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AN IDENTIFICATION-BASED CONTROL METHOD FOR AN OVERHEAD CRANE WITH A NEW COMBINED SENSOR PLACEMENT

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Abstract. This paper is devoted to an automatic control method for an overhead crane under the current parametric uncertainty of the crane, transported cargo, and exogenous disturbances. The control objective is to move the cargo in the horizontal plane to a point ensuring the final delivery of the cargo to the designated place while parrying the angular oscillations of the suspension and reaching given dynamic characteristics. The approach is based on a control scheme with a current parametric identification algorithm, an implicit reference model, and "simplified" adaptability conditions to track cargo movements directly. The control law generates a given trolley speed for the servo drive. The passport data of the crane installation are used to select the control law parameters. Unlike previous publications on the topic, the solution proposed below is simpler, more reliable in terms of operation, and less expensive. This is achieved by placing a combined sensor (an accelerometer with an angular rate sensor (ARS)) on a suspension cable near the crane trolley and applying, first, an algorithmic solution without the preliminary calculation of the ARS drift and, second, a current parametric identification procedure of higher efficiency. Computer simulation results are provided to confirm these advantages of the new solution. A similar example is implemented on an experimental installation.

Keywords: overhead crane, control automation, current parametric uncertainty, parametric identification algorithm, approximation.

INTRODUCTION

During the operation of different types of cranes, including overhead ones, the automatic transfer of suspended cargos is a topical problem. This is due to the high prevalence of such cranes and the need to damp pendulum oscillations, which reduce safety, productivity, the accuracy of work, etc. It is especially important to implement automation under the current parametric uncertainty of the crane installation, transported cargo, and exogenous disturbances, caused by a wide variety of crane operating modes.

There exist many automatic control methods for overhead cranes, which solve this problem in different formulations:

• approaches based on PID, PI, and PD controllers (for example, see [1–3]). These methods suffer from the drawback that the controllers have to be pre-tuned for a particular range of crane and load parameters.

• sliding mode control methods for overhead cranes (for example, see [4, 5]). Their main shortcoming is the appearance of high-frequency components in drive control, which worsens the performance characteristics of the control system.

• neural network controllers for overhead cranes [6, 7]. They are disadvantageous due to a considerable training time of the neural network.

• an adaptive control method for an overhead crane trolley that involves a Lyapunov function with the tuning of controller parameters based on the gradient identification algorithm [8, 9]. The main disadvantage of this method is the problem of selecting the parameters of the discrete identification algorithm for a particular case to make the closed-loop control system stable.

Another control approach for an overhead crane under current parametric uncertainty was considered in [10, 11]. It is based on an adaptive control scheme with a parametric identifier, an implicit reference model, and “simplified” adaptability conditions. It was proposed to use an asynchronous servo drive, which is fast enough in the full range of working loads [12]. The control law generates a given speed for the actuators to track linear cargo movements directly. This paper is a logical continuation of the studies mentioned to simplify the control system design and reduce the operating costs of an overhead crane by improving the control approach.

1. PROBLEM STATEMENT

We will consider the motion of the crane and the cargo only along one axis (the horizontal movements of the cargo) corresponding to the movements of the crane trolley; see the diagram in Fig. 1. (The other axis corresponding to the movements of the crane beam can be treated by analogy.)

Figure 1 has the following notations:

m_T and m_c are the masses of the crane trolley and transported cargo, respectively, considering the inertia of the rotating masses (the cargo consists of the load transferred and the gripping device, i.e., a hook);

r_c is the radius of the cargo inertia;

l is the length of the cargo suspension (the distance between the attachment point of the suspension on the trolley and the cargo’s center of gravity); by assumption, $l_{\min} \le l \le l_{\max}$, where $l_{\min}$ and $l_{\max}$ are the minimum and maximum values of the suspension length, respectively;

x_T is the horizontal movement of the trolley along the Ox axis;

$v = \dot{x}_T$ is the trolley speed with the restriction $|v| \le v_{\max}$, and v_{giv} is a given value of this speed;

f_{con} is the control force generated by the servo drive of the crane trolley using the signal v_{giv};

f_{fri} is the friction force counteracting trolley movements;

f_w is the wind force applied to the center of the cargo mass (the wind load);

φ is the deviation angle of the cargo suspension from the vertical axis;

$x_c = x_T + l \sin\varphi$ is the horizontal movement of the cargo;

S is the location point of a combined sensor (an accelerometer and an angular rate sensor (ARS)) on the suspension cable (see below);

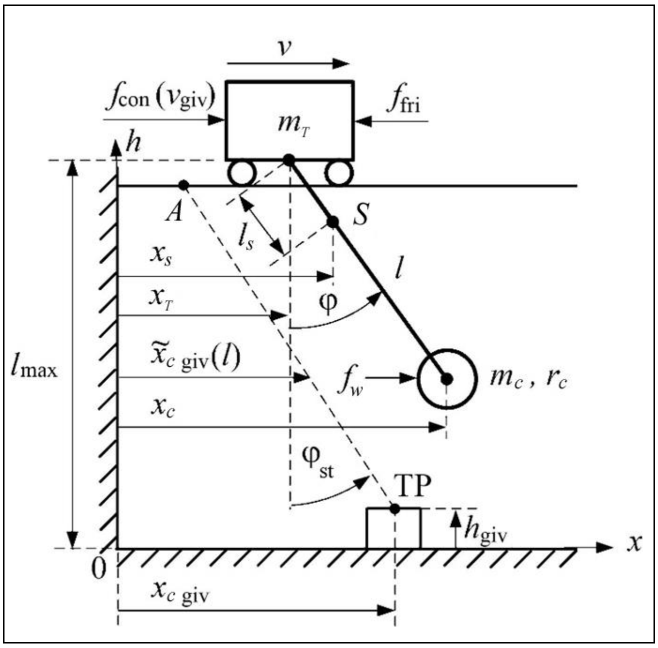

Fig. 1. Overhead crane movements with a cargo along the horizontal axis corresponding to crane trolley movements.

l_s is the suspension length to the combined sensor (the distance between the attachment point of the suspension on the trolley and point S), $0 < l_s < l_{\min}$

$x_s = x_T + l_s \sin\varphi$ is the horizontal movement of the combined sensor;

TP is the target point of cargo delivery with coordinates $(x_{C\ \text{giv}},\ h_{\text{giv}})$, where $x_{C\ \text{giv}}$ and h_{giv} (loading height) are its coordinates along the horizontal Ox and vertical Oh axes, respectively;

$\varphi_{\text{st}} = \operatorname{arctg}(f_w/(m_C g))$ is the steady-state constant value of the angle φ expected at the end of the control process, where g is the free fall acceleration;

$\tilde{x}_{c\ \text{giv}}(l) = x_{c\ \text{giv}} - (l_{\max} - h_{\text{giv}} - l)\sin\varphi_{\text{st}}$ is a given position of the cargo on the horizontal axis at the current suspension length (the distance between the zero point and the A–TP line in the Ox axis.

Assume that the exogenous disturbance formed by the friction force and wind load is a step signal with limited amplitude.

The mathematical model of the mechanical system (Fig. 1) was described in [13], with the cable mass and angular motion friction neglected due to their smallness. Based on this model and the initial values of the variables characterizing position and velocity, by applying Poinsot’s theorem (to include the wind effect), we obtain the following differential equations for the dynamics of this mechanical system:

$$\begin{cases}(m_T+m_c)\ddot{x}_T+(m_c l\cos\varphi)\ddot{\varphi}\\ \quad = f_{\text{con}}-f_{\text{fri}}+f_w+m_c l\dot{\varphi}^2\sin\varphi\\ (m_c l\cos\varphi)\ddot{x}_T+m_c\left(l^2+r_c^2\right)\ddot{\varphi}\\ \quad =-m_c gl\sin\varphi+lf_w\cos\varphi\\ x_c=x_T+l\sin\varphi.\end{cases} \tag{1}$$

Usually, the angle φ has a small range (not exceeds $10-20^\circ$), and its rate of change is also low. In view of the motion kinematics, we can take $\sin\varphi\approx\varphi$, $\cos\varphi\approx 1$, and $\dot{\varphi}^2\sin\varphi\cong 0$. Then system (1) can be linearized to

$$\begin{cases}\ddot{x}_T\approx a_x^{f_{\text{con}}}f_{\text{con}}+a_x^{\varphi}\varphi+a_x\\ \ddot{\varphi}\approx a_{\varphi}^{f_{\text{con}}}f_{\text{con}}+a_{\varphi}^{\varphi}\varphi+a_{\varphi}\\ x_c\approx x_T+l\varphi,\ \dot{x}_c\approx\dot{x}_T+l\dot{\varphi},\ \ddot{x}_c\approx\ddot{x}_T+l\ddot{\varphi},\end{cases} \tag{2}$$

where $a_x^{f_{\text{con}}}=\gamma^{-1}m_c\left(l^2+r_c^2\right)$, $a_x^{\varphi}=\gamma^{-1}g\left(m_c l\right)^2$, $a_x=\gamma^{-1}m_c\left[-\left(l^2+r_c^2\right)f_{\text{fri}}+l^2 f_w\right]$, $a_{\varphi}^{f_{\text{con}}}=-\gamma^{-1}m_c l$, $a_{\varphi}^{\varphi}=-\gamma^{-1}\left(m_T+m_c\right)m_c gl$, $a_{\varphi}=\gamma^{-1}l\left[m_c f_{\text{fri}}+\left(m_T+m_c\right)f_w\right]$, and $\gamma=m_c\left[m_T l^2+\left(m_T+m_c\right)r_c^2\right]$.

We rewrite the third equation of the third line of system (2) using its first two lines and substitute the signal f_{con} expressed from the first equality of (2) into the resulting formula. Consequently, the cargo motion is described through the trolley speed as follows:

$$\ddot{x}_c\approx a_c^{\dot{v}}\dot{v}+a_c^{\varphi}\varphi+a_c, \tag{3}$$

where $a_c^{\dot{v}}=r_c^2\big/\left(r_c^2+l^2\right)$; $a_c^{\varphi}=l\left(a_{\varphi}^{\varphi}-a_x^{\varphi}a_{\varphi}^{f_{\text{con}}}\big/a_x^{f_{\text{con}}}\right)=-g\mu$; $\mu=l^2\big/\left(l^2+r_c^2\right)$ is the dimensionless coefficient of the impact of the cargo's radius of inertia; $a_c=l\left(a_{\varphi}-a_x a_{\varphi}^{f_{\text{con}}}\big/a_x^{f_{\text{con}}}\right)$ is the value equal to the cargo acceleration due to exogenous disturbances; by assumption, the values of the variables x_c and $\dot{x}_c$ at the initial time instant are known.

For further considerations, we suppose that $l^2>>r_c^2$, which holds in most practical cases. Then the parameter a_c depends only on the wind and $a_c^{\dot{v}}\approx 0$. Hence, equation (3) can be written as

$$\ddot{x}_c\approx a_c^{\varphi}\varphi+a_c\approx -g\varphi+a_c; \tag{4}$$

here the parameter a_c represents the acceleration generated by the wind force alone.

According to this equality, under the above assumptions, the linear movement of the cargo is mainly affected only by the angular position of the suspension and the wind load. We will use equation (4) as a controlled object. Despite its substantially simplified form valid only under the above assumptions, this equation will be employed to design a control law (an approximation of the linear movement of the cargo), similar to the approach described in [11], albeit with some modification.

The natural frequency ω_0 of the angular oscillations of the cargo was found in [11] based on the parameters in Fig. 1 and equality (4). It satisfies the relations

$$\omega_0\approx\sqrt{-a_c^{\varphi}/l}=\sqrt{\mu g/l}\approx\sqrt{g/l}\,;\omega_0\geq\sqrt{g/l_{\max}}. \tag{5}$$

Given the crane passport data $l_{\min}$, $l_{\max}$, $v_{\max}$, the coordinates of the cargo's target point $x_{c\,\text{giv}}$, h_{giv}, and the initial positions of the trolley and cargo on the horizontal axis (they are considered to be known), we pose the following problem: under the current parametric uncertainty of the crane, cargo, and exogenous disturbances (with unknown onset times and intensities), it is required to design a control law generating a given speed of the crane trolley to be executed by the servo drive so that the cargo's horizontal movement will satisfy the conditions

$$\dot{\varphi}\to 0;\quad x_c\to\tilde{x}_{c\,\text{giv}}(l). \tag{6}$$

If $\varphi_{\text{st}}\neq 0$, the second condition in (6) requires positioning of the cargo not over the target point but so that, by the end of the control process, the cargo will be delivered to the target point by lowering without additional movements of the trolley (cargo movement along the A–TP line). In addition, conditions (6) must be fulfilled with motion dynamics close to the required one considering the speed characteristics of the servo drive. Modern asynchronous servo drives are fast enough: they have signal processing delays at a level of hundredths and tenths of a second. Therefore, we suppose that $\dot{x}_T\cong v_{\text{giv}}$ [12].

Let the main data about the suspension and cargo be provided by the combined sensor (an accelerometer and an ARS) placed on the suspension cable (similarly to the patent [3]) close to the crane trolley. See point *S* in Fig. 1 and a more detailed diagram in Fig. 2.

This placement of the combined sensor on the cable is advantageous over its location on the hook with remote information transmission proposed in [11]:

– It is possible to organize wired information transmission from the sensor and wired power supply for the sensor, a much more reliable and simple solution with reduced operating costs.

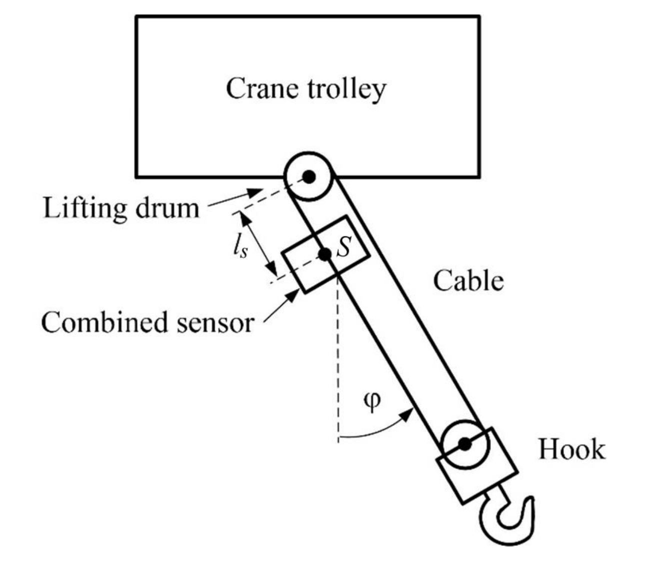

Fig. 2. The combined sensor placement on the suspension cable.

– The sensor is located in a safe place without exposure to external mechanical impacts.

– It is easy to fix the sensor sensitivity axes. (The suspension hook often rotates around the vertical axis.)

– The data obtained from this sensor are universal regardless of the kinematic scheme of the lifting mechanism [14].

– The cargo's additional secondary high-frequency motions (including the unmodeled ones) associated with the double-pendulum motions of the hook and cargo are transferred to the sensor to a lesser extent.

2. CONTROL ALGORITHM

To achieve conditions (6) with a given speed of the crane trolley, we apply the approach [11] based on the control scheme with a current parametric identification algorithm, an implicit reference model, and "simplified" adaptability conditions to track the cargo movement directly. Within this approach, the implicit reference model is an oscillating link with the values of the variables at the initial time instant t_0 equal to those of the original object:

$$\begin{gathered}\ddot{x}_m = -2\omega_m \xi_m \dot{x}_m - \omega_m^2 \left(x_m - \tilde{x}_{c\,\text{giv}}(l)\right),\\ x_m(t_0) = x_c(t_0), \quad \dot{x}_m(t_0) = \dot{x}_c(t_0),\end{gathered} \tag{7}$$

where x_m is the variable describing the reference dynamics of the cargo motion (corresponding to the variable x_c); ω_m and ξ_m are the reference values of the natural frequency and relative damping coefficient, respectively, representing assigned parameters of the reference model [15].

First, let the values of the parameters a_c^{φ}, a_c and the variables φ, φ_{st} be known. We equate the right-hand sides of equation (4) and the first equality in (7), replacing x_m and $\dot{x}_m$ by x_c and $\dot{x}_T$, respectively, to find the control law generating the given speed v_{giv} of the crane trolley:

$$\begin{aligned}\dot{x}_T \approx v_{\text{giv}} &= T_x^{-1}\left[\left(\tilde{x}_{c\,\text{giv}}(l) - x_c\right) - \omega_m^{-2}\left(a_c^{\varphi}\varphi + a_c\right)\right]\\ &= T_x^{-1}\left(\tilde{x}_{c\,\text{giv}}(l) - x_c\right) - 0.25 T_x \xi_m^{-2}\left(a_c^{\varphi}\varphi + a_c\right),\end{aligned} \tag{8}$$

where $T_x = 2\xi_m/\omega_m$ is the time constant of the linear movement of the trolley and cargo (the control time constant), set by the reference model.

Indeed, based on equation (4), $\ddot{x}_c$ can be substituted in formula (8) instead of the term $(a_c^{\varphi}\varphi + a_c)$. Then, in view of the notations introduced in (7), we arrive at the equality describing the cargo's behavior in the closed-loop control system with the control law (8):

$$\ddot{x}_c \approx -2\omega_m \xi_m \dot{x}_T - \omega_m^2\left(x_c - \tilde{x}_{c\,\text{giv}}(l)\right). \tag{9}$$

It matches the assigned reference (7), except that the variable $\dot{x}_c$ is replaced by $\dot{x}_T$ to eliminate internal instability in the closed-loop control system.

According to Fig. 1, the tracking error of the cargo position can be represented as

$$\tilde{x}_{c\,\text{giv}}(l) - x_c = x_{T\,\text{giv}} - x_T, \tag{10}$$

where $x_{T\,\text{giv}} = \tilde{x}_{c\,\text{giv}}(0) = x_{c\,\text{giv}} - \left(l_{\max} - h_{\text{giv}}\right)\varphi_{\text{st}}$.

Therefore, we write equality (8) in the form of the dynamics equation of the closed-loop control system:

$$T_x \dot{x}_T + x_T \approx x_{T\,\text{giv}} - \omega_m^{-2}\left(a_c^{\varphi}\varphi + a_c\right). \tag{11}$$

In the absence of angular motion $(\varphi \equiv 0)$ and disturbances $(a_c \equiv 0)$, it follows that $x_T \to x_{T\,\text{giv}}$ and hence $x_c \to \tilde{x}_{c\,\text{giv}}(l)$ by the aperiodic law with the time constant T_x (which explains the name of this parameter).

Due to formulas (4) and (11), if a steady state $\dot{x}_T \to 0, \dot{\varphi} \to 0, \dot{x}_c \to 0, \ddot{x}_c \to 0$ is reached in the closed-loop control system, it can only be the case when $\varphi \to \varphi_{\text{st}} = -a_c/a_c^{\varphi}$ and $x_T \to x_{T\,\text{giv}}$. As a result, by the expression (10), $x_c \to \tilde{x}_{c\,\text{giv}}(l)$.

For the closed-loop control system (2), (4), (8) with $\omega_m < \omega_0$, we have $\dot{x}_T \to 0, \dot{\varphi} \to 0, \dot{x}_c \to 0$, $\ddot{x}_c \to 0$ as $t \to +\infty$; this fact was established in [11]. Hence, the control objective (6) is achieved.

In crane operations, the obvious desired movement of the cargo to a given point is the process with the minimum possible control time and overshoot. As is known, for the oscillating link, this requirement corresponds to the values of the relative damping coefficient not less than 0.71 [15]. Also, the parameters ω_m and T_x in (8) should be chosen depending on the maximum speed of the drive and the required movement of the cargo. Due to these provisions, the control law (8), and equality (11) without the last term (only the aperiodic process and its maximum speed are considered [15]), we write the requirements for these parameters in the following form:

$$0.71 \le \xi_m < 1, \quad \omega_m = \frac{2\xi_m v_{\max}}{k_{\mathrm{tr}} \Delta x_c} < \omega_0;$$
$$T_x \triangleq T_x(\Delta x_c) = k_{\mathrm{tr}} \frac{\Delta x_c}{v_{\max}} > \frac{2\xi_m}{\omega_0} = \frac{\xi_m T_0}{\pi} \overset{\xi_m = 0.8}{\approx} \frac{T_0}{4}, \tag{12}$$

where k_{tr} is the positive coefficient of change of the transient time, chosen considering inequalities (12); $\Delta x_c = \left| x_{c\,\mathrm{giv}} - x_{c_0} \right|$ is the cargo transfer distance, where x_{c_0} is the cargo's initial position on the Ox axis; $T_x(\Delta x_c)$ is the dependence of the parameter T_x on Δx_c (further denoted by T_x under a fixed value of the argument); T_0 is the period of natural pendulum oscillations of the cargo suspension.

Let us explain the choice of the coefficient k_{tr}. If $k_{\mathrm{tr}} \ge 1$, we have $|v| \le v_{\max}$ in the transients; the higher the value of this coefficient is, the greater the control time will be. Under $k_{\mathrm{tr}} < 1$, the transient time will decrease but the trolley speed can reach the value $v_{\max}$; at such time instants, there is no oscillation-damping control, which (of course) worsens the quality of control.

In view of the relations (5), the third inequality in (12) will hold if

$$T_x(\Delta x_c) = k_{\mathrm{tr}} \frac{\Delta x_c}{v_{\max}} > \frac{2\xi_m}{\sqrt{g}} \sqrt{l_{\max}} \overset{\xi_m = 0.8}{\approx} \sqrt{l_{\max}} / 2, \tag{13}$$

where the units of measurement correspond to the SI system.

Figure 3 shows the admissible domain of the parameter T_x (13) depending on $l_{\max}$ in the case $\xi_m = 1$ (the upper unshaded area). The boundary in this figure represents the minimum value of T_x.

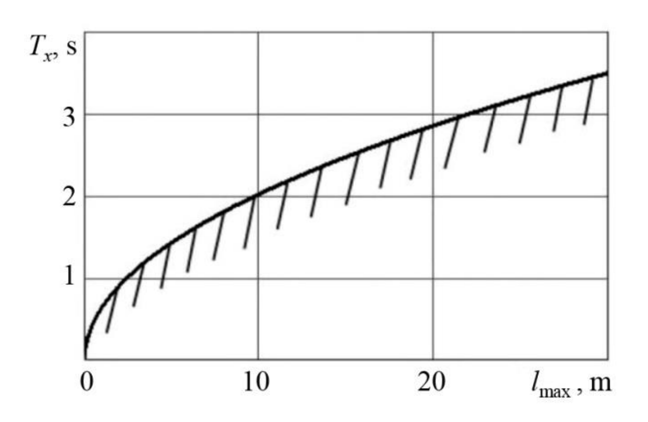

Fig. 3. The admissible domain of the parameter T_x of the control law (8) depending on the maximum length of the crane suspension under the unit relative damping coefficient of the reference model (the upper unshaded area).

Hence, the expression (13) can be used to determine the value of the parameter T_x for the control law (8) from the passport data $l_{\max}$, $v_{\max}$ of the crane installation and the cargo transfer distance Δx_c, which are known in advance. Also, this parameter can be refined based on the period of the natural oscillations of the suspension using inequality (12).

The relations (12) and (13) reflect the stability bounds of the parameter T_x in the control law (8). Obviously, the further its value lies from the bounds (i.e., the greater the value of T_x is), the more stable the closed-loop control system will be. However, the transients become longer.

Consider an example of choosing the value of the parameter T_x based on these relations. Let $l_{\max} = 10$ m, $v_{\max} = 1$ m/s, $\Delta x_c = 8$ m, $\xi_m = 0.8$, and $k_{\mathrm{tr}} = 1$. Then, by formula (13), we obtain $T_x > 1.6$ s and $T_x = 8$ s. Due to equation (11), this solution ensures the control time $t_{\mathrm{con}} \approx 3T_x = 24$ s to transfer the cargo to the given point [15]. (By assumption, the pendulum oscillations will be damped by the end of the transients.) When the cargo transfer distance is reduced, the values of these parameters decrease: $T_x \ge 1.6$ s, $t_{\mathrm{con}} \ge 4.8$ s. These performance indicators agree with the time standards [16], e.g., for medium overhead cranes with a lifting capacity of up to 50 tons. Such cranes will be considered in the model example below.

The control law (8) generating the given speed is based on the exact value of the expression ($a_c^{\varphi}\varphi + a_c$), i.e., the parameters a_c^{φ}, a_c of the controlled object (4). With various types of transferred cargos, suspension lengths, and exogenous disturbances, the values of these parameters are often unknown in practice. Another problem is the exact determination of the angle φ. The variable $\ddot{x}_c$ can be used in the control law (8) instead of $\left(a_c^{\varphi}\varphi + a_c\right)$. However, this acceleration cannot be directly measured using an accelerometer placed in the cargo's center of gravity: the accelerometer measures the sum of the apparent acceleration (generated by the support reaction due to gravity) and the desired acceleration. Application of the dependence $\ddot{x}_c \approx \ddot{x}_T + l\ddot{\varphi}$ from system (2) requires knowledge of the suspension length l and, moreover, results in a highly noisy signal due to the accelerations.

To simplify the crane control system, we propose replacing the cargo's linear acceleration in the control law (8) with the linear acceleration $\ddot{x}_s$ of the combined sensor placed at point *S* at the distance l_s from the trolley $\left(0 < l_s < l_{\min}\right)$; see Figs. 1 and 2. Similar to [11], we will use not the signal of this acceleration but its approximation based on the current parametric identification.

According to the above assumptions and formulas (2) and (4), the linear acceleration of the cargo is $\ddot{x}_c \approx \ddot{x}_T + l\ddot{\varphi} \approx a_c^{\varphi}\varphi + a_c \approx -g\varphi + a_c$. Obviously, the same acceleration for the combined sensor is equal to $\ddot{x}_s \approx \ddot{x}_T + l_s\ddot{\varphi}$. By comparing these dependencies, we easily find

$$\begin{aligned}\ddot{x}_s &\approx \ddot{x}_T + l_s\ddot{\varphi} \approx \ddot{x}_c - \left(l - l_s\right)\ddot{\varphi} \\ &\approx -g\varphi + a_c - \left(l - l_s\right)\ddot{\varphi}.\end{aligned} \qquad (14)$$

In view of this equality and formula (10), we write the control law (8) by replacing the variable x_c with x_s:

$$\begin{aligned}\dot{x}_T &\approx v_{\text{giv}} = T_x^{-1}\left[\left(x_{s\,\text{giv}} - x_s\right) - \omega_m^{-2}\,\ddot{x}_s\right] \\ &\approx T_x^{-1}\left(x_{c\,\text{giv}} - \left(l_{\max} - h_{\text{giv}}\right)\varphi_{\text{st}} - x_T\right) \\ &\quad -0.25T_x\xi_m^{-2}\left(-g\varphi + a_c - \left(l - l_s\right)\ddot{\varphi}\right),\end{aligned} \qquad (15)$$

where $x_{s\,\text{giv}} = \tilde{x}_{c\,\text{giv}}(l_s) = x_{c\,\text{giv}} - \left(l_{\max} - h_{\text{giv}} - l_s\right)\varphi_{\text{st}}$.

The first part of (15) directly implies the equation describing the horizontal movement of the combined sensor; it is analogous to equality (9), where the variable x_s should be substituted for x_c. Making the same substitution in the proof from [11] under the conditions of achieving the control objective will give the same conclusions for the linear movement of the combined sensor. Recall that this sensor is placed between the attachment point of the suspension on the trolley and the cargo's center of gravity on their line. Hence, based on formula (10), the control law (15) is similar to the expression (8).

To implement the control law (15) generating the given speed, in particular, it is necessary to determine the current values of the variable φ_{st} and the approximated value of $\ddot{x}_s$. Consider some possible ways to do it.

The value $\varphi_{\text{st}} \neq 0$ can be caused only by the wind effect. If an accelerometer is placed at point *S* to measure the sum of the acceleration of the support reaction due to gravity and $\ddot{x}_s$, its value will be given by $\ddot{x}_s^{\text{acc}} \approx g\varphi + \ddot{x}_s$ due to formula (14). According to the stated above, we have $\ddot{x}_s \cong 0$ for the steady-state process in the closed-loop control system. Therefore,

$$\widehat{\varphi}_{\text{st}} \approx \left(\ddot{x}_s^{\text{acc}}/g\right)_{\text{lp}}, \qquad (16)$$

where the upper cap "∩" indicates an estimate; $\ddot{x}_s^{\text{acc}}$ is the readings of the accelerometer placed at point *S* whose sensitivity axis is orthogonal to the suspension cable; the subscript "lp" denotes low-pass filtering to eliminate the transient components.

To approximate $\ddot{x}_s$, we will use the integral of the signal from an ARS placed at point *S*. Due to the known drift of the sensor, this procedure will generate the dependence

$$\int_t \omega_{\text{ARS}}dt \approx \varphi + \Delta\omega_{\text{dr}}t + \varphi_0, \qquad (17)$$

where ω_{ARS} is the data from the ARS with the sensitivity axis parallel to the suspension rotation axis; $\Delta\omega_{\text{dr}} \approx \text{const}$ is the drift of this sensor; φ_0 is the initial value of the angle φ.

We build an approximating current parametric identification algorithm based on equations (14) and (17), generating the equality

$$\begin{aligned}z &\triangleq \ddot{x}_s + g\int_t \omega_{\text{ARS}}dt = \ddot{x}_T + l_s\ddot{\varphi} + g\int_t \omega_{\text{ARS}}dt \\ &\approx \widehat{a}_1 t + \widehat{a}_2 = \widehat{\boldsymbol{\theta}}^{\mathrm{T}}\mathbf{y},\end{aligned} \qquad (18)$$

where z is the response of the identification object; $\widehat{a}_1$ and $\widehat{a}_2$ are the estimates of the parameters $a_1 = g\Delta\omega_{\text{dr}}$ and $a_2 = g\varphi_0 + a_c$, respectively; $\widehat{\boldsymbol{\theta}}^{\mathrm{T}} = \left[\widehat{a}_1,\ \widehat{a}_2\right]$ is the vector of the estimated parameters; $\mathbf{y}^{\mathrm{T}} = \left[t,\ 1\right]$ is the vector of the factor variables; finally, T denotes transpose.

For the signal $\ddot{x}_s$, the approximating estimate $\hat{\ddot{x}}_s$ is given by

$$\hat{\ddot{x}}_s = -g\int_t \omega_{\text{ARS}} dt + \hat{a}_1 t + \hat{a}_2 . \quad (19)$$

Note that the expression (18) neglects the term $(l_s - l)\ddot{\varphi}$ of the original equality (14). This is dictated by experiments with the closed-loop control system with the control law (20) generating the given speed under different parameter values: due to the rapid convergence $\ddot{\varphi} \to 0$ and the self-tuning properties of the control system, the estimate of this term has little effect on the resulting motion dynamics. Moreover, with this solution, the estimate $\hat{\ddot{x}}_s$ is ahead of $\ddot{x}_s$, making the closed-loop control system with natural delays more stable.

We apply the recurrent least-squares method with the forgetting factor [17] as a current identification algorithm:

$$\begin{cases} \hat{\boldsymbol{\theta}}_i = \hat{\boldsymbol{\theta}}_{i-1} + \mathbf{P}_i \mathbf{y}_i \varepsilon_i, \ \varepsilon_i = z_i - \hat{\boldsymbol{\theta}}_{i-1}^{\mathrm{T}} \mathbf{y}_i \\ \mathbf{P}_i = \left[\mathbf{P}_{i-1} - \mathbf{P}_{i-1} \mathbf{y}_i \mathbf{y}_i^{\mathrm{T}} \mathbf{P}_{i-1} \left(\beta + \mathbf{y}_i^{\mathrm{T}} \mathbf{P}_{i-1} \mathbf{y}_i \right)^{-1} \right] / \beta \\ \mathbf{P}_0 = \vartheta \mathbf{E}_2, \ \beta < 1, \ \beta \to 1, \end{cases} \quad (20)$$

where $i = 1, 2, 3, \ldots$ denotes the ith time instant with a step Δt; ε is the identification residual; $\mathbf{P}_i$ is the covariance matrix of the parameter estimation errors, of dimensions 2 × 2; β is the assigned forgetting factor of previous measurements to track the time-varying desired parameters; ϑ is a large positive number determining the initial rate of variation of the parameter estimates; finally, $\mathbf{E}_2$ is an identity matrix of dimensions 2 × 2.

Note that the identification algorithm (20) uses linearly independent factor variables on any time interval. This makes the resulting estimates stable, generally ensuring the stability of the identification algorithm as well [18].

Therefore, we replace the control law (15) generating the given speed with another one based on the current estimates $\hat{\varphi}_{\text{st}}$, $\hat{a}_1$, $\hat{a}_2$, and $\hat{\ddot{x}}_s$ using the expressions (16), (18)–(20):

$$\dot{x}_T \approx v_{\text{giv}} = T_x^{-1}\left(x_{c\ \text{giv}} - \left(l_{\max} - h_{\text{giv}} \right) \hat{\varphi}_{\text{st}} - x_T \right) \\ -0.25 T_x \xi_m^{-2} \left(-g\int_t \omega_{\text{ARS}} dt + \hat{a}_1 t + \hat{a}_2 \right). \quad (21)$$

Following recommendations [18], a low-pass filter should be applied to the law (21) to eliminate high-frequency components in the closed-loop system while maintaining control accuracy. Such motions may arise due to the double pendulum suspension formed by the hook and cargo in some cases [10].

For the linearly independent elements of the vector function $\mathbf{y}_i$ on a sliding time interval, the identification residual of the algorithm (20) very quickly—in a few iterations—converges to the neighborhood of zero under a sufficiently small step Δt and an appropriately chosen value of the parameter β; moreover, it will remain therein, despite that the parameter estimates may be far from the true values [18]. Thus, the algorithm (20) ensures the condition

$$\hat{\ddot{x}}_s \to \ddot{x}_s , \quad (22)$$

i.e., approximates the variable $\ddot{x}_s$ even if the current parameter estimates are inaccurate.

In other words, the estimates can be substituted into the law (21) from the very beginning of the identification algorithm. In addition, the control objective is achieved for the current parameter estimates and $\hat{\ddot{x}}_s$, as was proved in [11].

Considering this algorithmic support of the adaptive control system of the crane trolley, we propose the following information sensors and additional algorithmic operations:

– an encoder on the crane trolley or $x_T \approx \int_t v_{\text{giv}} dt + x_{T_0}$ due to the servo drive properties mentioned above, where the last term is the initial position of the trolley;

– the combined sensor (an accelerometer and an ARS) placed on the suspension cable close to the crane trolley at point *S* at the distance l_s (Fig. 3), by analogy with the patent [3]; the accelerometer is used to obtain the signal $\hat{\varphi}_{\text{st}}$ by formula (16), whereas the ARS provides the signal ω_{ARS};

– filtering on the real differentiating link of the signal $(v_{\text{giv}} + l_s \omega_{\text{ARS}})$ to form the signal $\ddot{x}_s$, used in equality (18); it gives an approximate values of the desired signal without the ARS drift effect [15].

Note that these formulas correspond to the relative smallness of the cargo's radius of inertia, . Computer simulations show that otherwise, e.g., in the case , the closed-loop control system also provides high-quality control. In particular, the reason is the good approximation properties of the identification algorithm (20), which ensures condition (22).

3. A MODEL EXAMPLE

To compare the new approach with the previously known solution [11], we modeled the proposed control system under the same conditions as in the cited paper, even with a somewhat expanded variation range of the crane and cargo parameters; for this purpose, the dependencies (1), (18)–(21) and the expressions (5)–(7), (12), (13), and (16) were used. In particular, numerical simulation was carried out in Matlab/Simulink/SimMechanics, and the differential equations were solved by the Runge–Kutta method of the fourth and fifth orders with a step of 0.01 s.

Consider control of the trolley of a typical medium crane with the following parameters: $m_T = 450\,\text{kg}$, $m_c = 100 - 50\,000\,\text{kg}$, $l_{\min} = 3\,\text{m}$, $l_{\max} = 15\,\text{m}$, $l_s = 1\,\text{m}$, $r_c = 0.2 - 5\,\text{m}$, $x_{c\,\text{giv}} = 10\,\text{m}$, and $h_{\text{giv}} = 0$. The friction force is viscous: $f_{\text{fri}} = k_{\text{fri}} v$, where $k_{\text{fri}} = 0.3\,\text{N}\cdot\text{s/m}$. The servo drive generating the speed $\dot{x}_T$ of the crane trolley according to a given value v_{giv} is described by an aperiodic link with the unit gain and a time constant of 0.1 s. It has additional nonlinearities: a time delay of 0.03 s and the output signal constraints $v_{\max} = 0.67$ m/s and $|\dot{v}| \le 3\,\text{m/s}^2$. Many of these parameters match the standard [19] and the variety of typical cargo. The speed control law (21) is filtered on an aperiodic link with the unit gain and a time constant of 1 s.

Assume that a step wind disturbance with an intensity of 5% of the cargo weight affects the cargo at the time instant 50 s, corresponding to $\varphi_{\text{st}} = 2.9^\circ$. (This disturbance is smoothed by an aperiodic link with a time constant of 1 s.)

The angular velocity and linear acceleration were measured using an MPU-6050 micromechanical sensor. The data contain the centered Gaussian noise with RMS errors of 0.1 deg/s (angular velocity) and $0.1\,\text{m/s}^2$ (acceleration), $\Delta\omega_{\text{dr}} = 0.03\,\text{s}^{-1}$ [20]. The accelerometer readings had a constant bias (the accelerometer's "zero") of $0.17\,\text{m/s}^2$, which corresponds to an accelerometer's angular setup inaccuracy of about 1°. The linear movement x_T of the trolley was determined by an encoder with similar noise with an RMS error of 0.01 m.

The identification algorithm (20) has the following parameter values: $\Delta t = 0.01$ s (the same for the law (21)), $\vartheta = 10$, and $\beta = 0.985$. In the identification algorithm, the variable $\ddot{x}_s$ is replaced by its approximate description $\ddot{x}_s(s) \approx \frac{s}{0.5s+1}\left(v_{\text{giv}}(s) + l_s\omega_{\text{ARS}}(s)\right)$, where s denotes the Laplace transform variable.

In the operating modes presented, the natural frequency of the crane (the relations (5)) varies in the range $\omega_0 = 0.8 - 1.8\,\text{s}^{-1}$ or $T_0 = 3.49 - 7.85$ s. Therefore, in view of the expressions (12) and (13), the parameter values of the law (21) are taken as $\xi_m = 0.9$, $k_{\text{tr}} = 2/3$, $\omega_m = 0.2\,\text{s}^{-1}$, and $T_x = 10\,\text{s}$. The dependence (16) has the form $\hat{\varphi}_{\text{st}}(s) \approx \frac{1}{2s+1}\left(\ddot{x}_s^{\text{acc}}(s)/g\right)$.

We compared the behavior of the closed-loop control system on the variables x_T and x_c with the variable x_m, representing the output of the model corresponding to the reference motion (7) with the above parameters: $\ddot{x}_m = -2\omega_m\xi_m\dot{x}_m - \omega_m^2\left(x_m - \tilde{x}_{c\,\text{giv}}(l)\right)$.

Figure 4 shows the simulation results under the average values of the crane operating mode parameters with the minimum suspension length $l = 3\,\text{m}$: $m_c = 5000\,\text{kg}$ and $r_c = 2\,\text{m}$. For the other values of the last two parameters from their ranges (see above), the curves turn out to be almost the same, deviating merely by units of percent.

Next, Fig. 5 presents the simulation results under the average values of the crane operating mode parameters with the maximum suspension length $l = 15\,\text{m}$: $m_c = 5000\,\text{kg}$ and $r_c = 2\,\text{m}$. Just as in the previous case, the other values of the last two parameters from their ranges (see above) lead to almost the same curves. The constant positioning error of the cargo at the target point is about 0.28 m. If the angular setup of the accelerometer in the combined sensor has no error, this positioning error will disappear.

For other suspension lengths, the results are of intermediate character. They confirm the theoretical considerations: with a large variety of cargo parameters, its movement is close to the behavior of the assigned reference with reaching the target point at the loading height with a small error proportional to the angular setup inaccuracy of the accelerometer in the combined sensor (the presence of the uncompensated "zero" of the accelerometer). When a step wind disturbance occurs, it is parried. Note that these properties are obtained under the current parametric uncertainty. The linear movement of the cargo is close to the reference. The transient time for a 10 m load transfer is about 25 s. The angular deviation of the cargo suspension does not exceed 2°. Such properties were obtained for other values of the crane parameters without changing the control algorithm.

According to the simulation results, compared to the approach described in [11], the control system design method proposed above has similar properties in terms of the quality of cargo transfer control, the damping of pendulum oscillations, and disturbance parrying.

Also, the crane control algorithms under consideration have been validated on an experimental setup to investigate cargo calming on overhead cranes with asynchronous servo drives [21], demonstrating similar properties.

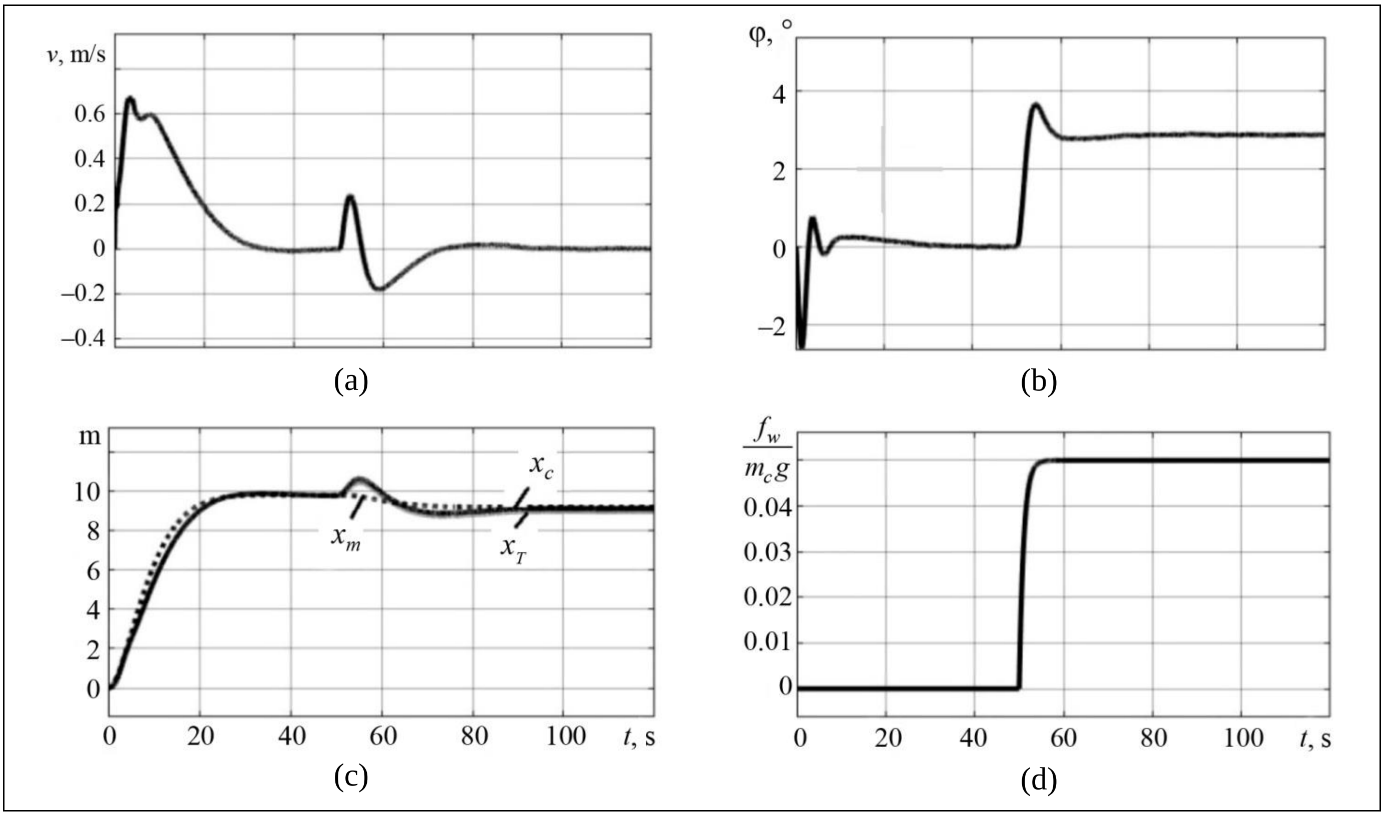

Fig. 4. Simulation results for the adaptive control system of a medium overhead crane with the minimum suspension length (3 m): (a) trolley speed, (b) the deviation angle of the suspension, (c) the linear movements of the trolley, cargo, and reference output, and (d) the relative wind force.

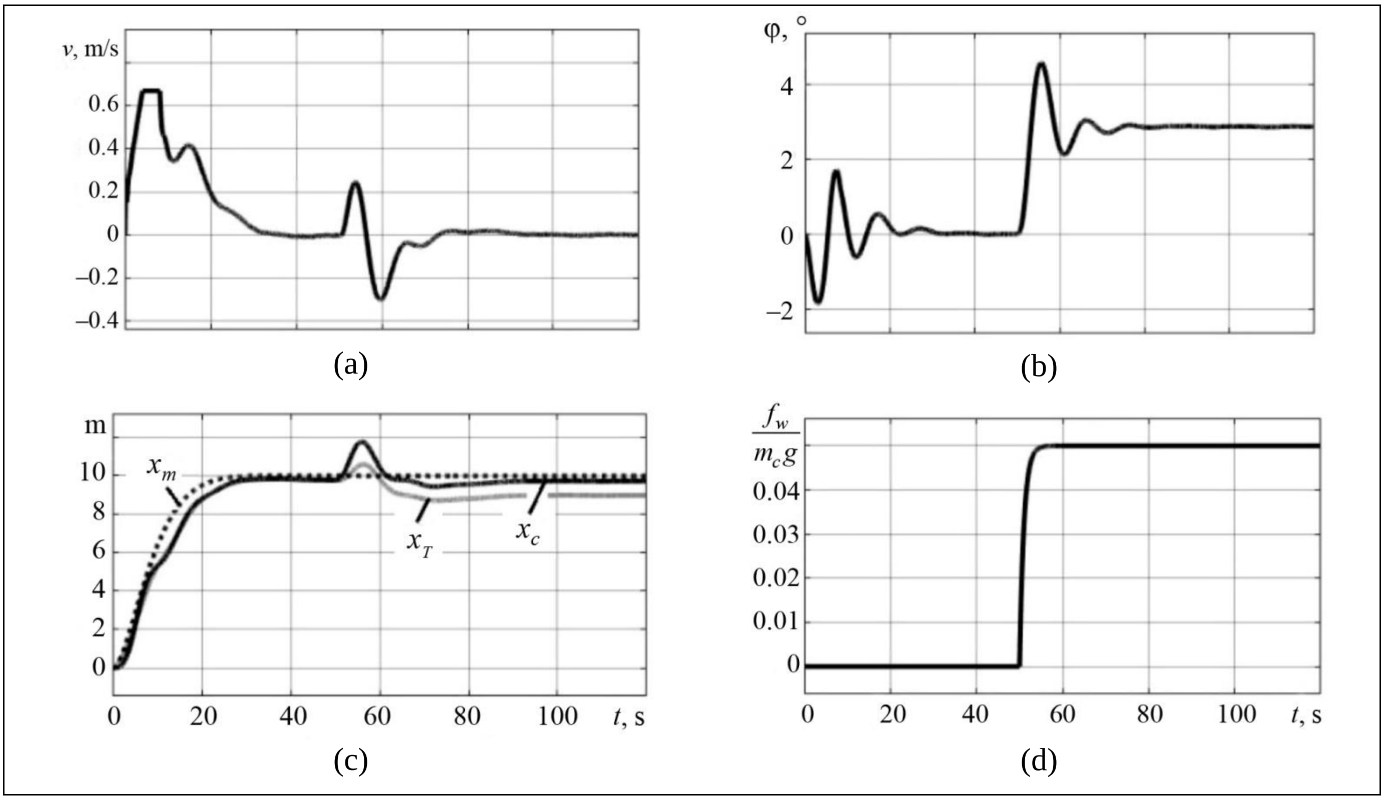

Fig. 5. Simulation results for the adaptive control system of a medium overhead crane with the maximum suspension length (15 m): (a) trolley speed, (b) the deviation angle of the suspension, (c) the linear movements of the trolley, cargo, and reference output, and (d) the relative wind force.

CONCLUSIONS

The studies have shown the effectiveness of the presented identification-based control solution for an overhead crane and its implementability using modern equipment. The approach proposed in this paper yields a control law with similar properties as in [11], but it provides additional advantages:

– the safer, more reliable, and less costly (in terms of maintenance operations) placement of information sensors;

– application of an effective current parametric identification procedure based on the least-squares method with the guaranteed stability of this algorithm;

– possibility to select control law parameters based on the passport data of a crane installation;

– no requirement to pre-tune the control system before each start (to determine the ARS drift). Therefore, inexpensive mid-class sensors can be used in the control system.

If the crane control system is intended for automated crane control, it has to be tuned at the mounting stage as follows:

– determining the "zero" of the accelerometer for further subtraction from the current readings to eliminate the constant cargo positioning error;

– selecting the value of the parameter and determining the dependence by the expressions (12) and (13) using the crane passport data for the control law (21) generating the given speed;

– selecting and validating the parameter values of the current identification algorithm.

– setting the parameter values of the low-pass filters.

(Note that periodic fine-tuning during scheduled maintenance work is also possible.)

If the crane control system is intended for fully automatic operation, an accelerometer with the highly stable "zero" should be used, or accelerometer readings should be periodically corrected through special procedures.

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