deadweight and increased integrity of the system.

Figure 3 - Bridge or Overhead Cranes

b. In Jib crane, load is carried on a hoist which is supported on a horizontal jib. The jib cranes are commonly used for carrying of material to all kind of floors.

Figure 4 - Jib Crane

c. Whereas, tower cranes are fixed with a concrete slab on ground type of cranes are fixed on the ground with a concrete slab to give the best balance to the crane and its operator cabin.

Figure 5 - Tower Cranes

d. For cargo operations, usually Deck cranes placed on the ships are used. These deck cranes are also employed for unloading ships on the shores if no other alternated means is available.

Figure 6 - Deck Cranes

e. Aerial cranes have the advantage of accessing even those difficult areas where other cranes cannot reach. These cranes, built on aerial platforms, can be used for disaster relief and rescue operations as well.

Figure 7 - Aerial Crane

f. Mobile recovery crane vehicles are hydraulically operated boom cranes which are built on mobile platforms. These mobile cranes are employed to extricate equipment or vehicle casualty from the site of accident / casualty and evacuate it to a suitable place where repairs can be affected.

Figure 8 - Mobile Recovery Crane Vehicle

g. Bulk handling cranes are equipped with bucket and designed for carriage of scrap material or minerals etc. and do not use any hook or sling.

Figure 9 - Bulk Handling Crane

3.4 **Safety of Crane Operation**

There are certain requirements which must be adhered upon in every recovery crane for ensuring safety of the crane operation. The main requirements are mentioned as under:-

a. An overload safety switch is equipped for safety.

b. The marking of maximum lifting capacity of the hoisting cable and the hook must be identified on the crane so that payload in excessive payload is not loaded on the boom crane.

c. The availability of safety equipment like gloves for safety of the crane operator, helmets for the recovery crane crew and shackles etc. should be ensured.

d. The marking of bridging capacity should be endorsed on front of the recovery crane.

e. A minimum safety distance of 50 meters must be maintained from the recovery crane vehicle.

f. There should be a well-trained signaler standing at a safer distance but in the line of sight of the recovery crane operator giving hand signals during the recovery operation. Some of these signals may be described as follows:-

g. Calculation of payload must be carried out before attaching it with the crane and the appropriate tackle layout be planned / prepared beforehand.

3.5 **Case Study of a Mobile Rotary Crane**

For the purpose of research, case study of Euro Cargo IVECO 7.5 Ton, a Light Recovery Vehicle (LRV), will be carried out in this research paper. This mobile rotary crane was designed as a joint venture between three major international commercial vehicle manufacturers: IVECO - New Holland – Cummins. The aim of developing this EFI (Electronic Based Fuel Injection) based recovery truck was to meet new and more demanding customer expectations and also tougher EU regulations on environment safety. The Light Recovery Vehicle is being employed both in military and commercial outfits as a mobile rotary crane for extrication and evacuation of equipment casualty from the site of extrication to the nearest place where repairs can be effected.

Figure 10 - IVECO Euro Cargo Recovery Crane

The above figure 10 shows a IVECO Euro Cargo parked in a unit Military Transport Park and is available online[47]. However, there is as such no anti say system developed by the manufactures to regulate the payload oscillations during recovery operation being performed by this crane. However, in this paper, a suggested adaptive controller will be discussed utilizing he onboard sensors integrated with the EFI system in addition to newly suggested sensors required for anti-sway system. In order to develop a logical buildup, it is essential to present a brief overview of the hydraulic rotary crane installed on this recovery truck.

The wrecker equipment is mounted on a six-wheeled drive truck. When stabilized on its four stabilizers, the crane truck is suitable for hoisting loads up to 7500 kg. If it is not stabilized, it can be operated- with the vehicle at a standstill or moving very slowly -to lift and move loads compatible with the complex stability. When correctly equipped, it can perform all road off-road recovery and towing operations involving broken-down heavy vehicles.

3.6 **Operation of Hydraulic Crane**

A rotary type hydraulic operated crane is fitted on IVECO LRV 7.5 tone Euro Cargo. A hydraulic circuit is used to regulate the crane operation as per the commands given by the crane operator through various operating levers in the operator cabin. This operator cabin is fitted on a rotating turret type platform on recovery truck as shown in figure 11[48]. The circuit is supplied by one hydraulic pump driven by power take-off which receives power from the vehicle main gearbox. The switch units allow independent or simultaneous movement of all the crane components. The stabilizer control levers are situated on the right and left-hand body sides. The levers aimed at controlling the raising of jib and hook and rotating turret and rear pull winch are located in the turret. Front pull winch control levers are located on the front left-hand side of the cabin. Piloted lock valves guarantee stability even if the dynamic system's pipes are accidentally broken. Safety devices avoid interference of the hook-bearing block with the crane boom head and the complete unwinding of the load-lifting winch. Piloted lock valves guarantee stability even should the dynamic systems pipes are accidentally broken. The stabilizers are horizontal/vertical type and can be positioned even with partial extension reducing proportionally the payloads.

Figure 11 - Recovery Operation in Field Using a Mobile Rotary Crane

3.7 Functions of Main Components in the Hydraulic Crane System

The crane operation is based on the performance of miscellaneous mechanical sub-systems and hydraulic circuits. The mechanical power for deriving the various accessories is generated from the six-cylinder inline type engine which further derives the hydraulic vane type pump. It further supplies power to a control valve bank from where the hydraulic fluid is circulated with high pressure to operate any of the following:-

a. Hoist cable motor

b. Boom lift cylinder

c. Boom Swing motor

d. Winch motor

3.7.1 Power Divider Assembly

The power divider assembly is a single speed gearbox with one input shaft and output shafts. A bracket to the bottom of the hydraulic reservoir mounts it. The drive shaft yoke is connected by a propeller shaft and universal joint to the power-divider mounted on the rear of the transfer case. The winch output shaft yoke is connected by a universal joint to the front end of the rear winch. A coupling to the hydraulic pump input shaft connects the hydraulic pump output shaft. When the hydraulic pump control lever is in the disengage position, the air passage inside the governor control valve controlled by the governor valve mounted on the rear of the distributor drive housing. When the hydraulic pump control lever is in the engage position, the air passage inside the governor-valve control valve are arrange so that the engine speed governor is controlled by the governor valve mounted on the front of the power divider. The power divider, hydraulic pump and relief valve are removed from the vehicle as a single unit.

3.7.2 **Hydraulic Pump and Relief Valve**

The flange mounted van type hydraulic pump is bolted to the hydraulic pump adapter mounted on the rear of the power divider. The power divider output shaft derives the hydraulic pump output shaft coupling. The adjustment relief valve assembly is connected to the hydraulic pump outlet port. The purpose of this valve is to protect the crane hydraulic system from excessive 1450±50 PSI pressures.

3.7.3 **Swing Motor**

The swing motor mounted on the rear of the base plate consists of a pair of double acting hydraulic cylinders. The front end of both rods is connected to the pivot post derive pinion crank. The drive pinion at the lower end of the crank drives the ring gear at the bottom of the pivot post through an idler gear. The rear end of both cylinders is anchored by a pin to a bracket welded to the base plate, controls the flow of hydraulic oil through the cylinder.

3.7.4 **Boom Lift Cylinder**

The boom lift cylinder is vertically mounted to the rear of the pivot post. The lower end of the cylinders is pivoted on shift installed between the sides of the shipper support. The upper end of the lift cylinders is piston rods are pirated on a shaft installed between the sides of the shippers. Weight of the lift cylinders is approximately 530 ponds. By using overhead hoisting equipment to raise the front end of the boom to its position of maximum elevation, the boom lift cylinders can be removed without removal of the boom and shipper assembly.

3.7.5 **Boom Hoist hydraulic Motor and Cable Drum Assembly**

The boom hoists hydraulic motor and cable drum oil assembly is bolted to the rear of the shipper. Either the oil motor or the cable drum can be removed separately. However, the removal procedures in the section cover removal of both assemblies as a single unit. Weight of the boom hoist hydraulic motor and cable drum assembly is approximately 730 pounds.

3.7.6 **Boom Crowd Cylinder**

The boom crowd cylinder is mounted horizontally inside the boom and shipper assembly. Two nuts and a locking plate secure the rear end of the crowd cylinder piston rod to the anchor (welded to the rear end of the shipper). A collar welded the crowd cylinder at a point midway between the end is secured to this boom by two pin inserted to the sides of the boom and the collar.

3.7.7 **Power Divider Control**

The power divider controls consist of the rear winch control linkage and hydraulic pump control linkage is comprised of the hydraulic pump control lever rear control rod relay lever front control rod is connected to the lower end of the control lever by a yoke and pin. The front end of the rear control rod is connected to the left lever of the relay lever assembly by a yoke and pin. The rear end of the front control rod is connected to the pump-output shifter shaft arm and to the rear end of the governor-valve-control-valve control rod. The front end of the governor-valve-control-valve control rod. The front end of the governor-valve-control valve control rod is connected to the valve lever.

3.7.8 **Clutch Control Valve**

It is a two-way air valve connected in the air compressed air system between the air supply line and the roto chamber. The valve is bolted to bracket attached to the front of the wrecker body floor plate. When the valve lever is in the disengage position, compressed air is permitted to pass through the valve and control valve to roto-chamber airline into rear end of the roto chamber.

3.7.9 **Roto Chamber**

The roto chamber a single-acting air cylinder having a spring loaded piston, which causes the piston to move to and remain at the rear end of the cylinder whenever the clutch control valve lever is in the engage position. An adjustable yoke and pin to the clutch release outer lever connect the front end of the roto chamber push rod. Compressed air admitted into the rear end of the roto chamber that through the clutch control valve causes the piston inside the roto chamber to move forward. This causes the push rod and outer lever to contact and push the clutch release inner lever forward, thereby disengage the clutch.

3.7.10 **<u>Base Plate and Pivot Post Assembly</u>**

. The base plate and pivot post assembly as referred to in this paragraph consist of the crane base plate pivot post shipper support component and control valve bank assembly. The combined weight of these units is approximately pounds. The base plate is bolted to the crane body, which is bolted to the left and right frame side rails. The pivot post, which is hollow, is internally supported at the top and bottom by tapered roller bearings. Which are installed on tubular support member attached to the toe of the pivot post support, anchors the pivot post to the support which permitting the pivot post to rotate freely on its vertical axis. The shipper support, on which the boom and shipper assembly is pivoted, is bolted to mounting bosses cast on the sides of the pivot post.

3.7.11 **<u>Boom and Shipper Assembly</u>**

The boom and shipper assembly consists of the boom and shipper, which is telescoping tubular steel members having a rectangular-shaped cross section, held together by the boom crowed cylinder. The rear end of the shipper is pivoted on a pin installed at the top rear of the shipper support, which permits raising and lowering the front end of the boom. Weight of the shipper and boom assembly is approximately 2,150 pounds. Although the boom and shipper assembly removal procedure in this section require removal of the boom hoist hydraulic oil motor and cable drum before removal of the boom and shipper assembly, both assemblies can be removed as a single unit.

3.7.12 **<u>Control Valve Bank Assembly</u>**

The control valve bank assembly is bolted to a shelf at the front of the operator comportment. Wrecker crane operating instruction caution plate is mounted on the control valve bank cover.

3.7.13 **<u>Hydraulic Reservoir and Equipment Box</u>**

The hydraulic reservoir and equipment box assembly is bolted to the brackets attached to the frame side rails. Although the reservoir and equipment box removal procedures require removal of the power divider, hydraulic pump and relief valve before removal of the reservoir and equipment box, these assemblies can be removed with the reservoir as a single unit.

3.7.14 **Crane Body**

The crane body is bolted at the rear of left and right frame side rails by two U-bolts, one on each side, attached to the left and right frame side rails. The approximately weight of the crane body, included the outer riggers is 2100 pounds. The base plate and pivot assembly must be removed before the removal of crane body. A schematic description of the crane is presented in the figure with the help of a side and top view of the wrecker cane.

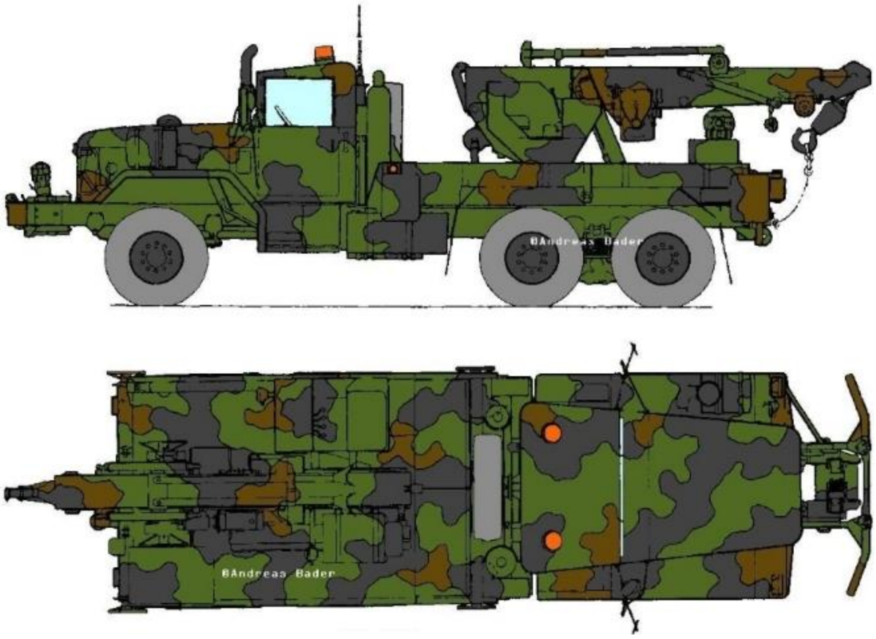


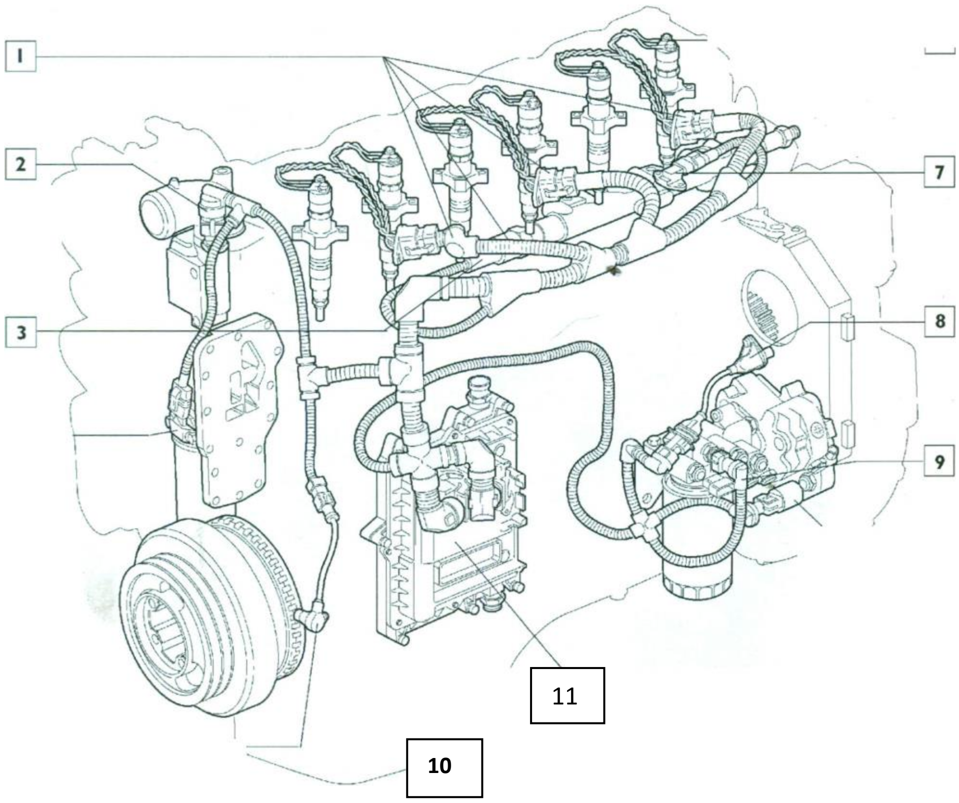
Figure 12 - Side and Top View of a Typical Recovery Crane Vehicle

3.8 **EFI System in the Recovery Truck**

IVECO 7.5 tonne Euro Cargo utilizes EFI system to provide optimal consumtion of fuel supply in accordance with operating conditions of the vehcile. This EFI system takes input from various sensors installed at different locations of the vehicel and feeds electronic pulses to and Electronic Control Unit (ECU) The control unit, depending on signals coming from various sensors, computes the optimum injection point according to an intemal mapping.The ECU thus turn regulates the fuel supply to common rail diesel fuel injection system thus eunsuring optimum fuel economy and reduced emissions of the burned fuel. Through the sensors, present on the engine, the ECU controls the engine operation. The description of these various sensors will be discussed in detail in subsequent lines as few of these sensors may be utilized for the feedback control of payload oscillations in the later chapeters of this research paper. In order to reduce particulates emissions, very high injection pressures are required. The Common Rail system allows injecting the fuel up to pressures reaching 1450 bar. at the same time, the injection precision, obtained by the electronic system control, optimizes the engine perfonmance, reducing emissions and consumption.

3.9 **<u>Desciption and Function of EFI System in the Recovery Truck</u>**

An EFI system in the Euro Cargo Recovery Truck consists of various sensors which are located on different assemblies / sub asseblies of the engine compartment. The location of most commonly used sensors is identified in the figure 13 given in Technical Manual of Euro Cargo LRV [49] as mentioned below.



<u>Figure 13 - Location of Different Sensors in Euro Cargo LRV</u>

The location of various sensors and related EFI components is mentioned as follows:-

a. Injectors connections.

b. Engine coolant temperature sensor.

c. Fuel pressure sensor.

d. Engine sensor.

e. Output shaft sensor.

f. Injector.

g. Air pressure/temperature sensor.

h. Camshaft sensor.

i. Fuel heater and fuel temperature sensor.

j. Pressure regulator

k. EDC 7 control unit

3.9.1 **Air Pressure / Temperature Sensor**

It is a component integrating a temperature sensor and a pressure sensor. Fitted on the intake manifold, it measures the maximum inlet air capacity to calculate precisely the fuel quantity to inject at every cycle.

3.9.2 **Fuel Pressure Sensor**

The fuel pressure sensor which is located on the end of the common rail of diesel fuel line, determines the pressure of diesel to measure the pressure of fuel injection. The injection pressure value is used to control the pressure and to determine the electric injection control length.

3.9.3 **Output Shaft Sensor**

It is an inductive sensor placed on the front engine part. Signals generated through the magnetic flow that is closed on the phonic wheel change their frequencies depending on output shaft rotation speed.

3.9.4 **Timing Sensor**

It is an inductive sensor placed on the engine rear left part. It generates signals obtained from magnetic flow lines that are closed through holes obtained on the keyed gear on the camshaft. The signal generated by this sensor is used by the ECU as injection phase signal. Though being equal to the flywheel sensor, it is not interchangeable since it has a different outside shape.

3.9.5 **Idle Speed Control**

The control unit processes signals coming from various sensors and adjusts the amount of injected fuel. It controls the pressure regulator and changes the injection time of injectors. Within certain thresholds, it also takes into account the battery voltage.

3.9.6 **Coolant Temperature Sensor**

It is a variable-resistance sensor suitable to measure the coolant temperature to provide the control unit with an index of the engine thermal state.

3.9.7 **Fuel Temperature Sensor**

It measures fuel temperature to provide the control unit with an index of the diesel fuel thermal state.

3.9.8 **Control Unit Microprocessor**

This control unit microprocessor is the brain of the EFI system. It receives feedback from the various sensors in the form of electronic signals. The pulse width of the these signals determine the injection timing and duration of the electronic fuel injectors thus ensuring optimum fuel supply to the engine combustion chamber. The control unit microprocessor allows storing certain EPROM data, among which failure memory and Immobilizer information, so as to make them available upon the subsequent start-up.

3.10 **Role of EFI in the Suggested Anti-Sway Control Scheme**

Euro Cargo Light Recovery Truck is not equipped with any anti-sway. However, a sensor based feedback control system can be designed for the LRV based on digital sensors in addition to the already available sensors of the EFI system. For instance, once the payload is being extricated with the help of recovery crane from the site of casualty to the repair site, then the hanged payload may observe increased oscillations due to the momentum and speed of the crane vehicle. In this case, the speed of the vehicle will be continuously sensed by the Speed Sensor. The signals generated through the magnetic flow that is closed on the phonic wheel change their frequencies continuously depending on the vehicle speed. The anti-sway control system, depending on the change in vehicle speed as determined by the vehicle speed sensor, dampens the payload oscillations to keep them in safe limit so as to ensure collision free path from the nearby traffic and road installations.

CHAPTER – 4

ANTI-SWAY CONTROLLER FOR RECOVERY CRANE VEHICLE

4.1 General

This chapter will discuss the phenomenon of sway observed in the payload of recovery crane vehicle. In addition, types of sway and reason for its suppression will be mentioned in this part of the study. It will be followed by the existing techniques to regulate the sway phenomenon. The suitability of controlling these oscillations with the help of a classic PID controller will also be deliberated along with the theoretical rationale.

4.2 Sway Phenomenon

The undesirable to and from motion observed in the payload once it is being lifted / recovered with the help of a recovery crane vehicle is called as sway. These payload oscillations are generated due to the inertia of the payload which causes the hanged mass to swing even when the crane has halted its operation. Resultantly, in the absence of a suitable control method to regulate these payload oscillations, time is wasted while waiting for ceasing of this sway motion before it can be place unloaded from the boom crane. These payload oscillations are dangerous for workers, operators, and cranes and load itself. Additionally sway decreases productivity of casualty handling and increases working cycle.

4.3 Types of Sway

There are different types of sway as illustrated in figure 14[50]. The main types are mentioned as follows:-

a. Primary Sway. In this casualty the casualty movement in an arc below the actuator, parallel to the actuator travel direction.

b. Secondary Sway. In the actuator travel direction there is so-called secondary sway, where the casualty rotates about the head block sheaves.

c. Longitudinal Sway. Move of the casualty back and forth in the gantry travel direction.

d. Rotational Sway. Rotation of the casualty about the vertical axis extended down from the actuator. This rotational sway is caused primarily by eccentrically loaded casualty.

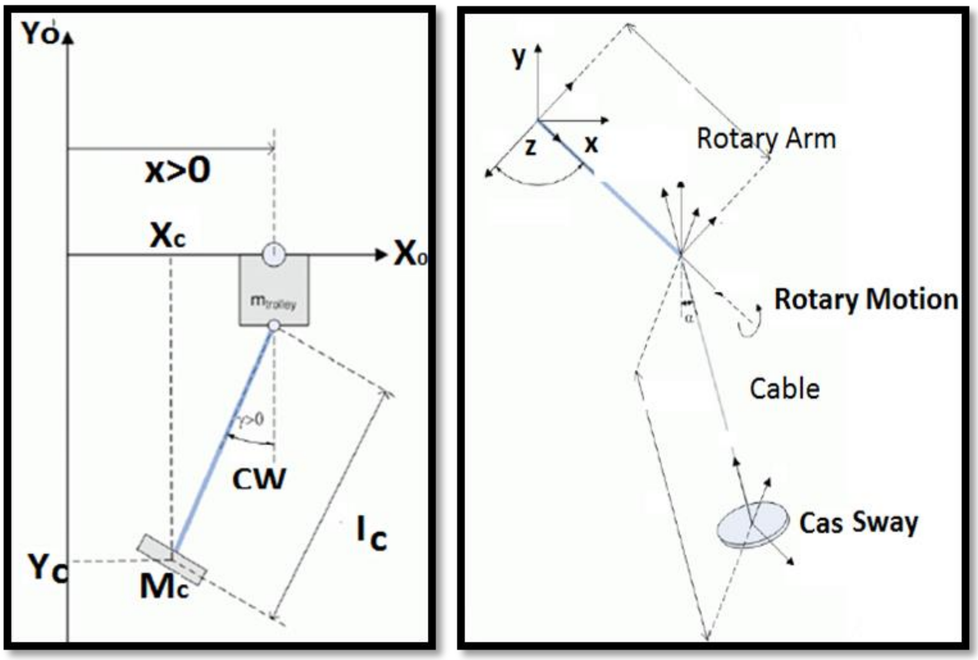


Figure 14 – Types of Sway

4.4 Requirement to Suppress Sway

The sway produced in a mobile recovery crane vehicle is detrimental for the speed and safety of the recovery operation and needs to be regulated due to following reasons:-

a. The oscillations are undesirable since a payload being lifted from a mobile crane vehicle, while oscillating, cannot be safely transported to its desired destination until its sway is dampened and it is brought to rest. Thus a significant amount of time is wasted while waiting for these oscillations to be suppressed. This phenomenon retards the speed of the crane operation.

b. A vehicle or equipment which is being hoisted from a mobile boom crane may collide with the other objects in the surrounding causing damage not only to its safe transportation but to other objects in the surroundings e.g. passing by traffic, other containers or personnel. Thus the safety of the recovery crane process is compromised which needs to be regulated by some control scheme.

c. The payload oscillations increase the magnitude of lateral forces causing increased impact of shear stresses on the boom crane metallurgy and the associated crane equipment including hoisting cable, hooks, chains, shackles spreaders etc. Thus the wear rate of these equipment increases if these oscillations are not regulated reducing their useful life period.

d. In the absence of a control scheme, sway suppression in a mobile boom crane is highly dependent on the skill and expertise of the crane operator. Thus more seasoned and experienced crane operators are preferred recovery operations which demand significant training of these operators. However, even a new inexperienced crane operator may be trained after a shorter time scan for performing delicate recovery operations if a suitable robust controller is installed in a mobile boom crane.

4.5 **Sway Damping Techniques**

In case of overhead or fixed craned, state of art sway damping techniques has been applied all over the world[51, 52]. However, for hydraulic operated rotary cranes employed in mobile recovery vehicles like IVECO Euro-Cargo LRV, sway is up till now manually controlled and depended on the skills of the recovery operator or manual control of sway by employing recovery personnel. However, we can go through the different sway damping techniques used in other types of cranes for building our proposed control scheme. Hydraulic sway damping with high reliability dissipates the sway energy of the load but do not prevent the sway from starting. This technique of hydraulic sway damping dissipates the sway but doesn’t prevent sway initiation. Inclined auxiliary ropes damp the sway, since the head block is subject to very heavy mechanical shock loads and increases the weight of the lifted load and decrease the free space available for maintenance purposes, see fig. 4.4.

Electronic anti-sway prevents the sway by manipulating the acceleration and deceleration of the actuator or gantry motion. Math algorithms modify the velocity of the actuator or gantry so that the sway is compensated but the crane driver loses control of the travelling motions; they are not capable of preventing rotational sway, because the systems manipulate the acceleration/ deceleration of the actuator / gantry motions, not each individual hoist rope fall.

4.6 Proposed Anti-Sway Controller for Rotary Crane of IVECO Euro Cargo

Since type of crane mounted on the recovery vehicle of IVECO Euro Cargo is a rotary one, therefore it can be safely assumed that this rotary crane motion will consist of load hoist and boom hoist. The control of load hoist is manageable for the crane operator as it does not result in load sway. However, for boom hoist there is a requirement for designing a control system to dampen the payload oscillations. The control model should be robust enough to ever varying parameters such as length of hoist cable, mass of the payload friction forces. The need to measure the rope tension can be eliminated if the controller is robust with respect to the change in rope length. It is noteworthy that as the calculating of the rope plays an important role in the natural frequency of the load–rope system, therefore, control scheme must be rigid with changes in the natural frequency of the load–rope.

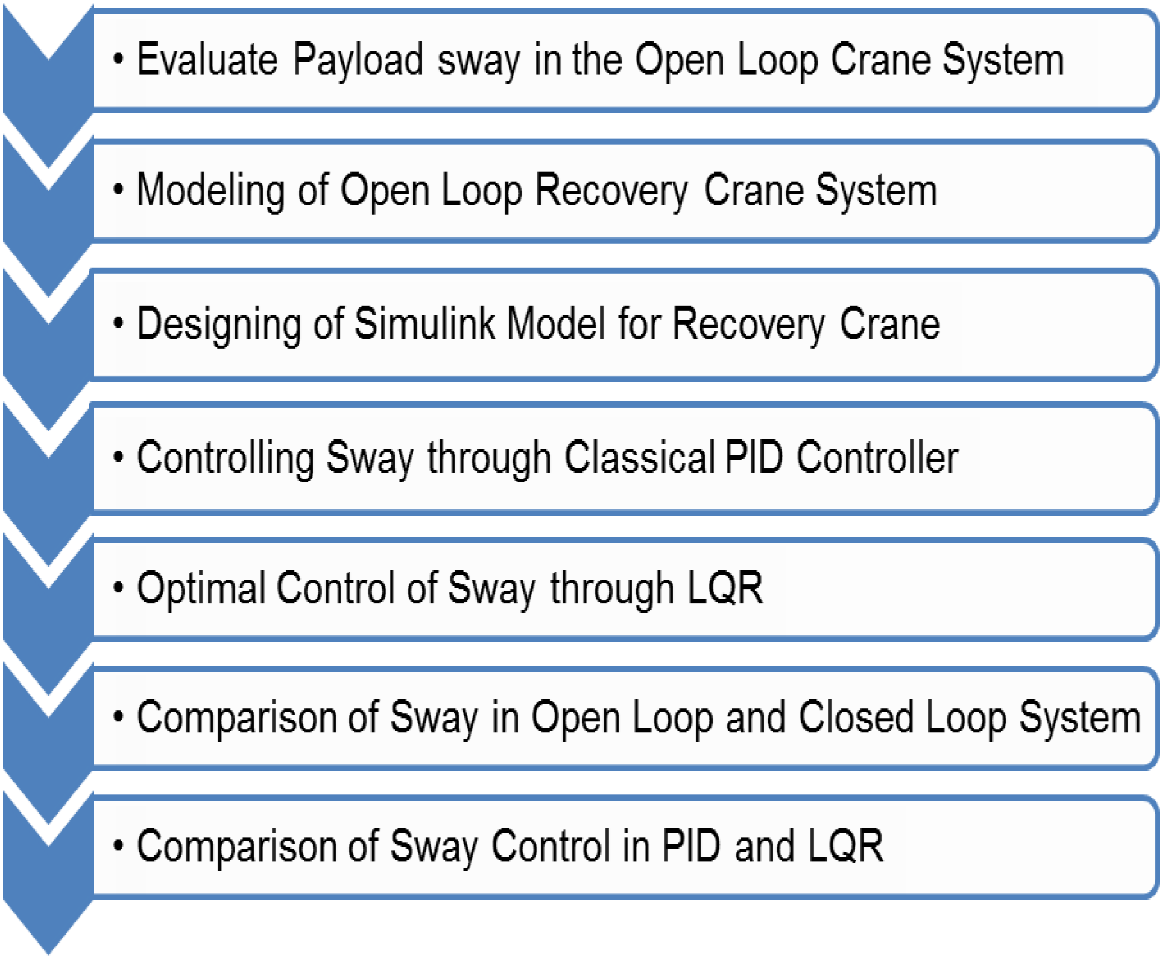


Figure 15 - Flow Chart for Optimal Controller Design

4.7 **Controller Design Basis**

The crane is equipped with an installed diesel engine of 7.5 ton capacity. The luffing motion of the boom crane is operated by a hydraulic cylinder mounted between crane tower and boom. All hydraulic drive systems are closed systems and each hydraulic circuit is operated by a separate pump at the power divider gear box. The crane can be operated by a crane cabin 5 meters above the ground with a good visibility on the payload. The cabin provides hand levers for the operation of the crane which delivers the signal directly to the central control unit via a CAN bus. The central control unit connects all the other inferior hardware systems. Various sensors such as engine output shaft sensor for measuring speed of the crane vehicle, optical sensors for measuring positions and change in swing angle of the payload are installed on the engine and boom crane respectively. In order to limit the maximum torque on the mobile crane, a load cell is installed to measure the actual crane load. The system is integrated in the winch, measuring in fact the force on the rope. By CAN bus, all feedback signals from sensors and actuator are connected to the central control unit. For the control system the sensors are provided. To measure the slewing position of the crane tower, the position of the winch respectively the rope length, the position of the boom respectively the outreach of the boom tip, encoder systems are installed. The resolution of the encoders is calculated to reach a control performance of a few centimeters concerning to the position accuracy of the crane load.

4.8 **Motor Encoder Arrangement**

The cart is equipped with a DC motor and an encoder to move the payload up and down. The motor output shaft is connected to a pinion pulley that drives an output pulley, through a belt. This rotates the pulley or spools. The pulley is wound with a cable and the bottom of the cable is attached to a payload. The pulley therefore acts as a spool that can reel in or reel out and thus change the height of the payload. A gimbal is attached near the bottom of the cart and it allows the cable to sway in any direction. There is a limit switch, i.e. fastened to the bottom of gimbal. The sensor is triggered when the payload pressed down on the switch. This signal can be used to prevent the load from not being risen any further and for calibration.

The optical encoder measures the swing of the cable when it swings lengthwise with the jib. The optical encoder, measures the swing of the cable when the payload moves perpendicularly to the jib. The potentiometer can also be installed along with the encoder system which can give the variation of the voltage change due to the change in the position of the cable with respect to the position of the casualty being hooked.

4.9 **Intelligent Control of Payload**

The frequency and amplitude of payload oscillations increases with the increase in speed of the recovery crane vehicle while the vehicle is on move. In the conventional mobile boom crane vehicles, where there is no control scheme to regulate the payload oscillations, the payload oscillations also reaches a dangerous level with the increase of crane operation. Therefore, in order to regulate the sway in the payload within a safe level, the speed of crane operation has to be kept in a safe limit. In the design of robust controller for regulating these payload oscillations, a pushbutton or joystick type radio remote transmission may be used which employs controllers with carrying frequency drive. This arrangement which caters for active load control is equipped with a switch for adjusting sling length for fine tuning on distant transmission sent by radio.

The requirement to actively control the payload oscillations demands that the operation is automatically controlled and the rates of acceleration and deceleration balanced automated; crane acceleration and deceleration rates are optimized in accordance with the frequency converter derive. This configuration permits the crane operation speed to enhance as per the requirement with full focus on transporting the supported mass without requiring manual regulation of. Thus the safety of the crane operation is less dependent on the experience of the crane operator.

4.10 **PID Controller System**

The Control system on the basis of which the system designed is the implementation of PID (Proportional-Integral-Derivative) Control System which would cater the sway in the recovery operation. PID Control system will be elaborated in this section. PID is a closed loop feedback based controller widely used on a large scale of industrial control applications. A PID controller receives the measured system response and compares it with a Preset Value (PV). The difference between the measured and reference values is forward as an input to the controller for adjusting the system response as per the desired response through use of a manipulated variable (MV). A PID controller is made of three distinguished parameters namely proportional, integral and the derivative values, symbolized as P, I, and D. These parameters can be simply interpreted in form of time. The proportional terms are dependent on the present error. The integral parameter is based on accumulation of past errors and the term derivative predicts on the basis of existing rate of change the estimated future errors. The system response is adjusted on the basis of aggregated sum of these three parameters through a controller based element such as a damper or an actuator or a control valve position.

PID controller has been regarded as the most useful kind of controller in those situations where the information and knowledge with respect to the underlying process is not available. The three integral parameters of the PID controller can be tuned as per peculiar process requirements to in the algorithm of the PID controller. The response speed of the controller to the error between the reference and measured valued and the extent to which it overshoots to the reference PV, become the basis for determining the responsiveness of the controller. However, the optimum control of the system or its stability is not ensured by a PID controller. In few applications, only two of the three parameters may be tuned for controlling the system. In such situations, a PID controller will be termed as PD, PI, I or P controller in the absence of other parameters. The most common of such forms of PID controller is PI as derivative action is sensitive to measurement noise and absence of an integral term may obstruct the system from reaching its intended value.

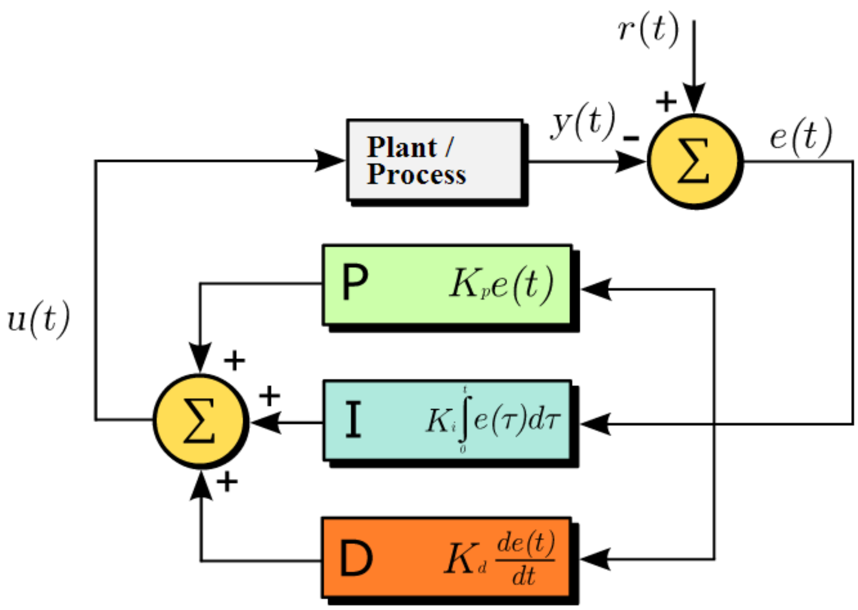


Figure 16 – Schematic of a PID Controller in a Closed Loop

4.11 PID Algorithm

A PID controller is named so because of its three constituting proportional, integral, and derivative terms whose aggregate determines the input for the Manipulated Variable (MV) output of the PID controller. If $u(t)$ is the output of the controller then the PID algorithm takes the following form:-

$$u(t) = MV(t) = \mathrm{K}_p e(t) + K_i \int_0^t e(\tau)d\tau + K_d \frac{d}{dt}e(t) \tag{4.1}$$

Where,

K_p can be termed as Proportional gain.

K_i can be termed as the Integral gain.

K_d can be termed as derivative gain

t can be termed as instantaneous time

e can be termed as the error

T can be termed as the variable of integration

4.12 Proportional Term

The output value is produced by a proportional term in relation to the current error value. The tuning of the proportional response can be carried out by taking the product of error with the proportional gain constant, K_p. Thus this term is expressed as follows:-

$$\mathrm{P}_{out} = K_p e(t) \tag{4.2}$$

If there is a high proportional gain then it forces a significant change in the output response for a giving change in the error. The stability of the system is affected if there is an increased value of the proportional gain. But a small output response if to a large input error is seen if the value of the proportional gain is raised. As per the industrial practices, the proportional term will result in majority of the output change. It is noteworthy that as the value of the error is always non-zero to derive a PID controller therefore, it usually generally operates with a steady-state error termed as droop or offset. Droop is equal to the process gain and is inversely proportional to proportional gain. It may be lessened with addition of an integral term.

4.13 Integral Term

The contribution from the integral term is proportional to both the magnitude of the error and the duration of the error. The integral in a PID controller is the sum of the instantaneous error over time and gives the accumulated offset that should have been corrected previously. The accumulated error is then multiplied by the integral gain (K_i) and added to the controller output.

The integral term is given by:

$$I_{out} = K_i \int_0^t e(\tau)d\tau \tag{4.3}$$

The integral term accelerates the movement of the process towards set point and eliminates the residual steady-state error that occurs with a pure proportional controller. However, since the integral term responds to accumulated errors from the past, it can cause the present value to overshoot the set point value (see the section on loop tuning).

4.14 Derivative Term

It is determined with the rate of change found from the error curve slope with respect to time and multiplying this by the derivative gain K_d. The magnitude of the contribution of the derivative term to the overall control action is termed the derivative gain, K_d. The derivative term is mathematically expressed by the following expression:-

$$D_{out} = K_d \frac{d}{dt} e(t) \tag{4.4}$$

Derivative action predicts system behavior and thus improves settling time and stability of the system. A typical PID controller requires a low pass filter for the derivative term to restrict disturbance and High Frequency gain. Due to carrying effect of derivative term on the stability of the system, it is rarely employed in practical applications.

4.15 Loop Tuning

In loop tuning control parameters, for the purpose of desired control response, including integral and derivative gain, and proportional band etc. are optimized.

Different systems and different applications may vary in requirements which may be conflicting with each other , however, the basic requirement of stability of oscillations is a pre-requisite for all. Though there are only three parameters in a PID but its tuning is a complicated task since it should meet difficult criteria keeping in view the limitations posed by PID control. There exist different methods for PID tuning. For instance, a PID controller can be tuned, using default tuning. However, computer simulations are carried out again and again for tuning the initial designs, until closed loop system performances are achieved. Few systems have a degree of non-linearity involved therefore, the parameters which give satisfactory results under full-load operating conditions don't give the same performance when the system is staring from zero load condition. This shortcoming can be corrected by employing different parameters for different operating conditions that is the process of gain scheduling.

4.16 **Stability of System Response**

The incorrect selection of parameters for PID control e.g. the gains of If the PID controller parameters including the gains of the three terms of PID can lead to unstable input of control process. It can result from an exceeding gain and can be restricted by saturation. Although marginal stability may be acceptable in few cases, stability of the system response is required and system must not oscillate for any combination of operating conditions and set points.

4.17 **Optimized System Behaviour**

As per the requirements of an application, the optimum behaviour is different on the change of set point or process change. There are two requirements involved i.e., the suppression of the noise / disturbance and implementation of variation in the desired value or in simpler words, the capability of the manipulated / control variable to track the set value. is done by the control variable. The rise time and settling time are the two particular pre-requisites for command tracking. There are few processes which do not permit a system variable to go beyond an overshoot value for the sake of safety whereas others may reduce the energy expended in reaching a new desired point.

4.18 **Tuning Methods**

Out of the various methods available for PID tuning, the selection of a particular method will depend on system response time and whether the loop can be taken offline for tuning or not. In the scenario where the system can be taken offline, then in such a situation, the most suitable method will involve the step input and then determining the system response over time to use this output for finding control parameters.

4.19 **Tuning by Manual Methods**

In those situations, where the system cannot be taken off-line, then the suitable tuning method will be first setting K_i and K_d values to zero and then increasing the value of K_p until the output of the integral in offline oscillates. This is followed by setting K_p to half of that value approximately. Subsequently, K_i is made to rise (provided the system response remains stable) till correction of any offset in sufficient time for the process. In the last, the value of K_d is increased, provided that the loop is prompt enough to reach its referred value after a load disturbance. It is noteworthy that excessive overshoot may result from exceeding K_d unnecessarily. The set point can be reached promptly if a fast PID loop tuning is adopted. Yet, closed loop over-damped system (with setting of K_p less than half of the K_p setting resulting in oscillations) may be essential in cases where a system cannot accept overshoot.

4.20 **Limitations of PID Control for Rotary Crane**

PID controllers which are widely employed in many control schemes and give acceptable results with coarse tuning. However, they may not be successful in providing optimum results. Since PID is based on feedback system without any knowledge regarding direction of the process, thus as a whole performance may have to be compromised. Although PID controller in our control scheme for Euro Cargo may be the most suitable controller without modeling of the system, improved results may be achieved by explicitly modeling the process actor and that to not relying to the presence of an observer.

In those cases, where PID loop gains have to be reduced for preventing overshoot of the control system or its oscillation around set point value, the alone use of PID controller may yield unsatisfactory results. In cases of non-linear system response as well, a PID may fail to adapt to varying system response and thus a balance between regularity and response time may be compromised. In such cases it is advisable to use feed forward control, with the employment of PID for controlling error only. As a substitute measure, modification of PID can be carried out for e.g., changing of Alternatively, PIDs can be modified in more minor ways, such as by changing the parameters like gain scheduling or on the basis of performance, this modification can be carried out. Improvement in measurement can also be carried out for e.g., high sampling rate, precision and accuracy along with use of low pass filter. Cascading of different PID controllers can also be adopted alternatively.

4.21 **Linearity**

One of the shortcomings of the PID controller is their linearity and symmetric, therefore, their performance in non-linear systems varies. For instance, in the applications involving temperature control, active heating along with passive cooling may be desirable thereby slowing the adjustment of overshoot. The solution in such a scenario is overdamping the tuning of the PID for preventing the overshoot at the cost of increase in settling time..

4.22 **Disturbance in Derivative**

One of the commonly faced issues with the term of derivative is its amplification of high frequency measurements or the disturbance in the system thus causing the output of change drastically. The extent of this amplification is to such a larger scale that a true derivative term may not be present in a physical controller but only an approximation with a restricted bandwidth. The use of a low pass filer may be helpful in few cases for eliminating noise / disturbance with higher frequency. There may be a restriction on the extent of filtering since, a derivative control and low pass filter may cancel each other thus giving rise importance of instrumentation of a low pass filter. The performance and capability of filtration may be improved by employing a non-linear median filter. Alternatively, switching off the differential band may be adopted which implies the use of PI controller instead of PID controller.

CHAPTER 5

MODELING OF RECOVERY CRANE SYSTEM

5.1 General

In this chapter, a suitable mathematical model will be developed for the rotary type crane in a hydraulically operated boom crane vehicle. In addition, the luffing motion of boom crane will also be modeled to design and testify a suitable control scheme in the subsequent chapters. Since the mobile crane vehicle is expected to operate in a field environment, therefore, a relevant model for catering the effect of varying environmental parameters including wind speed, air drag etc. will also be included.

5.2 Dynamic Model of a 3-D Recovery Vehicle Hoisting Payload

In order to implement the Control System upon the operation of a Recovery Vehicle hoisting a casualty we first need to formulate the equation of motion for the casualty being hoisted by the recovery. Since type of crane mounted on the recovery vehicle of IVECO Euro Cargo is a rotary one, therefore it can be safely assumed that this rotary crane motion will consist of load hoist and boom hoist. The control of load hoist is manageable for the crane operator as it does not result in load sway. However, for boom hoist there is a requirement for designing a control system to dampen the payload oscillations[53]. The modeling of system is necessary to introduce anti-sway controller in the overall system.

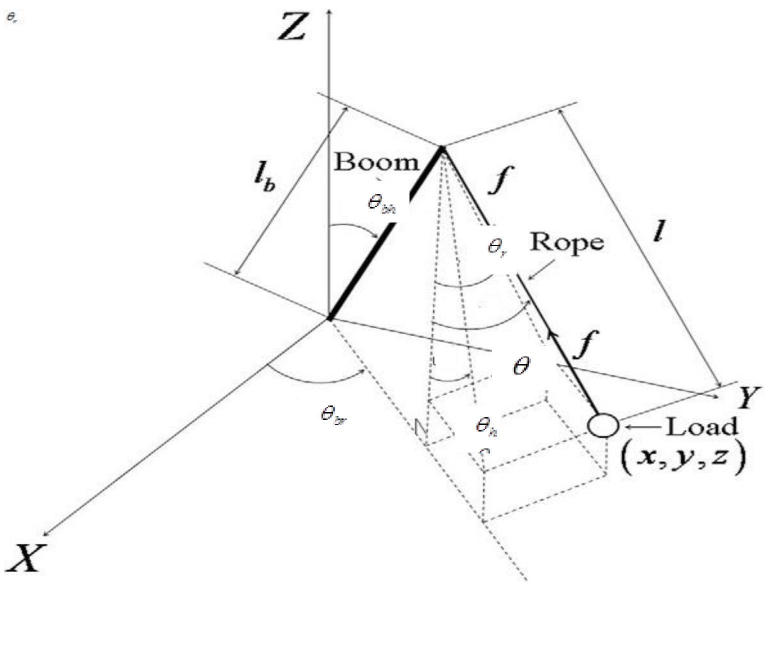


Figure 16 - Schematic Model of the Rotary Crane

The above figure shows the rotary type hydraulic crane model of our Euro Cargo Recovery vehicle. In order to model the system, few notations will be used.

x, y and z are the payload's 3 D positional coordinates

θ is payload sway angle as measured from the vertical,

θ_h is the component of payload sway angle along boom hoist direction

θ_r is the component of payload sway angle along boom rotation directions

θ and $\theta_{br} = ku$ is the boom hoist and rotation angle.

m is payload weight

l is the length of the rope

l_{bm} is the boom length

f is the rope tension

If the controller gains is donated by c_g and c_i is the boom rotational motion's control input then following mathematical expressions can be derived for the rotary crane as under:-

$$\ddot{x} = -\frac{a_g}{l}(x - l_{bm}\sin\theta_{bh}\cos\theta_{br}) \tag{5.1}$$

$$y^{\cdot\cdot} = -\frac{a_g}{l}(\text{y} - l_{bm}\sin\theta_{bh}\cos\theta_{br}) \tag{5.2}$$

$$\theta_{br} = c_g c_i \tag{5.3}$$

It is noteworthy that the dynamics of the boom hoist motion may be integrated during modeling of the system. In addition, load position x and y are represented by load sway angle θ_h and θ_r for driving controller for the payload oscillations. Few assumptions are adopted during modeling for instance we may ignore the rope, twisting and stretching effect under rope tension. The payload may be taken as a point mass. Now the dynamics for hoist and rotation of the boom are represented as follows

$$J_{i+2}\theta^{\cdot\cdot}_{i+2} + d_{i+2} = \tau_i, \qquad i = 1, 2 \tag{5.4}$$

In above eq,

J_{i+2} is the nominal inertia,

τ_i is the input torque

d_{i+2} indicates the disturbances.

It is worth mentioning that our case study model that is the Euro Cargo mobile boom crane will be operating in a field environment where the effect of noise and other disturbances cannot be ignored. This situation is in contrast to the overhead / fixed cranes working in a controlled / structured environment. Thus in order to negotiate these disturbance effects, we will need to design an observer with a configuration as mentioned below:-

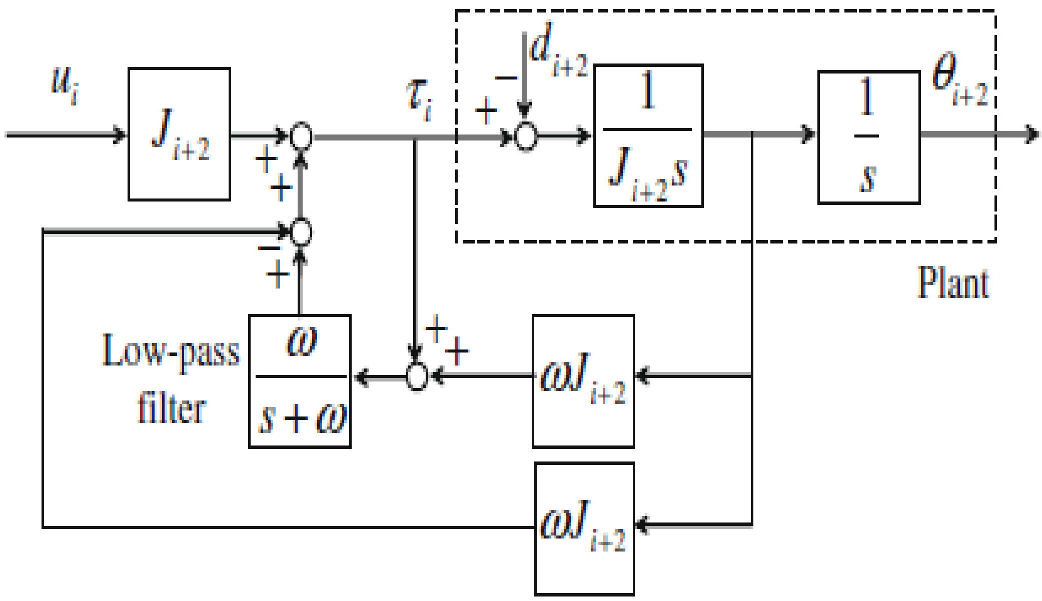


Figure 17 – Observer for Addressing Noise in the Boom Crane Control Scheme

In the Figure,

s is operator for the derivative

u_i is the control input of the low pass filter

ω is the cutoff angular frequency of the low-pass filter.

However, in the presence of EFI sensors and newly suggested sensors in our case study of Euro Cargo, the system states can be accurately determined with the feedback achieved through these sensors. Therefore, there will be no need for estimating the states with the help of a controller designed for the system under study. For purpose of simplifying our case for modeling the system, we can make an assumption that the sway angle is so small that it is reasonable to approximate

$\sin\theta_h$ = $\sin\theta_r \simeq$ 0 and $\cos\theta_h$ = $\cos\theta_r \simeq$ 1. This assumption leads to the following mathematical expression with respect to the sway mechanics of the payload:-

$$\mathrm{m}\ddot{x} = -f(\theta_h \cos\theta_{br} - \theta_r \sin\theta_{br}) \tag{5.5}$$

$$\mathrm{m}\ddot{\mathrm{y}} = -f(\theta_h \sin\theta_{br} + \theta_r \cos\theta_{br}) \tag{5.6}$$

$$\mathrm{m}\ddot{\mathrm{z}} = -mg + f \tag{5.7}$$

Thus equation for locating coordinates of the hung mass in 3D can be formulated as:-

$$x = l_{bm} \sin\theta_{bh} \cos\theta_{br} + l(\theta_h \cos\theta_{br} - \theta_r \sin\theta_{br}) \tag{5.8}$$

$$y = l_{bm} \sin\theta_{bh} \sin\theta_{br} + l(\theta_h \sin\theta_{br} + \theta_r \cos\theta_{br}) \tag{5.9}$$

$$z = l_{bm} \cos\theta_{bh} - l \tag{5.10}$$

The following equations can be derived from eq (1) to eq (6) :-

$$\ddot{x} = -\frac{g + \ddot{z}}{l}(x - l_{bm} \sin\theta_{bh} \cos\theta_{br}) \tag{5.11}$$

$$\ddot{y} = -\frac{g + \ddot{z}}{l}(\mathrm{y} - l_{bm} \sin\theta_{bh} \sin\theta_{br}) \tag{5.12}$$

by differentiating equations (4), (5) and (6) with respect to time twice and substituting them into eq (7) and (8) followed by the process of linearization of the obtained terms with respect to desired states and inputs, we have the following state space representation of the model:-

$$\dot{x}_i = A_i x_i + b_i \omega_i \qquad \text{Where i =1 and 2}$$

$$[x_i \quad x_{i+4} \quad x_{i+6}]^T$$

$$A_i = \begin{pmatrix} 0 & 1 \\ -\xi & 0 \end{pmatrix} \tag{5.13}$$

$$B_i = \begin{bmatrix} \alpha_i \\ 1 \end{bmatrix} \tag{5.14}$$

$$C_i = \begin{bmatrix} 1 & 0 \end{bmatrix} \tag{5.15}$$

$$D_i = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \tag{5.16}$$

Where, $$\xi = \frac{g}{l}$$

$$\alpha_1 = -\frac{l_{bm} \cos\theta_{bh} f}{l}$$

$$\alpha_2 = -\frac{l_{bm} \sin\theta_{bh} f}{l}$$

It is noteworthy that for the system to be controllable $\sin\theta_{bh} \neq 0$ and $\cos\theta_{bh} \neq 0$.

5.3 Simplified Model for the Luffing Movement

After modeling the rotary crane in our hydraulic operated Recovery Vehicle, we now draw a model for expressing luffing motion of the hydraulic boom. The modeling of the luffing motion is essential since this luffing motion of the boom crane plays a crucial role in the resultant payload oscillations. For the purpose of modeling this motion, the motion of the hydraulic boom crane can be simplified with the help of a simplified diagram as shown below:-

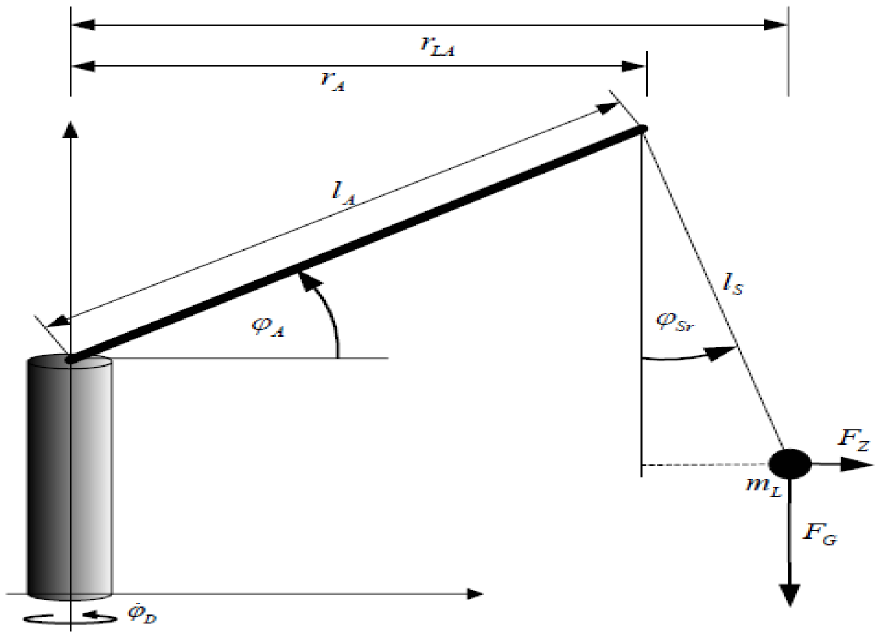


Figure 18 - Model for Luffing Motion of a Boom in a Hydraulic Crane

As shown in Figure 5,

ϕ_{Sr} is the radial angle for the rope,

$\ddot{\phi}_{Sr}$ the angular acceleration in radians

$\dot{\phi}_{D}$ the cranes rotational angular velocity

l_S is length of rope,

r_A is vertical distance till the boom end

$\ddot{r}_A$ is radial acceleration boom end

F_z is the centrifugal force

Now for catering the hydraulic cylinder / actuator dynamics, a first order system will have to be modeled. It can be noticed that the payload oscillates with the radial rope angle φr in the radial direction and can be mathematically expressed as follows:-

$$\ddot{\phi}_{Sr} + \frac{g}{l_s}\sin(\phi_{Sr}) = -\frac{1}{l_s}\ddot{r}_A + \frac{1}{l_s}(r_A + l_S\phi_{Sr})\dot{\phi}_D^{\,2} \tag{5.13}$$

Above equation is derived from the simple pendulum equation. Keeping into account the dynamics of the hydraulically actuating cylinder the motion for the hydraulic operated boom cylinder can be expressed as follows:-

$$\ddot{z}_{zyl} = -\frac{1}{T_W}\dot{z}_{zyl} + \frac{K_{VW}}{T_W A_{zyl}}u_l \tag{5.14}$$

Where

$\dot{z}_{zyl}$ is the boom cylinder velocity

$\ddot{z}_{zyl}$ is the boom cylinder acceleration

T_W is the time constant

A_{zyl} is the area of cross-section for the cylinder

u_l denotes the input voltage of the servo valve

K_{VW} represents the constant of proportion for the flow rate.

5.4 <u>Effect of Wind and Environment During Recovery Operation in Field</u>

Since a Euro Cargo Recovery Vehicle is designed to perform recovery crane operations in open field environment, therefore, the payload oscillations observed during field operations are doomed to be affected by the tough field environments.

Unlike the operation of overhead fixed cranes, which are used in industry, where mostly the cranes operate in a closed loop controlled environment, our subject of interest that is the boom crane vehicle will be functioning in open loop uncontrolled environment. Since, these field conditions are liable to variation, thus there is a need to model the effect of these changing environmental parameters during our modeling process. It is noteworthy that while carrying out recovery operation in the field conditions, the rotary crane of the hydraulic crane is affected by the varying wind speed and the prevailing environment. Now on the basis of aerodynamics of the drag force[54], the wind induced drag force acting on the crane structure can be determined as follows:

$$F_D(t) = \frac{1}{2}\rho C_D A U^2(t) \tag{5.15}$$

Where,

ρ is the density of air,

C_D is the coefficient of drag,

A is the projected area of the system structure

U(t) represents the speed of wind

$\mathrm{U(t) = \bar{U}(t) + w(t)}$

Where

Ū represents the constant speed of wind as per the sea level

w(t) represents the varying wind speed.

The wind-induced drag force, $F_D(t)$ can therefore be mathematically modeled as under:

$$F_D(t) = \frac{1}{2}\rho C_D A\bar{U}^2 + \rho C_D A\bar{U}w(t) + \frac{1}{2}\rho C_D A w(\mathrm{t})\left|w(t)\right| \tag{5.16}$$

Here the first term in the above equation is called the average for the mean drag force and is usually constant at a given speed of wind. However, the second and the third term in the same equation those which are relevant with the factors of turbulent winds. We can neglect this term while determining the effect of wind on structures. For ease of simplification , the projected area of the hydraulic crane can be assumed as negligible.

5.5 System Modeling and Control Design

Based on the research carried out on the various control system being implemented by the industrial section and a detailed study of the Euro Cargo Recovery Vehicle Operation, we have proposed a feedback system to control the sway of the casualty. It controls the sway angle while keeping the payload oscillations that occur in the hoisted casualty, at a minimum. A feedback control system is designed to rotate the boom to a desired angle, i.e. movement the boom to and fro or back and forth, and try to decrease / dampen the swings that are perpendicular to the actuator in the casualty. The modeling of Recovery Crane Cabin is carried out in sketcher software with its isometric and top views are shown in figure 18 below:-

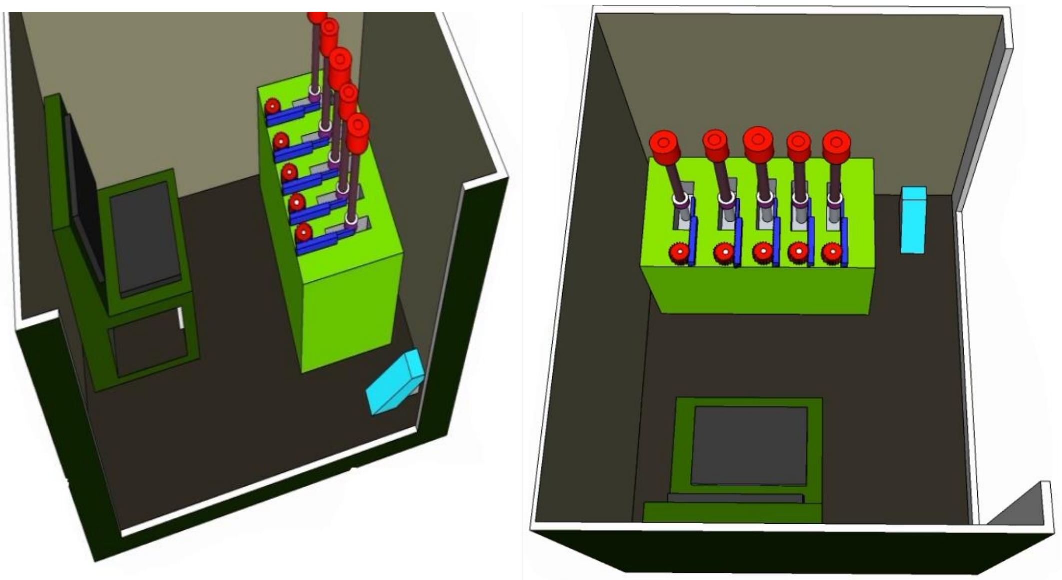


Figure 19 - Modeling of Recovery Crane Cabin

The above figure also shows the modeling of the improvised spool valve bank assembly where a suitable control scheme can be implemented. The closed loop feedback from various sensors is received in the form of electronic pulse at the control unit which then adjusts the motion of the crane boom levers at the spool valve assembly as per the sway of the payload. The position of the payload is control using a proportional-integral-derivative compensator, or PID. There are sensors which measure the speed of the payload directly, therefore their feedback is used for calculating the difference between the ideal response and actual response i.e. the error in this case. For control design purposes, the straight derivative is used. A high-pass filter is used to obtain the velocity from the encoder.

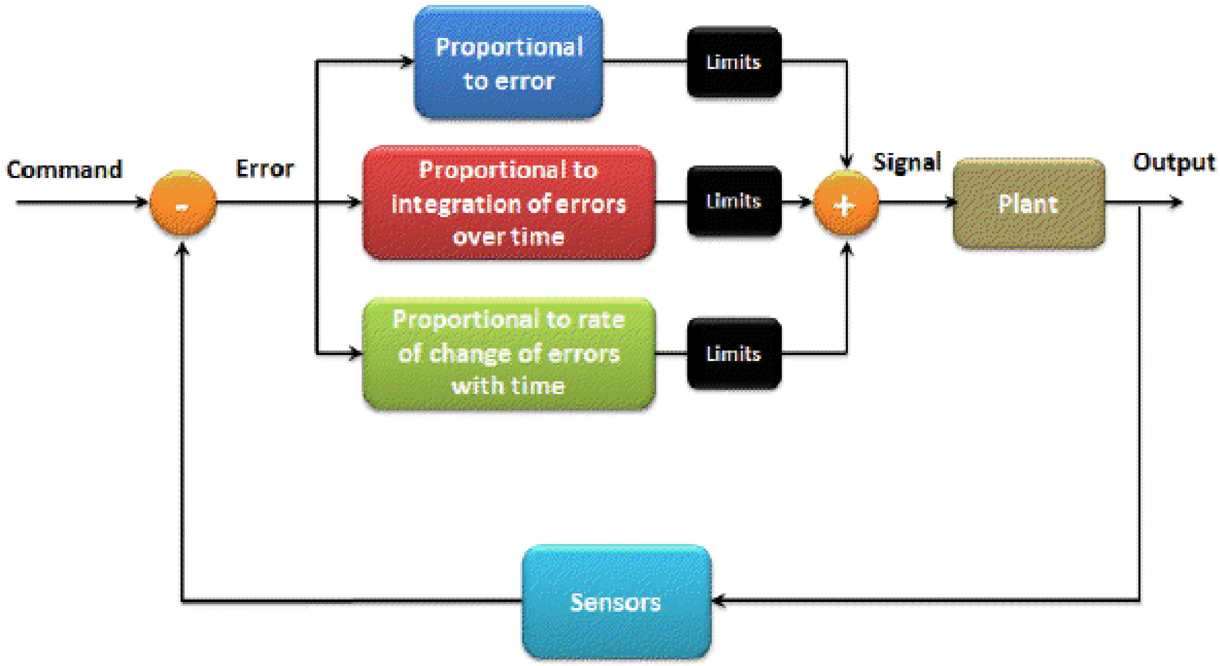


Figure 20 - Overview of the PID Control System being Implemented

It can be seen from the figure below that the main cause of sway initiation is the change in the velocity the casualty / payload that is being hoisted by the recovery crane/boom. In order to cater for the sway once the casualty moves in forward direction, the boom is extended to move in forward direction. Whereas the boom is retracted in the reverse direction once the sway in the backward direction exists in a casualty motion. Therefore, an optical sensor will be used in the proposed control scheme to measure the rate of change of speed and velocity in the payload oscillations during the crane operation.

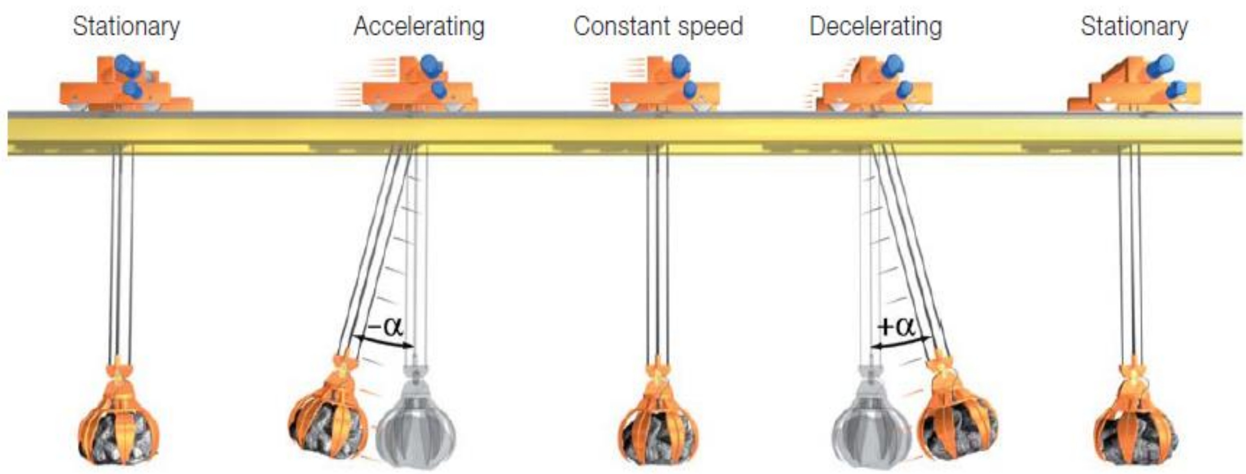


Figure 21 - Change of Velocity and Acceleration in Payload Oscillations

The Simulink model of the mobile boom crane to control the payload oscillations is given in the figure below. A step input is given in the control system that is further changed in a continuous term by using the function of sigmoid so as to attain a gradual ascending / descending value. As the Recovery Vehicle lifts the casualty via hoist cable through the boom, there is a sudden change in the voltage of encoder that becomes a cause for the sway initiation in the casualty. As the geometry of our recovery vehicle`s boom is somewhat different from the over-head cranes so we cannot simply use the technique of damping the sway by moving the boom actuator in the direction of sway being produced that is extending and retracting the boom.

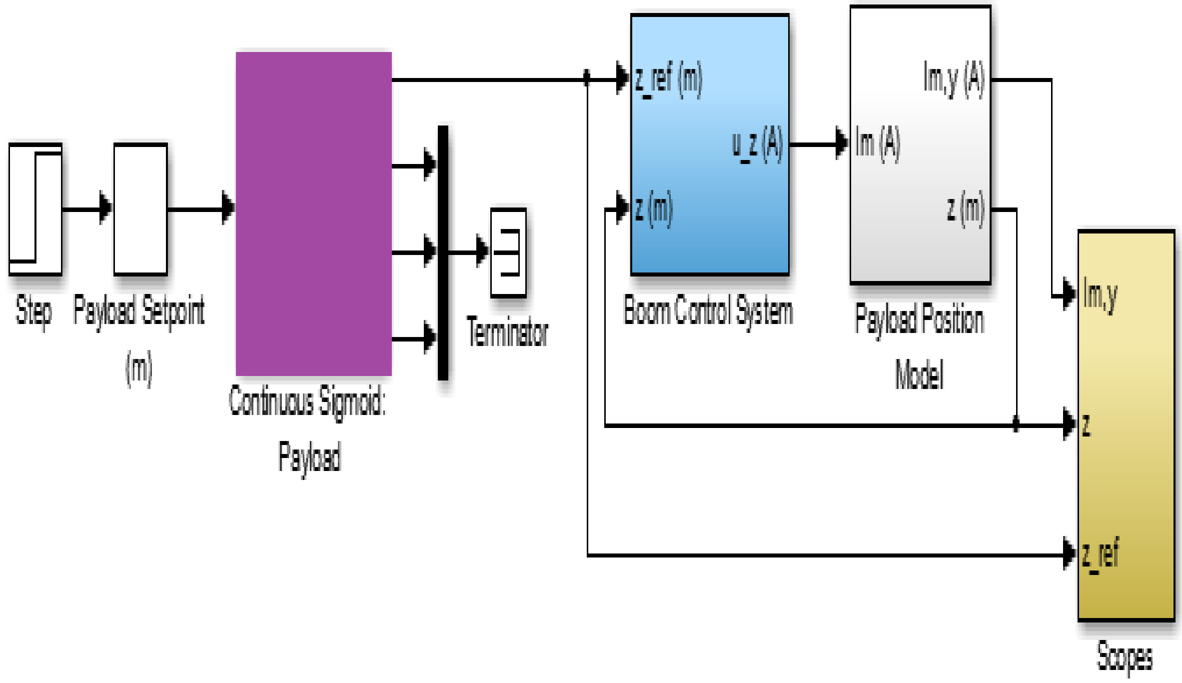


Figure 22 – Simulink model for Mobile Boom Crane

The controller design for boom crane is given in the figure below. As the boom is at an angle and we know that the lifting capacity of the vehicle is maximum when the boom is fully retracted and gradually decreases as the boom is extended. To cater for that problem we have devised such a system, that would keep the motion of the recovery operation at a constant velocity so we may not face the problem of acceleration and deceleration that is the main cause of sway initiation

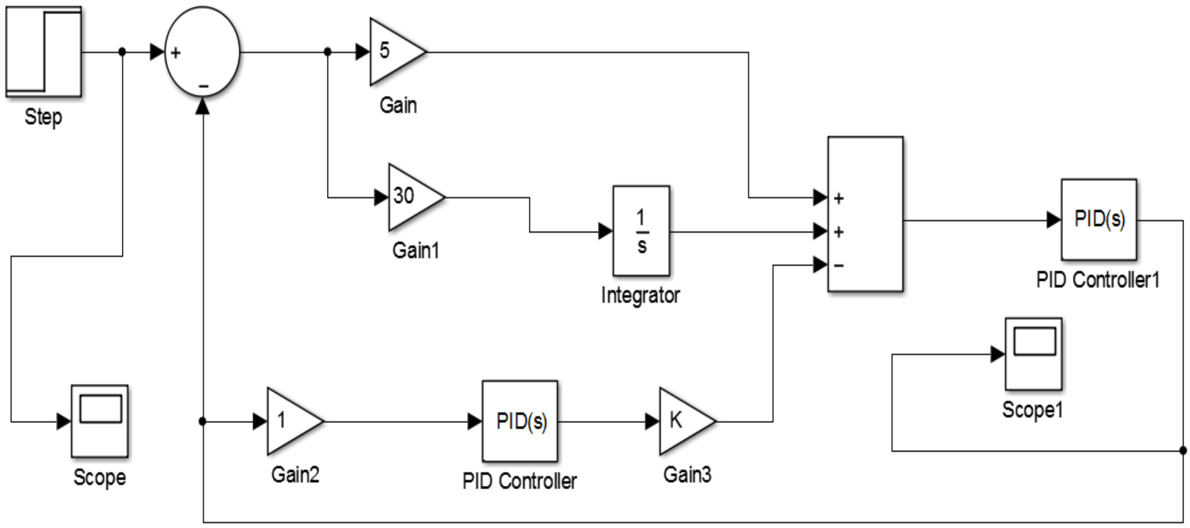


Figure 23 - Control Design for Boom Crane

It can be seen that as the casualty is hoisted, there comes a sudden change in the value of the potentiometer that becomes a cause of vibration initiation. So after the implementation of the control system, the disturbance in the value of the potentiometer falls, that helps in damping the initiation cause. The implementation of control system for the random sway that comes within the system is damped by using the PID controller system .However the performance of the proposed control scheme is little affected by the load mass, initial load swings, and slow hoisting motion.The payload position model is given in the figure as follows:-

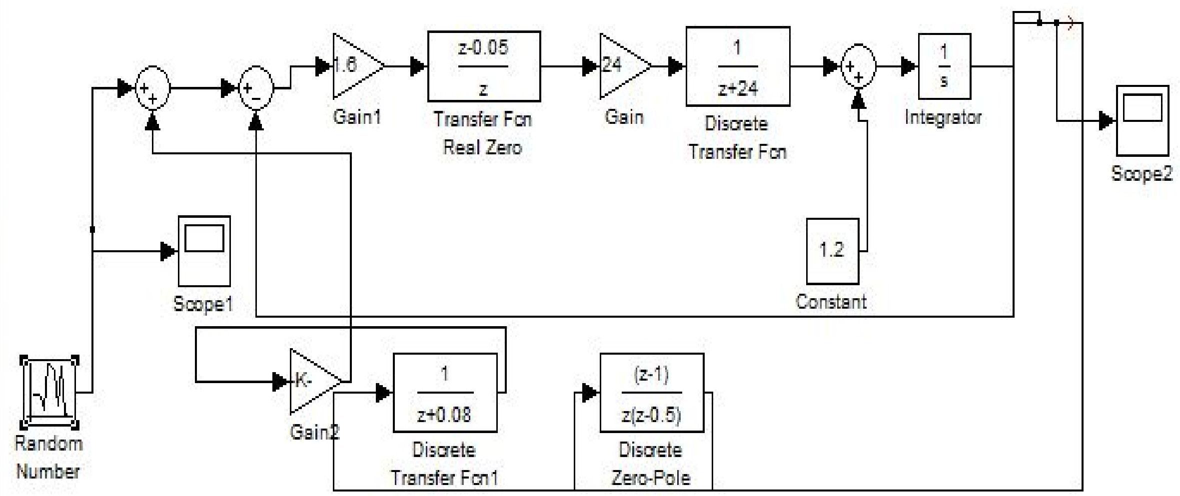


Figure 24 - Payload Position Control Model

The input of the system as shown in the figure that is predicting the sway in the system which is calculated instantaneously by the position sensor being placed with the system. The output of the system can be shown in the figure which is indicating sway control.it can be seen that the vibration in the system is nearly reduced to zero which proves an effective way of sway damping.

5.6 **LQR Controller**

The proposed control scheme based on classical PID controller can be compared with another control scheme based on optimized controller LQR. The control of a dynamic system like Recovery Boom crane model can be optimized with the help of a Linear Quadratic Regulator at a minimized cost. The optimized setting for the regulating LQR controller are found by using a mathematical algorithm which reduces the cost function by on the basis of provided weighing factors. The mathematical algorithm reduces the undesired variations from process parameters for example amplitude of oscillations in this case. The quantum of control action is also contained in this sum to for limiting the amount of energy it consumes. Although the LQR optimizes the control action, however, during the process, the weighing matrices have to be selected and the desired goals are then compared with the achieved results. In order to optimize the control scheme for the Recovery Boom Crane, there exists a need to select the control matrices Q and R. The following Q and R matrices will define the regulating matrices for the Recovery Boom Crane Model :-

$$Q = \begin{pmatrix} 2 & 0 \\ 0 & 1 \end{pmatrix}$$

$$R = 0.5$$

Thus the control matrix Q is a 2x2 symmetric positive semidefinite matrix and Q is 1x1 symmetric positive definite matrixes.

CHAPTER 6

SIMULATION RESULTS

6.1 General

In this chapter, the results of the change in payload oscillations after use of the proposed two control schemes will be discussed. The two control methods i.e., a classical PID Controller and LQR will be compared after their application on the Euro Cargo's recovery boom crane model will be discussed. The simulations were carried out on the SIMULINK model presented in the previous chapter and the results will be tabulated subsequently.

6.2 Simulation Results

The simulations in MTLAB / Simulink were carried out to compare the payload oscillations produced in the recovery crane in the three cases. The first case involved the current method of manual controlling the payload oscillations in the absence of any control scheme. The second case deals with the sway control or the control of payload oscillations in the Euro Cargo Recovery Crane after application of the proposed control scheme based on a classic PID controller. The third case covers the control of payload oscillations in the Euro Cargo Recovery Crane through LQR. The comparison of the trend of the oscillations in the hung mass will be carried out in this section and their results will be discussed for evaluating the proposed control scheme. The simulation results before the application of control scheme are presented in figure 25.

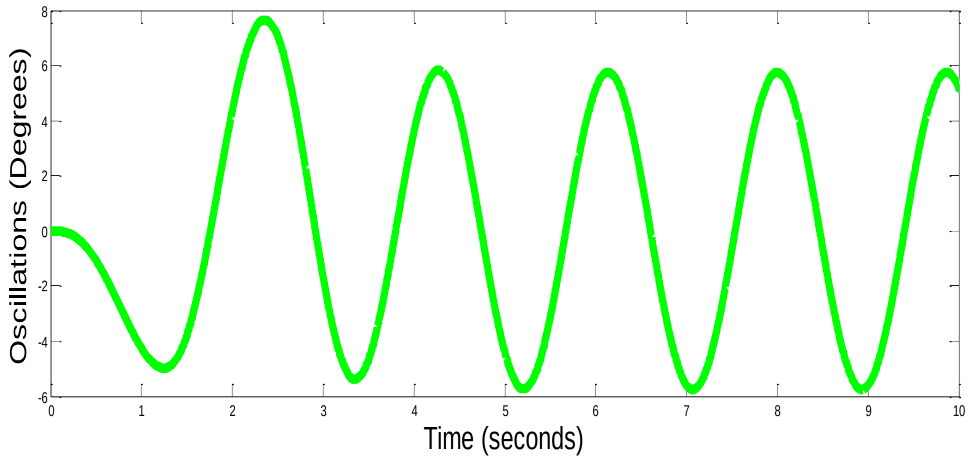


Figure 25 - Response of Payload Oscillations without Control Scheme

Whereas, the results of the pattern in payload oscillation after adoption of the suggested control arrangement are shown in figure 24 and figure 25. By comparing the results of the simulations it can be easily deduced that much improvement in the overshoot, undershoot and settling time is observed after application of proposed control scheme based on LQR. For instance, the change in payload oscillations after application of PID based controller is shown in figure 26 as follows:-

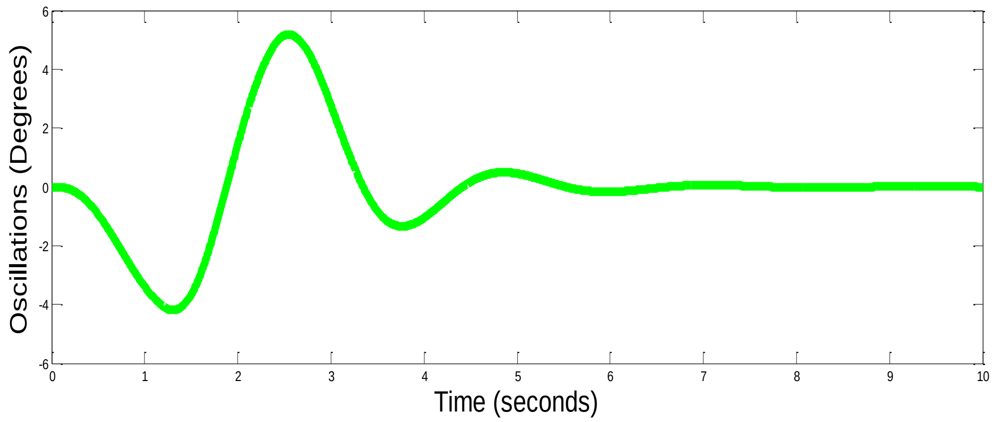


Figure 26 - Response of Payload Oscillations with PID Controller

The results of the simulations of the Simulink model after the application of LQR is shown in figure 27 as follows:-

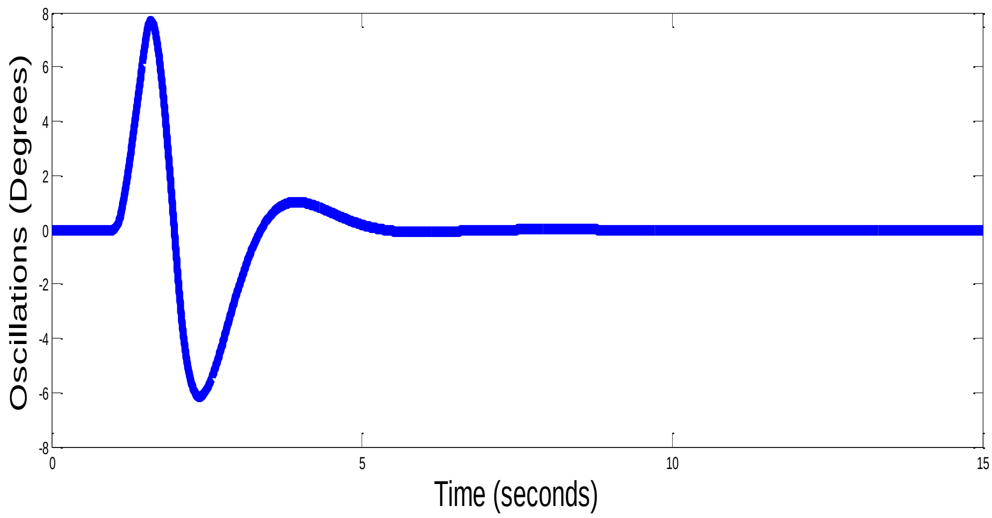


Figure 27 - Response of Payload Oscillations with LQR

By comparing the trend of curves in figure 25 with those in figure 26 and 27, it can be seen that the oscillations' amplitude is regulated after the application of the two control schemes. The amplitude of oscillations in case of manual control (i.e., in the absence of any control scheme) gives a sinusoidal plot over a finite interval of time.

However, in order to compare the improvements of these results gained through PID classical controller with that of LQR, it is essential that the results of the two controllers be plotted on the same curve.

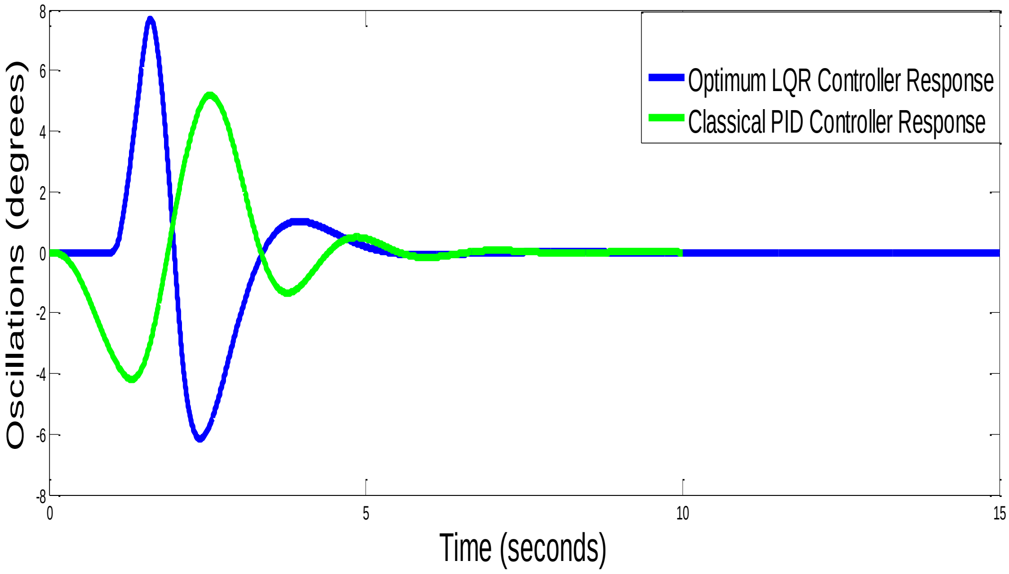


Figure 28 - Comparison of PID and LQR Results

The figure 27 shows the comparison of the two plots. The change in the amplitude in oscillations after the application of classical PID controller as shown in green whereas the blue curve shows the response of oscillations in recovery crane after LQR. It can be seen that the settling time of sway motion which was decreased to 7.35 seconds after application of PID has been further reduced to 5.1 seconds. Thus there is an overall improvement in the settling town after application of LQR. It can be further observed that in case of LQR, an initial increase in the peak amplitude is observed which then help to dampen the settling time to 5.1 seconds as in contrast to PID classical controller. The settling time in case of POD controller is 7.35 seconds. Therefore, it can be validated from the simulation results that the LQR controller provides the optima controller to dampen the sway motion in the payload of IVECO Euro Cargo Recovery Vehicle. .The evaluation of the performance of the two cases is also tabulated in the table 1. After the application of suggested control schemes in the Euro Cargo Recovery Vehicle, there is a marked improvement in the settling time results when PID controller is replaced by LQR . These results are tabulated as in table no 1 and table no 2. The table 1 shows the comparison of sway motion in the payload oscillations in terms of settling time, rise time and steady state error. The Settling time in terms of the payload's oscillations is the time passed from the application of an instant step input till the time when the output has reached and confined to a specific error band about the final value. Whereas, the rise time is required for the response of the recovery vehicle's payload to rise from 10% to 90% of its final value.

Ser	Control Technique	Settling Time	Rise Time	Steady State Error
1.	Without Controller (Open Loop System)	$\propto$	0.15 sec	0
2.	Control by PID	7.35 sec	0.65 sec	0
3.	LQR Controller	5.1	0.35 sec	0

Table 1 - Comparison of Performance Parameters

The response in terms of payload's amplitude of oscillation has also been tabulated in Table no 2:-

Ser	Control Technique	Amplitude Diversion (Degrees)	
		Min	Max
1.	Without Controller (Open Loop System)	-5.751	7.656
2.	Control by PID	-4.195	5.179
3.	LQR Controller	-5.9	7.8

Table 2 - Comparison of Payload Oscillations' Amplitude

6.3 **System Limitations and Risks**

There are following limitations in control of payload oscillations through the proposed control scheme which are as follows:-

a. There are certain forces like gravitational force, potential energy possessed by the payload and resistive forces which affect the operation of crane and these factors have not been considered while modeling our proposed control scheme.

b. For the control of payload oscillations effectively, ideally a levelled ground is required for the recovery crane vehicle which usually is not available during field recovery operations. The irregularity of a ground on which a recovery crane is being performed may affect the amplitude of payload oscillations and thus a suitable arrangement may have to be catered for while designing a control scheme for this raised or irregular terrain of ground.

c. It is supposed for ease of modeling that length of the cable hoisting the payload remains same. However, in actual scenario it varies during crane operation while the payload is being dropped on the ground after reaching the required destination point.

d. Tension in the hoist cable can vary with the repeated mechanical wear, if proper and regular maintenance is not being carried out. It will ultimately affect the results of the payload oscillations.

CHAPTER 7

CONCLUSION AND FUTURE RECOMMENDATIONS

7.1 Conclusion

During military operations involving the movement of large military convoys, whenever a vehicle gets bogged down, falls from a ridge or has some mechanical fault due to which the vehicle is unable to movement on its own, this is where recovery operation performed by recovery staff comes into play. Keeping in view the present Low Intensity Conflict (LIC) environment and present security situation on our western front of the operations, the recovery tasks should be performed swiftly to minimize the chance of being ambushed or raid by the enemy forces. Recovery operation should be thus time saving and to be performed at the earliest to improve the mobility of the convoy. If one of the vehicles moving in a convoy gets damaged and needs recovery, the whole convoy stops and gets into danger in the prevailing threat spectrum posed by the prevailing LIC environment. Sway caused in the recovery operation increases the recovery operation time. In-experienced recovery operator would take much time catering for the sway that would again be dangerous. Thus the application of the propose control scheme is useful for regulating the swing angel of Euro Cargo Recovery Vehicle. The suggested scheme resulted not only in reducing the settling time of the system but has done it without any overshoot and undershoots. It must be kept into consideration that the anti-sway control schemes for overhead, fixed and tower cranes have now matured enough over the passage of time due to their wider scope of application on industrial scale. Various intelligent control schemes have been applied successfully on these types of cranes for the benefit of these industrial processes. However, for mobile recovery vehicles equipped with hydraulic operated boom cranes operating in an open / unstructured field environment, a little research has been focused to find a suitable control scheme. Therefore, the focus of this research thesis has remained on a suitable control scheme based on PID which can be configured with the varying parameters of the operating an EFI based mobile recovery crane. Though, the results of the PID controller have been compared through LQR, nevertheless, it should be kept into consideration that the classical PID controller cannot sustain the online variations of the system due to disturbances and noises. Only the adaptive based or intelligent controller has the quality and capability of adjusting PID gains even online. These limitations can however, be addressed if the tuning of PID controller is done through various intelligent methods. The intelligent methods e.g. Fuzzy Logic, Adaptive Neuro Fuzzy Interference System (ANFIS) and Genetic Algorithm (GA) achieve best performance in terms of settling time, overshoot and undershoot as compared to conventional PID tuning methods. Therefore, these intelligent control schemes, may be undertaken in the future research work as mentioned in the following lines.

7.2 **Future Recommendations**

This research was the first effort of its kind aimed at designing a suitable control scheme for regulating the payload oscillations observed during the recovery operations performed mostly be our military recovery resources in addition to those executed by the civil recovery cranes. From the successful implementation of the control scheme in our case study, following recommendations can be deduced for the future research work on the topic:-

a. In this research work, a control scheme utilizing PID controller has been implemented. These results are compared by using an optimized LRQ controller for controlling the sway angle. However, these results can be further tuned by using intelligent controllers like Fuzzy logic controller (FLC), or hybrid controllers for instance; Neuro fuzzy or any other suitable combination.

b. In order to further fine tune the results in future research, the sophistication of applied sensors can be enhanced for calculating the instant rate of change of swing angle, change in weight of the payload, experience of crane operator to vary the speed and sway of the recovery crane operation accordingly. The sophistication of applied sensors will obviously be accompanied by the increase in the cost of the control scheme.

c. This research work took into consideration the effect of important field parameters like wind speed and drag coefficient countered during the recovery operations carried out during the open loop field environment. However, in future research work, the uncertainties and disturbances like noise can also be catered for to evaluate the effectiveness of the proposed control scheme.

d. In future work, the due consideration can be given to the dynamics factors of the system modeling for instance kinetic and potential energies etc. for evaluation the effectiveness of the control scheme on actual non-linear system.

e. Different control techniques like Neuro-fuzzy controller and H-infinity etc. can also be applied for comparing the results of this proposed control scheme results with these novel sophisticated controllers.

f. The hook mechanism of the boom can be further made more effective by using the Spreaders that would increase the weight of the lifting mechanism but would dampen the sway.

g. The control scheme may be designed in a way to monitor surrounding workspace for unsafe operating conditions. The system should stop the operation instantly as any unsafe condition detected.

h. The further research can be carried out to calculate optimum path of operation automatically as done in automated CNC machines.

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