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Payload Oscillations Minimization via Open Loop Control

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The results of tests of payload oscillations, forced by linear control function which allows to minimize payload sway after acceleration phase and after overhead crane stopping are presented in this paper. The analysis of solution of this problem has been carried out. The algorithm of operation for real drive system which takes into account the possibilities of driving of an overhead crane is also presented. The impact of inaccuracies of measurement of the ropes length on minimizing a displacements of payload during the duty cycle is shown as well. The correctness of the method is confirmed by results both simulation and experimental tests.

Keywords: payload oscillations, oscillations minimization, underactuated, systems, crane drives.

1. Introduction

Nowadays increasing of payload transport efficiency using overhead cranes is a very important problem. One of the limitations is flexible (on ropes) suspension of the payload. Adverse effects of this are payload oscillations after stopping the overhead crane and an added time which is necessary to their damping. Tries of payload sway minimization, are presented in [11], [14], [26] and [31]. In [2] authors paid attention to the difficulties in the measuring of payload oscillations and they presented the method of determining dynamics of oscillating payload based on changes in driving torques. Authors of paper [9] reviewed the methods of modelling and control of the payload. Paper [22] present the different types of crane drives with fuzzy logic control, using the system of control and measurement STK (Takagi-Sugeno-Kang) which allowed to analyse the operating parameters of the crane. Tests confirmed the improvement of the operating features, such as positioning or decrease payload displacements.

In [5] we can find references to the works confirming analogy of suspended payload movement to the mathematical pendulum movement. It confirmed observations taken during research of payloads dynamics in our Division. Authors of the paper proposed the methods of payload oscillations damping during deceleration from constant speed to stop. Basing on developed model of the pendulum, the digital, analog and adaptive (with deceleration time dependent from oscillation period of a mathematical pendulum) methods of payload sway damping were presented. In [3] the control system of an overhead crane trolley movement with disturbances (such as wind or friction forces) basing on observer was presented. This system allowed mapping the assumed trajectory of trolley. In the presented system appeared the deviations from assumed trajectory during unsteady movements, with a good accuracy of payload positioning at the end of duty cycle. The linearized model described in state space was used.

The issue of wind disturbance compensation with state simulator was presented in [24]. The non-linear system of overhead crane was controlled by authors of [6] using fuzzy logic, basing on experiences of crane operators. This method gave satisfactory effects with payload positioning and minimization of payload sway for the case of system driven by DC motor. In the same paper, the authors reviewed the papers of this scope and drew attention to the complexity of the methods contained therein and hence the difficulties in introducing them to use in the industry. In [13] and [16] two drive systems were presented. The first, based on two subsystems (for positioning and payload sway damping) and the second, which used a neural nets to avoid complex mathematical description or in case of lack of identification of models. In both systems PD controllers were used.

Overhead cranes are one of typical kind of underactuated systems and such they are presented in some papers e.g. [32], [33] and [34]. Authors agreed, that in cases when the number of degrees of freedom of a system was higher than the number of actuators (underactuated systems), the analyze of kinematic and dynamic dependencies between the crane and the payload suspended on ropes was necessary to develop the proper system of crane control. Crane control, proposed in these papers were developed basing on trajectory planning method, which refers to trajectory of displacement, acceleration, speed or even jerk.

One of payload minimization method is the input shaping control. This method has been known for many years and has had different applications in vibration limitation [19]. In [21] input shaping was defined as "a feedforward control technique for reducing vibrations of oscillatory systems controlled by computers. The method works by creating command signals that cancels the vibration caused by the first part of the command signal". The examples of the input shaping method for damping of payload sway were presented e.g. in papers [4], [18], and [25]. Input shaped control was also implemented to the system with the payload hoisting with satisfactory effects, what was presented in paper [20]. Input shaping techniques were also used for controlling the payload swaying, where payload and its suspension system were treated as a double pendulum as described in [1], [15] or [23].

More emphasis were put on safety and reliability issues that are becoming increasingly important in overhead cranes operation. Paper [17] was focused on safe transport of the payload from one point to another one without any damage of devices working under crane or injury of employees as effective and economical way.

Payload movements on high levels (overhead) were presented as one of methods of safe transport.

A lot of papers, for example [27], [28], [29] and [30] were presenting dynamic models of laboratory devices simulating operation of overhead cranes as models of real overhead cranes. This can be suitable for research with suspended point trajectory or input function for minimization of payload sway.

The advantages of inverters usage, especially: speed control capabilities, freedom in inverters programming, possibility of energy recuperation, mechanical brakes control and ease in communicating with other devices as the reasons for wide use of cranes driven by motors equipped with inverters were presented in [10].

Inverters allow to free shaping of speed during unsteady motions, what allows to use them for minimizing of payload sway during duty cycle of overhead cranes also in open loop systems.

One of features of inverters is the ability to set of acceleration or deceleration values of the drive (by adjust times of speed ramps). The paper presents the easiest way to minimise payload sway using this feature, developed by the author, confirmed by analytical calculations and experimentally verified. This method needs only information about ropes length and co-ordinates of starting and ending points. Method looks to be the most suitable for overhead cranes working indoor, what means without wind acting on payload.

2. Dynamic model of the payload

Movement of the payload suspended on ropes was described as a movement of mathematical pendulum in vertical plane, parallel to the track, based on the model presented in Fig. 1. This model was described, among others, in [8].

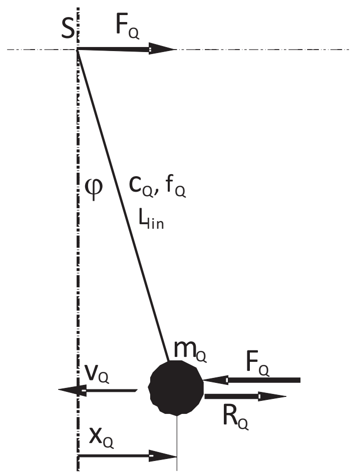

Figure 1 The interaction between ropes and payload

During development the model, the following assumptions were made:

- the payload oscillates in vertical plane, parallel to the track,
- speed of the payload suspension point was shaped freely,
- the payload oscillations were small. The following dependences were assumed:

$$\sin\varphi \cong \frac{x_Q}{L_{lin}} \qquad \text{and} \qquad \varphi \cong \frac{x_Q}{L_{lin}}$$

- ropes had constant length and were weightless (rope systems were introduced into the model as one equivalent rope),
- force of the air resistance acting on the payload was proportional to its speed and placed in the centre of gravity of the payload,
- wind acting on the payload was not considered.

The known rules of mechanics were used in mathematical description. The force of ropes impact to the payload is described by equation:

$$F_Q = c_Q x_Q \text{ [N]} \tag{1}$$

where:
c_Q – horizontal stiffness of the payload suspension (where a displacement of the payload in respect to the suspension point was taken as a deflection):

$$c_Q = \frac{m_Q g}{L_{lin}} \text{ [N/m]} \tag{2}$$

m_Q [kg] – mass of the payload,
L_{lin} [m] – length of the ropes,
g [m/s^2] – acceleration due to gravity,
x_Q [m] – displacement of the payload in respect to the suspension point in the direction of ropes suspension point movement.

Force of air resistance was described by equation:

$$R_Q = -\rho_Q v_Q \text{ [N]} \tag{3}$$

where:
ρ_Q [Ns/m] – air resistance coefficient for the payload,
v_Q [m/s] – horizontal speed of the payload in direction of the payload suspension point movement.
Equation of the payload motion was as follows:

$$m_Q \frac{dv_Q}{dt} = F_Q + R_Q \tag{4}$$

Dependence between relative movement of the payload and speed of the payload suspension point was calculated as follows:

$$\frac{dx_Q}{dt} = v_s - v_Q \tag{5}$$

The description in the space of state variables, which allows to solve equations using numeric methods was developed as follows:

$$\begin{aligned}\frac{dv_Q}{dt} &= -\frac{\rho_Q}{m_Q}v_Q - \frac{c_Q}{m_Q}x_Q \\ \frac{dx_Q}{dt} &= v_s - v_Q\end{aligned} \tag{6}$$

The control signal for the system was the speed of the payload suspension point v_s (eg bridge speed).

3. The algorithm for determining the control function

Research results of the methods of optimum control functions allowing the payload sway damping were presented among others in [7] and [8].

Out tests confirmed that payload oscillations could be damped after acceleration and deceleration phases using proper shaping of the payload suspension point speed during unsteady motions. Exemplary control function is presented in Fig. 2.

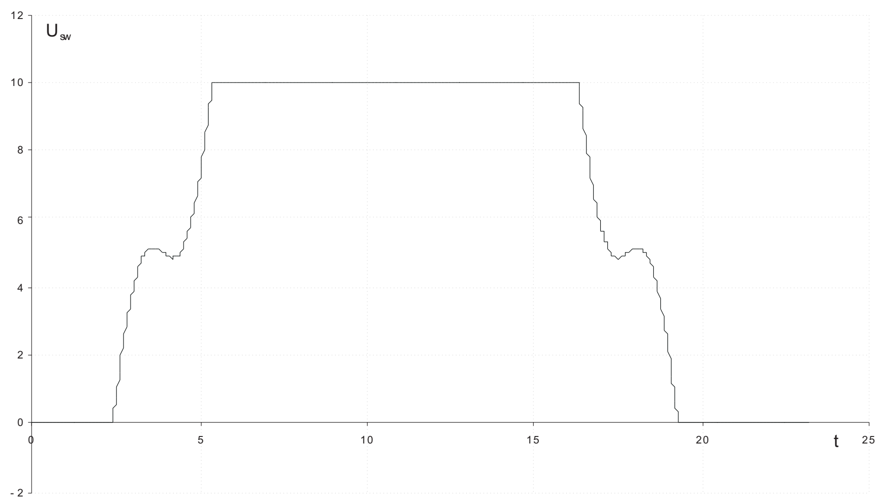

Figure 2 Exemplary control function

The most presented in literature methods of the payload sway minimization were based on closed loop systems. These methods often use very complex algorithms of input signal definition or determining a correction to input signals. The trials of implementation a fuzzy logic control or systems with observer to minimize the payload sway also appeared.

Every system using observers and loops requires detailed information from measurement systems (e.g. payload displacement) during all duty cycle time.

Basing on experiences, the assumption was made, that it was possible to determine a linear control function allowing the damping a payload sway after the acceleration and deceleration phases. Therefore the open loop control of overhead cranes traversing and travelling mechanisms and also one of features of inverters (namely the possibilities of setting the fixed and known accelerations - ramp times) could be used to damping a payload sway.

Basing on model shown in Fig. 1 and described by equation (6), excluding as small the damping, an equation in the form (7) was obtained:

$$\begin{aligned} \frac{dv_Q}{dt} &= \frac{c_Q}{m_Q} x_Q \\ \frac{dx_Q}{dt} &= v_s - v_Q \end{aligned} \tag{7}$$

from the second equation (7):

$$v_Q = v_s - \dot{x}_Q \tag{8}$$

after differentiating:

$$\frac{dv_Q}{dt} = \frac{dv_s}{dt} - \ddot{x}_Q \tag{9}$$

substituting (9) to the first equation (7), equation (10) was obtained:

$$\ddot{x}_Q + \frac{c_Q}{m_Q} x = \frac{dv_s}{dt} \tag{10}$$

assuming the relationship:

$$\frac{c_Q}{m_Q} = \omega_0^2 \tag{11}$$

the following equation was obtained:

$$\ddot{x}_Q + \omega_0^2 x = \frac{dv_s}{dt} \tag{12}$$

Because the linear rise of speed during acceleration time was assumed, as presented in Fig. 3, therefore:

$$v_s = \frac{v_u}{t_r} t \qquad \Rightarrow \qquad \frac{dv_s}{dt} = \frac{v_u}{t_r} \tag{13}$$

Considering (13) in (12), the new equation was as follows:

$$\ddot{x}_Q + \omega_0^2 x = \frac{v_u}{t_r} \tag{14}$$

The sum of general integral x_{Qg} of homogenous equation and particular integral x_{Qp} of inhomogenous equation was the solution of equation (14):

$$x_Q = x_{Qg} + x_{Qp} \tag{15}$$

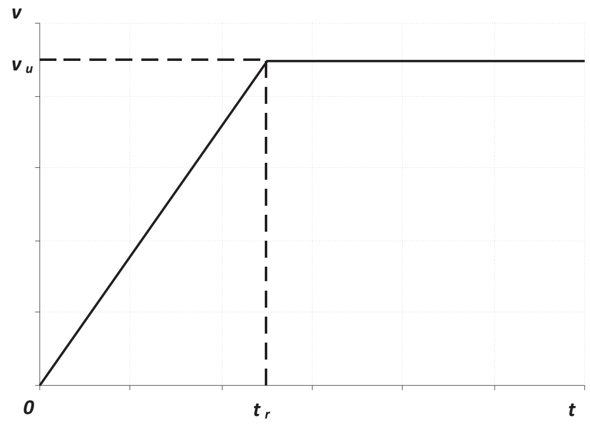

Figure 3 Control function during acceleration

where:

$$x_{Qg} = A\sin(\omega_0 t) + B\cos(\omega_0 \cdot t) \tag{16}$$

particular integral x_{Qp} was assumed constant by method of prediction according to the right side of equation (14):

$$x_{Qp} = C \tag{17}$$

A, B, C – constants

Using equation (17) and equation (14) a constant C was determined:

$$0 + \omega_0^2 C = \frac{v_u}{t_r} \qquad \Rightarrow \qquad C = \frac{v_u}{\omega_0^2 \cdot t_r} \tag{18}$$

basing on equation (15):

$$x_Q = A\sin(\omega_0 t) + B\cos(\omega_0 t) + \frac{v_u}{\omega_0^2 t_r} \tag{19}$$

for t = 0:

$$x_Q = 0 \qquad \Rightarrow \qquad B = -\frac{v_u}{\omega_0^2 t_r} \tag{20}$$

$$\frac{dx_Q}{dt} = 0 \qquad \Rightarrow \qquad A = 0 \tag{21}$$

Substituting A and B to (19) the following equation was obtained:

$$x_Q = \frac{v_u}{\omega_0^2 t_r}\left(1 - \cos(\omega_0 t)\right) \tag{22}$$

for $t = t_r$ we assumed absence of oscillations, what meant that $x_Q = 0$:

$$x_Q = 0 \qquad \Rightarrow \qquad \cos(\omega_0 t_r) = 1 \tag{23}$$

if

$$\cos(\omega_0 t_r) = 1 \qquad \Rightarrow \qquad \omega_0 t_r = 2\pi n \tag{24}$$

knowing that:

$$\omega_0 = \frac{2\pi}{T_Q} \tag{25}$$

where: T_Q – period of mathematical pendulum oscillations
from (24) the following dependence was obtained:

$$\frac{2\pi}{T_Q} t_r = 2\pi n \qquad \Rightarrow \qquad t_r = T_Q n \tag{26}$$

That means, payload oscillations after time of linear acceleration equal to multiple of mathematical pendulum oscillation period are equal zero. The speed of payload suspension point after this period is constant. These assures the minimization (or absence) of a payload oscillations during steady motion. A similar dependence can be derived for deceleration time. Multiplication of the oscillation period of mathematical pendulum can be useless in case of short ropes where periods of pendulum oscillation are short and there is a danger of slippage of the drive wheels.

Proposed method requires only the measurement of line length and information about the limitations associated with the lack of slippage. To verify the possibilities of the payload sway minimization using linear control function the real overhead crane was used.
The following symbols were used in the algorithm of control function determination:
x_p, y_p [m] – initial coordinates of the payload (actual position),
x_k, y_k [m] – final coordinates of the payload (position at the end of the cycle),
l_Q [m] – ropes length,
v_u [m/s] – speed during steady motion,
v_{max} [m/s] – maximum speed of the drive,
t_r, t_h [s] – acceleration and deceleration times,
T_Q [s] – mathematical pendulum oscillation period,
S [m] – assumed displacement,
a [m/s^2] – acceleration of ropes suspension point,
a_{gr} [m/s^2] – limited acceleration resulting from limitations associated with the lack of slippage.

The algorithm of determining the acceleration and deceleration times for single mechanism to minimise payload sway was developed as follows:

1. Obtaining the final coordinates x_k, y_k (from operator or system) and calculation of the displacement S:

$$S = x_k - -x_p[\text{m}] \tag{27}$$

2. Measurement of the ropes length l_Q and calculation of the oscillation period T_Q of mathematical pendulum according to the relation:

$$T_Q = 2\pi\sqrt{\frac{l_Q}{g}}[\text{s}] \tag{28}$$

3. Checking whether the calculated displacement and times of acceleration and deceleration allow to reach the maximum speed in steady motion ($v_u = v_{max}$) according to Fig. 4a.

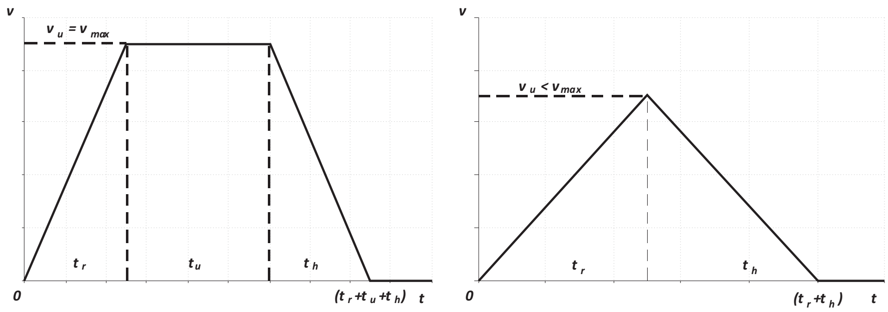

Figure 4 Control function. a) with steady motion, b) without steady motion

Assuming equality of acceleration and deceleration times respectively $t_r = T_Q$ and $T_h = T_Q$, the time of steady motion was determined as follows:

$$S_m = \frac{1}{2} t_r v_u + t_u v_u + \frac{1}{2} t_h \cdot v_u \tag{29}$$

$$S_m = \frac{1}{2} \left(t_r + t_h \right) v_u + t_u v_u \tag{30}$$

$$t_u = \frac{S_m}{v_u} - \frac{1}{2} \left(t_r + t_h \right) \tag{31}$$

The movements of the mechanism with maximum speed were not possible when assumed displacements were small. Then the steady motion time t_u could be calculated as negative. In this case, the algorithm determines time $t_u = 0$ and new speed $v_u < v_{max}$ is calculated as follows:

$$v_u = \frac{S_m}{\frac{1}{2} \left(t_r + t_h \right)} \tag{32}$$

4. Calculation and verification whether the acceleration was less than limited acceleration associated with the lack of slippage:

$$a = \frac{v_u}{t_r} < a_{gr} \tag{33}$$

In case of positive answer, the data of duty cycle parameters such as t_r, t_h, t_u, v_u were obtained and sent to control system of traversing mechanism. A negative answer meant determining the new values of the times t_r and t_h respectively equal to $t_r = 2T_Q$, $t_h = 2T_Q$ and recalculation of the whole procedure.

Further increasing of the times of acceleration and deceleration was aimless. For example double period of mathematical pendulum oscillations with ropes length $l_Q = 1$ m reached 4 s. However for every overhead crane this question should be considered individually and eventually the times of acceleration and deceleration could be extended to multiple of T_Q.

The additional criterion used to evaluate the method of the payload sway minimization was energy E_K of payload at the moment of the duty cycle ending, described by equation:

$$E_k = \frac{m_Q v_Q^2}{2} + \frac{c_Q x_Q^2}{m2} \tag{34}$$

Example waveforms of the maximum payload displacement and the energy at the moment of overhead crane stopping for different ropes length with respect to multiple of T_Q are presented in Figs. 5 and 6.

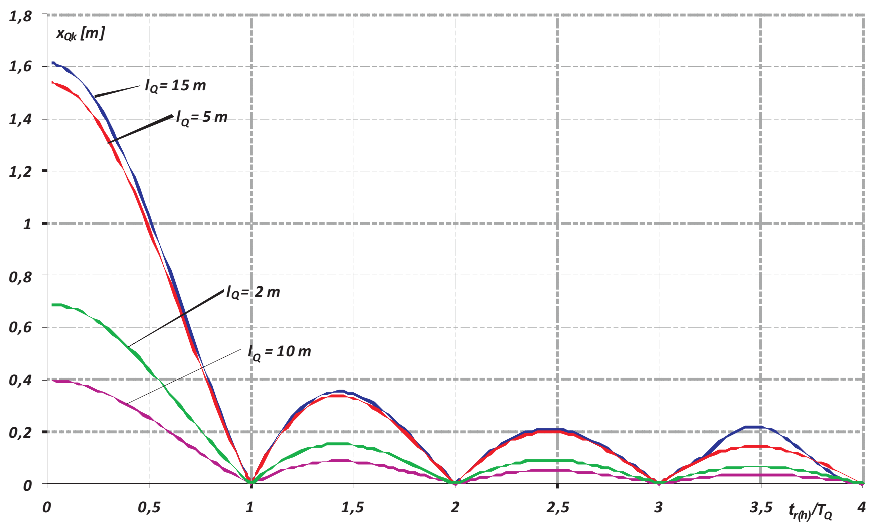

Figure 5 Maximum payload displacement at the moment of overhead crane stopping for different ropes length with respect to multiple of T_Q

Figure 5 shows dependence between minimum values of the payload displacement and multiple of the period of the payload oscillation. Energy presented in figure 6, reaches a minimum in points which are multiple of T_Q similarly to waveforms presented in Fig. 5.

Fig. 7 presents results of simulation tests for cases of different times of acceleration and deceleration, depending on multiple of the period of the payload oscillations. These are examples showing cases of 10 m displacement with the payload mass equal 5000 kg which is suspended on the ropes with length $l_Q = 10$ m. Times of acceleration and deceleration were assumed respectively 0,5T_Q, T_Q, 2T_Q,

$3T_Q$. In case of $3T_Q$, the maximum speed is lowered according to the algorithm described above. Figure presents waveforms of the payload suspension point speed v_s (upper) and the payload sway x_Q (lower). In case of $t_r = t_h = 0,5T_Q$, when the steady motion starts when the payload swing reached maximum and next continuation of large payload oscillations after overhead crane stopping is visible. In other cases at the beginning of the steady motion and at the duty cycle ending, the payload oscillations are minimized and also the payload sway are smaller during unsteady motions.

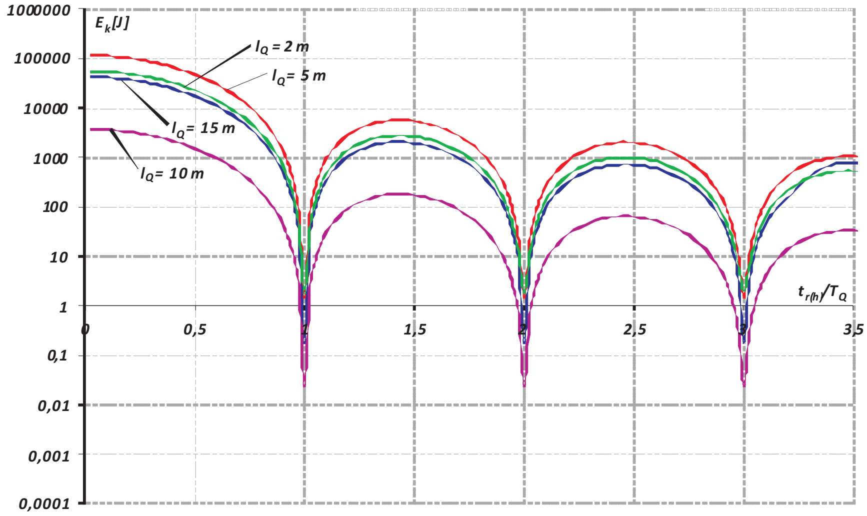

Figure 6 Energy of the payload at the moment of overhead crane stopping for different ropes length with respect to multiple of T_Q

Correctness of the operation of the system depends on the accuracy of measuring the length of the payload ropes, the assumed computational model of ropes length or elongation of ropes. The influence the measuring accuracy of the ropes length on maximum displacement of the payload at the moment of duty cycle ending is presented in figure 8. To determine this influence, for the control function it was assumed different length (L'_{lin}) of rope than for the model (L_{lin}). The new lenght was calculated as follows:

$$L'_{lin} = L_{lin} \pm \Delta_{lin} \tag{35}$$

where:

Δ_{lin} – difference between model and assumed length of the ropes.

The measurement error of the ropes length, e.g. at level 0.1 m for ropes length 5 m gives oscillations with amplitude at the end of the cycle at the level about 1 cm.

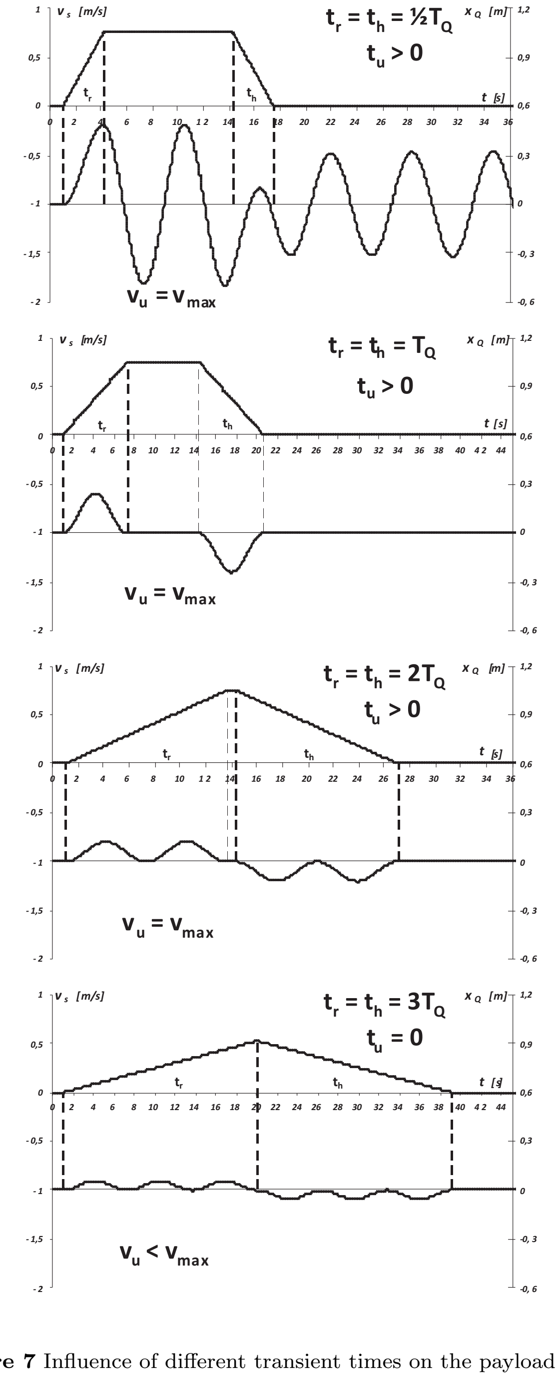

Figure 7 Influence of different transient times on the payload sway

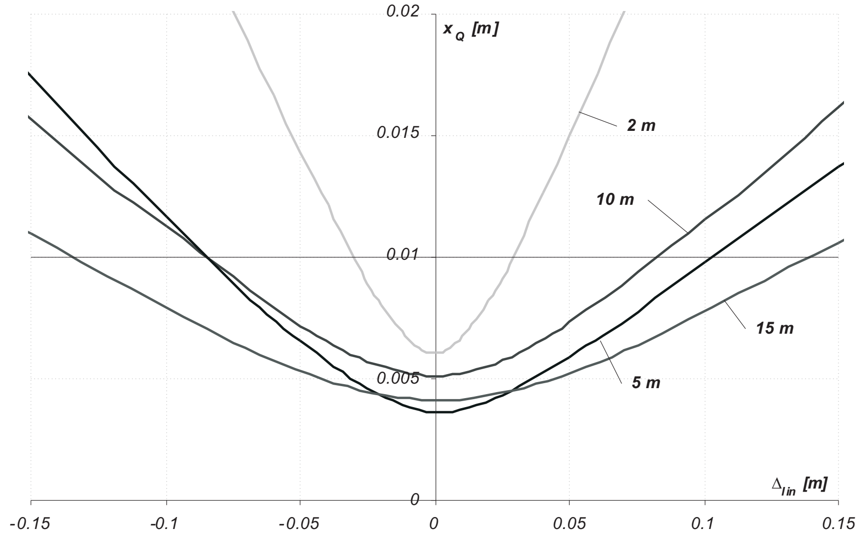

Figure 8 The influence of measuring accuracy of ropes length on the payload sway

4. Experimental and simulation tests

For experimental tests the real overhead crane with span 10 m and capacity 5000 kg was used. The overhead crane was equipped with ac squirrel-cage motors supplied by the inverters. The experimental stand is presented in figure 9. Overhead crane is equipped with several measuring systems among others such as:

1. measuring system of the ropes length (using absolute angle decoder installed on winch shaft),

2. measuring system of the payload sway (using potentiometric system shown on Fig. 10),

3. measuring system of speeds (using encoders co-operating with idle wheel of trolley and trailing wheels installed on end trucks).

The following designations were assumed:
v_{ms} – adjustable speed of the crane bridge,
v_{ws} – adjustable speed of the trolley,
v_{mr} – real speed of the crane bridge,
v_{wr} – real speed of the trolley,
x_{Qm} – displacement of the payload in direction of the bridge movement,
x_{Qw} – displacement of the payload in direction of the trolley movement.

Figure 9 Experimental stand – overhead crane

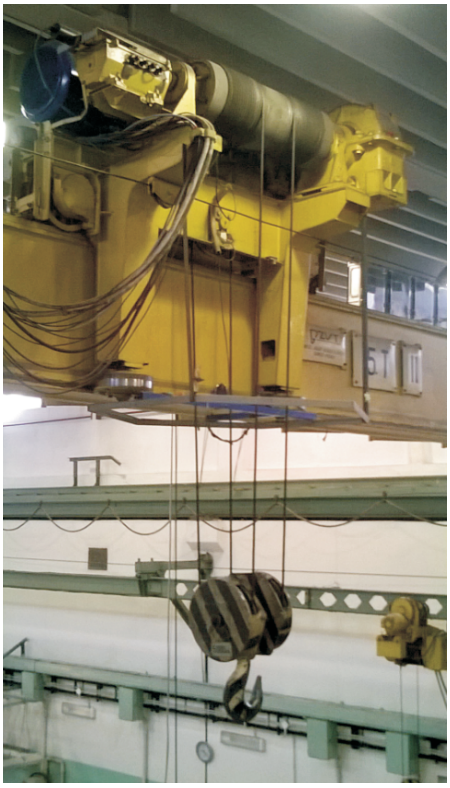

Figure 10 Rope system of the crane with measuring systems

Exemplary waveforms of the payload sway in one directions are presented in Fig. 11 and in two perpendicular each other directions in Fig. 12. The presented tests were carried with the payload mass 5000 kg.

Better damping of the payload oscillations in direction of the trolley than bridge movement is visible in Fig. 12. Differences between the adjustable and real speed were the main cause of larger payload oscillations in direction of bridge movement. The next cause of differences was design of ropes system, which was characterized by different parameters in directions of the bridge and trolley movements. The problem was described in [12]. The ropes equipped with measurement systems of the payload sway and the ropes length are presented in Fig. 10.

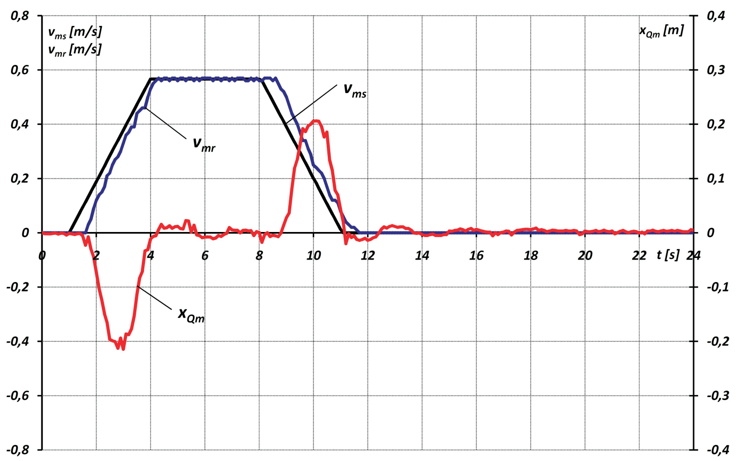

Figure 11 Experimental waveforms of control function, speedand payload displacement during the bridge movement

The simulation tests using experimentally verified model of the overhead crane, which was detailed described in [12], were carried for different transport levels (what means different length of the ropes), different masses of the payload and different waveforms of particular mechanisms speeds.

Exemplary waveforms on four different transport levels with payload mass 5000 kg suspended on the ropes with length respectively 3, 6, 9 and 12 m are presented in figures 13 and 14. Figure 13a presents the trajectory of last phase of the payload movement after the overhead crane stopping, where the times of transient movements are $t_r = t_h = 4$ s for every case, while Fig. 13b shows the same last phase trajectory for times $t_r = t_h = T_Q$. Both figures are presented using the same scale. Fig. 14 is the magnification of Fig. 13b. The multiple reduction of the payload sway relative to waveforms presented in Fig. 13a is visible in Figs. 13b and 14.

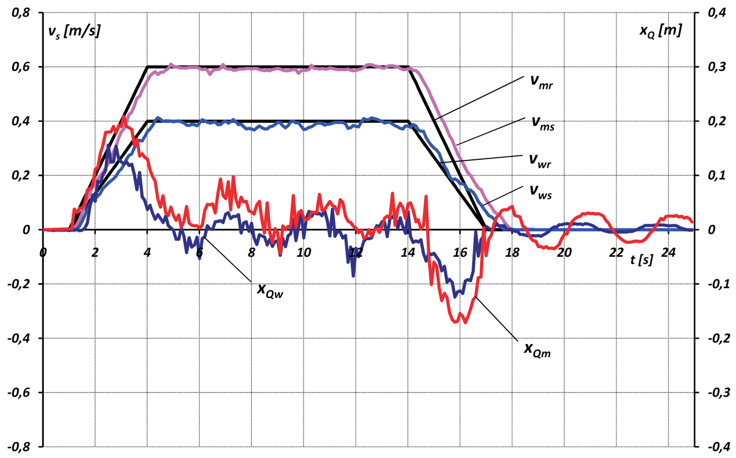

Figure 12 Experimental waveforms of control function, speed and payloaddisplacement during simultaneous bridge and trolley movement

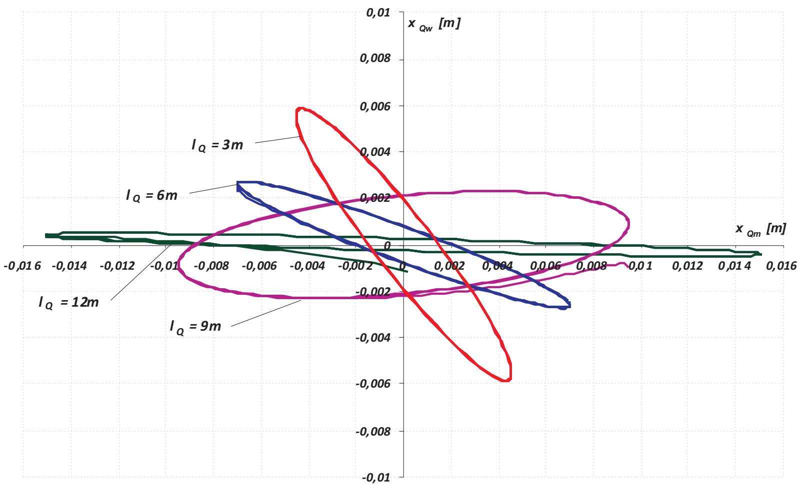

Figure 13 Trajectories of the payload movements after the overhead crane stoppingfor different transport levels. a) $t_r = t_h = 4$ s, b) $t_r = t_h = T_Q$

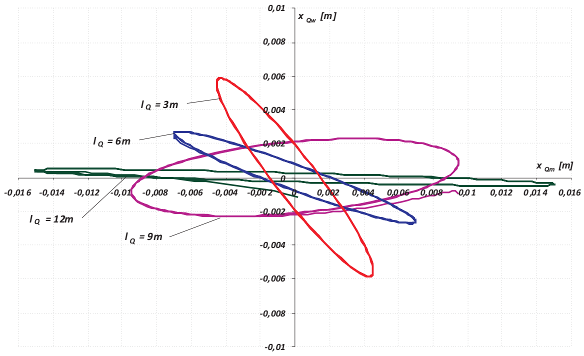

Figure 14 Trajectories of the payload movements after the overhead crane stoppingfor different transport levels. $t_r = t_h = T_Q$

5. Conclusions

Presented algorithm of calculation of acceleration and deceleration times of traversing mechanisms of overhead cranes allows to determine duty cycle parameters with effect of minimum payload oscillations during steady motion and after overhead crane stopping. Because open loop control is assumed, this algorithm is more effective for overhead cranes operating within buildings, without wind acting on the payload. Presented analyses confirmed the possibility of using the multiple of mathematical pendulum periods as transient times of travelling and traversing mechanisms of overhead cranes with good effect. Experimental tests on the real overhead crane and simulation tests on verified model confirmed the effectiveness of the algorithm and also showed sensitivity to the accuracy of mapping a control function by both mechanisms.

Decisive for minimize of the payload oscillations during the transportation cycle was accurate mapping of the control function, which determined changes of the speed and measurements of the ropes length. Developed signals allow adjust the parameters of inverters supplying travelling and traversing mechanisms by inputting only times of acceleration and deceleration (ramp time) and level of voltage which is proportional to adjustable speed in steady motion.

The proposed method allows to simultaneous applying other methods of improvement of operational features of overhead cranes, such as e.g. overhead crane position control system.

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