Parabolic Angle-based Anti-sway Control for Container Cranes with Limited Dynamic Loads

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Abstract:

A novel anti-sway control system for cranes is proposed. The proposed method is based on the formation of a predetermined angle of deviation as a piecewise-continuous parabolic function, the second derivative of which does not experience discontinuity, and its third derivative is a rectangular pulse of certain variable-sign amplitude, which eliminates the mechanical stress in the crane construction and kinematic gears. The proposed method provides damping of the oscillations of the suspended cargo during its horizontal motion (in two orthogonal coordinates) and hoisting/lowering. The control method is invariant to the mechanism/payload mass ratio, lift height and hoisting speed.

Keywords: Anti-sway system, motion control, oscillations damping, container crane.

Introduction:

Rapid and reliable operation of harbor and ship-to-shore container cranes has a significant impact on a port terminal efficiency. The improvement of a crane performance requires studying and deep analysis of the crane operation in both steady-state and dynamic modes of operation. Thus, several mathematical models of different level of complexity have been created to simulate the operation of different types of cranes [1-11].

The major part of a loading/unloading cycle of such cranes is travelling; i.e. moving the load along the boom from the ship to the shore and vice versa. With lift heights 20-50 m, trolley travelling mechanisms are characterized by relatively long transients due to the sway of the suspended load, damping such oscillations may take 20-30 s. Thus, including anti-sway control is mandatory for reliable and efficient crane operation.

Various anti-sway control systems have been developed to mitigate the load oscillation during the operation of cranes, in the last decade a dynamical nonlinear model of the crane was implemented to suppress the cargo swaying [12, 13]. A rate-based control strategy was developed in [14] to directly compute required crane joint rates to isolate the payload from the ship motion using only inertial measurement unit information. Pontryagin's Principle with constant force and step-change switching technique was used implemented in [15].

The implementation of input-shaping in the anti-swinging control was achieved in [16] using the on-line two-dimensional inclinometer measurements and an on-line calculation of input impulse trains. The developed input-shaping control was divided into two phases: straight line motion and external perturbances cancellation. An adaptive fuzzy sliding-mode control (AFSMC) was presented in [17] for the robust antisway trajectory tracking of overhead cranes subject to both system uncertainty and actuator nonlinearity. In [18] a nonlinear tracking control law was implemented to eliminate the nonlinear characteristics of the system and achieve the satisfactory position control and swing suppression, even when the initial swing angle and the variation of payload weight exist. An integrated computer simulator tool of rotary crane with ship behavior was considered of ship sway and load sway in [19]. The integrated simulator of was realized by corporating an external force interface routine of a component with Fluid analysis software. In [20] an anti-swing controller using a dual stage control system is proposed. The system parameters are characterized by system identification process with respect to the 1/20 scale D4 crane model and the LQR controller for anti-swing was designed based on decoupled of X- and Y-axis and linearized crane model.

Recently, a sliding-mode control for container cranes is discussed in [21, 22, 23, 24]. A sliding surface is designed in such a way that the sway motion of the payload is incorporated into the trolley dynamics, the control law is of a varying gain. An adaptive version of the sliding mode control of a crane system is developed in [23], considering the case of no prior knowledge of the payload mass and damped elements. Using two inputs, namely, trolley driving force and cargo lifting force, the proposed adaptive robust controller simultaneously executes four duties, including tracking the trolley, hoisting the cargo, keeping the cargo swing small during transient state, and completely eliminating the payload angle at steady destination.

MPC (Model predictive control) controller, which provides fast transfer of cargo with sway reduction in [26]. The solution for criterion function of MPC controller was reached through multicriteria optimization. In [27] a consideration of the finite-time regulation controller for the underactuated crane systems in 2-dimensional (2D) space with both constant cable length and varying cable length. The designed controller can simultaneously suppress the payload swing and regulate the trolley to the desired destination within a finite time for the 2D bridge crane system in the case of constant cable length.

In [28], it is shown that the load oscillatory behavior depends on the length of the rope and thus a gain-scheduling control law is proposed to reduce such an effect. Specifically, to take into account the technological limits in the controller implementation, a fixed-order controller is tuned, by also enforcing robustness and performance constraints.

In [29], a three-dimensional model of a bridge crane is considered, the control law of which is based on generalized equations of the kinetic and potential energy of the crane and the swinging load. To compensate for oscillations, a controller is synthesized, the transfer function of which is inverse to the linearized pendulum model with two degrees of freedom. To control the motion of the crane, it is necessary to measure the sway angle of the load, the mass of the load and the movable part of the mechanism of horizontal motion of the bridge, and the trolley mass.

Also in [30], a precise and reliable cable angle measurement and a complete dynamic model of the crane system are required. Different techniques are then proposed to estimate the payload acceleration in order to increase the controller performances.

One possible method to solve the anti-sway problem is to implement Pontryagin's Principle [15, 25], which takes into account the constraints on the control inputs supplied to the control object, and gives the most effective results in the synthesis of the optimum response of a system. However, the resulting solutions with multiple switching control between the maximum and minimum values require accurate measurement of the mass ratio of the load and mechanism, the instantaneous height of the suspended load and its persistence during the transition process. In practice, such restrictions impose additional requirements on the used equipment, and hinder the freedom of action of the operator. Therefore, successful damping of the load oscillations requires the use of such control methodology that allows the simultaneous operation of multiple mechanisms (horizontal travelling and lifting). On the other hand, the control methodology should be least sensitive to the errors of the measured parameters of the payload and any possible random disturbances.

The antisway control method proposed in this paper requires measuring the length of the rope only; all other parameters can be found from the mechanisms ratings and don not required measurement. Antisway control is carried out by applying the speed reference signal to the travelling electric drive input.

Motion model of trolley and load:

The mathematical model of the travelling mechanism of the trolley or the horizontal motion of the crane can be represented by a two-mass mechanical system with an inextensible imponderable rope with a length of L (The distance between the suspension point and the center of gravity of the load). Taking into account the simplified assumption made for most industrial cranes $(\sin\alpha \approx \alpha, \cos\alpha \approx 1)$, the dynamics in this model can be described using the following system of differential equations:

$$\begin{aligned} \ddot{p}_C &= \frac{F(t)}{m_C} - g\frac{m_L}{m_C}\alpha, \\ \ddot{p}_L &= g\alpha, \\ \alpha &= \frac{p_C - p_L}{L}, \end{aligned} \tag{1}$$

where p_C, p_L – position of the suspension point on the carriage and the load, respectively. α – the angle of deviation (sway) of the rope from the vertical on axis x or y, $F(t)$ – force produced by the electric drive; which is the control action on the suspension point (Fig. 1).

When the trolley is subjected to force $F(t)$, the denominator of the equivalent transfer function of the system is written as:

$$Q(s) = \frac{L}{g}\frac{m_C}{m_C + m_L}s^2 + 1.$$

The dynamic properties of the system depend on the mass ratio.

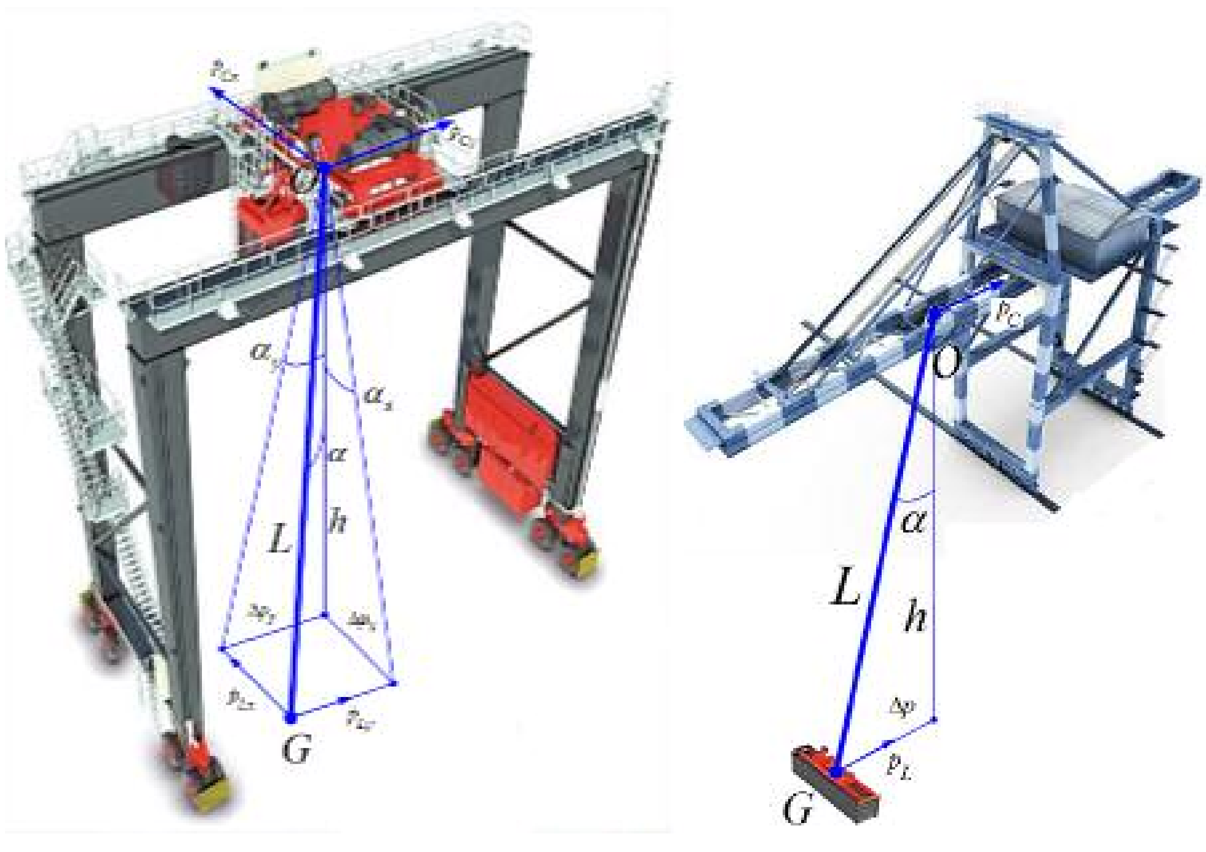


Fig.1. The main parameters that characterize the process of swinging in cranes:

a) with longitudinal movement of the cargo along the console (Ship to Shore Container Crane);

b) with combined movement operations (Rear Container Crane).

But when the speed of the suspension point is controlled, the dynamic processes are described in accordance with the block diagram shown in Fig. 2.

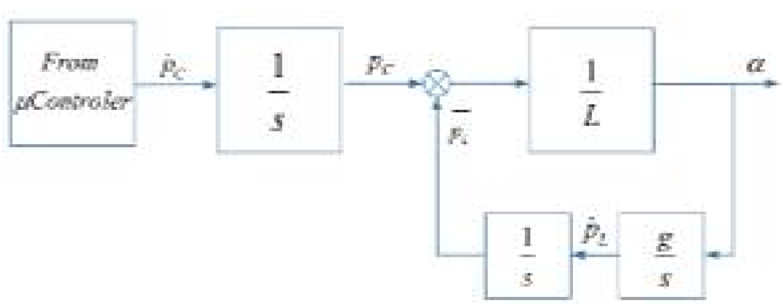


Fig. 2. Block diagram of the suspension point speed control loop.

In this case, the denominator of the transfer function does not depend on the masses of the load and the crane:

$$Q(s) = \frac{L}{g}s^2 + 1.$$

Anti-sway control:

The control objective is to create such a coordinate control law that ensures the absence of the load swinging by the end of a transient of the travelling process. One of the invariant methods to the ratio of the masses of the moving part of the crane and the load is controlling the speed of the load suspension point with a certain function. In this method with a predefined function of α, from equation (1), the load speed $v_2(t)$ can be defined as the integral of $\alpha(t)$:

$$\dot{p}_L(t) = g\int_0^t \alpha(t)\,dt, \tag{2}$$

Differentiating the equation $\alpha L = p_C - p_L$, we obtain the function of the speed of the suspension point:

$$\dot{p}_C(t) = \dot{p}_L(t) + \frac{d\big(\alpha(t)L(t)\big)}{dt} = g\int_0^t \alpha(t)\,dt + \alpha\dot{L} + L\dot{\alpha}. \tag{3}$$

For (3), a control method in which the angle α is defined by a smooth continuous twice differentiable function is chosen, the function should have zero initial and final conditions, including the first derivative.

The most important feature of such control law is the independence of the functions of the speed of the load and the suspension point on the masses. This feature makes it possible to implement the control law in mechanisms where the load mass varies widely, which is typical for container cranes. At the same time, depending on the type of the load, the center of gravity of the container may slightly shift from the geometric center of the container – up to $\pm 0,5\ m$.

Cosine law of angle for anti-sway control:

One of the set of such functions is, for example, the harmonic function:

$$\alpha = \alpha_m\left(1 - \cos\left(\frac{t}{T_G}\right)\right), \tag{4}$$

where α_m – half of the maximum angle of vertical deviation of the rope, T_G – time constant, inverse of the angular frequency of the load oscillations.

According to (4), with small values of deviation angle $(\sin\alpha \approx \alpha, \cos\alpha \approx 1)$ the speed of the load from the initial value V_0 till the final V_E will be changed according to the following function:

$$\dot{p}_L(t) = V_0 + \alpha_m g t - \alpha_m g T_G \sin\left(\frac{t}{T_G}\right), \tag{5}$$

where $\alpha_m g = a$ – the average acceleration of the load and the trolley during one period of oscillations.

The speed of the suspension point in general must satisfy the condition:

$$\dot{p}_C(t) = \dot{p}_L(t) + \frac{d(\alpha L)}{dt}. \tag{6}$$

Thus, if the acceleration can be calculated according to the following formula:

$$a = \frac{V_E - V_0}{2\pi T_G}, \tag{7}$$

Then, after a time $t = 2\pi T_G$, we obtain $\alpha = 0$, $\frac{d\alpha}{dt} = 0$, $\dot{p}_C = \dot{p}_C = V_E$.

If the value of *2a* exceeds the acceleration limit, then it should be noted that at $t = \pi T_G$ the angle α reaches its maximum value $\alpha = 2\alpha_m$, and the acceleration of the load is maximum and equals $2\alpha_m g$. If at this time the acceleration of the trolley is changed to the value of the load acceleration, then the deviation angle α will remain constant.

Thus, we obtain the control law of the travelling mechanism, consisting of three stages: 1) acceleration from initial state $\alpha = 0$ to the predetermined set point speed, which is achieved with zero deviation of the load from the vertical; 2) steady-state operation with to the set point speed with $\alpha = 0$; 3) braking till standstill, which is reached with zero load deviation from the vertical.

The duration of the transient is smallest if the duration of the first and third intervals are minimized, as the average acceleration of the mechanism and the load during these intervals is less than in the second stage in two times.

Based on the maximum permissible force $F_{\max}$, developed by the travelling drive, T_G should satisfy the following condition:

$$T_G \geq \sqrt{\frac{Lm_C}{2g\left(m_C + m_L\right)}}, \tag{8}$$

where m_L, m_C – nominal mass of the load and the moving part of the crane in selected direction,

So, the duration of the second interval

$$t_2 = \frac{V_E - V_0}{2a} - \pi T_G, \tag{9}$$

The linear component of the acceleration during the first and third intervals $a = \frac{F_{\max}}{2\left(m_C + m_L\right)}$.

With fixed length of the ropes during the transient process, from eq. 5 and 6 we can obtain the relationship of the speed of the suspension point for each of the three time intervals:

$$\dot{p}_C(t) = \begin{cases} V_0 + at - \left(aT_G - \frac{aL}{gT_G}\right)\sin\left(\frac{t}{T_G}\right) \ \forall\ 0 \leq t \leq \pi T_G, \\ V_0 - a\pi T_G + 2at \ \forall\ \pi T_G < t \leq \pi T_G + t_2, \\ V_0 + a\left(t + t_2\right) - \left(aT_G - \frac{aL}{gT_G}\right)\sin\left(\frac{t - t_2}{T_G}\right) \ \forall\ \pi T_G + t_2 < t \leq 2\pi T_G + t_2. \end{cases} \tag{10}$$

The force developed by the traveling mechanism can be derived from eq. 1:

$$\begin{aligned} F(t) &= m_C \ddot{p}_C + m_L \ddot{p}_L = \\ &= m_C\left(g\alpha + \alpha\ddot{L} + 2\dot{\alpha}\dot{L} + \ddot{\alpha}L\right) + g\alpha m_L, \end{aligned} \tag{11}$$

Respectively, for each of the three time intervals we obtain:

$$F(t)=a\begin{cases} m_C\left(\frac{L}{gT_G^{\ 2}}\cos\left(\frac{t}{T_G}\right)+\frac{v_L}{gT_G}\sin\left(\frac{t}{T_G}\right)\right)+ \\ +m_L\left(1-\cos\left(\frac{t}{T_G}\right)\right)\forall\ 0\le t\le \pi T_G, \\ 2\left(m_C+m_L\right)\forall\ \pi T_G<t\le \pi T_G+t_2, \\ m_C\left(\frac{L}{gT_G^{\ 2}}\cos\left(\frac{t-t_2}{T_G}\right)+\frac{v_L}{gT_G}\sin\left(\frac{t-t_2}{T_G}\right)\right)+ \\ +m_L\left(1-\cos\left(\frac{t-t_2}{T_G}\right)\right)\forall\ \pi T_G+t_2<t\le 2\pi T_G+t_2. \end{cases} \tag{12}$$

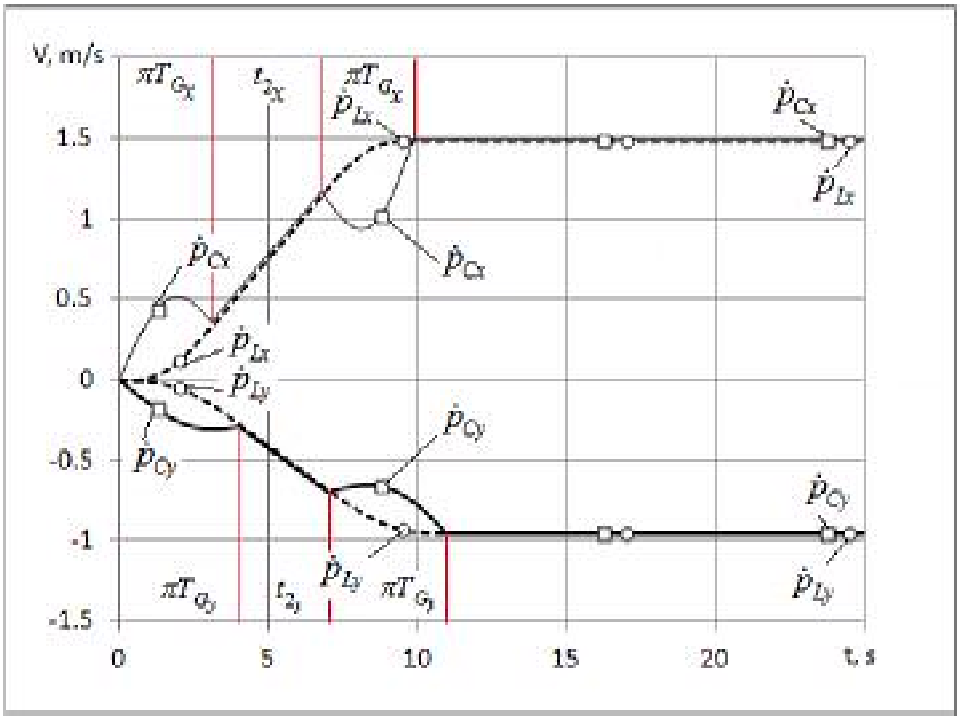
where $v_L=\frac{dL}{dt}$ is the speed of change of the ropes length, i.e. the hoisting/lowering speed.

Taking $v_L=0$, we obtain several characteristic values on the basis of which the required parameters of the transient process can be defined:

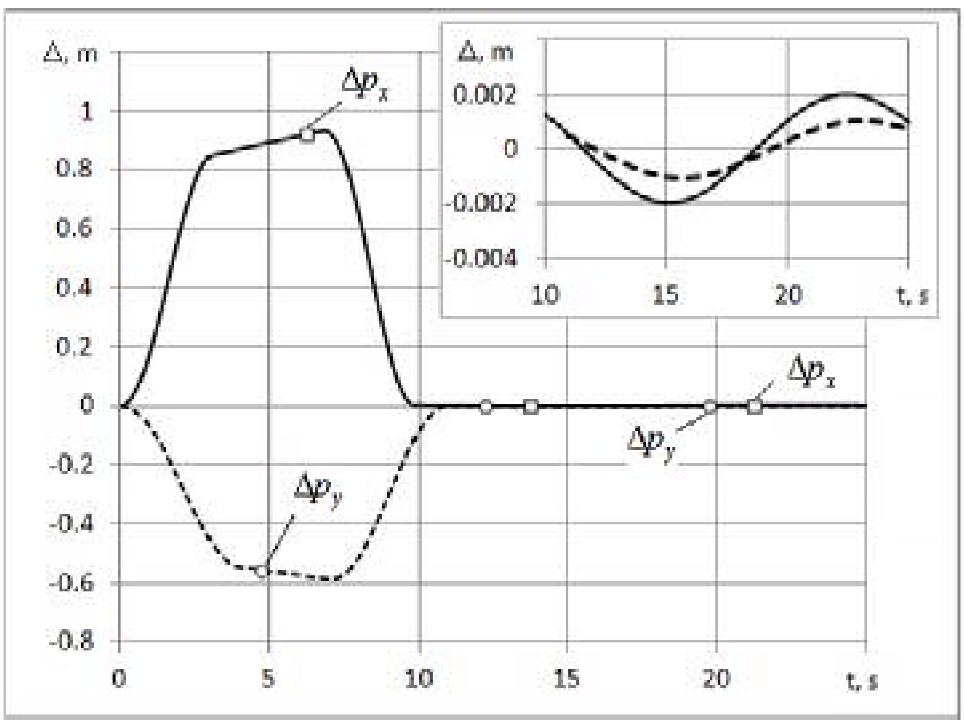
$$F(t)=\begin{cases} am_C\frac{L}{gT_G^{\ 2}}\forall\ t=0, t=2\pi T_G+t_2, \\ 2a\left(m_C+m_L\right)\forall\ \pi T_G<t\le \pi T_G+t_2, \\ -am_C\frac{L}{gT_G^{\ 2}}+2am_L\ \forall\ t=\pi T_G, t=\pi T_G+t_2. \end{cases} \tag{13}$$

According to the maximum permissible force in the second interval $F_{\max}=2a\left(m_C+m_L\right)$, we get $a=\frac{F_{\max}}{2\left(m_C+m_L\right)}$ and the minimum duration of the first and third intervals $T_{G\min}=\sqrt{\frac{Lm_C}{2g\left(m_C+m_L\right)}}$.

Thus, the control law of the speed of the suspension point is obtained. Such control law gives the opportunity of combining the hoisting and traveling operations, and does not require an accurate measurement of the load mass. T_G can be chosen according to some limitations. The deviation of the mass from the calculated value leads to the change of the force only during the transient. Theoretically, the load oscillations do not exist by the end of the mechanism acceleration, but the real speeds of the load and its suspension point are coincided with the calculated values. Fig. 3 illustrates an implementation of the obtained speed control law (eq. 10) in a lowering transient process at 1m/s with an initial length of the ropes 35 m and horizontal motion in x, y coordinates simultaneously.



a



b

Fig. 3. Illustration of the proposed method of damping the load oscillations when the operations of the load horizontal motion and lowering are combined: a) waveforms of the speed of the crane and the load; b) waveforms of the load deviation from the equilibrium (vertical) position.

The time intervals πT_{G_x}, t_{2_x}, πT_{G_y}, t_{2_y}, which can vary due to different parameters of electric drives and mechanisms are shown. Obviously, residual vibrations (Δp_x, Δp_y) are absent.

It is obvious that the predicted behavior of the load allows for the automation of the crane operation, by implementing positioning at predetermined points.

It should be noted that the use of the cosine function (4) for the angle of deviation of the load results in a necessity of step change in the motor torque/force by a value of $\pm F_{\max}$ several times during the transition (13). Such control law inevitably leads to a mechanical surge that may cause damage and reduce the life of the mechanical transmission and the motor.

Parabolic anti-sway control:

Thus, the function of deviation of the load angle should be modified to eliminate the impermissible mechanical stresses on the motor shaft, use a physically realizable torque law which can be implemented in modern variable frequency electric drive (VFD) or direct torque control (DTC) systems, and guarantees the condition of damping the oscillations of the suspended load when any combined vertical and horizontal motion of the load is possible in container cranes.

To prevent such negative phenomena as well as to produce a physically realizable torque law, compatible with VFD or DTC, it is necessary to form such a function $\alpha(t)$ that meets the condition of continuity and twice differentiable. For this condition, ideal rectangular pulses of variable-sign torque are not required to be produced by the motor.

In accordance with (2), (3) and (12), a control method based on the formation of a predetermined deviation angle, as a piecewise continuous function, the second derivative of which is continuous can be proposed.

Then, forming the third derivative of $\alpha(t)$ in a rectangular form of fixed amplitude and width and a variable sign on the initial sections of acceleration and deceleration, the change of the second derivative can be provided in a form of a trapezoid. This in turn determines the shape of the first derivative of a curve consisting of parabolas at the initial and final sections and a middle linear section.

Thus, the shape of the curve $\alpha(t)$ contains cubic parabolas in the initial and final sections in conjunction with linear sections during the increase and maintaining the deflection angle of the load. Fig. 4 shows the initial part of this process, corresponding to the interval πT_G of Fig. 3.

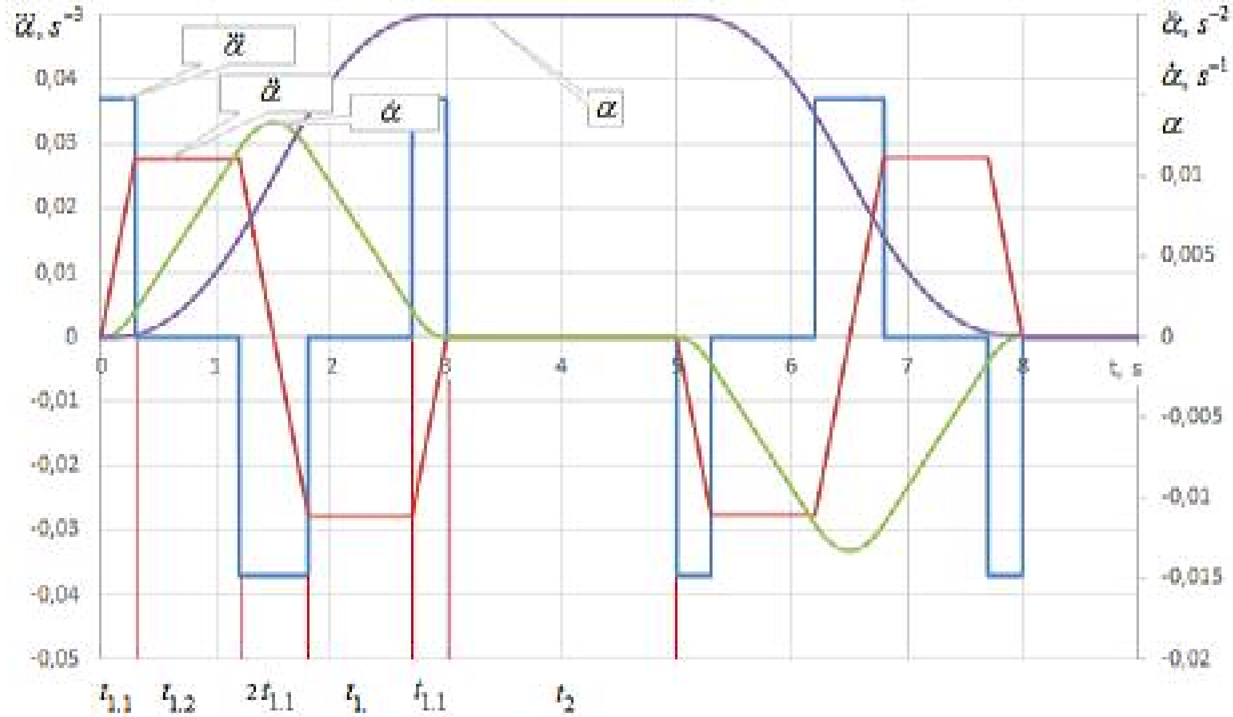


Fig. 4. Diagrams of the angle of deviation of the rope α and its derivatives at the beginning of the acceleration process.

It is important that the control system must set the duration of intervals $t_{1.1}, t_{1.2}$ and the amplitude of the third derivative $\dddot{\alpha}_{\max}$, based on the permissible rate of change and the maximum torque on the motor shaft. The other coordinates should be calculated by the controller using any method of numerical integration:

$$\ddot{\alpha} = \int_0^t \dddot{\alpha} dt, \ \ \dot{\alpha} = \int_0^t \ddot{\alpha} dt, \alpha = \int_0^t \dot{\alpha} dt. \tag{1}$$

To calculate the necessary parameters, the time intervals $t_{1.1} - t_{1.2} - t_{1.1}$ should be analyzed.

For this interval should be satisfied the following conditions:

$$\begin{gathered} 0 \le \left|F(t)\right| \le F_{\max}, \\ \alpha\big|_{t=2t_{1.1}+t_{1.2}} = \frac{\alpha_{\max}}{2}. \end{gathered} \tag{2}$$

Based on the relationship $\alpha'''(t)$:

$$\dddot{\alpha} = \begin{cases} \dddot{\alpha}_{\max} \ \forall \ 0 \le t \le t_{1.1}, \\ 0 \ \forall \ t_{1.1} \le t \le t_{1.1} + t_{1.2}, \\ -\dddot{\alpha}_{\max} \forall \ t_{1.1} + t_{1.2} \le t \le 2t_{1.1} + t_{1.2}, \end{cases} \tag{3}$$

We get:

$$\ddot{\alpha} = \ddot{\alpha}_{\max} \times \begin{cases} t \; \forall \; 0 \le t \le t_{1.1}, \\ t_{1.1} \; \forall \; t_{1.1} \le t \le t_{1.1} + t_{1.2}, \\ t_{1.1} - \left(t - \left(t_{1.1} + t_{1.2}\right)\right) \; \forall \; t_{1.1} + t_{1.2} \le t \le 2t_{1.1} + t_{1.2}, \end{cases}$$

$$\dot{\alpha} = \ddot{\alpha}_{\max} \times \begin{cases} \frac{t^2}{2} \; \forall \; 0 \le t \le t_{1.1}, \\ \frac{t_{1.1}{}^2}{2} + t_{1.1}\left(t - t_{1.1}\right) \; \forall \; t_{1.1} \le t \le t_{1.1} + t_{1.2}, \\ \frac{t_{1.1}{}^2}{2} + t_{1.1}t_{1.2} + t_{1.1}\left(t - \left(t_{1.1} + t_{1.2}\right)\right) - \frac{\left(t - \left(t_{1.1} + t_{1.2}\right)\right)^2}{2} \; \forall \; t_{1.1} + t_{1.2} \le t \le 2t_{1.1} + t_{1.2}, \end{cases}$$

$$\alpha = \ddot{\alpha}_{\max} \times \begin{cases} \frac{t^3}{6} \; \forall \; 0 \le t \le t_{1.1}, \\ \frac{t_{1.1}{}^3}{6} + \frac{t_{1.1}{}^2\left(t - t_{1.1}\right)}{2} + \frac{t_{1.1}\left(t - t_{1.1}\right)^2}{2} \; \forall \; t_{1.1} \le t \le t_{1.1} + t_{1.2}, \\ \frac{t_{1.1}{}^3}{6} + \frac{t_{1.1}{}^2 t_{1.2}}{2} + \frac{t_{1.1}t_{1.2}{}^2}{2} + \left(\frac{t_{1.1}{}^2}{2} + t_{1.1}t_{1.2}\right)\left(t - \left(t_{1.1} + t_{1.2}\right)\right) + \frac{t_{1.1}\left(t - \left(t_{1.1} + t_{1.2}\right)\right)^2}{2} - \\ - \frac{\left(t - \left(t_{1.1} + t_{1.2}\right)\right)^3}{6} \; \forall \; t_{1.1} + t_{1.2} \le t \le 2t_{1.1} + t_{1.2}, \end{cases} \tag{4}$$

In the boundary points we get:

$$\ddot{\alpha} = \ddot{\alpha}_{\max} \times \begin{cases} t_{1.1} \; \forall \; t = t_{1.1}, \\ t_{1.1} \; \forall \; t = t_{1.1} + t_{1.2}, \\ 0 \; \forall \; t = 2t_{1.1} + t_{1.2}, \end{cases}$$

$$\dot{\alpha} = \ddot{\alpha}_{\max} \times \begin{cases} \frac{t_{1.1}{}^2}{2} \; \forall \; t = t_{1.1}, \\ \frac{t_{1.1}{}^2}{2} + t_{1.1}t_{1.2} \; \forall \; t = t_{1.1} + t_{1.2}, \\ t_{1.1}t_{1.2} + t_{1.1}{}^2 \; \forall \; t = 2t_{1.1} + t_{1.2}, \end{cases}$$

$$\alpha = \ddot{\alpha}_{\max} \times \begin{cases} \frac{t_{1.1}{}^3}{6} \; \forall \; t = t_{1.1}, \\ \frac{t_{1.1}{}^3}{6} + \frac{t_{1.1}t_{1.2}{}^2}{2} \; \forall \; t = t_{1.1} + t_{1.2}, \\ t_{1.1}{}^3 + \frac{3t_{1.1}{}^2 t_{1.2}}{2} + \frac{t_{1.1}t_{1.2}{}^2}{2} \; \forall \; t = 2t_{1.1} + t_{1.2}. \end{cases} \tag{5}$$

When $t = t_{1.1} + t_{1.2}$, the force reaches a maximum value, and when $t = 2t_{1.1} + t_{1.2}$, the deviation angle is equal to half of the maximum value. Given that $F_{\max} = \left(m_C + m_L\right)\alpha_{\max} g$, at a constant length of the rope $\left(v_L = 0\right)$, from (11) we get:

$$F\left(t\right) = \left(m_C + m_L\right) g\alpha + m_L \ddot{\alpha} L, \tag{6}$$

Then we get a system of equations:

$$\begin{cases} F_{\max} = (m_C + m_L)\alpha_{\max} g = \ddot{\alpha}_{\max}(m_C + m_L) g\left(\frac{t_{1.1}^3}{6} + \frac{t_{1.1}t_{1.2}^2}{2}\right) + \ddot{\alpha}_{\max} m_C L t_{1.1} \ \forall\, t = t_{1.1} + t_{1.2}, \\ \frac{\alpha_{\max}}{2} = \frac{F_{\max}}{2(m_C + m_L) g} = \ddot{\alpha}_{\max}\left(t_{1.1}^3 + \frac{3t_{1.1}^2 t_{1.2}}{2} + \frac{t_{1.1}t_{1.2}^2}{2}\right) \forall\, t = 2t_{1.1} + t_{1.2}, \\ \dot{F}\big|_{t=t_{1.1}} = m_C \ddot{\alpha}_{\max} L + (m_C + m_L) g \ddot{\alpha}_{\max}\frac{t_{1.1}^2}{2}. \end{cases} \tag{7}$$

The first two equations can be rewritten as follows:

$$\begin{cases} \frac{F_{\max}}{\ddot{\alpha}_{\max} t_{1.1}} = (m_C + m_L) g \frac{t_{1.1}^2}{6} + (m_C + m_L) g \frac{t_{1.2}^2}{2} + m_C L, \\ \frac{F_{\max}}{\ddot{\alpha}_{\max} t_{1.1}} = (m_C + m_L) g \left(2t_{1.1}^2 + 3t_{1.1}t_{1.2} + t_{1.2}^2\right). \end{cases} \tag{8}$$

Equating the right-hand sides, we get:

$$11t_{1.1}^2 + 18t_{1.1}t_{1.2} + 3t_{1.2}^2 - 6\frac{m_C}{(m_C + m_L) g} L = 0. \tag{9}$$

$$t_{1.1} = \frac{-18t_{1.2} + \sqrt{192t_{1.2}^2 + 264\frac{m_C}{(m_C + m_L) g} L}}{22}. \tag{10}$$

Substituting (23) in the first equation of (20), taking into account the third equation, the solution for $t_{1.1}$ can be found in an iterative manner as follows:

$$\frac{F_{\max}\left(m_C L + (m_C + m_L) g \frac{t_{1.1}^2}{2}\right)}{F'_{\max} t_{1.1}} = \\ = \frac{22}{3}(m_C + m_L) g t_{1.1}^2 - 3(m_C + m_L) g t_{1.1}\sqrt{\frac{32}{6}t_{1.1}^2 + 2\frac{m_C}{(m_C + m_L) g} L} + 2m_C L. \tag{11}$$

Setting the maximum permissible value of the rate of change of the force $\dot{F}_{\max}$, from the third equation of (20) $\ddot{\alpha}_{\max}$ equals:

$$\ddot{\alpha}_{\max} = \frac{\dot{F}_{\max}}{m_C L + (m_C + m_L) g \frac{t_{1.1}^2}{2}}. \tag{12}$$

And finally, from equation (22) $t_{1.2}$ is set to:

$$t_{1.2} = -3t_{1.1} + \sqrt{\frac{32}{6}t_{1.1}^2 + 2\frac{m_C}{(m_C + m_L) g} L}. \tag{13}$$

Thus, the transient parameters $\ddot{\alpha}_{\max}$, $t_{1.1}$, $t_{1.2}$ are determined.

Further, eq. (2) is used to calculate the load speed and eq. (3) – the speed of the suspension point, respectively, which in turn will be used as a speed setpoint signal in the control system of the traveling mechanism.

Fig. 5 shows diagrams corresponding to the acceleration of the traveling mechanism at a constant length of the rope, and Fig. 6 – when combining the operations of the horizontal motion and lifting, respectively.

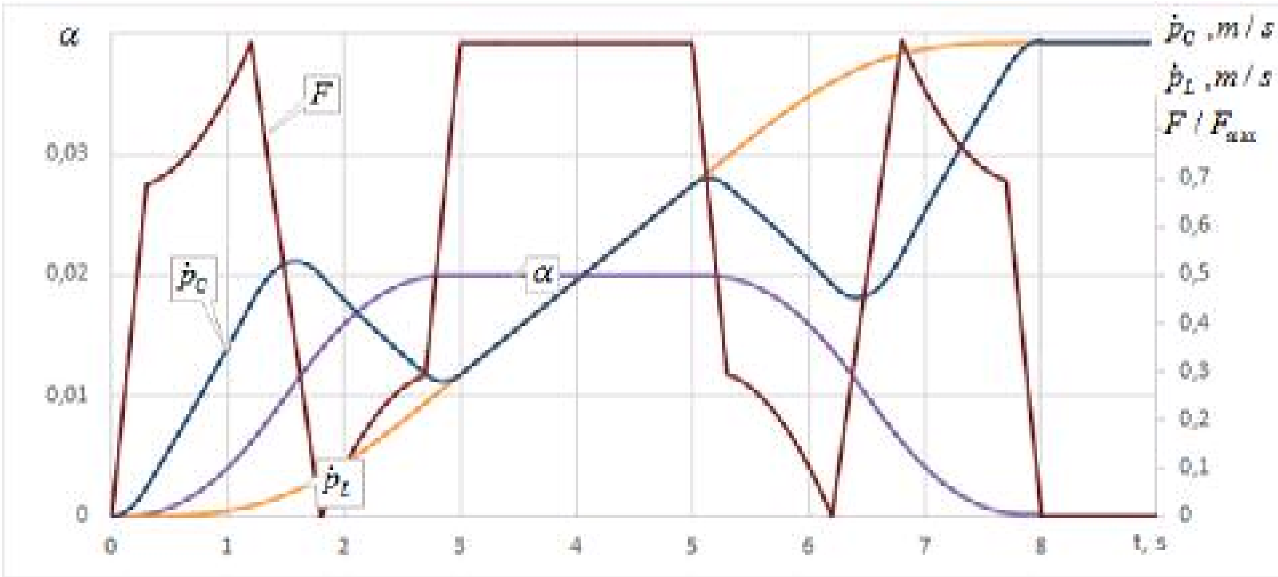


Fig. 5. Waveforms of the load and suspension point speed, force and the load angle of deviation and its derivatives in accordance with Fig. 3.

As seen from Fig. 4 and 5, no step-changes in the force is required for anti-sway control. This helps reduce shock loads in the drive of the crane, increase the accuracy of control, and coordinate the rate of change of force (motor torque) with the capabilities of the drive system.

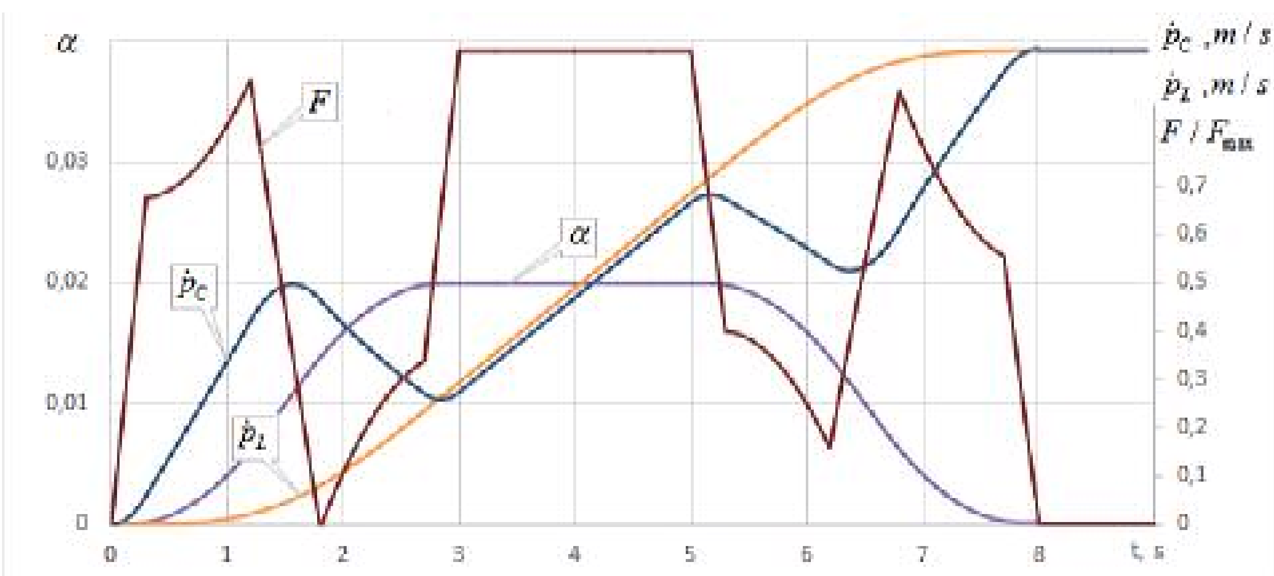


Fig. 6. Waveforms of the speed of the load and suspension point and motor torque/force for a lowering process at 1m/s.

A distinctive feature of the proposed anti-sway control methodology is the constancy of the sign of the force (motor torque), which makes this method physically realizable in modern VFD or DTC. At the same time, measuring the length of the rope in container cranes is easy (most of them have the necessary sensors to ensure comfortable operation of the operator). Therefore, the proposed control method can be implemented precisely in this class of hoisting machines.

Experimental Setup and Results:

In order to validate the simulation results, a container crane prototype with a lifting height of 1m and a jib length of 1.5m is designed. A photograph of the experimental setup of the container crane is shown in Fig. 7. Stepper motors are used in the trolley and hoist drives. The motors are connected through a 4-q and 2-phase drivers to the controller using the protocol Modbus, which is connected to SCADA Citect 7. The measurement and control of the experimental setup are implemented on a DSP board with a sampling time of 2ms.

The motion algorithm is generated in the SCADA system, and the motion coordinates are demonstrated online on the screen. In addition to that, the visual observation of the load sway is a major factor in estimating the control system performance.

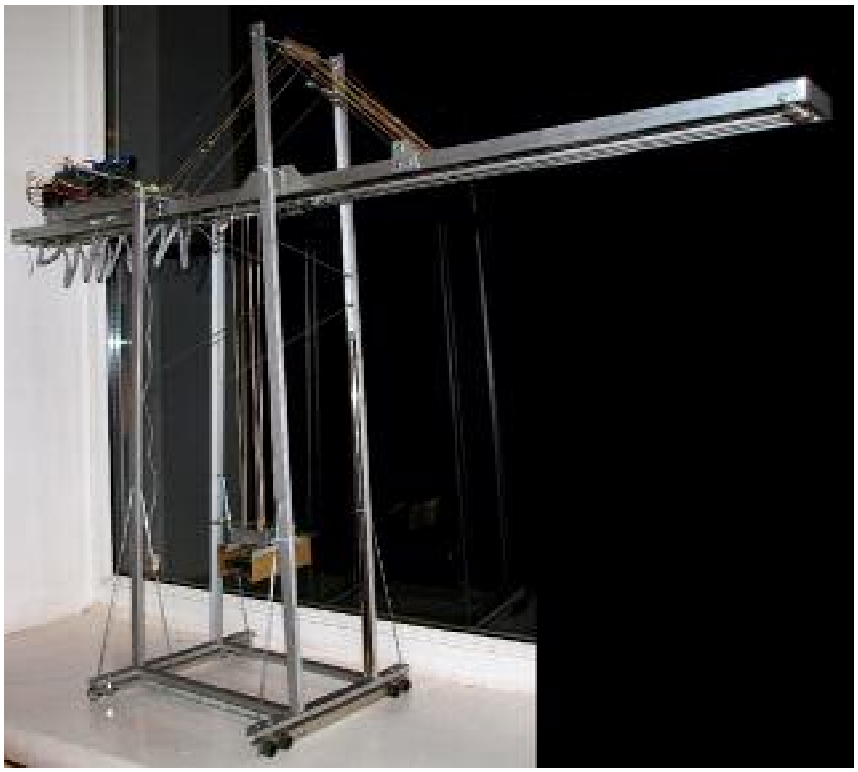


Fig. 7. Photograph of the experimental setup of the container crane.

In Fig. 8 a complete motion cycle including a starting of the hoist/trolley drives, load translation, and stopping is performed using the proposed anti-sway control method. As seen from the corresponding waveforms, the load sway is significantly damped by the end of the starting and stopping. The visual observation of the load behavior confirmed the absence of the load sway as well.

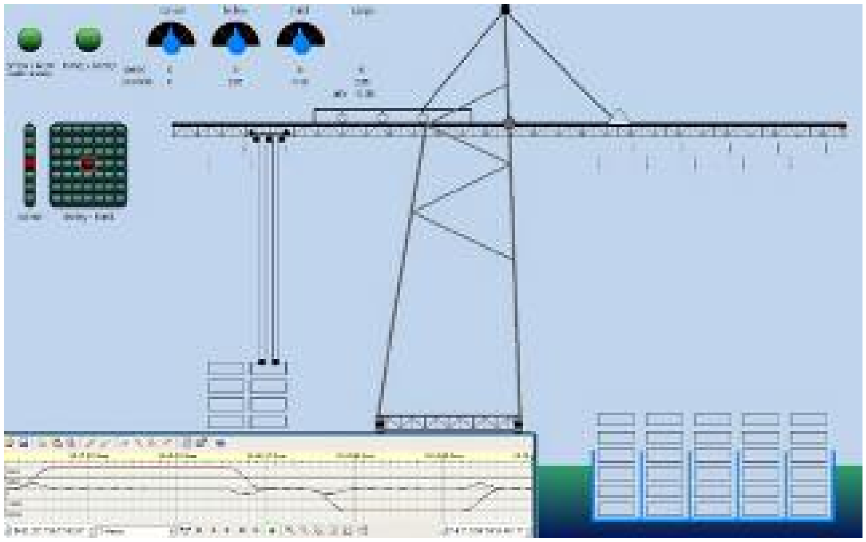


Fig. 8. Experimental results for the crane complete stroke: starting, load translation, and stopping with anti-sway control.

Conclusion:

1. A parabolic angle-based Anti-sway control for container cranes is developed. The proposed control method is based on the speed control of the suspension point of the payload. The oscillations of the suspended load during its horizontal motion (in two orthogonal coordinates) and lifting/lowering are damped. The control method is invariant to the mechanism/cargo mass ratio and to the hoisting speed.

2. Experimental and simulation results show that the load sway disappears by the end of any transient process with simultaneous horizontal load motion (in two orthogonal coordinates) and vertical (lifting/lowering).

3. The coincidence of the theoretical and experimental results proves the validity of the mathematical model on which the simulation is based. It also proves the effectiveness of the proposed anti-sway control methodology.

4. Controlling the speed of the load suspension point, enables selecting the parameters of the transient processes in a wide range, based on the requirements and limitations of the dynamic loads and response.

5. The proposed control method is based on the formation of a predetermined angle of deviation as a piecewise-continuous function, the second derivative of which does not experience discontinuity. Its third derivative is a rectangular pulse of a certain variable-sign amplitude, which eliminates the mechanical stress in the crane construction and kinematic gears.

6. The distinctive features of the proposed anti-sway control methodology are the constancy of the sign of the force (motor torque) during the transient; this makes this method physically realizable in modern VFD or DTC systems. Also, the antisway control requires measuring the length of the rope only; all other parameters can be found from the mechanisms ratings and don not required measurement.

7. Based on the formulated speed diagrams of the mechanism and the load during starting and braking, the system's safety is increased, and the smoothness of motion permits the crane automated operation, including precised positioning of the load in predetermined points. Such automated operation can be implemented in the design of container cranes working in automated terminals.

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