

Design, construction, and control of a novel tower crane.

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Abstract— This paper presents a new concept of a tower crane, which greatly reduces the ballast and eliminates the anchoring, as will be “self-balancing” which implies removing the anchoring replaced by a sliding counterweight trolley. This paper presents the design of the mechanics as well as the final assembled self-balancing tower crane. Its dynamical model has been obtained and controlled by means of classical and anti-sway controllers. Simulations are conducted in order to show the effectiveness of the proposed tower crane.

Keywords— Crane Control, Mechanical Cranes, Modeling, PID Control.

I. INTRODUCTION

THE general definition of a crane is a mechanical system designed to lift and move loads through a hook suspended from a movable arm.

In the construction industry, where it is needed lift, it is often used cranes that are anchored or subject to ballast [1], [2]. In most designs, the moment is generally several times greater at the base than at the top of the crane, it is intended to have the center of gravity of the system located at the base, and so the center of mass obligates the system to remain in balance, regardless of the magnitude of the load, which is much minor to the total mass of the crane. In the cranes tower, the load lifted must be able to move, an action that is done by a trolley along the boom, reaching the counterweight in the counter boom a fixed distance regard to the tower. This principle of operation limits the weight can be lifted by making it dependent on the distance you want to move in the boom. This limitation is related to the balance generated by the counterweight, ballast and load. Fig. 1 shows the weights

that cranes tower can lift according to the length of displacement along the boom.

A new concept of tower crane is presented here, as will be self-balancing crane, anchoring is replaced by a sliding counterweight trolley in order to provides a counterbalancing moment exactly equal to boom moment in order to maintain the balance.

As mentioned in [2] at present, control cranes are moving from manual operation, which depended on the ability of an operator, to an automatic control, especially when they are very big and move loads at high speeds. Hence, new methods of automation are being developed. For example combinations of classic control laws with modern control laws [3], [4] and [5], also fuzzy logic controllers are used [6], [7], [8] and [9]. To control automated cranes can be seen several techniques:

The first technique is based on the generation of trajectories [10] and [11] for transporting load from one point to another with minimal sway of the load. That path will depend on the type of crane used, because for some types such as bridge cranes will be relatively easy over the tower crane. These techniques can be used optimal control.

The second technique is based on the feedback of the position and sway angle of the suspended load [10].

The third technique is based on divide the controller in two parts, control anti-sway and another control of the position of the load [10]. Each of them designed separately and combined to ensure the performance and stability of the system.

Although much research is concentrated on the generation of trajectories, since cranes are being fastened by any anchoring system or excessive ballast, in this work we use the second technique with the application of classical control laws to achieve a self-balancing and proper operation. The self-balancing tower crane mechanical elements have been constructed and assembled, its dynamical model is also obtained in order to verify the proper operation and to check optimal performance of the proposed crane by means of simulation using sway and anti-sway controllers.

II. DESIGN OF THE SELF-BALANCING TOWER CRANE

A tower crane as shown in Fig. 2 consists basically [1] by the elements named in figure. In this section a description of the proposed crane is presented.

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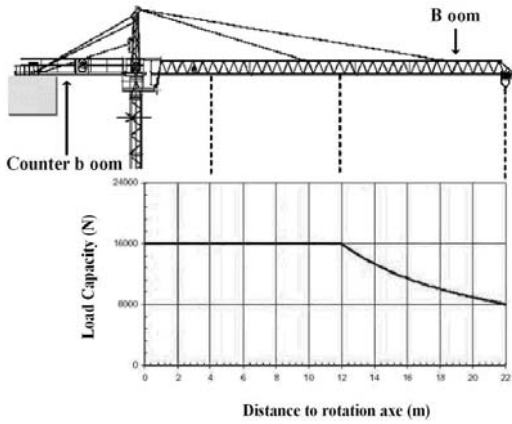


Fig. 1 Displacement respect to the load.

As the proposed crane will be self-balancing crane,

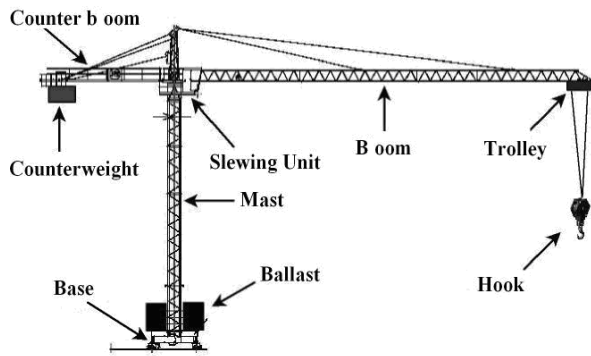


Fig. 2 Overview of a crane parts

anchoring is replaced by a sliding counterweight trolley in order to provide a counterbalancing moment exactly equal to boom moment in order to maintain the balance. The operation of the proposed crane is based on the balance of moments as shown in Fig. 3 and it is given by the relationship

$$F_1 \times D_1 = F_2 \times D_2 \quad (1)$$

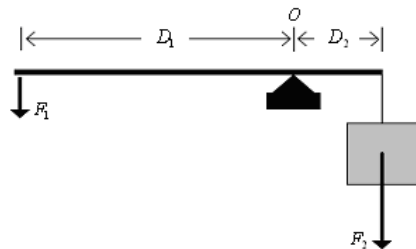


Fig. 3 Relationship moments in balance

The proposed crane in this paper reduces the ballast and eliminates the anchoring, since it will be "auto balancing". The concept of self-balancing implies eliminating anchoring system, which is replaced by a movable counterweight as shown in Fig. 4.

A. Mechanical Design of the proposed crane.

To achieve the mechanical design of the tower crane, initially technical specifications are required, both the mechanical and the electrical and electronic components, such as motors, sensors, circuit cards, etc. these general specifications are in

Table I.

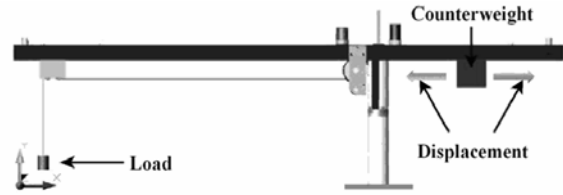


Fig. 4 General Outline of the crane self-balancing

TABLE I
GENERAL SPECIFICATIONS

Specification	Value
Type	Tower
Material	Aluminium and iron
Length of the boom:	160 cm
Length of counter boom	60 cm
The tower height	45 cm
Maximum height of load displacement	130 cm
Maximum weight of the load	1000 g
Speed lifting of the load:	5 cm / sec
Angle maximum load balancing	3 °
Rotation angle of the tower	360 °

Specifying the dimensions desired, the experimental tower crane designed is shown in Fig. 5, being a good start to apply control laws classical and modern.

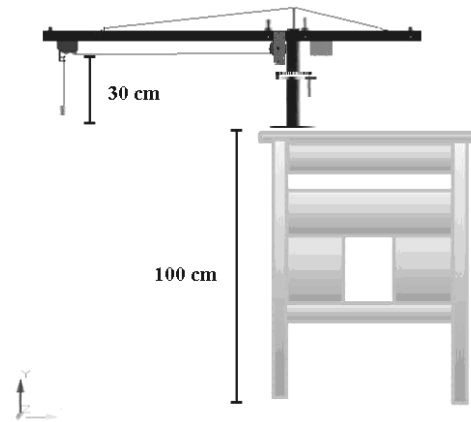


Fig. 5 Experimental crane tower on a table

One of the technical specifications indicates that the tower height is 45 cm, choosing this length so that the crane is not very high to increase the stability, it will also be on a structure as the building, which in our case is a table, which has an height of 100 cm as it is shown in Fig. 5. This crane does not have mechanical system of displacement. Another specification is that the maximum height of the vertical displacement of the load (m_l) should be 130 cm, noting that in Fig 5, it is the sum of the height of the table plus 30 cm in lifting crane, this load is going to be at least 15 cm below the highest point of the boom.

A few pieces for the design of the novel crane will be described below.

The mast and slewing mechanism.

The mast consists of a group of parts, so that half is fixed to the base, and the other half can be rotated. The parts are displayed in Fig. 6.

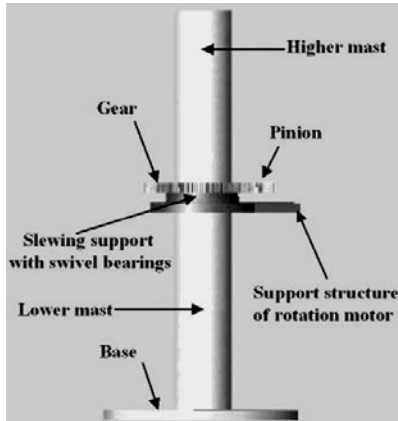


Fig. 6 Mast and base

The reason they are two sections with almost the same length, it is because at half mast are the mechanical elements of rotation as shown in Fig. 7, gear, (small gear) pinion and motor, which by its dimensions occupy considerable space, and if these traction elements are placed near the junction of the mast and the boom, there will not be enough space for the hoisting elements and mass displacement m_1 , as well as the displacement of counterweight (mass m_2).



Fig. 7 Lower mast and motor stand

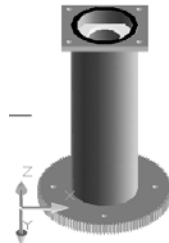


Fig. 8 Gear assembled at the upper mast

The second part of the mast is joined to gear, forming one piece. As shown in Fig. 9, as well as the basis, the gear has a slot where the upper mast is assembled.

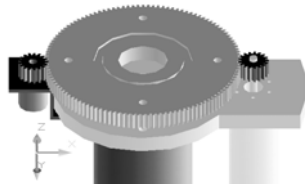


Fig. 9 Gear and pinion in detail

An important element to give rigidity to the mast is a screw 2.54 cm ($\frac{1}{2}$ in) in diameter and 40 cm long, which is placed in the middle and along the mast coupled with bearings that are shown in Fig.10 and they are assembled by pressure on the internal supports of the mast as shown in Fig. 11, in the slewing gear and rotation motor support, preventing torsion forces affect the slewing support.

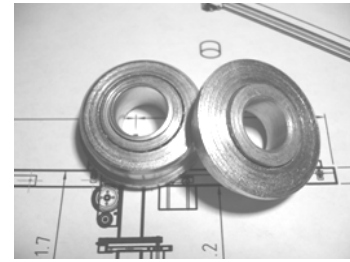


Fig. 10 Bearings for vertical support

Boom and counterboom.

In the commercial crane, the boom, counter boom and mast are built with tubular steel bars and / or angular, forming a lightweight structure and strong at the same time. In the case of the experimental crane, we decided to use aluminium bar, because it avoids manufacture every component of the boom and counter boom, satisfying with the characteristics of weight and resistance to be desired. As shown in Fig. 12, taking advantage of the length of aluminium bar, the boom and the counter boom are the in the same bar.

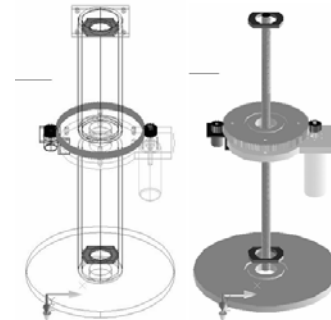


Fig. 11 Internal supports of the mast and internal screw

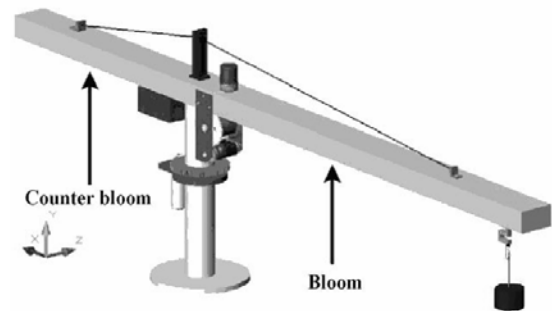


Fig. 12 Boom and counterboom of the crane

Trolleys for the movement of the load m_1 and counterweight m_2 .

The trolley shown in Fig. 13, consists of a chassis or metal base and four wheels which are for moving into two rails. The rails that belong to the boom are longer than those of the counter boom. The trolley will be moved by means of a steel cable and a pair of sheaves, one of which is coupled to the displacement motor and the other is coupled to the sensor (encoder) for measuring the distance travelled in one direction or another.

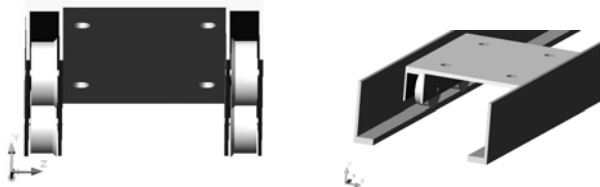


Fig. 13 a) Bottom of the car, b) A car on rails

System of horizontal displacement of the masses m_1 and m_2 .

The displacement system of mass m_1 is constituted by the pieces displayed in Fig.14.

Likewise, the displacement system of mass m_2 consists of the pieces displayed in Fig. 15.

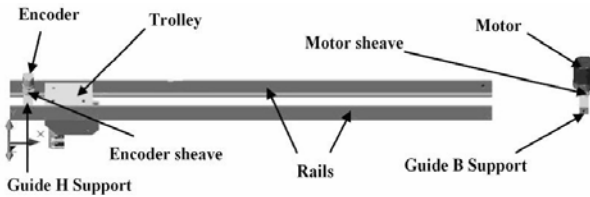


Fig. 14 Displacement System of mass m_1 .

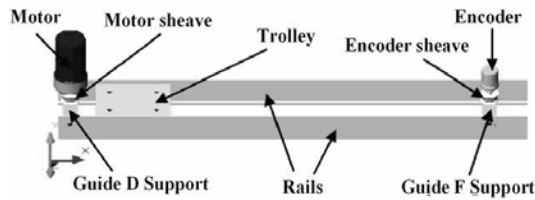


Fig. 15 Displacement System of the mass m_2

The hoisting mechanism and measurement of load weight (mass m_1)

The hoisting mechanism of mass m_1 consists of the following elements showing in Figs. 16 and 17.

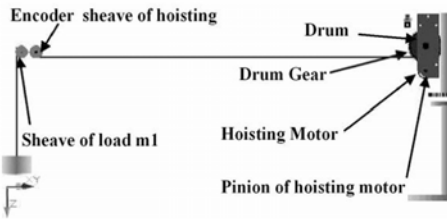


Fig. 16 The hoisting mechanism of mass m_1

Elements of measurement of load weight are formed basically by: load cell, load sheave of mass m_1 and sheave support of load. The load sheave is fixed to load cell through a support, also the load cell is fixed to trolley as shown in Fig.18.

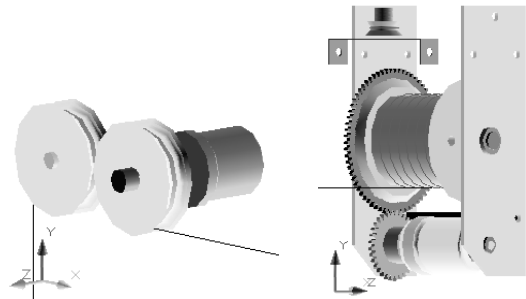


Fig. 17 View of hoisting system of the mass m_1

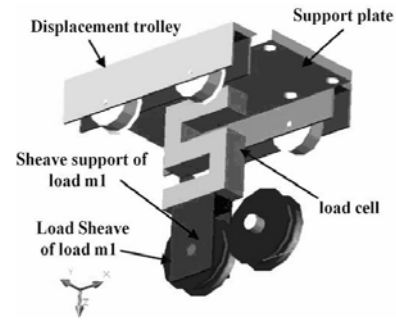


Fig. 18 System of weight measurement

Tower Crane

The crane assembled is shown in Figs. 19 and 20

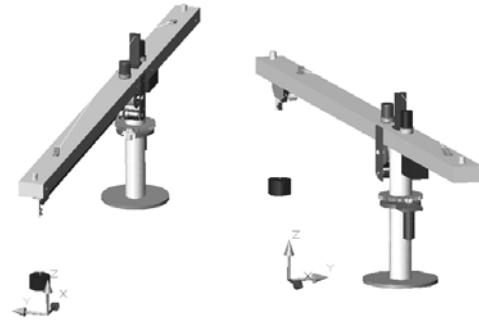


Fig. 19 Front and back of the crane

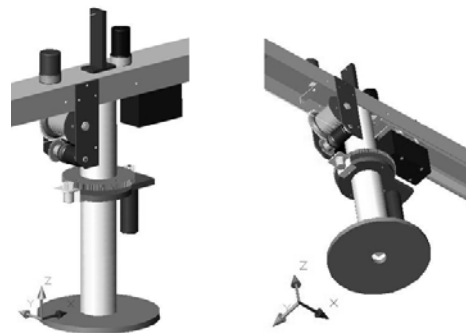


Fig. 20 Mast and additional elements

Finally, in this section the construction of the novel tower crane is presented. Fig. 21 shows the mast, Fig. 22 sliding trolley, Fig. 23 the trolley into the rail, and Fig. 24 shows the whole system of the self-balancing tower crane.



Fig. 21 The mast



Fig. 22 The sliding trolley

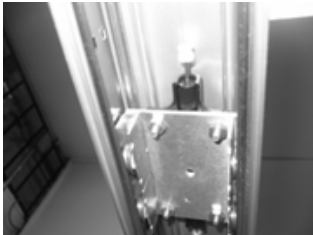


Fig. 23 The sliding trolley in rails

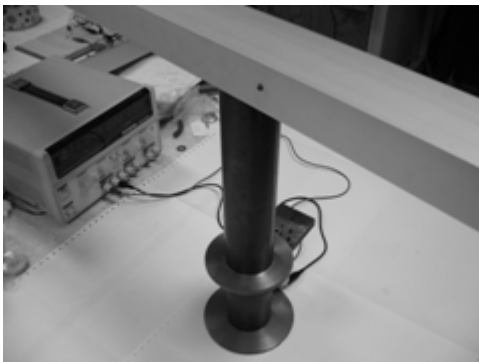


Fig. 22 The whole system of the self-balancing tower crane

III. MODEL OF THE PROPOSED TOWER CRANE

In this section is considered the lifted load (mass m_1) and the horizontal displacement of the counterweight (mass m_2) to maintain balance. The operation of the proposed crane is based on the balance of moments given by the relationship as in (1).

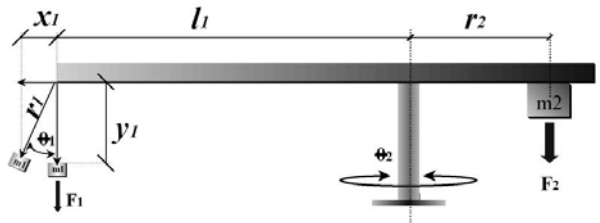


Fig. 23 Balancing moments in the crane, when the distance of the counterweight is adjusted.

The Lagrangian is given by:

$$L = K_T - V_T \quad (2)$$

Considering linear movements, from Fig. 23 the following equations are obtained:

$$\begin{aligned} x_1 &= -r_1 \sin \theta_1 \\ y_1 &= r_1 \cos \theta_1 \\ x_2 &= r_2 \\ y_2 &= h_2 = 0 \end{aligned} \quad (3)$$

Therefore Euler Lagrange equation for r_1 is given by

$$\frac{\partial}{\partial t} \left(\frac{\partial L}{\partial \dot{r}_1} \right) - \left(\frac{\partial L}{\partial r_1} \right) = \tau_{r_1} \quad (4)$$

The model is obtained as:

$$\begin{bmatrix} \frac{r_2 m_2}{l_1 \cos \theta_1} & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & J_1 + \frac{r_2 m_2 r_1^2}{l_1 \cos \theta_1} \end{bmatrix} \begin{bmatrix} \ddot{r}_1 \\ \ddot{r}_2 \\ \ddot{\theta}_1 \end{bmatrix} + \begin{bmatrix} 0 & 0 & -\frac{r_1 r_2 m_2 \dot{\theta}_1}{l_1 \cos \theta_1} \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} \dot{r}_1 \\ \dot{r}_2 \\ \dot{\theta}_1 \end{bmatrix} + \begin{bmatrix} -\frac{r_2 m_2 g}{l} \\ 0 \\ \frac{r_1 r_2 m_2 g \tan \theta_1}{l} \end{bmatrix} = \begin{bmatrix} \tau_{r_1} \\ \tau_{r_2} \\ \tau_{\theta_1} \end{bmatrix} \quad (5)$$

IV. ANALYSIS AND SIMULATION RESULTS WITH THE CONTROL LAWS

The control laws that are applied to the proposed crane are: P without anti-sway control with velocity feedback, PID without anti-sway control, and finally PID with anti-sway control. An oscillation angle $= 3^\circ = 0.054$ rad is considered.

The parameters controller were chosen to obtain an overdamped response, these parameters are in the Table II. It is important to mention that parameters provide a better response because avoid overshoots in the movement response of the two masses m_1 and m_2 . These overshoots can generate considerable forces due to inertia of the masses, causing an imbalance in the moments and therefore the inevitable collapse of the crane.

A. P without anti-sway control.

The P control law with velocity feedback is given by the following equation

$$\tau = K_p e - K_v \dot{q},$$

where $K_p, K_v \in R^{n \times n}$ are symmetric matrixes defined positive.

In Fig. 24 it can be seen that by lifting the load from an initial value of 0.8 m to 0.4 m a very small permanent error is

presented, this error is a characteristic of proportional control, it can be reduced by increasing gain K_p , in order to avoid overshoot it is necessary to increase the constant K_v . In the same way, the displacement of the mass m_2 shown in Fig. 25, the parameters are chosen to provide a over damped response, in order to keeping the balance of the crane. The load oscillation is shown in Fig. 26, with an oscillation angle equal 3° .

TABLE II
PARAMETERS VALUES OF THE P CONTOLLER

Symbol	Quantity	P Controller with sway	PID Controller with sway	PID Controller with anti- sway
K_{pr1}	proportional gain of r_1	1000	3000	1500
K_{vr1}	derivative gain of r_1	75	200	100
K_{ir1}	integral gain of r_1	0	100	50
K_{pr2}	proportional gain of r_2	750	1000	750
K_{vr2}	derivative gain of r_2	100	100	120
K_{ir2}	integral gain of r_2	0	95	30
$K_{p\theta_1}$	proportional gain of θ_1	0	0	350
$K_{v\theta_1}$	derivative gain of θ_1	0	0	15
$K_{i\theta_1}$	integral gain of θ_1	0	0	3
m_1	load mass	1 Kg	1 Kg	1 Kg
m_2	counterweight mass	4 Kg	4 Kg	4 Kg
r_1	Cable length of the load	0.4 m	0.2 m	0.2 m

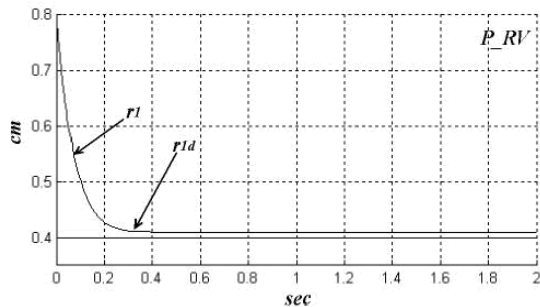


Fig. 24 Response of r_1

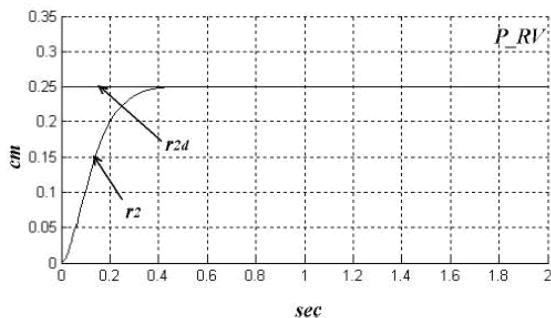


Fig. 25 Position of the counterweight (mass m_2)

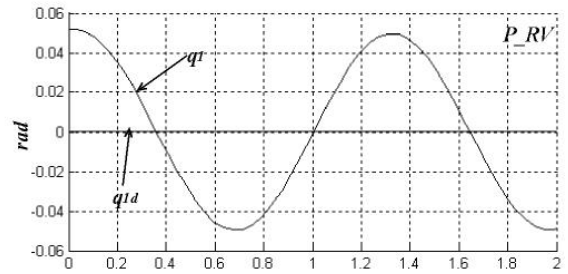


Fig. 26 Oscillation angle of the lifted load (angle q_1)

B. PID without anti-sway control.

The PID control law is given by the following equation

$$\tau = K_p e + K_v \dot{e} + K_i \int_0^t e(t) dt,$$

where $K_p, K_v \in R^{n \times n}$ are symmetric matrixes defined positive.

In Fig. 27 it can be seen the lift of the load from an initial value of 0.8 m to 0.2 m It has a better response that P controller. In Fig. 28, it can be seen as r_{2d} has an oscillation due to the compensation being made by the control law, by the forces generated due to the oscillation of mass m_1 ; r_2 tries to follow r_{2d} . The oscillation of the mass m_1 is shown in Fig. 29, and the oscillation is not attenuated.

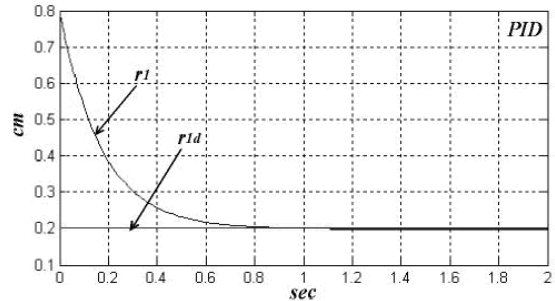


Fig. 27 Response of r_1

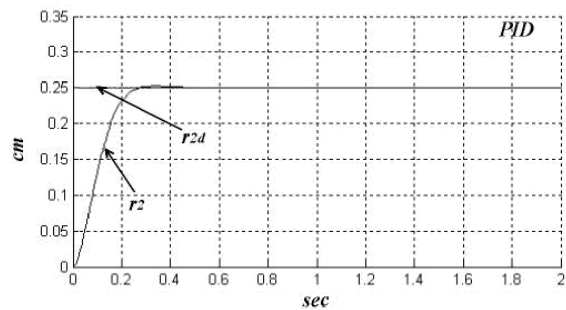


Fig. 28 Position of the counterweight (mass m_2)

C. PID with anti-sway control.

The control anti-sway of the suspended load of a tower crane is very important [12], [7], [3] and [13], increases safety, simplifies its operation, and increases the speed of load positioning, thereby saving time and money, however, excessive oscillation in a tower crane, can be a cause of the imbalance and hence the collapse of the system.

The responses obtained by applying PID control are shown in Figs. 30 and 31, which avoid overshoots. In Fig. 32 an oscillation damped is shown because an anti-sway control law is applied in order to get angle $\theta_1 = 0$. This is achieved by moving the trolley with m_1 along the boom, being this third degree of freedom. In the simulation with oscillation of m_1 the trolley remains fixed, and therefore there is no control over the angle θ_1 .

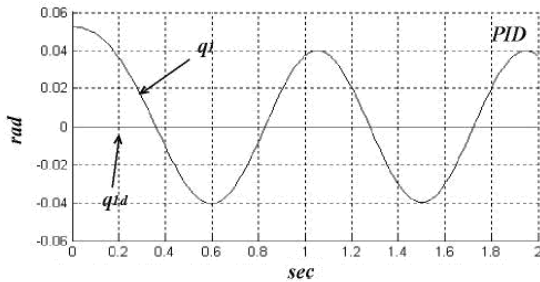


Fig. 29 Oscillation angle of the lifted load (angle q_1)

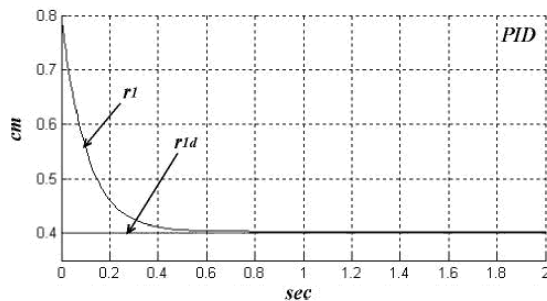


Fig. 30 Response of r_1

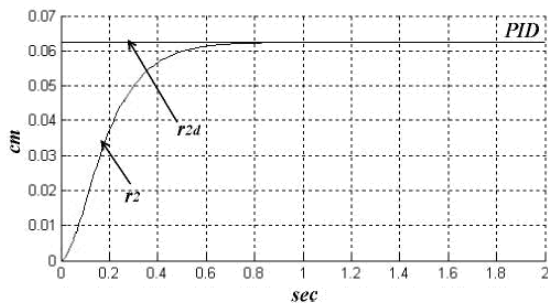


Fig. 31 Position of the counterweight (mass m_2)

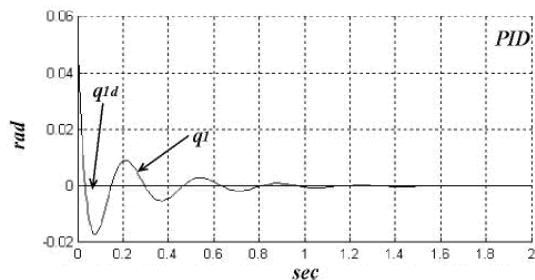


Fig. 32 Oscillation angle of the lifted load (angle q_1)

V. CONCLUSIONS

The dynamic model used and is described by (5), which

consists of three equations, where each one corresponds to each degree of freedom of the crane. This means that if $\tau_{\theta_1} = 0$, there is no control over the angle θ_1 , then only have control over two degrees of freedom, r_1 and r_2 , and as a consequence a free oscillation in the load is expected.

If τ_{θ_1} is a control law, then we have a third articulation that will compensate the oscillation of the load m_1 , which corresponds to the displacement of the trolley that moves the load along the boom.

Therefore the dynamic model founded is very representative of the proposed tower crane on different forms of control, which was checked with the simulation.

In the simulation of each control law and with different values of the parameters, it is found that the dynamic model is correct and covers the most important features of a mechanical crane. The manufactured tower crane is presented following the general specification required, the future work it will be the validation of the model with the real process.

The future work is the development of an anti-sway controller consists of some steps: modelling the dynamic crane behaviour, selection of an appropriate control structure, tuning of the control parameters and validation based on the developed crane model, implementation of the controller on the proposed self-balancing tower crane, fine tuning and experimental validation of the performance.

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