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Propeller thrust tower crane slewing mechanism model identification

Abstract. Any study of the dynamics and control of mechanical systems is based on adequate mathematical models that contain the dynamic parameters of the system under study. Their evaluation, in particular for the tower crane boom system, is a particularly relevant scientific and practical problem, the solution of which will provide the basis for further calculations of the optimal modes of movement of the tower crane slewing mechanism. The research aims to determine the dynamic parameters of the installation (moment of inertia of the slewing mechanism J, torque of dry friction forces M_0, driving torque coefficient K_1) and to plan experimental studies. The experimental method, numerical optimization methods (in particular, the modified Rot-Ring-PSO method), and statistical methods were used to conduct the research. Based on the results of the experiments, the dynamic parameters of the mathematical model of the laboratory installation of the tower crane slewing mechanism with propeller thrust were identified. The criterion that evaluates the identification error of the parameters K_1, M_0, and J was formed and minimized using the Rot-Ring-PSO algorithm. Plots of the kinematic characteristics of the movement of the boom system in terms of the angle of rotation of the boom and the speed of rotation of the boom were constructed. When processing the experimental data, the dependence of the error values on the supply voltage of the propeller drive was revealed. The error in the boom rotation speed at the drive supply voltage of 90% (compared to the voltage variant of 40%) decreased by almost 15%, and the error in the boom rotation angle at the drive supply voltage of 90% (compared to the supply voltage variant of 40%) decreased by almost 3 times. The regularity has been confirmed that with an increase in the supply voltage, the error value of the system decreases. In the course of processing the experimental studies, the dynamic parameters of the installation were identified: $K_1 = 4.80 \cdot 10^{-8}$ V/(rpm)^2, $M_0 = 34.519$ Nm, $J = 24.21$ kgm^2. The obtained results will be used to optimise the plant's motion modes, and the developed identification algorithm can be used for other similar problems

Keywords: error; criterion, equipment; algorithm; crane

INTRODUCTION

Tower cranes are used in the construction of high-rise buildings and to move various loads on construction sites. These cranes must, first and foremost, be reliable, and have sufficient performance and high safety standards. This determines the relevance of studying the dynamic characteristics of tower cranes and optimising their operation for improving production processes.

One of the mechanisms of a tower crane is the slewing mechanism. Many scientists studied the optimal movement modes of this mechanism. As such, Patent of Ukraine

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No. 131788 "The method of controlling the movement of the slewing mechanism of the tower crane" (2019) synthesises the optimal mode of movement of a tower crane. The peculiarity of this study is determined by the change in the speed of the crane's slewing mechanism being implemented using the derived optimal law of movement of the mechanism, which eliminates pendulum oscillations of the load and increases the reliability and capacity of the tower crane. V. Loveikin *et al.* (2020) analysed the process of starting the tower crane slewing mechanism, provided that the crane moves at a steady-state speed, and identified the most significant factors that affect the energy, dynamic, and kinematic processes of the system. A. Trąbka (2016) presented a comprehensive method for determining the parameters characterising the main structural components of cranes. Its novelty lies in the fact that the analysis ensures the detection of deficiencies even before the installation of components on the crane structure. F. Liu *et al.* (2021) determined the oscillation and oscillation characteristics of a tower crane. They proved that the mass of the load has little effect on the spatial angle of slewing during the lifting and tilting motion and that the intensity of the change in the angle of rotation increases with the height of the tower crane. M. Zhang *et al.* (2020) developed an adaptive crane motion controller based on a dynamic model of a tower crane that ensures accurate load positioning while eliminating pendulum oscillations of the load. R. Čápková *et al.* (2019) presented a study that considers the control of crane motion using a PID controller. R. Gao *et al.* (2013) presented a method for evaluating the braking process of a tower crane. It was based on a model of the dynamics of an elastic system. V. Loveikin *et al.* (2023) considered a method for minimising the driving torque of the trolley movement mechanism during a steady-state crane slewing. It is based on the developed mathematical model of the "tower crane-load" system and minimization of the optimisation criterion by synthesising optimal control. V. Loveikin *et al.* (2022), based on a dynamic system, investigated a method for minimising trolley motion oscillations during a steady crane slewing using the movement duration criterion. W. Chen *et al.* (2020) studied a tower crane scheme to assess the stability of the structure under variable wind loads using CFD (Computational Fluid Dynamics) and time-domain analysis of wind reactions. O. Grigorov *et al.* (2020) analysed the use of a hydrodynamic drive for the movement and slwing mechanism. As a result, compared to other drives, they proved that it is effective in operation. I. Gorbatyuk & O. Bulavka (2022) presented a method for determining dynamic loads during load hoisting using calculation schemes that reflect the actual operation of the mechanism.

Optimal control of the movement of tower crane mechanisms requires a mathematical model. Its definition is a mandatory problem. However, the identification of model parameters can be quite difficult. Therefore, the research aims to develop a methodology for identifying the parameters of the model of the tower crane slewing mechanism using a laboratory installation as an example.

MATERIALS AND METHODS

An experimental laboratory installation of a tower crane (Fig. 1) with a boom length ($L = 3570$ mm) at the National University of Life and Environmental Sciences of Ukraine was used for the study. In the installation, the worm gear slewing mechanism was replaced by a propeller-driven slewing mechanism.

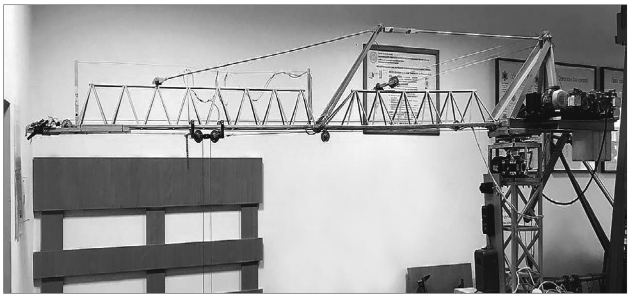

Figure 1. Experimental laboratory installation of a tower crane
Source: authors' photo

The propeller-driven slewing mechanism (Fig. 2) is installed at the end of the boom, which allows the experimental unit to reach the optimum speed and angle of rotation faster with less driving force and, accordingly, at a lower propeller speed.

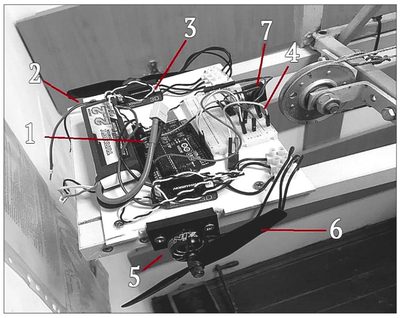

Figure 2. The structure of the propeller-driven slewing mechanism
Note: 1 – Arduino UNO board, 2 – Turnigy battery, 3 – Hobbywing XRotor Pro 25A driver, 4 – control block, 5 – iFlight XING-E Pro brushless motor, 6 – propellor, 7 – engine speed driver
Source: authors' photo

The list of equipment and its main technical characteristics used for the experiments: iFlight XING-E Pro brushless motor (China) (motor size 28.5x19.7 mm; weight 33.8 g; no-load current at 12.6 V 1.2 A; maximum power 800.5 W); Turnigy battery (China) (capacity 2200 mAh; voltage 11.1 V; dimensions: 104x27x35 mm); Hobbywing XRotor Pro motor driver (China) 25A (rated current 25 A; peak current for up to 10 seconds 40 A; frequency 50-500 Hz); AUTONICS E40S6-5000-3-T-24 incremental encoder (South Korea) (Fig. 3) (accuracy 5000 imp/rev, supply voltage 12-24 V).

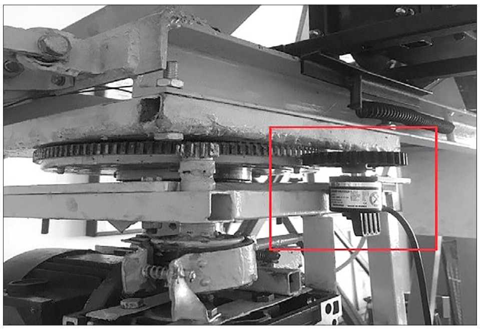

Figure 3. Incremental encoder location

Source: authors' photo

To record the data (supply voltage of the boom slewing mechanism drive, boom rotation angle, time), the Arduino UNO board (China) was used. The code was created using the Arduino IDE programming system. All data were obtained in numerical form and then processed in Wolfram Mathematica.

A data acquisition device m-DAQ 14 (Ukraine) and a laboratory power supply were used for data collection (Fig. 4).

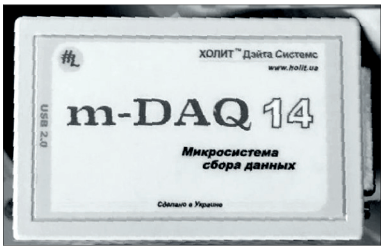

Figure 4. Data recording device m-DAQ 14

Source: authors' photo

To conduct the experiments, the supply voltage to the actuator was increased from 10% of the nominal value to 100% in 10% increments. At 10 and 20% of the nominal voltage, the dry friction forces of the boom were greater than the thrust of the propeller, so these values were not used in the future. A total of 16 experiments were conducted with the motor supply voltage ranging from 30% to 100% of the nominal (12V) value.

It was proposed to use the following system of equations (mathematical model) to describe the motion of the laboratory installation:

$$\begin{cases} M_D = K_1(PU_i)^2 L; \\ M_D - M_0 = J\ddot{\varphi}, \end{cases} \quad (1)$$

where K_1 – driving torque coefficient, H/(rpm)^2; P – drive speed ratio, rpm/V; U_i – nominal voltage that was applied to the motor on the i-th control cycle, B; L – boom length, m; M_0 – torque of dry friction forces, Nm; J – moment of inertia of the boom system, kgm^2; $\ddot{\varphi}$ – angular acceleration of the boom system, rad/s^2.

Model (1) was chosen as it well describes the physical phenomena occurring in the experimental setup (inertial, hydraulic, etc.). For example, as the motor supply voltage increases, the thrust of the propellers increases quadratically, which is the reason for the quadratic expression in the first equation (1). Unknown and requiring determination in the model (1) are the values of K_1, M_0, J. To determine them, a criterion was developed to describe the prediction error of model (1), i.e., the total deviation of the predicted data from the experimental data. This criterion is described by the following expression:

$$I = I_{\varphi.RMS} + I_{\varphi.max} + I_{\dot{\varphi}.RMS} + I_{\dot{\varphi}.max} = \sqrt{\sum_{i=1}^{690}\left(\varphi_{m.i} - \varphi_{e.i}\right)^2} + \\ + \max\left(\varphi_{m.i} - \varphi_{e.i}\right) + \sqrt{\sum_{i=1}^{690}(\dot{\varphi}_{m.i} - \dot{\varphi}_{e.i})^2} + \max\left(\dot{\varphi}_{m.i} - \dot{\varphi}_{e.i}\right), \quad (2)$$

where $I_{\varphi.RMS}$ and $I_{\dot{\varphi}.RMS}$ – root mean square error values for the boom position and rotation speed; I$I_{\varphi max}$ and $I_{\dot{\varphi}.max}$ – maximum (modulo) error values for the boom position and rotation speed; $\varphi_{m.i}$ and $\varphi_{e.i}$ – i-th values of the boom position obtained based on model (1) and experimental data, respectively; $\dot{\varphi}_{m.i}$ and $\dot{\varphi}_{e.i}$ – i-th values of the boom rotation speed were obtained based on model (1) and based on experimental data, respectively.

Criterion (2) was minimised using the Rot-Ring-PSO algorithm (Romasevych *et al.*, 2021) to identify the parameters K_1, M_0, and J.

RESULTS AND DISCUSSION

To illustrate the results, the plots (Fig. 5 and 6) and Table 1 are presented at 40% and 90% of the nominal supply voltage of the drives.

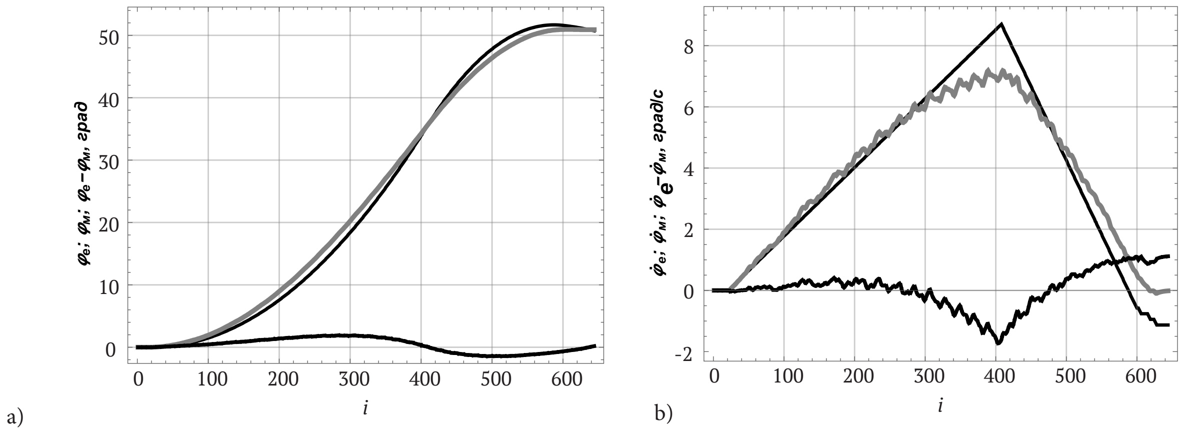

Figure 5. Plots of the kinematic characteristics of the boom system movement at a motor supply voltage of 40% of the nominal value

Note: a) by the angle of rotation of the boom, b) by the speed of rotation of the boom

Source: compiled by the authors

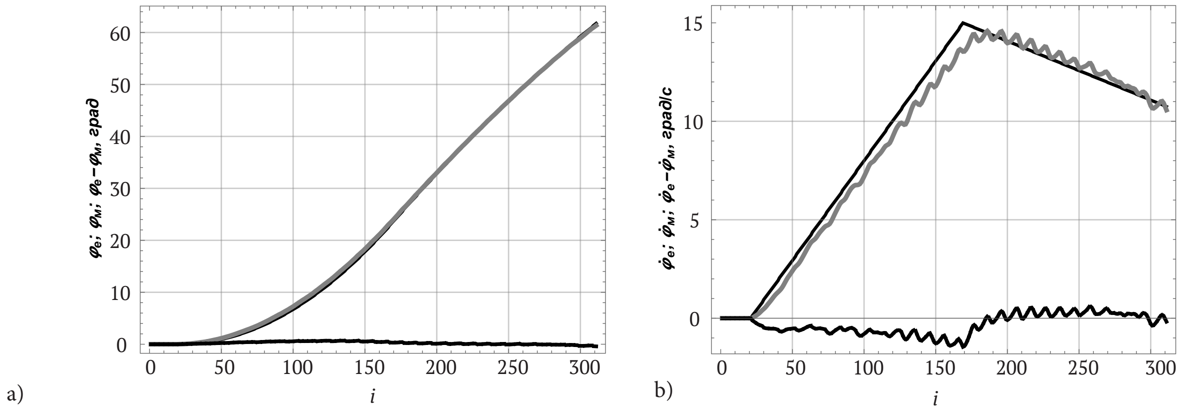

Figure 6. Plots of the kinematic characteristics of the boom system movement at a motor supply voltage of 90% of the nominal value

Note: a) by the angle of rotation of the boom, b) by the speed of rotation of the boom

Source: compiled by the authors

The black curve in Figure 5a shows a plot of changes in the angle of rotation of the boom system obtained using model (1). The grey curve represents the values obtained during the experiment. The dashed curve shows the plot of the prediction error of model (1), i.e., the difference between theoretical (predicted by model (1)) and experimental values. Figure 5b shows similar plots but for the boom slewing rate. Plots for the drive supply voltage level of 90% of the nominal value are shown in Figure 6. Figure 6a shows that the curves for the theoretical and experimental values of the boom angle of rotation coincide and their error is insignificant. A slight discrepancy between the curves is shown in Figure 6b. For a more detailed explanation of the results obtained, numerical data are provided (Table 1).

Table 1. Error-values at changes in drive voltage

Error name	Scale	Drive supply voltage, % of nominal value	
		40	90
Maximum (modulo) values			
Boom rotation speed	degrees/s	1.7317	1.4531
Boom position	degrees	1.9246	0.6965
Root mean square values			
Boom rotation speed	degrees/s	0.6374	0.5881
Boom position	degrees	1.1440	0.3503

Source: compiled by the authors

Table 1 shows that at 90% of nominal supply voltage, the maximum error in boom slewing speed is reduced by 15% (compared to 40% of nominal supply voltage). The maximum error in the boom slewing position at the same levels of drive supply voltage decreased by almost 3 times. Increasing the supply voltage of the drive has a negligible effect on the value of the RMS error in the boom slew rate. However, the factor of supply voltage has a significant effect on the RMS value of the boom position error, which decreased by almost 3 times when the supply voltage increased from 40% to 90% of the nominal value.

Comparing the values of the errors (Table 1), the increase in the motor voltage shows a tendency to reduce the error in the speed and angle of slewing of the installation. An increase in the supply voltage of the drive does not greatly affect the change in the error in the boom slewing speed, but at the same time, the error in the boom position decreases significantly with an increase in voltage.

Based on the data presented in Table 1, the error values at a drive supply voltage of 90% of the nominal value are small. This allows us to determine parameters $K_1 = 4.80 \cdot 10^{-8}$ W/(rpm)^2, $M_0 = 34.519$ Nm, $J = 24.21$ kgm^2 for this particular motor voltage level for further calculations of optimal driving modes.

H.M. Omar & A.H. Nayfeh (2005) developed a method for controlling crane mechanisms by installing two controllers. One of them provides system oscillation damping and the other provides trajectory control of the crane mechanisms. B. Jerman *et al.* (2004) presented a study of the dynamics of crane movement based on the developed mathematical model of a slewing crane.

V. Kovalenko *et al.* (2022) developed a control law for the crane control mechanism to rotate the boom with a load to a given angle φ in the shortest possible time, and at the end of the working cycle, the load should not oscillate. The problem of synthesising the laws of motion of the mechanism with optimal performance is solved by the methods of optimal control theory, namely, using the maximum principle. The basis is the differential equation of the mechanical system "boom crane – load". The common feature of this work is that it was conducted on a laboratory setup, the difference being that in our case the slewing mechanism is propeller-driven, while in the study mentioned above it is of the standard type, i.e., with a worm gearbox.

Yu. Romasevych *et al.* (2023) revealed the solution to the problem of trajectory avoidance when constructing the trajectory of a load moved by a tower crane by developing a methodology for synthesising consistent laws of motion of mechanisms. The commonality with the present study is the laboratory setup, even though the experiment was carried out using an accelerometer and a m-DAQ 14 data recording device. Thus, the laboratory setup is a fairly versatile tool for implementing the results of solving the problems of controlling tower crane mechanisms.

J. Ye & J. Huang (2022) solved the problem of controlling the movement of a tower crane boom based on the developed model of the installation. Moreover, the pendulum oscillations of the load and the influence of the movement (vibration) of the boom on them are considered. All experiments were carried out under ideal conditions in a laboratory setup, i.e., neglecting the aerodynamic effect and the influence of external weather factors. This confirms the applicability of the approach used in this study.

G. Yao *et al.* (2018) proposed to apply the finite element method and the Saint-Venant principle to conduct experiments to determine the stability of the installation structure. In contrast to this study, a 1:2 scale model of a tower crane installation was used instead.

The study of I. Doçi & S. Lajqi, (2018) aimed to determine the dynamic parameters of the tower crane slewing mechanism. This study is similar to the one cited above, although the authors additionally considered the oscillations of the plant.

CONCLUSIONS

The study involved the selection of measuring and recording equipment for experimental studies to identify a model of a laboratory installation of a tower crane with an improved slewing mechanism with propeller thrust. To determine the dependence of the angle and speed of the boom slewing from time, as well as the value of the voltage applied to the propeller drive, a code was developed in the Arduino IDE programming system.

Using the Rot-Ring-PSO algorithm, the minimum RMS values of the prediction errors of the model of the tower crane slewing mechanism (in terms of speed and angle of rotation of the boom) were determined. All calculations were performed at different motor supply voltages. It was found that at a lower motor supply voltage, the error increases, and at a higher one, it decreases. In particular, the error values are as follows: maximum boom rotation speed -1.45 deg/s, RMS -0.59 deg/s, maximum boom rotation angle -0.69 degrees, RMS -0.35 degrees.

The data obtained in the course of processing the experimental studies allowed us to identify the dynamic parameters of the plant $K_1 = 4.80 \cdot 10^{-8}$ W/(rpm)^2, $M_0 = 34.519$ Nm, $J = 24.21$ kgm^2, which will be used to implement the optimal modes of movement of the laboratory tower crane installation.

In the future, it is necessary to carry out theoretical calculations of the dependence of the boom slewing speed on the parameters that affect it: the mass of the load, trolley position on the crane boom, and the length of the rope.

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None.

CONFLICT OF INTEREST

None.

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Ідентифікація моделі установки механізму повороту баштового крана із пропелерною тягою

Анотація. Будь-які дослідження динаміки і керування механічних систем ґрунтується на адекватних математичних моделях, які містять динамічні параметри досліджуваної системи. Їх оцінка, зокрема для стрілової системи баштового крана, представляє собою окрему актуальну науково-прикладну задачу вирішення якої дасть підстави для подальших розрахунків оптимальних режимів руху механізму повороту баштового крана. Метою статті є визначення динамічних параметрів установки (момента інерції механізму повороту J, момента сил сухого тертя M_0, коефіцієнта рушійного момента K_1) та проведення планування експериментальних досліджень. Для проведення досліджень використано експериментальний метод, методи чисельної оптимізації (зокрема модифікований метод рою часток Rot-Ring-PSO), а також статистичні методи. За результатами проведених експериментів було ідентифіковано динамічні параметри математичної моделі лабораторної установки механізму повороту баштового крана із пропелерною тягою. Сформовано та за допомогою алгоритму Rot-Ring-PSO проведено мінімізацію критерію, який оцінює похибку ідентифікації параметрів $K_{1,}$ $M_{0,}$ J. Побудовано графіки кінематичних характеристик руху стрілової системи по куту повороту стріли та по швидкості повороту стріли. При обробці експериментальних даних виявлено залежність величин похибки від напруги живлення приводу пропелера. Похибка по швидкості повороту стріли при напрузі живлення приводу 90 % (у порівнянні з варіантом напруги в 40 %) зменшилась майже на 15 %, а по куту повороту стріли при напрузі живлення приводу 90 % (у порівнянні з варіантом напруги живлення 40 %) зменшилась, майже в 3 рази. Підтверджена закономірність, що при збільшенні напруги живлення зменшується величина похибки роботи системи. У ході обробки експериментальних досліджень отримано ідентифіковані динамічні параметри установки $K_1 = 4{,}80 \cdot 10^{-8}$ В/(об/хв)^2, $M_0 = 34{,}519$ Нм, $J = 24{,}21$ кгм^2. Отримані результати будуть використані для проведення оптимізації режимів руху установки, а розроблений алгоритм ідентифікації може бути використаний для інших подібних задачах

Ключові слова: похибка; критерій; обладнання; алгоритм; кран