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ITERATIVE METHOD FOR CONTROLLING WITH A COMMAND PROFILE THE SWAY OF A PAYLOAD FOR GANTRY AND OVERHEAD TRAVELING CRANES

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Abstract. *In this work, a novel anti-sway control system, obtained with a command profile, for gantry or overhead crane is proposed. The proposed method is based on an iterative calculation of the sway angle and of the corresponding applied velocity profile as input to the crane motors. A comprehensive mathematical model is developed. Simulation results show that the proposed anti-sway method is able to damp the oscillations of the suspended load during the contemporary movement along the three perpendicular Cartesian axes (both horizontal axis and also vertical axis). The system is schematized as a double-pendulum. Besides, as a consequence of being an iterative method, it is possible to obtain the total damping of the sway changing many times the velocity set, also if the previous damping control of the oscillations is again in progress. That makes the method extremely useful in the manual control of the crane by an operator as well as with automatic control.*
Keywords: Anti-sway, Open-loop control, Iterative method, Overhead and gantry cranes

1. Introduction.

In general, it is possible to divide cranes in two categories: industrial cranes and building (slewing) cranes. Industrial cranes, such as gantry cranes or overhead traveling cranes, are typically used for the transferring of load suspended by cables in industrial sites.

The hoisting equipment for a traveling crane typically is made with:

1) A traveling system on the transverse length of the crane, which is defined introducing horizontal axis x (Trolley movement).
2) A traveling system movable on rails along a horizontal axis y perpendicular to the axis x, so allowing to the carriage to be movable both along the axes x and y (Translation movement or Crane movement).
3) A device able to hoist the load. This device is defined typically of one or more cables and a hooking mechanism. The hoisting mechanism lifts the payload. The length of the cables is variable in order to move the load on a vertical axis z (Hoisting movement).

As a consequence of this industrial apparatus, typically, a crane has three inputs as movement commands, which command in an independent way: the trolley movement, the translation movement and the hoisting movement.

However, inertial forces on these cranes due to commanded trajectories or operator commands can cause the payload to have sway angles. And, this problem is increased by the fact that cranes are typically lightly damped, which means that it is necessary a long time to damp the sway.

It is obviously desirable to reduce the sway of the payload in order to obtain two positive effects: increase the safety of the load transfer and reduce the operational time to the shorter possible time.

Today, only an experienced operator can effectively lead a container to a stop of the movement without swing. Typically, the operators must wait for a time necessary to stop the swing and this time, spent to wait for the sway stop, can reach also one-third of the total transfer time.

Already several solutions to the problem to reduce the swing of the suspended load are proposed in the past. However, these solutions are related to different ways to obtaining a reduction of the swing.

1.1. **Closed loop anti-sway.** everal solutions of anti-sway problem are related to closed-loop anti-sway, that is the situation where the angle of swing is measured by means of sensors associated to the crane. In example, the French Patent FR2698344 [1] or the French Patent FR2775678 [2] describes a control system of the above type.

1.2. **Open loop anti-sway.** Nevertheless, the most relevant part of the solution of the problem to reduce the swing is related to open-loop anti-sway method. That means the situation where the reduction of the sway is obtained by generating a convenient profile of the velocity during the acceleration and the deceleration of the horizontal movements of the crane.

1.3. **"Sway cancellation" and "Filtering".** In this way, it is possible to delete the sway generated from a pulse horizontal acceleration, with another short equal pulse generated at a time equal to the semi-period of the oscillation.

Several Patents are relative to the "sway cancellation" method. We can mention U.S. Patent 4,756,432 to Kawashima [3], U.S. Patent 5,219,420 to Kiiski and Mailisto [4], U.S. Patent 5,960,969 to Habisohn [5], U.S. Patent 5,785,191 to Feddema et al. [6], U.S. Patent 5,127,533 to Virkkunen [7], U.S. Patent 5,526,946 to Overton [8] and also Patent WO 02/070388 to Ong and Gilbert [9].

The recent Patent EP 2015/2896590 to Caporali [10], however, relative to slewing cranes, used the method of the "Filtering" in order to obtain the cancellation of the sway.

1.4. **"Input Shaping".** By using this method the system sway is reduced by the convolution of the command input signal with a sequence of impulses that has been defined upon the natural frequencies of the system.

Alsop et al. [11] presented a two-dimensional linear model of a gantry crane. Jones and Petterson [12] extended the work of Alsop et al.

More recent contributions to the method of "Input Shaping" come from Fujioka et al. [13], Hu et al. [14], Masoud and Alhazza [15], Samin et al. [16], Liu and Cheng [17]. Nevertheless, the limitation of the input shaping technique is due to the fact that it requires highly accurate values of the system parameters to achieve a satisfactory system response (Yoon et al. [18]). Furthermore, Singhose et al. [19] showed that input shaping techniques suffer important degradation in crane maneuvers that involve hoisting.

1.5. **"Adaptive control".** In this case, adaptability towards the parameter uncertainties and the external disturbances of the system has been widely investigated from numerous researchers. For example, relevant studies relative to the "adaptive control" recently come from Nguyen et al. [20], Fang et al. [21], Sun et al. [22], Sun et al. [23].

1.6. **"Neural networks" and "Fuzzy method".** In the work of Abe [24] an innovative radial function network was proposed for the sway elimination; in the work of Smoczek and Szpytko [25] a fuzzy logic-based robust feedback anti-sway control was proposed for an overhead crane.

1.7. **"Command smoothing".** In this class of solution, it is introduced of an additional damping into the system (Ramli et al. [26]).

In this approach, the crane acceleration is typically given in the general form: $\ddot{x} = r + k_1\,\vartheta + k_2\,\dot{\vartheta}$ or similar expression. Here above, r is a function of the time that depends on the movement profile of the crane. The obtained system will have the wanted damping ratio and the natural frequency using the convenient values of the coefficients k_1 and k_2.

In U.S. Patent 5,443,566 to Rushmer et al. [27], sway corrections are estimated using a model of the crane with fixed-length cable. Also in U.S. Patent 5,490,601 to Heissat and Lacour [28], in U.S. Patent 5,878,896 to Eudier and Demoustier [29] (where the values of the coefficients k_1 and k_2 are obtained via experimental) and in the recent Patent WO 2009/065808 to Stanchev [30] the used method is the "Input Shaping".

Henry et al. [31] and Masoud and Nayfeh [32] developed this strategy of the "command smoothing", relying on time-delayed position feedback of the payload cable angles.

Furthermore, several recent works were developed using the method of "command smoothing" to reduce the sway. Recent contributions, in particular, arrive from Xie et al. [33], Huang et al. [34], Tang and Huang [35], Alghanim et al. [36]. Particularly, in the work of Huang et al., "command smoothing" was used for oscillation suppression of a bridge crane that utilized a double pendulum with a distributed mass payload.

Also the method presented in this paper uses the "command smoothing" method for reducing the sway, introducing an additional damping into the system in order to control the sway. The presented method can be considered innovative compared to previous works above described in some fundamental aspects.

In this work, the solution of the full dynamic equation of the crane system is computed taking account of the double-pendulum geometry, obtaining the effective period T of the double-pendulum. Subsequently, the cable flexibility and the relative damping are introduced: in this way, this method allows to obtain more performances as regards the previous methods where these data are not taken into account.

This novel work is based on the calculation of the complete dynamical equations for the crane system. In this way, also the continuous variation of the hoisting height for the suspension point of the payload and the variation of the corresponding velocity and acceleration are considered. Today, a lot of the previously mentioned works consider a simplified solution of the problem, which is defined a fixed length of the pendulum. So, these ways are not able to eliminate sway when the cable length changes during the horizontal movements.

The calculation uses the fundamental equation obtained by the Lagrange equations in order to define the successive steps. The control process is realized by a control device and includes a calculation step for determining the angle of oscillation of the load and the time derivative of the sway angle. Practically, the control method calculates the described data through an iterative process using the velocity and acceleration of the sway angle. This method does not require means to measure the sway angle, but the control device includes means to obtain the "effective length" l_{eff} of the suspension cable. The values of the coefficients k, which need to introduce an additional damping, are obtained with the observer method.

The velocity of the trolley movement is obtained directly from the reference velocity supplied by the control device. It is assumed that the drive controls the speed reference with high rapidity. The device able to regulate the system includes an estimator module and a correction module. The estimator module receives as input the "effective length" l_{eff} of the cable and the velocity of the movement of the trolley relative to the x direction.

Finally, as a consequence of being an iterative method, in the present work it is possible to obtain the total damping of the sway changing many times the velocity set, also if the previous damping control of the oscillations is again in progress. In other ways, it is not necessary that the target velocity has been reached in order to carry out a subsequent command towards a subsequent target velocity. That makes this new method extremely useful in the manual control of the crane by an operator.

This paper is organized as follows.

The dynamical model of the crane is presented in Section 2, describing in detail the geometry of the double-pendulum relative to the suspended load and the cable flexibility. From Lagrange equations, the dynamical equation relative to the sway angle is obtained, where it is highlighted the functional dependence on the acceleration of the trolley movement. The equation solution is obtained with an iterative method, where the estimated values of the sway angle and the trolley acceleration are obtained step by step. After that, designing an observer method, it is possible to synchronize this estimated model with the real crane, verifying that the system is observable.

In Section 3 an implementation of the method and the most relevant results of the model simulation are presented, with particular emphasis to the fact that total damping is obtained changing many times the velocity set also during the previous ramp.

In the end, in Section 4 concluding remarks are given.

2. **Dynamic Model of the Crane System and Solution Description.** The following detailed description of this new method includes a description of an industrial crane and a derivation of the equation of the motion that represents the dynamics of the defined degree-of-freedom of the industrial crane.

Referring to Figure 1, an industrial crane (gantry or overhead traveling crane) includes a multi-body system with 3 independent degrees-of-freedom for positioning a pendulum suspended by spreader bar to the trolley. Specifically, the industrial crane includes a translating load-line, having length L and a payload attached to a moving or translating trolley. A crane operator or a computer (in automatic way) can position the payload using some available commands and changing the load length l. The crane configuration is characterized by one vertical axis (in Figure 1, axis z) and two translational horizontal independent axes between them (in Figure 1, axis x). Therefore, there are three controlled motions for an industrial crane: a trolley translation, a crane translation and a vertical motion considering the variation of the length l. The first two movements can be considered in an equivalent way.

Dynamics of the generalized crane can be represented using the independent Lagrange coordinates relative to the 3 independent degrees of freedom:

$q_1 = \varphi$ sway angle

$q_2 = x$ horizontal position of the crane

$q_3 = l$ hoisting vertical position of the payload.

So, in order to describe the motion of the industrial crane, an equivalent kinematics scheme with concentrated masses is represented in Figure 1.

In this figure, a geometric description of the industrial crane system as a double-pendulum is given. Particularly, we can see that a rectangular body (representing the superior part of the overhead traveling crane or of the gantry crane) moves along the x axis, starting from the origin O. The points A, T and D represent, respectively, the suspension points of the lateral and central cables (of length L and l, respectively). d

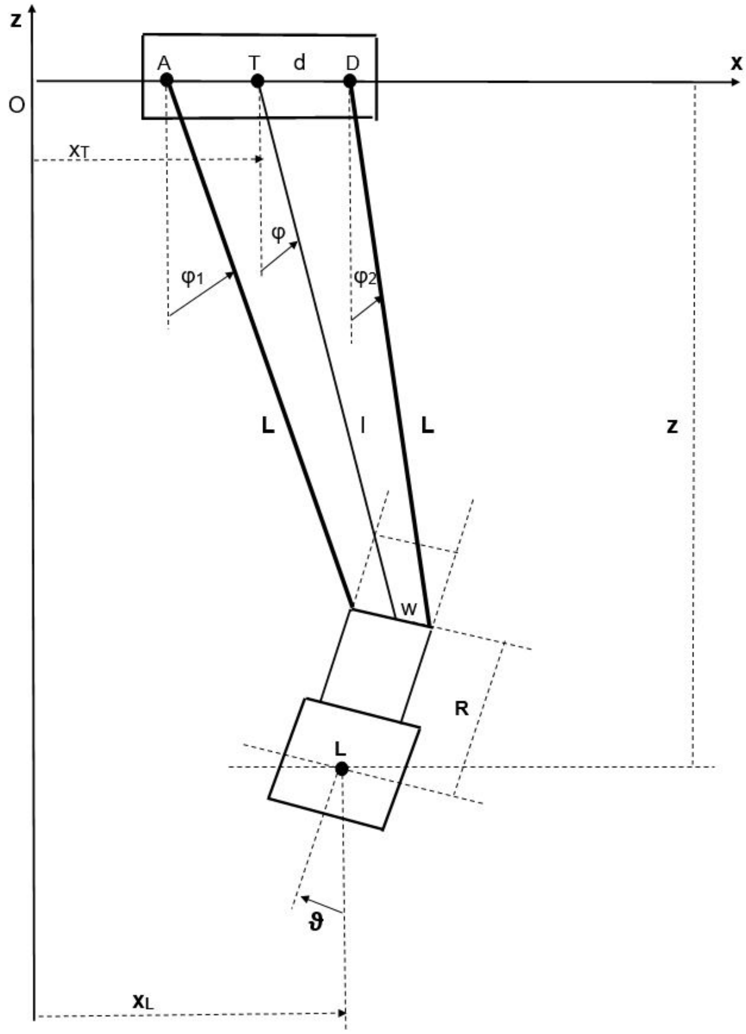

FIGURE 1. Geometric description of the industrial crane system, overhead traveling crane or gantry crane

is the distance between the points A-T, and T-D, w is the distance between the same cables, but in correspondence of the attack point on the payload. Finally, x_T and x_L represent the horizontal position of the crane mass center T (with mass M) and of the payload mass center L (with mass m), respectively. z is the height of the mass center of the payload (expressed as distance from the x axis).

Really, there is a correction on the cable length to consider. This is due to the fact that the spreader is suspended by two inclined cables and therefore there is a constraint between the sway angle of the pendulum φ and the roll angle of the spreader and possible container ϑ, as can be seen in Figure 1. In other words, here a model of the crane as a double-pendulum is used.

The derivation of this constraint is done with the help of the same Figure 1, considering $L_1 = L_2$.

We obtain the following relation between φ and ϑ:

$$d\sin(\varphi) = w\sin(\vartheta + \varphi) \tag{1}$$

It can be written when linearized for small angles as:

$$\vartheta = \frac{d - w}{w}\varphi \tag{2}$$

Besides, for small oscillations, we have:

$$x_L \approx x_T + l\varphi - R\vartheta \tag{3}$$

$$z_L \approx l + R \tag{4}$$

$$l^2 = L^2 - \frac{1}{4}(d-w)^2 \tag{5}$$

Now using Equations (2) in (3) yields:

$$x_L = x_T + l\varphi - R\frac{(d-w)}{w}\varphi \tag{6}$$

Therefore,

$$\varphi = \frac{x_L - x_T}{\sqrt{L^2 - \frac{1}{4}(d-w)^2} - R\frac{d-w}{w}} \tag{7}$$

In the end, the following relation holds for the period time T, taking account of the geometry of the load suspension:

$$T = 2\pi\sqrt{\frac{\sqrt{L^2 - \frac{1}{4}(d-w)^2} - R\frac{d-w}{w}}{g}} \tag{8}$$

Define the "effective length":

$$l_{\mathit{eff}} \equiv \sqrt{L^2 - \frac{1}{4}(d-w)^2} \tag{9}$$

and the geometric parameter "a":

$$a \equiv \frac{d-w}{w} \tag{10}$$

So, from (2), we have:

$$\vartheta = a\varphi \tag{11}$$

This length has to be considered in order to compute the effective period time of the double pendulum describing the sway of the load attached to the overhead crane.

$$T = 2\pi\sqrt{\frac{(l_{\mathit{eff}} - aR)}{g}} \tag{12}$$

2.1. **Double pendulum with cable flexibility.** Although the model of the double pendulum is able to describe better than the single ideal pendulum the real physical situation of the payload attached to the overhead crane, this model is not able to perform really well if compared with the realistic situation. Typically, this model estimates again the period time too low for all cable lengths. A too small period time can be the cause of incorrect corrections to the sway of the payload.

It is therefore likely that the assumption of an infinite stiffness of the cable is the major factor to consider for this difference.

In general, considering the equations of the motion that describe the overhead crane dynamics, some damping in the response of the model is obtained.

We must consider another model, which is exactly the same as the one presented in the previous section, but now extended with a certain cable stiffness k and damping b, as represented in Figure 2, where the cables have a parallel spring k and damper b. This model has two extra outputs, the stretch of cable 1 and cable 2. In this case, the period time T is increased compared to the model without cable flexibility. Part of this increase

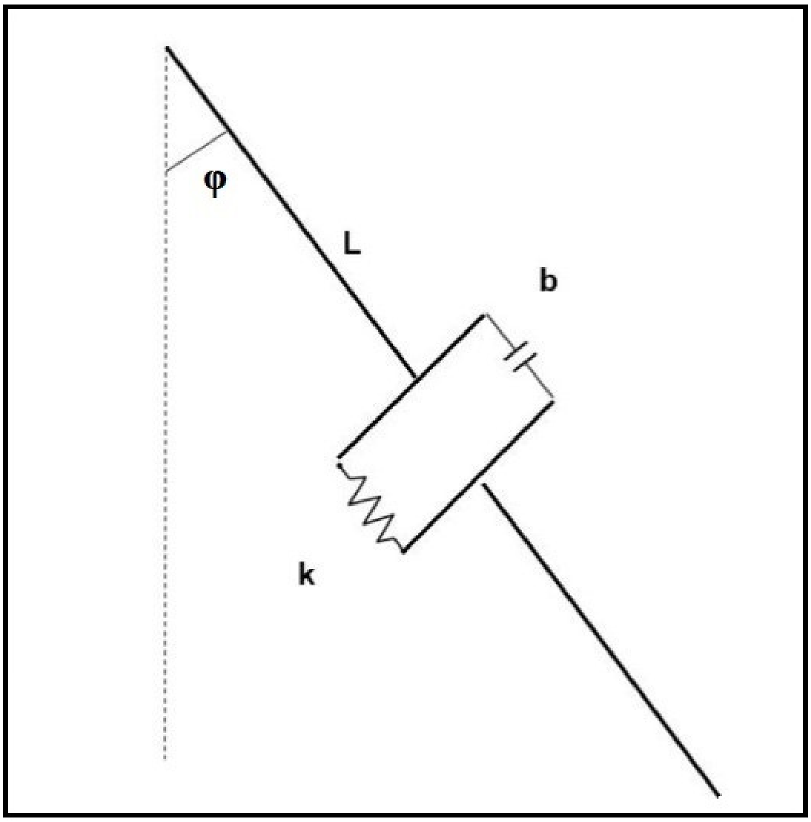

FIGURE 2. Representation of the cable flexibility

can be directed to the elongation of the cables due to the introduced finite cable stiffness. The cable stiffness and cable damping are:

$$k_{cable} = \frac{k_0}{L} \tag{13}$$

$$b_{cable} = \frac{b_0}{L} \tag{14}$$

Being the cable stiffness and damping dependent from the length, the correction on the period time T is not the same for every length. When the cable length is short, the stiffness will be larger and the model will approximate the model without cable flexibility.

However, the elongation of the cables is not the only cause of the increase in period time. Typically, the elongation of the cable with classic load parameters at a length of 20m is only 0.15m.

Therefore, the biggest part of the increase in period time comes from the phase delay introduced by the cable stiffness. In fact, the inclination of the cables decreases the period time compared to a pendulum. This is due to the fact that, as a consequence of the inclination of the cables, the last ones also exert a horizontal force on the load. When the cables have a finite stiffness, this force will be applied with a phase delay which will cause the period time to increase with respect to an infinite cable stiffness.

Also the damping parameter of the cables is an important factor in the total damping of the movement and has to be absolutely considered.

In order to compute the components of the elongation u_1 and u_2 for the cables 1 and 2 (both with initial length L), from Figure 1 we have:

$$\vec{u}_1 = u_1 \sin(\varphi_1) \cdot \vec{i} - u_1 \cos(\varphi_1) \cdot \vec{j} \tag{15}$$

$$\vec{u}_2 = u_2 \sin(\varphi_2) \cdot \vec{i} - u_2 \cos(\varphi_2) \cdot \vec{j} \tag{16}$$

being i and j the unit vectors in the direction x and y respectively.

Linearizing for small angles, we obtain:

$$L \cdot \varphi_1 - \frac{1}{2}d = l \cdot \varphi - \frac{1}{2}w \Rightarrow \varphi_1 = \varphi\left(\frac{l}{L}\right) + \frac{(d-w)}{2L} \tag{17}$$

$$L \cdot \varphi_2 + \frac{1}{2}d = l \cdot \varphi + \frac{1}{2}w \Rightarrow \varphi_2 = \varphi\left(\frac{l}{L}\right) - \frac{(d-w)}{2L} \tag{18}$$

Therefore, in the end, we have, considering that the two cables have the same length L and so the same absolute value of the elongation u:

$$\vec{u}_1 = u\left\{\left[\varphi\left(\frac{l}{L}\right) + \frac{(d-w)}{2L}\right]\vec{i} - \vec{j}\right\} \tag{19}$$

$$\vec{u}_2 = u\left\{\left[\varphi\left(\frac{l}{L}\right) - \frac{(d-w)}{2L}\right]\vec{i} - \vec{j}\right\} \tag{20}$$

where l is defined by (9).

Therefore, the total contribution of the elongation of the two cables with length L (see Figure 1) at the potential energy of the system is obtained as:

$$|\vec{u}_1|^2 = u^2\left\{\left[\varphi\left(\frac{l}{L}\right) + \frac{(d-w)}{2L}\right]^2 + 1\right\} \approx u^2\left(1+\varphi^2\right) \tag{21}$$

2.2. **Potential energy.** The potential energy of the system is given by two effects: the gravity on the mass payload m_L and the cable elongation with stiffness k.

The term due to the gravity is:

$$V_g = -m_L g \cdot (l\cos\varphi + R\cos\vartheta) \tag{22}$$

which for small oscillations reduces to (developing in series):

$$V_g = -m_L g \cdot (l+R) + \frac{1}{2}m_L g \cdot \left(l \cdot \varphi^2 + R \cdot \vartheta^2\right) \tag{23}$$

Considering Equation (21), the term of the potential energy relative to the cable elongation is:

$$V_k = \frac{1}{2}k_1 u_1^2 + \frac{1}{2}k_2 u_2^2 \approx k_0 u^2\left(1+\varphi^2\right) \tag{24}$$

Relatively to the term of the potential energy corresponding to the elongation, from (24) we can obtain, in the simple hypothesis of linear pendulum (that is, in Figure 1, $d = w \Rightarrow L_1 = L_2 = L = l$ and $k_1 = k_2 = k$):

$$\frac{\partial V_k}{\partial \varphi} \equiv \frac{1}{2}k_1 u_1^2 + \frac{1}{2}k_2 u_2^2 \approx 2kl^2\varphi \cdot \left\{1 - \frac{1}{\sqrt{1+\varphi^2}}\right\} \tag{25}$$

2.3. **Kinetics energy.** The total kinetics energy for the system considered in Figure 1, given by the crane with mass M and by the payload with mass m, is:

$$T = T_M + T_m \tag{26}$$

$$T = \frac{1}{2}Mv_M^2 + \frac{1}{2}mv_m^2 \tag{27}$$

where v_M is the velocity of the gravity center of the mass M and v_m is the velocity of the gravity center of the mass m.

We note also that the coordinates of the payload with mass m (Figure 1) are:

$$x_L = x_T + l\sin\varphi - R\sin\vartheta \tag{28}$$

$$y_L = -l\cos\varphi - R\cos\vartheta \tag{29}$$

2.4. **Lagrange equations.** If we establish the Lagrange function:

$$L = T + U \tag{30}$$

and applying the generalized Lagrange equations

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}_i}\right) - \frac{\partial L}{\partial q_i} = Q_i \tag{31}$$

we can obtain the system of equations in the generalized coordinates q_i.

Dynamics of the generalized crane can be represented as a multi-body system with 3 independent degrees of freedom:

$q_1 = \varphi$: sway angle

$q_2 = x$: horizontal position of the crane

$q_3 = l$: hoisting position of the payload.

Focusing our interest relatively to the Lagrangian coordinate φ, in the first approximation, from Lagrange Equation (31) we obtain:

$$\ddot{\varphi} = {}^1\!/_l\left\{-g\sin\varphi - \ddot{x}\cos\varphi + 2kl\varphi + \left(\dot{l} - k_f\right)\cdot\dot{\varphi}\right\} \tag{32}$$

Not considering, in the first approximation, the contribution of the elongations u_1 and u_2 for the cables 1 and 2 (and so considering the stiffness $k = 0$), we obtain from (32):

$$\ddot{\varphi} = {}^1\!/_l\left\{-g\sin\varphi - \ddot{x}\cos\varphi + \left(\dot{l} - k_f\right)\cdot\dot{\varphi}\right\} \tag{33}$$

where k_f is the friction coefficient with constant value, defined depending on the considered axis, $\dot{l}$ is the derivative of the length l (that is the velocity of the vertical movement along the axis z), $\ddot{x}$ is the acceleration of the trolley movement along the axis x and g obviously is the gravity acceleration.

2.5. **Equation solution.** In order to obtain the control of the load sway, this method requires a control device. This control device calculates the angle of oscillation of the load and the velocity of this sway angle using only the variable information of the length of the pendulum and of the speed and acceleration of the movement of the trolley along the axis x.

The control process is realized by the control device and includes a calculation step for determining the angle of oscillation of the load and the time derivative of the sway angle, starting from the knowledge only of the information above described.

Practically, the control method calculates the described data through an iterative process using the velocity and acceleration of the sway angle.

The calculation uses Equation (32) in order to define the successive step. This method does not require means to measure the sway angle. Nevertheless, the control device includes means to obtain the "effective length" l_{eff} of the suspension cable.

The velocity of the movement of the trolley $\dot{x}$ can be obtained directly from the reference velocity $V_{Ref(x)}$ that is the input to the drive able to control the motor causing the movement of the trolley in the direction x. Obviously, it is assumed that the drive controls the speed reference with high rapidity.

As we can see in Figure 3, the device able to regulate the system includes an estimator module and a correction module.

The estimator module receives as input the "effective length" l_{eff} of the cable and the velocity of the movement of the trolley relative to the x direction.

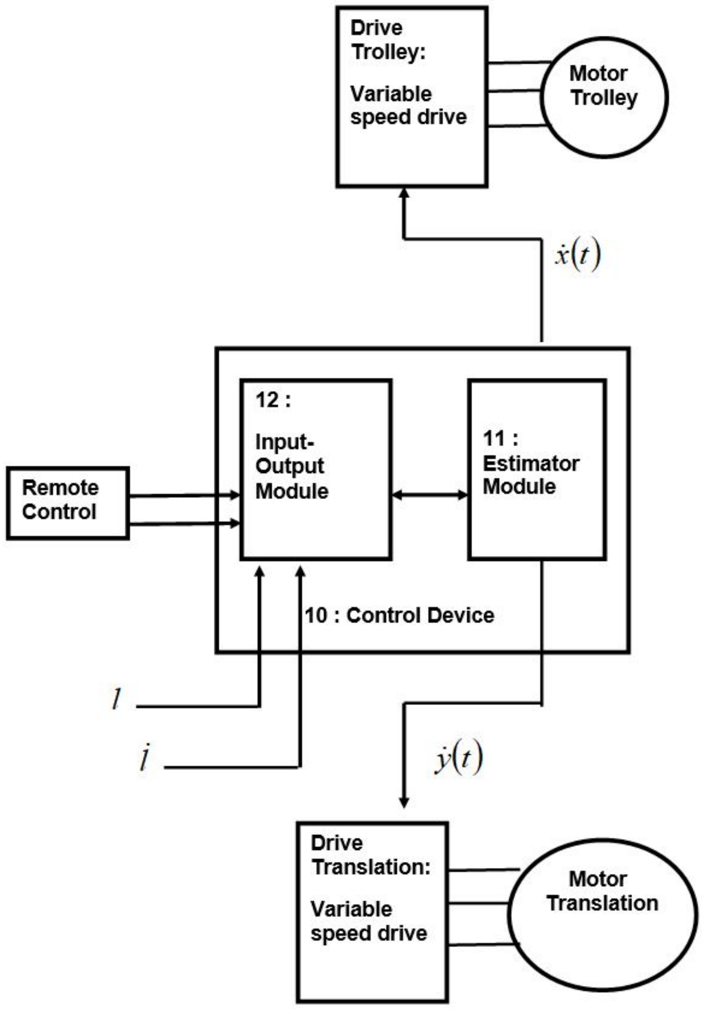

FIGURE 3. Simplified diagram of the control device, input-output, drives and motors

Starting from Equation (32), the solution is obtained with an iterative method that can be represented, at any time t, in the following way:

$$\dot{\varphi}_t = \dot{\varphi}_{t-1} + \ddot{\varphi}_{t-1} \cdot \Delta t \tag{34}$$

$$\varphi_t = \varphi_{t-1} + \dot{\varphi}_{t-1} \cdot \Delta t \tag{35}$$

$$\ddot{x}_t = (\dot{x}_t - \dot{x}_{t-1}) \cdot \Delta t \tag{36}$$

$$\dot{l}_t = (l_t - l_{t-1}) \cdot \Delta t \tag{37}$$

$$\ddot{\varphi}_t = {}^1\!/_{l_t} \left\{ -g \sin\varphi_t - \ddot{x}_t \cos\varphi_t + 2kl\varphi_t + \left(\dot{l}_t - k_f\right) \cdot \dot{\varphi}_t \right\} \tag{38}$$

where φ_t and φ_{t-1} represent the sway angle along the x direction respectively at the time t and at the previous time $t-1$, $\dot{\varphi}_t$ and $\dot{\varphi}_{t-1}$ represent the velocity of the sway angle along the x direction respectively at the time t and at the previous time $t-1$, $\ddot{\varphi}_t$ and $\ddot{\varphi}_{t-1}$ represent the acceleration of the sway angle along the x direction respectively at the time t and at the previous time $t-1$, $\ddot{x}_t$ represents the acceleration of the movement of the trolley relative to the x direction at the time t, $\dot{x}_t$ and $\dot{x}_{t-1}$ represent the velocity of the movement of the trolley relative to the x direction respectively at the time t and at the previous time $t-1$, $\dot{l}_t$ represents the velocity of the movement of the hoisting relative to the z vertical direction at the time t, l_t and l_{t-1} represent the length of the cable respectively at the time t and at the previous time $t-1$, and Δt represents the time difference between the instant t and the instant $t-1$.

The iterative process starts with the hypothesis that at the initial instant the values of the sway angle φ, of the sway angle velocity $\dot{\varphi}$ and of the sway angle acceleration $\ddot{\varphi}$ are equal to zero: that is the pendulum is initially in quiet conditions.

Therefore, for $t = 0$, we have:

$$\varphi_0 = \dot{\varphi}_0 = \ddot{\varphi}_0 = 0 \tag{39}$$

The considerations and formula written for the movement of the trolley along the axis x are equally valid for the movement of the crane along the perpendicular horizontal axis y. We can consider the x axis as the axis relative to the movement of the trolley and the y axis as the axis relative to the movement of the crane (translation movement).

The control device is able to control, at the same time, the movements along the axes x, y and z, being the axis z the vertical axis relative to the hoisting movement.

It is important to consider that the 3 axes are perpendicular among them. Therefore, the movements along these axes x, y and z are independent, being Equation (32) valid both for axis x and for axis y independently.

In other words, because in this kind of crane are not carried out rotational movements, in the equations of the movement are not manifested the centrifugal acceleration and the acceleration of Coriolis, which are responsible for the coupling of the equations.

In this way, a similar equation to Equation (32) is valid along the axis y, where only the sub-index x is substituted from the sub-index y.

Practically, the control device can control both movements along the axes x and y in an independent way, being 2 different and independent routines to control the said movements.

As described above, the iterative method allows to the estimator module to calculate in real time the estimated values of the sway angle and of the velocity of the sway angle.

To be able to synchronize this model with the real crane, an observer has to be designed, verifying that the system is observable.

Using the observer method (see, i.e., Srivastava et al. [37]) applied to Equation (32), it is possible to determine the eigenvectors and corresponding eigenvalues. As a consequence, the observer gain matrix can be obtained.

Specifically, we obtain the correction of the velocity of the movement along the axis x (and corresponding along the axis y) $\Delta\dot{x}$ and $\Delta\dot{y}$ according to the following equations:

$$\Delta\dot{x} = K_0 \cdot \varphi_x + K_1 \cdot \dot{\varphi}_x \tag{40}$$

$$\Delta\dot{y} = K_0 \cdot \varphi_y + K_1 \cdot \dot{\varphi}_y \tag{41}$$

wherein K_0 and K_1 are the observer gains applied respectively to the sway angle and to the velocity of the sway angle along the axes x and y, and $\Delta\dot{x}$ and $\Delta\dot{y}$ are the correction signals that are added to the velocity set-point $\dot{x}_{set}$ and $\dot{y}_{set}$ along the axes x and y respectively.

In other way, $\dot{x}_{set}$ and $\dot{y}_{set}$ are the set-point velocities defined either by the crane's operator during a manually controlled movement or by the automatic definition of the velocity during an automatic controlled movement.

Therefore, the references of velocity applied as an input to the inverter driving the motor relative to the axis x (or y) are the velocity set $\dot{x}_{ref}$ and $\dot{y}_{ref}$ (along with the axes x and y, respectively), that are obtained in the following way:

$$\dot{x}_{ref} = \dot{x}_{set} + \Delta\dot{x} \tag{42}$$

$$\dot{y}_{ref} = \dot{y}_{set} + \Delta\dot{y} \tag{43}$$

The K_0 and K_1 observer gains are functions of the cable length, in order to optimize the velocity corrections according to the length of the pendulum.

As consequence of its simplicity, the control device can be integrated either in a Plc or in a card controller internal to the drive relative to the movement x or y.

Therefore, this iterative method does not require any preliminary stage of modeling in order to know parameters as a measure of the sway angle or a measure of the current flux of the motor. Only, the method uses the calculated value of the sway angle and of the time derivative of the sway angle to compute the correction to the crane velocity to add to the set-point velocity (arriving from the operator or defined in automatic way). That is made in order to obtain the reference of velocity to send to the drive controlling the motor relative to the movement of the trolley (x axis) or of the crane (y axis).

3. **Implementations and Results.** As previously described, the control process relative to this new method does not require knowing variable parameters such as the measure of the angle of swing or a measure of the current flowing in the motor.

Therefore, the drive controller generates the speed profile of the movement relative to the trolley (Trolley movement) and it supplies the information of the speed profile to the drive able to control the corresponding motor. In this feature, a control device computes the speed profile of the movement relative to the trolley having as input the necessary information.

Figure 3 shows a simplified diagram of the control device according to this method relative to the movement of a load along the horizontal axis (trolley or/and translation axes). In general, the control device 10 is made with an estimator module 11 connected with an input-output module 12.

Practically, the estimator module 11 receives the inputs from the input-output module 12. These inputs are the length l of the cable, the vertical velocity of the cable, the information relative to the non-conservative component of the generalized force Q_φ that is the kf friction coefficient, the speed reference of the trolley $V_{\text{Set\_Trolley}}$ and of the crane translation $V_{\text{Set\_Transl}}$ from the operator or from the controller. After that, the estimator module 11 computes the speed profile to be supplied to the motors in order to obtain the anti-sway functionality.

The estimator module was practically realized with a Plc. In fact, in order to verify the theoretical results of the investigation, the whole system of governing equation was simulated in Codesys V3.5 SP7. That is because Codesys, as a structured language, is actually the most common way to realize function blocks in industrial motion control environment.

The cyclic task, where the function block of the used Plc was realized, had a time of updating equal to 30ms.

The function blocks (FB's) used were two: the first FB computes the speed profile necessary to obtain the anti-sway functionality and the corresponding actual sway angle, and the second FB is used to compute the actual length l of the cable and the corresponding vertical speed using the data coming from an external unit (i.e., an encoder) connected to the motor of the corresponding movement.

The speed reference is the target value to which the speed must reach, defined from the crane operator or from the automatic control. Usually, it is defined in Hz, as a consequence of the way in which the electric motors are defined. At the speed in Hz on the fast shaft (that is on the motor) a velocity corresponds on the slow shaft (that is on the wheels moving on the rails of the crane) depending on the reduction gearing from the motor to the wheel. Typically, the max speed of a crane can be from 0.2m/s to also more 2m/s.

The ramp set is the value of the time that would be necessary for the linear motor ramp to reach the speed reference. Based on the speed reference, the estimator module computes the real speed profile in order to have the anti-sway effect. Obviously, the speed profile is longer of the linear ramp set.

The cable length has a very important influence on the speed profile because the greater the height is (and so the cable length), the greater the time of the speed profile is. In general, it is very important to reduce the time of the speed profile in order to obtain fast answer to the commands of the crane movement.

In Figures 6 and 7, corresponding the first one to a command from 0Hz speed to 50Hz speed and the second one to a stop command, the differences in the length of the ramps with respect to the previous Figures 4 and 5 are highlighted. This greater length of the ramps is due to the greater cable length that, as described above, generates, as a result, a greater time to reach the speed target obtaining the anti-sway effect.

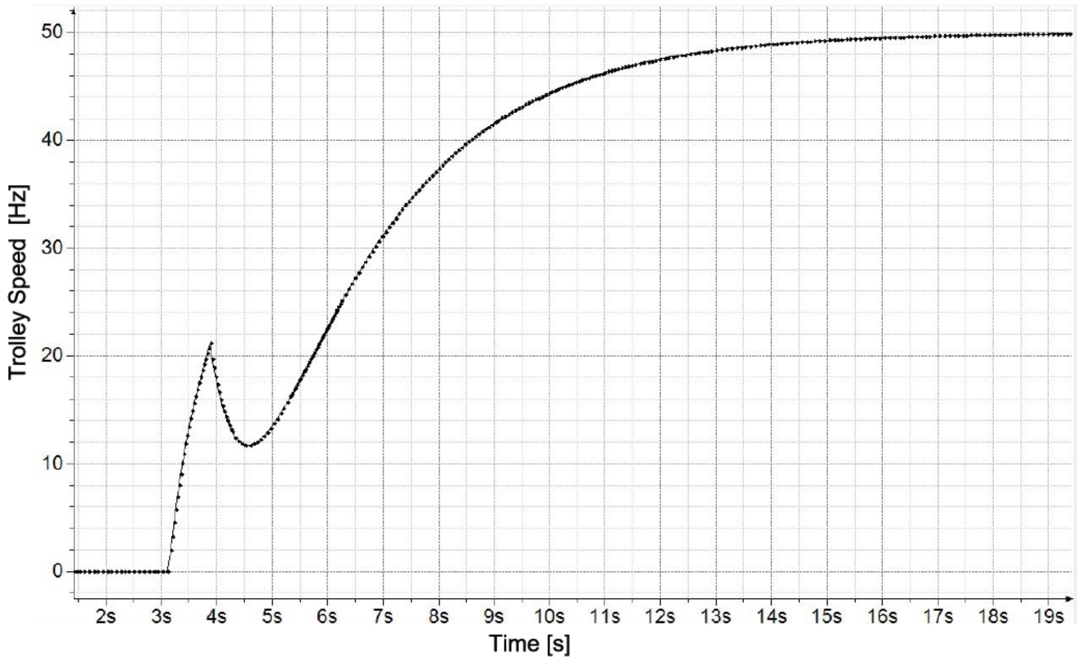

FIGURE 4. A graph relative to the speed profile of the considered movement (translation or trolley) to be supplied to the motor. The specific values of the inputs received from the estimator module are: speed reference = 50Hz (corresponding to 1.0m/s), ramp set = 1.5s, cable length = 11.5m. The cycle time of the realized task in the Plc was 30ms.

As above described and as possible to see in Figure 8, as a consequence of being an iterative method, in the present work it is possible to obtain the total damping of the sway changing many times the velocity set, also if the previous damping control of the oscillations is again in progress. In other words, it is not necessary that the target velocity has been reached in order to carry out a subsequent command towards a subsequent target velocity. On the contrary, the method to control the sway defined "Input Shaping" is not able to carry out this functionality. For example, in the work of Liu and Cheng [17], this method is well described and it is evident that it is necessary to finish the actual speed profile before to start with a subsequent command; otherwise the risk is not to get the correct attenuation.

Therefore, this feature makes this new method extremely useful in the manual control of the crane by an operator.

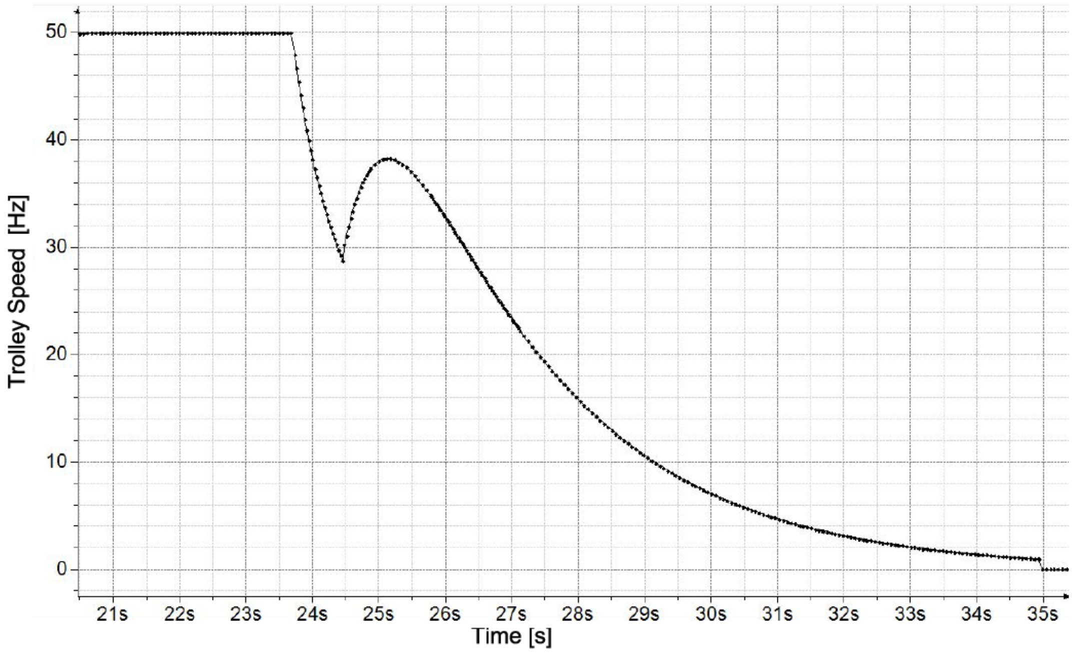

FIGURE 5. A graph relative to the speed profile of the considered movement (translation or trolley). The specific values are: speed reference = 0m/s starting from the speed of 50Hz (corresponding to 1.0m/s), ramp set = 1.5s, cable length = 11.5m, cycle time of the Plc task = 30ms. This speed profile is obtained in correspondence to a stop command of the crane movement.

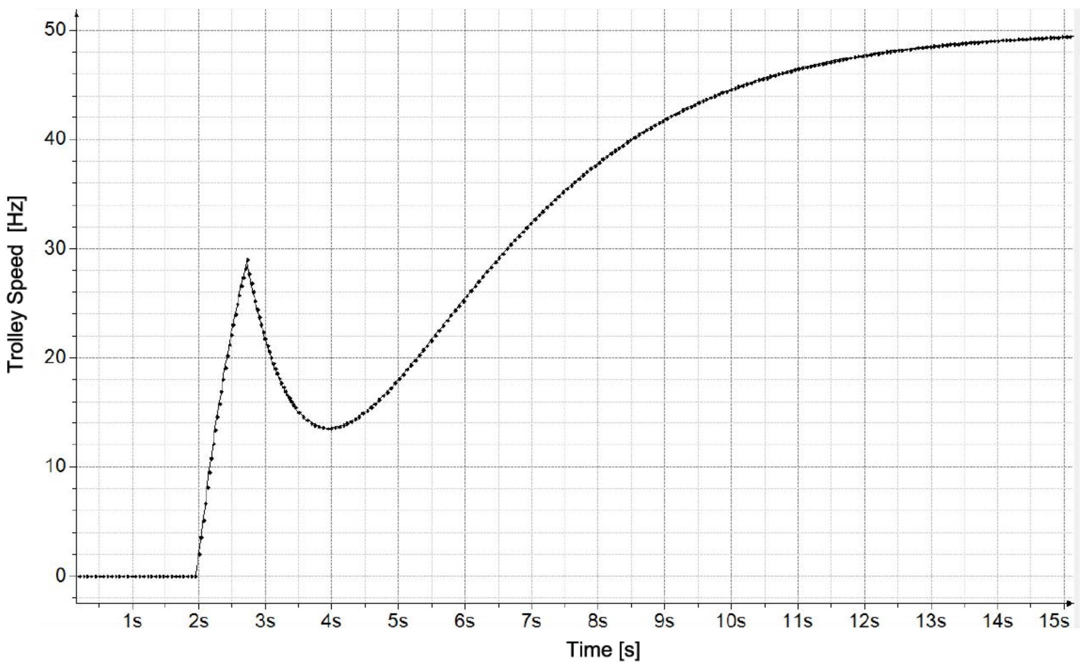

FIGURE 6. A graph relative to the speed profile of the considered movement (translation or trolley). The specific values are: speed reference = 50Hz (corresponding to 1.0m/s), ramp set = 1.5s, cable length = 25m, cycle time of the Plc = 30ms.

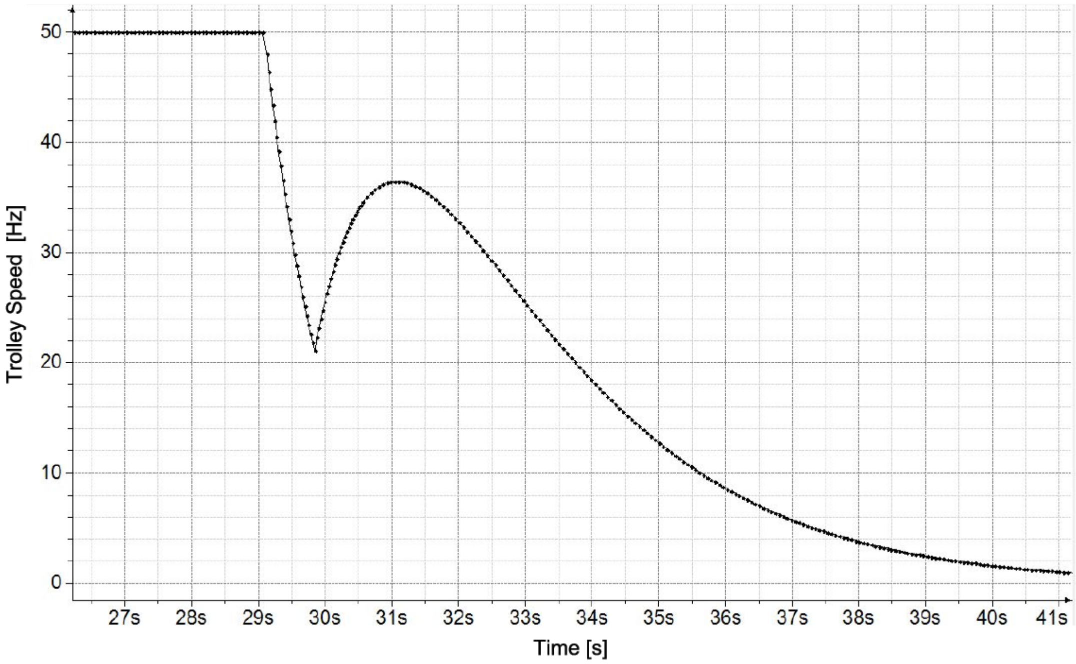

FIGURE 7. A graph relative to the speed profile of the considered movement (translation or trolley). The specific values are: speed reference = 0m/s starting from the speed of 50Hz (corresponding to 1.0m/s), ramp set = 1.5s, cable length = 20m, cycle time of the Plc = 30ms.

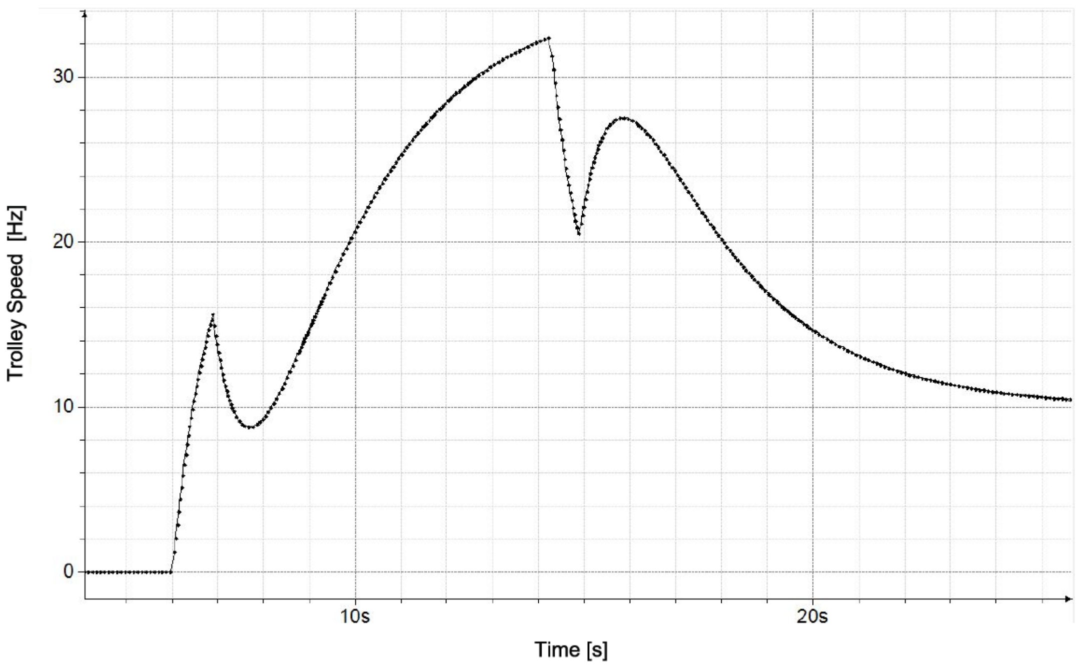

FIGURE 8. A graph relative to the speed profile of the considered movement (translation or trolley) when the speed profile in increasing is stopped, at a certain time, by a command to go to an inferior velocity. The specific values are: speed reference = 35Hz (corresponding to 0.7m/s), stopped during the increasing of the velocity for going to 10Hz (0.2m/s), ramp set = 2.5s, cable length = 15m, cycle time of the Plc = 30ms.

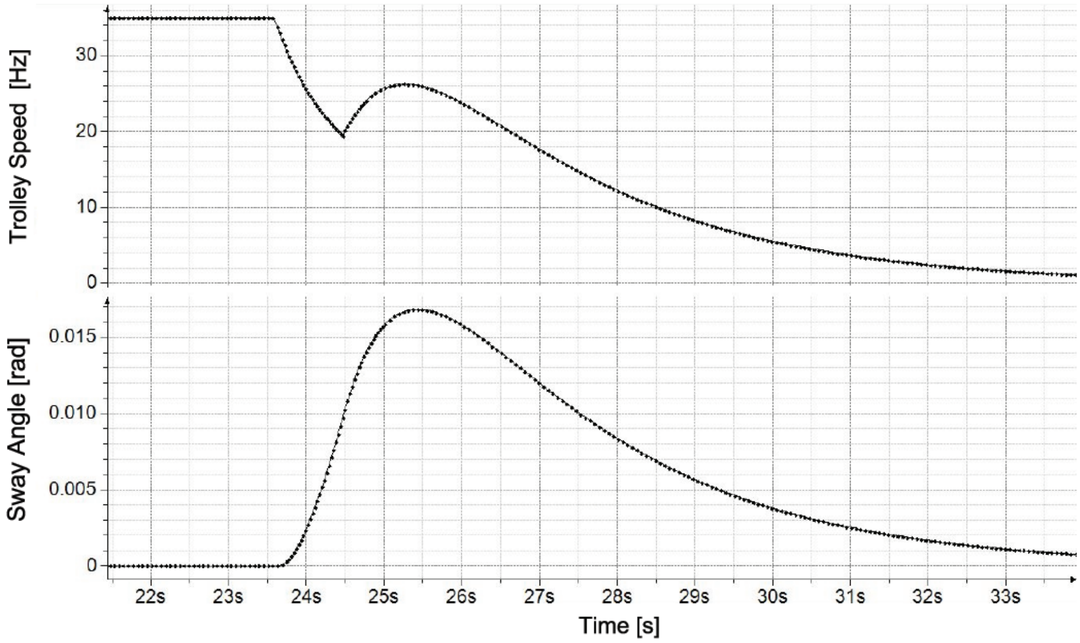

FIGURE 9. A graph relative to the speed profile of the considered movement (translation or trolley) and the corresponding profile of the sway angle φ. The specific values are: speed reference = 0m/s starting from the speed of 35Hz (corresponding to 0.7m/s), ramp set = 2.5s, cable length = 15m, cycle time of the Plc = 30ms.

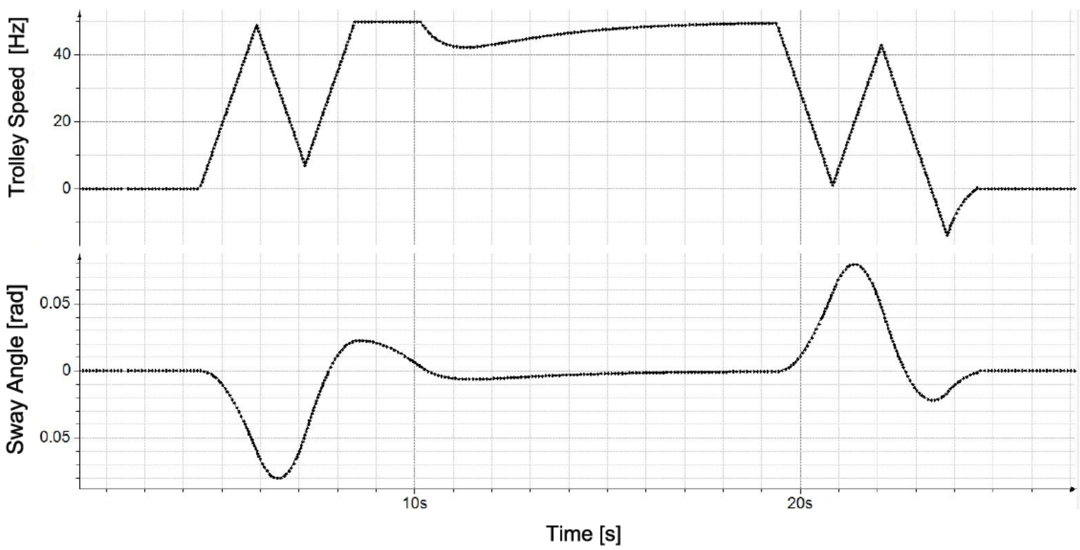

FIGURE 10. A graph relative to the speed profile of the considered movement (translation or trolley) and the corresponding profile of the sway angle φ. The specific values are: initial speed reference = 50Hz (corresponding to 1.0m/s), final speed reference = 0m/s, ramp set = 1.5s, cable length = 11.5m, cycle time of the Plc = 30ms.

In Figure 9, it is well highlighted, together with the speed profile, the sway profile, wherein the relative maximum of the speed profile corresponds to the relative maximum of the sway profile.

It is very important to observe in Figure 10 that, using in the function block different parameters for the profiles, the same profiles for the velocity can be very fast, particularly corresponding to a very fast stop with anti-sway. This is a very important result in order to control the crane with the high performance of the answer to the operator command. It is possible to note that, in order to obtain the wanted fast stop profile, the Trolley speed has to take little negative values. That corresponds to little slow movements in opposite direction to that of the commanded movement, necessary so that the corresponding sway angle profile reduces quickly to zero (as it is possible to see in Figure 10).

4. **Conclusions.** In this paper, the sway control for a gantry or overhead crane was investigated. It was obtained considering multiple ropes. A solution for the effective non-linear equations of motion is obtained defining an iterative method for the solution of the movement equation.

It was considered also the cable flexibility and stiffness that cause a sensible variation of the period time for the pendulum relative to the oscillations of the payload.

Besides, in the present work are considered also the continuous variation of the hoisting height for the suspension point of the payload and the variation of the corresponding velocity and acceleration.

A fundamental point is the novel computation of the exact period for the oscillation of the pendulum.

Another very important practical point is the obtained profile of the velocity with the activated functionality of the very fast stop because that allows fast control by the operator in case of manual control.

Possible future developments of this work may concern two fundamental aspects. On one side, it should be important to obtain a "command smoothing" solution also for slewing and tower crane, where the coupling between the translational and rotational motions involves a combined and complex payload sway in the radial and tangential directions. On the other side, the effects of external disturbances such as wind are to be studied because they have significant consequences on the crane's performance.

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