Analysis of the rational use of the elastic properties of the system of ropes and blocks for modeling the electric main lift bridge foundry crane with two drive motors

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Abstract
This scientific article considers the idea of the mechanical part of the main lift twin-engine electric overhead casting crane and construction of its logarithmic amplitude-frequency characteristics to determine the appropriateness of accounting elastic ties in the mathematical model of the drive with the help of some estimate parameters of the logarithmic amplitude-frequency characteristic (LAFC). It presents a structural scheme and mathematical model of mechanical part of electric drive for the three main states of movement: normal operation, the rise and descent of the load at failure of one engine. Modeling and calculation of the logarithmic amplitude-frequency characteristics of the system implemented in MATLAB (Simulink package). Generalized parameters have been calculated that allow analyzing the logarithmic amplitude-frequency characteristics response and giving a rationale for consideration of elastic ties in the drive operation simulation.

Keywords: casting crane; main hoist unit; dual-motor drive; mathematical model of the mechanical components; elastic coupling; simulation; log-magnitude and phase diagram.

1. Introduction

Crane facilities play an important part in the material handling. They provide supply with row materials and semi-finished products, transport of finished products and assist at mounting and repair of various equipments. Cranes are very essential units, their operation influences the yield of the whole process line.

All cranes are operated according to the set or non-predetermined cycle. Any lifting mechanisms are operated in the steady-state mode most of the time. At this, the period of transient processes is relatively short.

It should be noted that dynamic torques in transient modes result in a significant load of such kinematic assemblies as gear units, shafts, ropes, etc. Such elements as ropes are characterized by elasticity. Here, backlashes are also characteristic for power transmissions. Crane mechanisms with the electric drive shall be considered as a unified electromechanical complex. The multi-mass mechanical portion with elastic couplings and electric portion of these systems are in continuous interaction in dynamic modes. Control and disturbance actions in this system leads

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to transient processes accompanied by energy exchange between its elements. Similar events result in load swinging, vibrations and additional loads on kinematic assemblies, which deteriorates installation accuracy and quality of all operations. When positioning load, its swinging due to transient modes results in a "parasite" increase of the process cycle period [1]. If applicable, dynamic processes at simulation of electromechanical systems shall be investigated with the most accurate mathematical model of the mechanical portion of the electric drive. Unfortunately, even an accurate mathematical model considering elastic couplings cannot significantly provide accuracy and reliability of the simulation outcome. In some cases, the may reduce the calculation speed at system simulation. Furthermore, considering elastic characteristic and gaps in transmissions is unnecessary in some practical cases as they do not have any impact on operation of the whole electric drive.

2. Problem statement

This paper will examine the rationale for inclusion of the rope rigidity into a mathematical model. Furthermore, it will estimate the possibility to represent the unit as a rigid link, i.e. as a single-mass system. For this purpose, a mechanical portion should be represent as a controlled object. Then, the unit's log-magnitude and phase diagram is to be plotted, some estimation parameters should be determined and briefly analyzed. That is the mechanical portion of the electric drive that is of the greatest interest in the context of plotting the magnitude curve. The mathematical model considers elastic couplings between the cable reels and the beam. When simulating and computing, double-increase of the inertia of rotating parts during load descent in the emergency mode shall be taken into account. This event can be explained by the use of the ratchets in the gear units.

3. Main part

Estimation of the mechanical portion behavior is required to determine the practicability of considering elastic couplings at simulation and operation of the electric drive. To enable coupling analysis, a mechanical portion should be represent as a controlled object. A diagram of the mechanical portion of the dual-motor drive of the main hoist of the casting crane equipped with ratchets in the gear units has not been yet regarded. The overview literature pays attention to theoretical information on typical one-motor hoists only. The most investigation portion is devoted to mill-type ladle cranes related to the issues of mechanics and heavy engineering. Thus, [1] provides a general theoretical information on representation of the mechanical portion of the single-drum rope hoist as a double-mass system and general guidelines for calculation and analysis of amplitude-frequency characteristic. [2] gives a relatively deep insight into the issues of control of crane electromechanical systems. It determines some structural configurations of the mechanical portion of the electric drive. It also describes various magnitude characteristics and peculiarities of the unit operation. [3] provides very essential data on possible representation of the pulley lift system as a diagram with one rope, which rigidity is equal to the full pulley lift's one.

The paper considers an actual mechanical portion of the main hoist of the overhead casting crane No. 3 operated at the arc-furnace plant of Ural Steel JSC. The crane is mounted in the plant charging bay and intended for charging scrap metals into the arc steel furnace. Specifications of the main hoist of the casting crane are provided in [4]; its kinematic diagram is shown in Fig. 1.

The main hoist consists of two similar, symmetrically arranged single-drum hoist drives. Each of two hoist mechanisms is composed of an electric motor, two mechanical brakes, a double-reduction gear unit, an open gear set and a drum. Rims of two drums are constantly coupled. It provides their constant rotation rates and synchronism of the load lift. Drives are designed assuming possible operation with only one drive. It helps to complete an operation in the case of failure of one of the drives.

Two ends of two double blocks are attached to every load drum. The lower movable hoisting blocks are installed on the welded lifting beam, to which lamellar hooks are attached. When lifting loads, it is highly important to provide a greater reliability of the hoist mechanism. It is achievable due to the use of two motors and two brakes for each motor. Each drive has an ratchet wheel built-in into the gear unit. It helps to complete an operation in the case of failure of one of the drives. The ratchet device is operated as follows: the collar with pallets is attached to the output gear shaft. The ratchet gear wheel freely rotates at the collar boss assembly. Under normal operation conditions, the pallets are supported in the ratchet socket and engage the gear wheel. No power contact between the
drum gear wheels is present in this case. Any failures during load lift result in a power contact between the drum gear wheels and lead to pallets' slipping along the ratchet sockets of the lag ratchet gear wheel. In this case, the second motor takes the full load and rotates both drums [5].

Under emergency conditions, the load is distributed as follows: under influence of the load mass, the ratchet pallets are safely supported by the ratchet sockets. Changing rotation direction of the electric motor during descent results in turn of the ratchet gear wheel from the side of the operated motor in direction of the ratchet rotation. Load mass substantiates a constant ratchet pallet support at the side of the operated motor. The gear wheel and ratchet rotate co-directionally. Contemporary, a power contact between the drums causes the turn of the second drum to the side of load descent. The engaged gear of the drum-pinion unclosed pair is responsible for rotation of the ratchet neighboring on the shaft, which teeth are engaged in the sockets of the second gear wheel at the side of the failed electric motor. As a result, rotational components taking part at load descent will include complementary to the motor shaft, gear unit, pinion and drum of the functional motor also the drum, pinion, ratchet, gear unit and shaft of the second electric motor, which will result in duplication of inertia.
Under normal operation conditions, when both motors are functional and loads are accurately distributed between both motors, the gears between the drums are not loaded. The torque is transmitted at the failure of a drive only. In connection with the above, the design diagram has no mechanical link between the drums. As we can see, the system is three-mass under normal operation conditions. The first and second mass is formed by rotating parts: the drive motor, gear unit, unclosed pair and cable reel. Their inertia will be $J_1 = J_2 = 50.4 \text{ kg} \cdot \text{m}^2$. The third reciprocating mass is generated by the moving beam with the nominal load and ten pulley lift rolls, which inertia $J_3 = 1.579 \text{ kg} \cdot \text{m}^2$.

When developing the structural configurations, the pulley lift is replaced with the provided configuration with one rope, which rigidity is equal to that of the pulley lift [3].

$$C_{12,1} = C_{12,2} = \frac{E_K \cdot F_K \cdot m \cdot a \cdot p^2}{L} = \frac{1.4 \cdot 10^5 \cdot 10^6 \cdot 0.075 \cdot 2 \cdot 0.00251^2}{30} = 26460 \text{ Nm},$$

where $C_{12,1}$, $C_{12,2}$ – equivalent rigidity between the first and second mass corresponding to the first and second winches; $E_K = 1.4 \cdot 10^5 \text{ MPa}$ – elasticity modulus of the rope with a metal core to be applied according to [6]; $F_K$ – rope cross section area, $m^2$; $m$ – pulley lift rate; $a$ – pulley lift number; $\rho$ – reduction radius, $m$; $L$ – lift height, $m$.

The reduced pulley lift configuration is shown in Fig. 2. The above enables representation of the design diagram (Fig. 3) as a structural configuration of the mechanical portion shown in Fig. 4.

Under emergency conditions, the kinematic chine is opened at the side of the failed motor. The latter and a part of gear unit are switched off at lifting. The inertia of the drum, gear wheels of the unclosed pair and ratchet in the gear unit is added to the first mass. However, their own inertia practically does not effect on the total value of the rotating component inertia due to low own inertias and a high gear ratio of the unit. It is doubled at descent as a power contact between the drums at their rotation in the descent direction enables rotation of the second winch up to the electric motor.

In this case, the design diagram will have the variant shown in Fig. 5.
Fig. 3. Design diagram of the mechanical portion under normal operation conditions

\begin{align*}
M_{M1} & \rightarrow I/J_1 s \rightarrow \omega_1 \rightarrow C_{13} \rightarrow M_{13} \\
M_{M2} & \rightarrow I/J_2 s \rightarrow \omega_2 \rightarrow C_{23} \rightarrow M_{23} \\
& + \rightarrow \omega_3 \\
\end{align*}

- $J_1$ – inertia of the first mass (of the drum and gear unit of the first winch); $J_2$ – inertia of the second mass (of the drum and gear unit of the second winch); $J_3$ – load inertia; $\omega_1$ – resulting rate of the first mass; $\omega_2$ – resulting rate of the second mass; $\omega_3$ – generalized rate of the third mass expressed through the travel speed by means of the reduction radius; $M_{13}$ and $M_{23}$ – torques of tensile correlation between travelling masses of the system.

Fig. 4. Structural configuration of the main hoist of casting crane for normal operation conditions

Fig. 5. Design diagram of the ED mechanical portion under emergency conditions

As we can see, the mechanical portion is represented as a dual-mass system under emergency conditions. Here, the equivalent rigidity of the pulley lift system shall be calculated with the same formula, the only difference being duplication of the hoisting blocks; thus, $C_{12,em} = 52.921$ Nm. Obviously, $J_{1,em} \approx J_1$; at descent, $J_{1,emd} = 2J_1$. With due regard to the above, let us represent the design diagram in Fig. 5 as a structural configuration of the mechanical portion shown in Fig. 6.

To enable a further analysis of the mechanical portion under emergency conditions, it is rationally to determine some generalized parameters [2, 7, 8]. The correlation of the inertias for load lift and descent in the emergency mode is as follows:
\[
\gamma_{\text{lift}\_\text{em}} = \frac{J_1 + J_3}{J_1} = \frac{50.4 + 1.579}{50.4} = 1.031
\]

\[
\gamma_{\text{descent}\_\text{em}} = \frac{(J_1 + J_2) + J_3}{J_1 + J_2} = \frac{50.4 + 50.4 + 1.579}{50.4 + 50.4} = 1.015
\]

The resonance frequencies of the load lift and descent system:

- For normal conditions (provided, \( C_{12,1} = C_{12,2} \)):
  \[
  \Omega_{12\_\text{lift}} = \frac{J_1 + J_3}{J_1} = \sqrt{2C_{12,1}} \frac{50.4 + 1.579}{50.4 \cdot 1.579} = 185.9 \text{ 1/s .}
  \]

- For emergency conditions:
  \[
  \Omega_{12\_\text{lift}} = \sqrt{C_{12}} \frac{J_1 + J_3}{J_1 \cdot J_3} = \sqrt{52921 \frac{50.4 + 1.579}{50.4 \cdot 1.579}} = 186.0 \text{ 1/s ,}
  \]
  \[
  \Omega_{12\_\text{descent}} = \sqrt{C_{12}} \frac{(J_1 + J_2) + J_3}{(J_1 + J_2) \cdot J_3} = \sqrt{52921 \frac{50.4 + 50.4 + 1.579}{(50.4 + 50.4) \cdot 1.579}} = 184.5 \text{ 1/s .}
  \]

- The resonance frequencies relative to the load inertia for lift and descent:
  \[
  \Omega_{3\_\text{lift}} = \frac{\Omega_{12\_\text{lift}}}{\gamma_{\text{lift}}} = \frac{186}{\sqrt{1.031}} = 183.2 \text{ 1/s ,}
  \]
  \[
  \Omega_{3\_\text{descent}} = \frac{\Omega_{12\_\text{descent}}}{\gamma_{\text{descent}}} = \frac{184.5}{\sqrt{1.015}} = 183.0 \text{ 1/s .}
  \]

Based on the initial and calculated parameters and structural configuration in Fig. 4 and 6, let us make up a mathematical model of the mechanical portion. The log-magnitude and phase diagram is generated with the Control Analysis tool of the MATLAB's Simulink package. All log-magnitude and phase diagrams are located on one graph shown in Fig. 7. The obtained magnitude characteristics are used for analysis of main properties of the drive mechanical portion.

The log-magnitude and phase diagram has a slope of -20 dB/decade within the low-frequency region; discontinuity takes place at the resonance frequency only and tends to the asymptote with a slope of -40 dB/decade in the high-frequency region. An increase of the mass relation causes a small LAFC shift down along the amplitude axle and, thus, change of the system break frequency. Generally, the unit LAFC practically does not depend on conditions of the mechanical portion.

The motion of the first mass at low frequencies of the control action fluctuations is defined by the total inertia of the electric drive; at this, the mechanical portion behaves as an integrating factor. At \( M = \text{const} \), the rate is changed according to the linear law superimposed by fluctuations due to the elastic coupling. When the fluctuation frequency is approximate to the resonance fluctuation amplitude, the rates are increased and tend to infinity at \( \Omega = \Omega_{12} \). However, resonance manifestations greatly depend on parameters of the mechanical portion. Firstly, if the load
inertia is far less than the first mass \((J_3 \ll J_1, \gamma \rightarrow 1)\), the motion of the first mass is near to that determined by the integrating factor. Secondly, the motion of the first mass is determined by the same integrating factor at \(\Omega_{12} \rightarrow \infty\) in the low and medium frequency region.

Thirdly, the frequency of the change of input actions during unit operation prevents from reaching the resonance frequency region. In other words, the unit is frequently operated in the region of low frequencies of input actions.

Hence, an important practical conclusion can be drawn. As a rule, feedbacks of the motor variables are used at synthesis of crane electromechanical systems; the load inertia is far less than the first mass \((J_3 \ll J_1)\). An additional analysis of the generalized data obtained for emergency conditions proves together with the above that the elasticity negligibly influences on achievement of the first mass. This is due to the fact that \(\gamma \rightarrow 1\), and the resonance frequency relative to the load inertia is near to the resonance frequency of the complete system.

At the absence of the load motion or rate feedback, this fact enables representation of the drive mechanical portion as a rigid mechanical element without any appreciable error nor considering elasticity effect.

4. Conclusion

Investigation of dynamic processes in the mechanical portion of the electric drive of the complex crane facilities, considering specific parameters (gaps, elasticities, etc.), provision of rationale for their consideration and combined action of the improved drive mechanical portion and control system (over the longer term) are highly relevant objectives; this issue has been still insufficiently explored. Thus, this paper being an initial investigation stage has generated mathematical models of double-mass ED mechanical portion in different modes, LAFCs for various unit conditions and analysis of the generalized parameters of the double-mass mechanical systems and revealed impracticality of considering elasticities in the system as they will not have any serious impact on the operation of the investigated crane electromechanical system.

References
