Human Assisted Impedance Control of Overhead Cranes

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Abstract— *This paper presents an impedance control scheme through which a human operator can move an overhead crane in the x-y plane without applying excessive force. The human pushing force is not measured directly, but it is estimated from angle and position measurements. We show that this method is an effective way to provide intuitive control for overhead cranes.*

Keywords— **Crane, impedance control, human assisted control, anti-swing control**

I. Introduction

Traditionally, cranes with capacities below 1000 lbs. have been either manually operated or powered via motors and commanded through a pendant. These types of cranes, while inexpensive, provide limited performance in terms of speed and operator force (for manual cranes), or they suffer from a lack of precision and intuition (for pendant-driven cranes).

A pendant-free design concept is the main focus of this work, and involves achieving active compliance (assisted impedance) on the x, y axes of an overhead crane, e.g. the operator can push and pull the load and the crane follows.

The control of an overhead crane is a very interesting problem, due to the underactuated nature of the system.

Problems with the load swing have been reported as significant factors in limiting the maximum allowable velocities for overhead cranes. There has been significant research addressing this problem [1] [2] [3] [4], i.e. the main objective is to suppress the oscillations of the load, while the crane is accelerating or deaccelerating.

In contrast, in this paper we develop a control strategy using the estimation of the force applied by the operator to the load, assisted by a variable desired load impedance.

Previous research in the area has usually dealt with either 2D cranes (one axis of motion), or with two decoupled axes. A conventional overhead crane contains a bridge, actuated on both ends, and a trolley. Coupling effects between the bridge and the trolley are nonlinear in nature, as shown in [5], but they dissappear when the system is linearized around the stable equilibrium. In [13], a feedback linearization technique is introduced, and tracking of C^4 smooth reference trajectories are used to minimize oscillations. Passivity is used in [6] to minimize the Time-Absolute Error subject to actuator constraints. In [10], an observer based disturbance control and feedforward control are considered for a crane with variable cable length (z-axis), and Coulomb friction. Model Reference Adaptive Control is used in [11] on the linearized model of a crane. Pure non-linear controllers have also been looked at in [3], [12]. Finally, input shaping was used to stabilize both a single or a double pendulum crane in [7], [8].

In contrast with the previous work, we consider the swing suppression problem as secondary in nature to controlling the impedance felt by the operator pushing on the hoisted load. The concept of mechanical impedance is frequently used in force control and teleoperation applications [14], [15]. In this paper we present a linear control strategy which places the human operator in the outer control loop via an impedance block used for trajectory generation.

The paper is organized as follows. Section 2 contains the control problem statement; Section 3 presents the crane model and the modeling assumptions; Section 4 describes the linear controller-observer design; Section 5 presents numerical simulation results; Section 6 the shows experimental results, and, finally in Section 7 includes concluding remarks.

II. Problem statement

The control problem to be considered here is the motion of a 3D overhead crane depending solely on the force being applied to the load by the operator. To achieve this, we use a parametric linearized model for the crane, obtained through identification. We then use a model-based estimator for the operator pushing force. The need for estimation as opposed to direct measurement of the force is due to the desire to avoid installing any cables impending the movement of the operator within the crane workspace.

The observer model is linear, but it also includes a Coulomb friction term. The pushing force estimation is used to produce a desired trajectory for the crane, based on a desired mechanical impedance for the load. In effect this is the reference signal for a tracking controller.

There are some important challenges related to this control problem:

- The operator is inside the workspace of the crane, which stresses the need for stability of the system in all situations, e.g. in the presence of noise, disturbances, etc.

• There are robustness and performance issues because of using an imperfect model, because of measurement noise, because of external disturbances (such as friction), and because we assume that the hoisting cable can have different lengths, and the loads can have variable shapes (i.e., they are not point masses)

• The human operator pushing force is not considered a disturbance, but a generator of reference signals.

In the actual experiments we conducted, a lot of measurement errors were dealt with via the use of deadzones. At the same time, even though the nominal model used in the controller design was linear, all simulations were conducted using the actual nonlinear system dynamics.

III. System Model

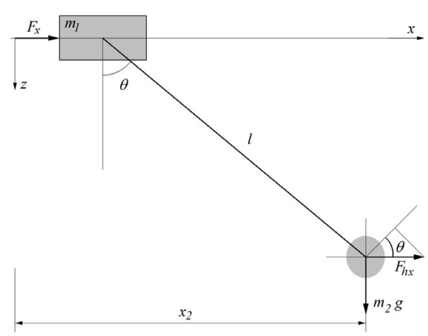

Fig. 1. Simple pendulum with a mass at the point of support that can move along the horizontal plane.

As a reasonable simplification to the original problem, we consider each axis of motion to be decoupled, i.e. the motion of the x and y axes are independent. Each axis can then be modeled as a simple pendulum with a point of support that changes its position along the specified axis.

The system on each axis is shown in Fig 1, and contains a load (m_2) attached through a rigid cable to a trolley/bridge mass (m_1) that can move along the horizontal axis. The nonlinear model for the x axis subsystem is given by:

$$M(q)\ddot{q} + C(q,\dot{q})\dot{q} + G(q) + F_r(\dot{q}) = \tau \tag{1}$$

where:

$$M(q) = \begin{bmatrix} (m_1+m_2) & m_2 l\cos(\theta) \\ m_2 l\cos(\theta) & m_2 l^2 \end{bmatrix}$$

$$C(q,\dot{q}) = \begin{bmatrix} 0 & -m_2 l\sin(\theta)\dot{\theta} \\ 0 & 0 \end{bmatrix}$$

$$G(q) = \begin{bmatrix} 0 \\ m_2 g l\sin(\theta) \end{bmatrix}$$

$$F_r(\dot{q}) = \begin{bmatrix} b_1\mathrm{sgn}(\dot{x}) + b_2\dot{x} \\ b_\theta\dot{\theta} \end{bmatrix}$$

$$\tau = \begin{bmatrix} F_x + F_{hx} \\ lF_{hx}\cos(\theta) \end{bmatrix}$$

$$q = \begin{bmatrix} x \\ \theta \end{bmatrix}$$

where l is the cable length, θ is the angle of the cable, b_2 is the viscous damping along the x axis, b_1 is the static friction along the x axis, b_θ denotes the viscous joint damping, F_x is the force applied to the cart (control signal) and F_{hx} is the force applied to the load by the human operator.

Expressing (1) in the form $\dot{X} = f(X,u)$, with $X = [x\,\theta\,\dot{x}\,\dot{\theta}]^T$ we have that:

$$\dot{X} = \begin{bmatrix} \dot{x} \\ \dot{\theta} \\ M^{-1}(q)\left(\mathcal{U}u - C(q,\dot{q})\dot{q} - g(q) - F_r(\dot{q})\right) \end{bmatrix} \tag{2}$$

where $\mathcal{U} = \begin{bmatrix} 1 & 1 \\ 0 & l\cos(\theta) \end{bmatrix}$ and $u = [F_x\; F_{hx}]^T$ so

$$\begin{aligned} \ddot{x} &= \eta m_2 l(l(F + F_h - b_1\mathrm{sgn}(\dot{x}) - b_2\dot{x} + \\ &\quad -F_h\cos(\theta)^2) + m_2 l^2\dot{\theta}^2\sin(\theta) + b_\theta\dot{\theta}\cos(\theta) + \\ &\quad +m_2 g l\cos(\theta)\sin(\theta)) \end{aligned}$$

$$\begin{aligned} \ddot{\theta} &= \eta(m_2 l(-(F - b_1\mathrm{sgn}(\dot{x}) - b_2\dot{x})\cos(\theta) + \\ &\quad -m_2 l\dot{\theta}^2\cos(\theta)\sin(\theta) - (m_1+m_2)g\sin(\theta)) + \\ &\quad +m_1 lF_h\cos(\theta) - (m_1+m_2)b_\theta\dot{\theta}) \end{aligned}$$

where $\eta = \frac{1}{m_2 l^2(m_1 + m_2\sin^2(\theta))}$.

Linearizing the equation (2) around $X^\star = [x,\,0,\,0,\,0]^T$ we obtain:

$$\dot{X} = AX + Bu = AX + [B_1|B_2]\,u \tag{3}$$

where:

$$A = \left[\begin{array}{cccc} \multicolumn{2}{c}{0_{2\times2}} & \multicolumn{2}{c}{I_2} \\ \hdashline 0 & \frac{m_2 g}{m_1} & -\frac{b_2}{m_1} & \frac{b_\theta}{m_1 l} \\ 0 & -\frac{(m_1+m_2)g}{m_1 l} & \frac{b_2}{m_1 l} & -\frac{(m_1+m_2)b_\theta}{m_1 m_2 l^2} \end{array}\right]$$

$$B = \left[\begin{array}{cc} \multicolumn{2}{c}{0_{2\times2}} \\ \hdashline \frac{1}{m_1} & 0 \\ -\frac{1}{m_1 l} & \frac{1}{m_2 l} \end{array}\right]$$

The measured states are the cable angle θ and the cart position x, therefore the output of the system is given by $Y = CX$,

$$C = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \end{bmatrix} \tag{4}$$

A simple rank check shows that this nominal control system is both controllable and observable.

IV. Controller description

In this implementation, each axis of movement is controlled independently, so we use two controllers with the same structure but with different parameters and settings. As a simplification, we will only reference the controller of the x axis in the understanding that all the descriptions also apply to the y axis controller.

The control input to the system is the force F_x and we assume that the force F_{hx} is not available through a direct measurement and that the outputs of the system are the position of the cart (mass m_1) and the cable angle, i.e. x and θ. The overall controller is shown in Figure 2.

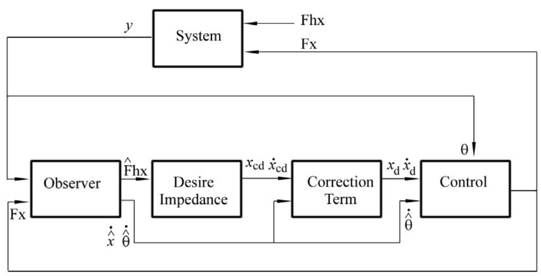

Fig. 2. Controller structure

A. Observer block

As can be seen in the Figure 2, a linear observer is used to obtain an estimate of the force F_{hx}. The dynamic equations of this observer are given by:

$$\dot{\hat{X}} = A_e\hat{X} + B_eF_x + LC_e(y - \hat{y}) \ \ ; y = [x\ ,\ \theta]^T \tag{5}$$

where: $\hat{X} = [\hat{x}, \hat{\theta}, \dot{\hat{x}}, \dot{\hat{\theta}}, \hat{F}_{hx}]^T$

$$A_e = \left[\begin{array}{ccc} A & | & B_2 \\ -- & + & -- \\ 0 & | & 0 \end{array}\right]; \quad B_e = B_1$$

$$C_e = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \end{bmatrix}$$

This system is also controllable and observable.

$$F_x = \begin{cases} F_x - F_{combx}; & |F_{combx}| < b_{1s} \text{ and } |\dot{\hat{x}}| < \epsilon \\ \\ F_x - b_1\, sgn(\dot{\hat{x}}); & \text{otherwise} \end{cases} \tag{6}$$

where:

$$F_{combx} = F_x - b_2\dot{\hat{x}} + \frac{b_\theta}{l}\dot{\hat{\theta}} + m_2 g\theta \tag{7}$$

b_{1s} is the stiction on the x axis and $\epsilon > 0$. Equations (6) and (7) describe the static friction compensation for the observer, taking into account two cases: first the static case when the cart is at rest and the observer is that of a simple pendulum, and the case when the cart is moving and the static friction is just subtracted from the control input F_x.

In addition to the pushing force estimate, the observer also generates filtered values for the cart position, velocity, cable angle and angular velocity.

B. Desired impedance block

We use the estimated operator force to generate the desired position of the load by passing it through a desired impedance block:

$$M_d\ddot{x}_{cd} + B_d\dot{x}_{cd} = \hat{F}_{hx} \tag{8}$$

where M_d is the desired mass, B_d is the desired damping and x_{cd} is the desired position of the load. Through the impedance block we can specify a particular performance for the motion of the load. At the same time, the "feel" of the load for the operator can be changed from very light with almost no damping, to heavy and viscous.

Since we don't have direct control on the position of the load, but on the position of the cart, we use a correction block to calculate the term x_{cd} and $\dot{x}_{cd}$ by:

$$x_d = x_{cd} + l\sin\theta \tag{9}$$
$$\dot{x}_d = \dot{x}_{cd} + \dot{\theta}l\cos(\theta) \tag{10}$$

where x_d is the desire position of the cart.

C. Controller block

The particular controller we employed is a simple pole-placement controller, which is used to track the reference trajectory $X_d = [x_d, 0, \dot{x}_d, 0]^T$. Therefore, anti-swing is achieved with a desired load impedance, if

$$F_x = K_1(x_d - x) + K_2(\dot{x}_d - \dot{\hat{x}}) + K_3\theta + K_4\dot{\hat{\theta}} \tag{11}$$

where K_i, $i = 1, 2, 3, 4$ are given by specific locations of the system poles.

V. Simulations results

Using Matlab/Simulink we simulated the control system (x-axis subsystem) with the following parameters:

$m_1 = 700$ Kg	$m_2 = 52.2$ Kg
$l = 3.352$ m	$b_\theta = 20$ N ·s
$b_1 = 280$ N	$b_{1s} = 300$ N
$b_2 = 1500$ N·s/m	$M_d = 1.162$ Kg
$B_d = 23.248$ N·s/m	

with the controller gains set as

$$K_x = \begin{bmatrix} 3089.1503 \\ 5032.0363 \\ -32260.15033 \\ 4040.5861 \end{bmatrix}$$

and a pulse of 4 N is the input pushing force. The results are shown in Figures 3 and 4.

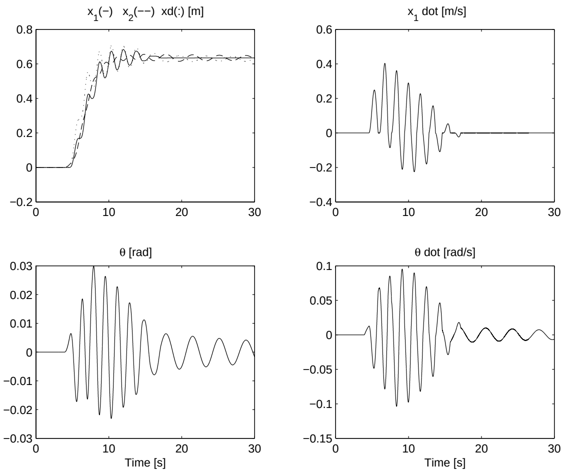

Fig. 3. States of the system

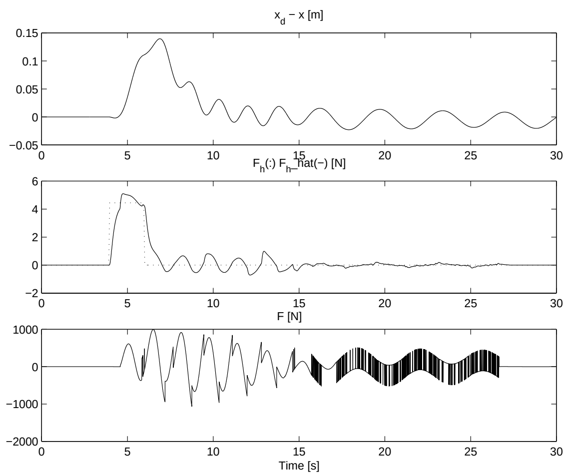

Fig. 4. Control signal

Figure 3 shows the system states, and it can be seen that the input force causes a step in the position of the cart. The shape of the reference trajectory x_d (dotted line in the plot) can be affected by adjusting M_d and B_d in the desired impedance block and in doing so the maximum velocity of the cart can be modified without retuning the and controller/observer gains.

Figure 4 shows the estimated force F_{hx} as well the position error for the cart and the control signal F_x. The errors in the observed force F_{hx} are due to the effects of the static friction along the x axis but they are small enough to be corrected by a deadzone around zero. All the simulation results are for a constant longitude of the wire (l) and a constant load.

VI. Experimental results

In the actual experimental implementation we have to deal with the uncertainties in the parameters of the system, the variation of the friction along the runways, the change of length of the cable, inaccuracies in the measurement of the angle, etc. All these differences between the model of the observer and the real system generate a non-zero observer force $\hat{F}_{hx}$ that can drive the crane in the absence of a pushing force. To fix this problem we used deadzones for some signals such as:

- The angle of the wire, θ.
- The estimated force applied to the load $\hat{F}_{hx}$.
- The control signal F_x.

The threshold for these deadzones are also a function of the angular velocity, such that there is a larger deadzone band when the load is swinging without any force applied to it and a lower value when the load is stationary and the operator is applying a force to it.

The parameters for the experimental setup (x axis) are very similar to the ones used in simulation except for the gains of the controller:

$$K_x = \begin{bmatrix} 9678.1489 \\ 15911.5604 \\ -68843.0295 \\ 35348.3095 \end{bmatrix}$$

For the y axis the parameters of the system were:

$$\begin{array}{ll} m_1 = 265 \text{ Kg} & m_2 = 52.2 \text{ kg} \\ l \;\; = 3.352 \text{ m} & b_\theta \;\; = 20 \text{ N} \cdot\text{s} \\ b_1 \;\; = 356 \text{ N} & b_{1s} = 360 \text{ N} \\ b_2 \;\; = 1600 \text{ N}\cdot\text{s/m} & M_d = 5.812 \text{ Kg} \\ B_d = 23.248 \text{ N}\cdot\text{s/m} & \end{array}$$

with the controller gains set as:

$$K_y = \begin{bmatrix} 916.9226 \\ 975.3221 \\ -16009.2891 \\ 799.5946 \end{bmatrix}$$

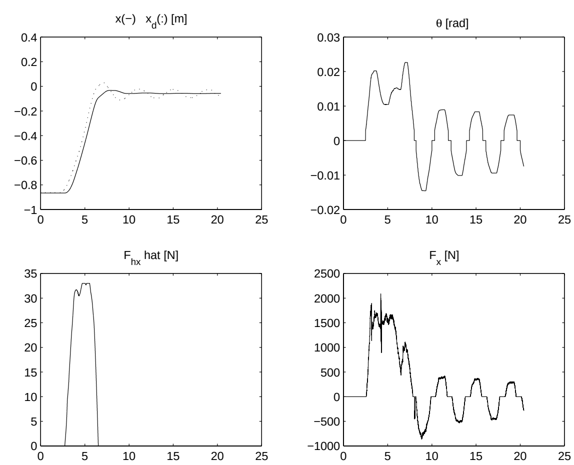

Fig. 5. Experimental results x axis

The experimental results for the x and y axes are shown in Figures (5) and (6) respectively. Each figure shows the estimated force applied to the load ($\hat{F}_{hx}$, $\hat{F}_{hy}$), along with the control signal (F_x, F_y) and the wire angle (θ, ϕ).

The actual position (x, y) and the desired position (x_d, y_d) of the crane are also plotted. All the experimental

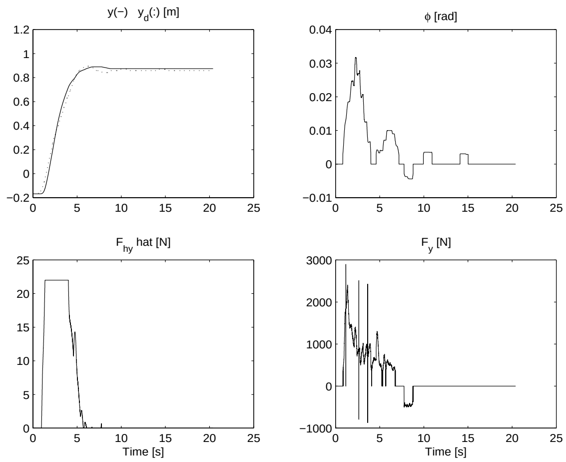

Fig. 6. Experimental results y axis

results are for a constant l, i.e., a constant longitude of the wire.

VII. Conclusion

In this paper we presented the problem of controlling an overhead crane using an estimation of the force applied to the load. Using a linearized system, a controller-observer was designed using the placement of the closed-loop poles for both the system and the observer. The controller structure was tested in both numerical simulation and then using an experimental setup. Due to parametric uncertainties and disturbances in the dynamical model of the system we used deadzones on the estimated applied force (F_h), the angle of the wire (θ, ϕ) and on the control signal (F). With the use of these nonlinear elements, we could work with a simple model of the system and yet obtain a relatively clean estimate of the force F_h, which is then used to generate a tracking signal defining the impedance of the crane.

Although the results shown are for a constant load and a constant length of the wire, we performed tests with different loads and different cable lengths and experimentally confirmed that the controller system is robust to variations to m_2 and l.

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