Optimal Control of Electric Drive Rotational Mechanisms Accounting for the Mechanical Components

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***Abstract* – Optimal control of drive rotational mechanisms accounting for the mechanical components is considered. A control method that provides damping of the suspended load and reduces the dynamic loads is proposed. Results of the system simulation are presented.**

***Index Terms* - Optimal control, rotational mechanism, electromechanical system, electric drive.**

I. INTRODUCTION

It is known the efficiency of crane mechanisms depend on a solution for the problem of damped suspended load oscillation [1]. A solution to this problem for the travelling mechanism utilises a drive torque law obtained on the basis of the maximum principle [2]; it optimises the operation of the electric drive control system by the criterion of high-speed performance: with the ac/deceleration of the travelling mechanism based on the shortest time subject to the suspended load behaviour constraints.

However, the analysis scheme for the rotational mechanism considerably gets complicated [3] because there are tangential (τ) and normal (n) acceleration components, and also Coriolis force components of both masses m_1 and m_2, as shown in Fig.1.

II. OPTIMAL CONTROL SYSTEM

According to Fig.1 the cable deviation angle of the suspended load (m_2) from the vertical direction can be decomposed into two components: α_τ - tangential, determined by the value of the acting force F_{y1} directed on a tangent, and α_n - normal, depending on the mechanical trajectory of the first mass. The control problem will be that to eliminate the component α_τ and maintain α_n constant.

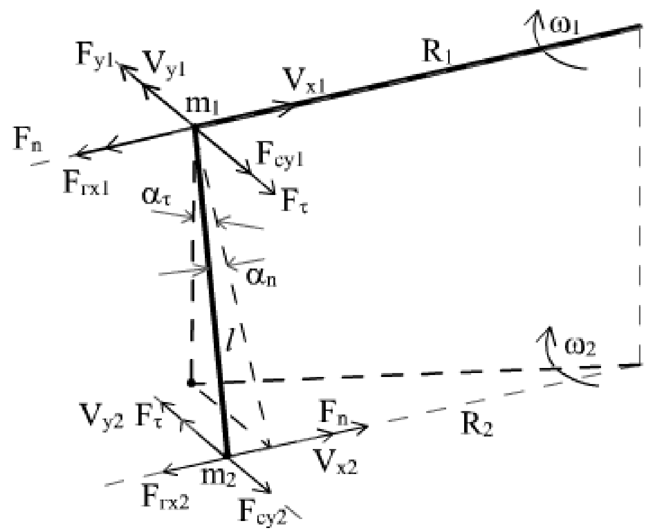

Fig.1 The calculating scheme of the rotational mechanism

The scheme depicted in Fig. 1 may be described by the formulas:

$$\begin{aligned}
\frac{dV_{y1}}{dt} &= \frac{F_{y1} - F_\tau - F_{cy1}}{m_1}, \\
\frac{dV_{x1}}{dt} &= \frac{F_{x1} - F_n - F_{rx1}}{m_1}, \\
\frac{dV_{y2}}{dt} &= \frac{F_\tau - F_{cy2}}{m_2}, \\
\frac{dV_{x2}}{dt} &= \frac{F_n - F_{rx2}}{m_2},
\end{aligned} \tag{1}$$

where F_{y1} is the force corresponding to the greatest possible from the electric drive, F_τ and F_n are tangential and normal force components. Formulas for calculation of these forces are:

$$F_\tau = \alpha_\tau \cdot m_2 \cdot g\,,$$
$$F_n = \alpha_n \cdot m_2 \cdot g\,, \qquad (2)$$

where accelerations

$$\alpha_\tau \approx \frac{s_1 - s_2}{l}\,,$$
$$\alpha_n \approx \frac{R_2 - R_1}{l}\,,$$

and were R_1 and R_2 – are the radiuses of the rotational mechanism and the load respectively.

Formulas for calculating the centrifugal forces F_{rx} of each mass and the tangential Coriolis force components F_{cy} are:

$$F_{rx1} = \omega_1^2 \cdot R_1 \cdot m_1\,,$$
$$F_{rx2} = \omega_2^2 \cdot R_2 \cdot m_2\,. \qquad (3)$$
$$F_{cy1} = 2 \cdot \omega_1 \cdot V_{x1} \cdot m_1\,,$$
$$F_{cy2} = 2 \cdot \omega_2 \cdot V_{x2} \cdot m_2\,. \qquad (4)$$

Since force F_{cy1} appears if the radius of the first mass is changed it can be used as the second control action for obtaining the optimal transient performance.

Using equations (1) and expressions (2), (3), (4) the algorithm for calculation of the rotational mechanism transient response is derived from the following suite of equations:

$$\alpha_{y1} = \frac{F_{y1} - F_\tau}{m_1}\,, \qquad V_{y1} = \int \alpha_{y1} dt\,,$$
$$\omega_1 = \frac{V_{y1}}{R_1} \qquad s_{y1} = \int V_{y1} dt\,,$$
$$\alpha_{y2} = \frac{F_\tau - F_{cy2}}{m_2}\,, \qquad V_{y2} = \int \alpha_{y2} dt\,,$$
$$\omega_2 = \frac{V_{y2}}{R_2}\,, \qquad s_{y2} = \int V_{y2} dt\,,$$
$$\alpha_{x2} = \alpha_n g - \omega_2^2 R_2\,, \qquad V_{x2} = \int \alpha_{x2} dt\,,$$
$$R_2 = R_1 + \int V_{x2} dt\,,$$
$$\alpha_n = \frac{R_2 - R_1}{l}\,, \qquad F_n = \alpha_n \cdot m_2 g\,,$$
$$F_{cy2} = 2 \cdot \omega_2 \cdot V_{x2} \cdot m_2\,, \alpha_\tau = \frac{s_{y1} - s_{y2}}{l}\,,$$
$$F_\tau = \alpha_\tau \cdot m_2 g\,. \qquad (5)$$

In this study the optimal control of the rotational mechanism electric drive performance is based on data appropriate for a real harbour crane ($m_2 = 20*10^3$ kg, $m_1 = 145*10^3$ kg, velocity $V_н$= 4,2 m/s, length of a cable l = 20m) driven by two motors (33 kW, 100 s^{-1}).

After compiling mathematical model and the program in accord with algorithm (5) the transient response of the system was predicted (Fig. 2a). Examination of this figure highlights the unfitness of the initial algorithm for the optimal control of the rotational mechanism because the second mass (m_2), via eqn (2), has a significant normal acceleration component. Therefore, by equating expressions F_n and F_{rx2} and accounting for eqn (2), it is possible to yield an expression for the change of the first mass radius R_1, to eliminate the normal acceleration of the second mass:

$$R_1 = R_{10} \cdot (1 - \omega_2^2 \cdot \frac{l}{g})\,. \qquad (6)$$

But if the radius R_1 changes the Coriolis force F_{cy1} is significant (via eqn (4)). This force, in turn, has influence upon the acceleration of the first mass (refer eqn (1)) and on the angle α_τ. Therefore by modifying the applied force F_{y1} (torque M_m) according to the law

$$F_{y1} = F_m + F_{cy1}\,, \qquad (7)$$

it is possible to compensate for the Coriolis force component.

The diagram for the variation in the dynamic force F_{y1}, radius R_1, velocities and angles using the improved control strategy is depicted in Fig.2b. Here the times for all operation stages t_1-t_3 are corrected because the force F_{y1} is variable.

Thus, using the law of optimal control obtained on the basis of maximum principle, and changing the first mass radius in accord with algorithm (6) and compensating for the Coriolis force component influencing the electric drive torque via (7) optimal control of the rotational mechanismis is possible.

However for dynamic similitude it is essential, that any rotational mechanism has spring connections for the shaft elements and an air-gap. Noting the dynamic torques are transmitted through spring elements implies the system dynamic loads increase in the acceleration phase.

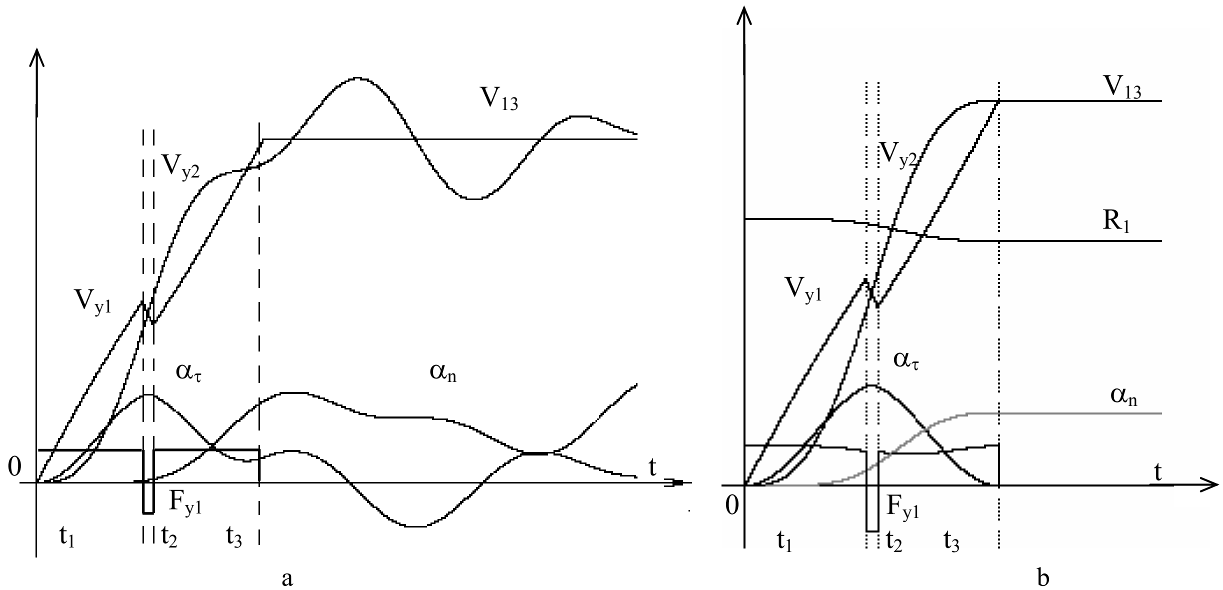

Fig. 2 Diagrams of response functions at optimal control of the rotational mechanism with application of the law for a travelling mechanisms (a) and the specified law with radius R_1 changing (b)

Obviously for optimal control of the mechanism, for a given application, knowledge of the motor torque peak is essential. One technique is to model the rotational mechanism of the crane as an electromechanical system (EMS) with three-mass mechanical components (Fig. 3) in order to limit the system kinematic complexity. Here J_1, J_2 and J_∂ are the moments of inertia of the mechanical rotational platform of the mechanism, the suspended load and the motor rotor, respectively; whereas δ is the air-gap parameter.

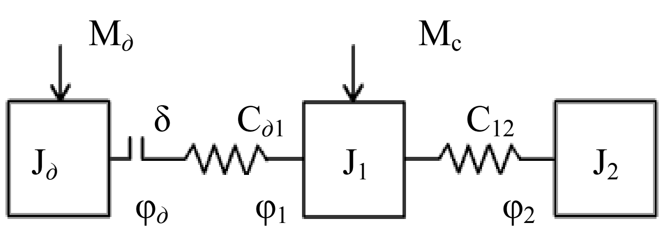

Fig. 3 Three-mass EMS model of the rotational mechanism

Equations of the two-mass $(J_\partial$-$J_1)$ EMS are:

$$M_\partial - C_{\partial 1}(\varphi_\partial - \varphi_1) = J_\partial \frac{d^2\varphi_\partial}{dt^2},$$

$$C_{\partial 1}(\varphi_\partial - \varphi_1) - M_c = J_1 \frac{d^2\varphi_1}{dt^2}, \qquad (8)$$

where φ_∂, φ_1 and $d\varphi_\partial/dt = \omega_\partial$, $d\varphi_1/dt = \omega_1$ are angular positions and angular speeds of the first and second masses; M_∂ is the motor torque, M_c is the load torque, $C_{\partial 1}$ is the connection spring rate between the EMS masses and $C_{\partial 1}(\varphi_\partial - \varphi_1) = M_{\partial 1}$ is the spring torque.

The equations (8) apply after the air-gap has closed. However when the air-gap is open $M_{\partial 1}=0$, $\omega_1=0$ and the motor idling. The instant the air-gap closes the drive torque is characterised by dynamic impact. During these transients the spring elements experience deformation oscillations which do not reduce the drive efficiency; however they raise the spring torque of the working equipment and promote system wear.

The greatest load above the average is characterised by the dynamic factor defined as

$$k_\partial = \frac{M_{\partial 1\max}}{M_{\partial 1cp}} \quad ,$$

where $M_{\partial 1max}$ and $M_{\partial 1cp}$ are the maximum and average values of the spring torque. It is known [4], for the travelling mechanisms $k_\partial \leq 2$. However the presence of the air-gap can raise the magnitude of the dynamic factor as it depends primarily on the speed differential of the first mass $\Delta\omega$:

$$k_\partial = 1 + \sqrt{1 + \frac{C_{\partial 1} \cdot_\Delta \omega^2}{M^2_{\partial 1cp}(J_\partial + J_1)} \cdot J_\partial J_1} \; .$$

Calculation of this transient response parameter reveals, that k_∂ at the first and second stages ranqes between 3 to 4. The amplitude of the spring torque practically does not change, because the electric drive has negligible damping ($M_1 \approx$ constant) and mechanical losses are likewise negligible.

For decreased torque it is proposed to close the air-gap at some fixed speed in order to minimise the differential between ω_1 and ω_2 $(\Delta\,\omega)$. In consequence of the ready availability of microprocessor control systems suggests the possibility to derive additional performances for the start up and drive reversal operational phases.

Fig. 4 depicts the transient responses to the input torque M_∂ utilising optimal control. An examination of Fig.4 reveals $k_\partial \leq 2$.

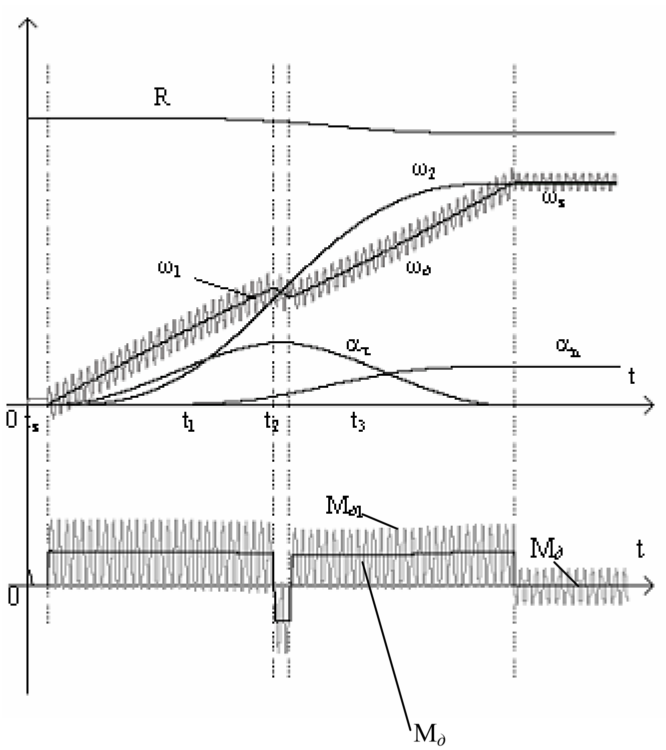

Fig. 4 Transient responses of the rotational mechanism EMS if optimal control is applied

III. Conclusions

Thus, applying the offered law it is possible to realize optimal control of the rotational mechanism, providing damping of oscillations of the suspended load and the minimal loadings in kinematic.

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