Load Swing Reduction in Manually Operated Bridge Cranes

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Abstract—In this paper, a procedure for control system design of an anti-sway system for manually operated bridge cranes is presented. Specifically, the human operator acts by means of a push-button panel to operate the crane and thus is considered as a disturbance to the closed loop control system that regulates to zero the angle of the load cable with respect to the vertical axis. Closed loop control has been made possible by the use of an inertial sensor and an Extended Kalman Filter which provides accurate angle estimation. An accurate multi-body model has been realized using the Automatic Dynamic Analysis of Mechanical System (ADAMS) environment. The model parameters have been tuned by system identification based on real measurements. Finally, the model is used to design a Sliding Mode Controller to suppress the load swing angle.

I. INTRODUCTION

An overhead bridge crane is typically used for challenging manipulation tasks in many industrial applications, e.g. in the refinement of steel or to handle raw materials in car industry. It consists of parallel runways with a travelling bridge spanning the gap, as described in [1]. A trolley, the lifting component of a crane, can travel along the bridge, as visible in Fig. 1.

![Bridge crane schematic representation](image)

The cranes considered in this paper are manually operated by means of a push-button panel. So, during the crane normal operating conditions, load oscillations can easily occur, mainly caused by the operator. The system proposed in this paper aims to remove the oscillation of the load while the bridge crane is moved by the operator. So, the anti-sway system operates in parallel to the operator commands. By eliminating undesired oscillations, safety is increased avoiding accidental impacts between the load and the operators. This problem has already been faced in many papers: the solutions that can be found in literature range from feedback control on linearized models, see [2], to neural network modeling for nonlinear control, see [3], to the application of sliding mode control techniques, see [4]. However, most of the literature is based only on simulation or small scale laboratory models. Only very few papers deal with real bridge cranes, but they are based on open loop techniques, like input shaping in [5][6], or path planning as in [7]. Such solutions suffer of two major limitations. The first one is related to the model quality: in fact, mismatches between the model and the real system are sometimes very large. The second one is that the controller usually needs an a priori knowledge of the path or of the operation that the bridge crane must do. This does not fit with manually operated bridge cranes. The proposed approach strongly differs from the standard trajectory-tracking control problems presented in the literature above, as it brings further complexity due to unexpected trajectory variations introduced by the human operator.

For this reason, we developed a closed loop control system based on a direct measurement of the steel cable angles with respect to the vertical. This has been obtained by means of a six DOF inertial platform and a Kalman filter in order to obtain a real-time measurement of the sway angle. So, the control system architecture (depicted in Fig. 2) aims to regulate to zero the sway angle, while the human operator is acting as a disturbance on the system, in a Human in the loop (HIL) framework, i.e. the human operator enters the loop and the control action provided by the control system is added to the human one.

So, as already proposed by [8], we designed a classical Sliding Mode Control but, moreover, with a real-time varying input provided by the human operator, with the final goal to allow real-time load swing minimization and relocation accuracy while moving at the maximum feasible speed. The choice of using Sliding Mode techniques is motivated by their intrinsic features: insensitivity to parameter variations and complete rejection of disturbances as stated in [9], together with flexible design and possibility to decouple high dimensional problems into sub-tasks of lower dimensionality. First, a sliding surface is defined to determine the desired state dynamics. Then, a standard First Order Sliding Mode Controller is designed together with chattering attenuation techniques achievement, following [10]. At the end, the
tuning of such control will be performed by the choice of only three parameters, each one having a clear well defined physical meaning (see [11]). In order to effectively tune the controller, a complete multibody model of the system has been developed and validated by means of ADAMS-Simulink co-simulation. The advantage of using such co-simulation lies also in the real-time visual description of the system dynamics. Moreover, as stated in [12], the method of co-simulation using Adams and Simulink can simplify the process of simulation and can make the results more accurate, increasing the control design reliability, providing a new approach for control system design of complex mechanical systems. A first attempt to model a bridge crane using ADAMS is showed in [13],[14], where the system is approximated as a 2 DOF damped pendulum. The strong approximation introduced by these works is due to the cable system modelling. Cable is modelled as a rigid link between trolley and payload and the pulley dynamics are then not considered. In fact, modelling the cable as a rigid body introduces strong limitation into the model, that grows up as much as cable length increases. In this work, the cable has been considered as a flexible body and its stiffness and damping have been considered.

The remainder of the paper is organized as follows:

In Sect. II the system layout is presented.

In Sect. III the multibody model of the three-dimensional bridge crane is built using Adams and then imported into Simulink environment for co-simulation. Then, the model parameters are tuned and the model is experimentally validated.

In Sect. V the Sliding Mode Controller is designed to decrease load swing. Then the designed controller is tested on the real bridge crane

In Sect. VI conclusion are drawn and future directions are derived.

II. SYSTEM LAYOUT AND ANGLE ESTIMATION

Fig. 3 shows the bridge crane used in this work. Its maximum load is 20 tons. The width is 10.45 m and the maximum cable extension is about 6 m. The bridge can move on a 70 m distance.

In this work, the operator moves the bridge crane using a push-button panel. There is no trajectory planning and and the operator move the load to the final destination.

Referring to Fig. 2 the components of the bridge crane are:

- The motors. They are two 1.5 kW AC motors for the bridge and two 1.1 kW DC brushless motors for the trolley. From the control design point of view they receive an input reference value of the speed and they can be considered ideal. The vertical movement is provided by a 18.5 kW AC motor.
- The system. I.e. the bridge and the trolley. This block has two outputs: the position of the bridge crane \((x(t), y(t))\) and the oscillation angles \((\theta_x(t), \theta_y(t))\).
- The push-button panel (operator “disturbance”). This is, strictly speaking, the only input of the fully manually-opeated bridge crane and it is unpredictable.
- The controller. The control system (visible in red) closes a feedback loop on the oscillation angle. The reference angle, since the aim is to remove the oscillation, is zero.

The cable angle is measured by a 6-DOF inertial sensor.
The sensing axes are oriented in the same direction of the bridge crane ones. The angle estimation \( \hat{\theta}(t) \) has been obtained using both the measurement from the accelerometer \( a_x, a_y, a_z \) and from the gyroscope \( \omega_x, \omega_y, \omega_z \) in the inertial platform by means of a steady state Kalman filter:

\[
\begin{align*}
\hat{\theta}_x(t+1) &= \hat{\theta}_x(t) + \Delta t \omega_x(t) + K_x(\theta_{a,x}(t) - \hat{\theta}_x(t)) \\
\hat{\theta}_y(t+1) &= \hat{\theta}_y(t) + \Delta t \omega_y(t) + K_y(\theta_{a,y}(t) - \hat{\theta}_y(t))
\end{align*}
\]

(1)

where:
- \( \Delta t = 1\text{ms} \) is the sampling time;
- \( \theta_{a,x}(t) = \tan^{-1}(\frac{a_x}{a_z}) \)
- \( \theta_{a,y}(t) = \tan^{-1}(\frac{a_y}{a_z}) \)

are the angle values computed on the basis of the accelerometer measurements.

The Kalman gains \( K_x, K_y \) are computed by solving the Discrete Algebraic Riccati Equation, where the following covariance matrices have been used:

\[
Q = \begin{pmatrix} 0.001 & 0 \\ 0 & 0.001 \end{pmatrix} \quad R = \begin{pmatrix} 0.1 & 0 \\ 0 & 0.1 \end{pmatrix}
\]

(2)

III. MULTIBODY MODELLING AND ADAMS-SIMULINK CO-SIMULATION

A multibody model of the bridge crane is designed for effective controller tuning. The model is realized on the basis of the real bridge crane described in the previous section and considering many additional DOF that are usually neglected. This increases the number of parameters describing the system properties. Some of the parameters, such that mass values and geometric dimensions, are directly derived from the real bridge crane. Those parameters that cannot be directly measured are tuned exploiting experimental measurements and system identification techniques. An overall view of the virtual model is shown in Fig.4.

![Fig. 4. Overall view of the multibody model of the bridge crane](image)

The model complexity is determined by a wide number of degrees of freedom. The six main DOF are:
- trolley motion along \( x \) axis;
- bridge translation along \( y \) axis;
- load vertical lift along \( z \) axis;
- load oscillations around these three axes \( \theta_x, \theta_y \) and \( \theta_z \).

Moreover, a wide number of additional DOF have been taken into account:
- \( \theta_w \) and \( x_w \) are the two additional DOFs that simulate the cable winding around the cylindrical drum placed on the trolley;
- \( \theta_t \) the angular rotation of the pulleys around their axes;
- \( \theta_h \) takes into account the DOF due to the rotational joint between hook and weight.
- \( \theta_{a,x} \) describes start anchor rotation about \( z \) axis. The goal is to simulate the connection between the cable fixed end with the trolley.

These secondary DOFs cannot be neglected because they themselves are among the load oscillation causes.

So, the multibody model is then composed of the following subsystems:
- linear guides;
- two bridges;
- trolley;
- lifting system made by:
  - pulleys;
  - two parallel cable systems;
  - hook;
  - load.

Each of these parts need to be modelled properly. The linear guides, the two bridges and the trolley are easily designed starting from the real bridge crane geometry. The lifting operation is based on a pulley transmission using two cable systems that link the trolley to the hook and consequently to the load. The cable system that supports the load is designed using two steel ropes. Each of the ropes starts from a different anchor point placed on the trolley and goes through a pulley system. At one end, the cables are winched around a drum placed on the trolley; on the other end, the fixed end they are connected to the start anchor, as shown in Fig.5.

Cables are modelled using Machinery ADAMS tool. The cable model can be imagined as a series of massless spherical linkages connected together using axial springs with their own stiffness and damping. Stiffness and damping of the whole cable are then derived from linkages series connection and are settable as a parameter in ADAMS. So, cable tension, cable-to-pulley contact (including friction) and winching length effects are captured. Cable winching is used in this model to simulate the winding and un-winding of the cable around the drum. Thus, due to the not rigid cable model, relative motion between trolley and load arises introducing non linearities into the system.

Attention must be paid also to the pulley system model. Real pulleys are well known, so it is possible to derive not only system geometry but also dimensions, masses and momentum of inertia. More studies are instead required to determine right friction coefficients. In particular, two friction torques have to be modelled:
- Friction torque due to the pulley rotation around its axis.
- It has been modelled as the sum of static component and a dynamic viscous friction.
Friction torque due to the contact between pulleys and cables, which plays an important role in the lifting simulation. In particular, a sliding torque between cable and pulleys has been modeled.

The fixed-end of the cable is linked to the trolley through a rotational anchor, which is able to rotate around x axis relatively to the trolley as visible in Fig. 5. This component introduces an additional rotational DOF between the load and the trolley, as said above. At the other end, the cable is wrapped around a drum.

Once the multibody model is completed, it can be exported to Simulink, by means of a dedicated tool. So, the controller can be tuned, tested and validated directly on the multibody model in the Simulink environment.

### IV. PARAMETER IDENTIFICATION AND MODEL VALIDATION

Once the multibody model of the real bridge crane has been created, some parameters are unknown: the anchor damping $D_a$, inertia $J_{a,x}$ and elasticity $K_a$; the rotational friction coefficients $\mu_{s,p}$ and $\mu_{d,p}$ of the pulley system; the cable damping $D_c$ and elasticity $K_c$. Thanks to the co-simulation environment it is possible to estimate the parameters by means of a standard identification process [15]. In particular, the identification process consists in the iteratively search for those parameter values which minimize the following cost function representing the distance between the simulated output signals (obtained from the model) and the experimental data

$$J(\vartheta) = \frac{\sum_{t=1}^{N}(\theta(t) - \hat{\theta}(t, \vartheta))^2}{\sum_{t=1}^{N}(\theta(t) - \frac{1}{N}\sum_{t=1}^{N}\theta(t))^2}$$

(3)

where $\vartheta$ is the vectorization of the unknown parameters that must be estimated; $\theta(t)$ is the measurement of the angle (single axis); $\hat{\theta}(t, \vartheta)$ is the angle provided by the multibody model; $N$ is the number of available data. The estimated parameters are

$$\hat{\vartheta} = \arg \min_{\vartheta} J(\vartheta)$$

(4)

Two data sets have been measured on the real bridge crane: an estimation data set, used to estimate the parameter values by minimization of the cost function in Eq. 3; a validation data set, that is used to assess the performance of the model with the estimated parameters.

Fig. 6 shows the validation results: the red line show the measured oscillation angle as it is estimated by the accelerometer and the blue line the simulated value. Relative standard error is then calculated to evaluate the goodness of fit between real and simulated data. Relative standard error values are under 10% in each of the performed tests.

![Fig. 6. Model validation results on z axis at H=3metres](image)

Finally, notice that the oscillation of the open loop system are really poorly damped.

### V. CONTROL DESIGN

Following the model validation, a First Order Sliding Mode Control (1SMC) is implemented on each axis, see [16]. The height of the load is fixed, in order to decouple the

1 $K_c$ is defined as the ratio of cable longitudinal stiffness with respect to reference beam formulation.

2 $D_c$ is defined as a ratio applied as multiplier of stiffness to define cable damping.
controller design in different subtasks, one for each degree of freedom. Thus, the following section will explain the control design process, from the Sliding Surface definition to the Sliding Mode Controller implementation. For the sake of simplicity, the attention is focused only on the bridge movement along a single axis. First, the sliding surface and the control action have been defined as:

$$\sigma(t) = a \cdot \dot{\theta}(t) + \dot{\theta}(t)$$

$$u(t) = u_d(t) + \beta \cdot \text{sign}(\sigma(t))$$

(5)

where: $u_d$ is the manual input provided by the operator through the push-button panel. The parameters $a$ and $\beta$ are the control tuning parameters. In particular, $\beta$ must be carefully tuned: the bigger it is, the faster the control action will reject disturbances and avoid oscillations but, on the other hand, it could lead to severe chattering and undesired load translation at the end of the transient. The parameter $a$ is used to balance the importance of the angle and of the angular velocity in the sliding surface. Their tuning is performed by trial and error. The control goal is to achieve a robust regulation to zero of the angle $\theta$ without loss of accuracy on the load positioning. In fact, the sliding mode control action keeps switching in order to maintain $\theta \approx 0$ as accurately as possible also at the end of the transient and this leads to a load undesired slow, continuous movement.

For this reason, the standard ISMC is modified by introducing a boundary layer, as suggested in [17]:

$$u(t) = u_d(t) + u_c(t)$$

(6)

where

$$u_c(t) = \begin{cases} 0 & \sigma(t) < \delta \\ -\beta \cdot \tanh \left( \frac{\sigma(t)}{\epsilon} \right) & \sigma(t) \geq \delta \end{cases}$$

(7)

where $\delta$ is experimentally properly tuned in order to achieve a smoother control action. The $\text{tanh}$ function is used instead of the $\text{sign}$ function to avoid chattering, being $\epsilon$ a proper smoothness coefficient.

Using the co-simulation tool described in the previous section, the designed controller has been validated through a realistic test. More specifically, the forward button is pushed three times consequently, providing the trolley speed reference shown in red in the upper plot in Fig. 7. If the controller is switched off, the load angle shown in the middle plot is obtained. If the controller is switched on, it provides the additional control action shown in blue in the upper plot and the load angle shown also in blue in the middle plot. Moreover, the load position is shown in the lower plot.

This test proves that accurate positioning of the load and oscillation reduction can be obtained with the proposed controller.

In the end, it is worth noting that, even if theory suggests that the use of boundary layers continuous functions in Eq. 7 could lead to poor robustness and possible instability, the intrinsic stability of the bridge crane helps avoiding this kind of risk, as shown in Fig. 8, where the phase portrait matches the expectations. In Fig. 8, the blue line describes the evolution of the states. The black continuous line is the sliding surface, given the input represented in blue in the upper plot in Fig. 7. The black dashed lines represent the effect of the boundary layer introduction on the states trajectories. In particular, three main large revolutions (labeled in the plot as 1,2,3) of the phase diagram are visible, associated to the three control action variations determined by the operator using the push-button panel (see the red plot in the upper figure in Fig. 7). Then, the evolution of the states is forced in proximity of the origin (the last and smaller revolution, labeled in the plot as 4), following the dynamics imposed by the sliding surface. Finally, the phase plot is constrained to the origin.

VI. CONCLUSIONS

In this paper, a procedure for control system design of an anti-sway system for manual bridge crane has been presented. Specifically, the human operator is considered has a disturbance to the closed loop control system that regulates to zero the angle of the load cable with respect to the vertical axis. Closed loop control has been made possible by the use of an inertial sensor and an Extended Kalman Filter which provides accurate angle estimation. An accurate multibody model has been realized. In order to describe many DOF, additional unknown parameters have been introduced in the model. Those parameters have been tuned by system identification based on real measurement. Finally, a Sliding Mode Controller has been designed and tuned using the multibody model thanks to co-simulation.
Fig. 8. Phase portrait of the controlled systems

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