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PID-controller tuning algorithm development for a "crane-load" dynamical system

Abstract. One of the main problems in the operation of cranes is the oscillation of the load on a flexible suspension. One of the ways to eliminate load oscillations on a flexible suspension is to use a proportional-integral-differential controller that generates a control signal for the crane movement. However, for it to function properly, it must be properly tuned. Standard approaches to tuning a PID controller, which is common in the practice of engineering calculations, do not allow solving this problem, and that is why it can be considered as a scientific and applied one. The research aims to develop an algorithm for tuning a proportional-integral-differential controller. For this purpose, a research issue was defined, which includes a mathematical model of the dynamic system, constraints on the overload capacity of the crane drive and the control function, conditions for achieving the steady-state speed of the crane and eliminating pendulum oscillations of the load on a flexible suspension. Using the modified particle swarm method, ME-D-PSO, the coefficients of the proportional-integral-differential controller were determined for a wide range of values of the load mass and the length of the flexible suspension. Based on the obtained values of the coefficients, an algorithm is presented that allows calculating the values of the coefficients for any values of the load mass and the length of the suspension. The dynamics of the movement of the crane-load system are analyzed for the smallest and largest selected parameters and for the case obtained by applying the developed algorithm. Practical application of the developed algorithm will allow obtaining optimal values of the proportional-integral-differential controller, which in turn eliminates oscillations of the load on a flexible suspension during crane operation, which in turn increases the safety of crane operation, structural durability, and increases the crane's capacity

Keywords: mathematical model; constraints; load pendulum oscillations; particle swarm method; capacity

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INTRODUCTION

It is difficult to overestimate the importance of cranes in modern manufacturing, agriculture, or logistics. To improve the quality of crane operation, it is necessary to ensure the high capacity of hoisting machines. One of the modern approaches to solving this problem is the use of PID controllers (proportional-integral-differential controllers) to control the movement of crane mechanisms. This allows optimizing crane operation by reducing dynamic loads in the crane's metal structure and drive, ensuring energy efficiency, and reducing wear and tear on crane components.

Many scientists worked on developing methods for tuning PID controllers. For example, in [1], the author proposed an algorithm for tuning the proportional, integral, and differential coefficients of a PID controller using fuzzy logic methods based on logical and linguistic models. The structure of an automatic control system based on a fuzzy PID controller was developed, the main components of which are a fuzzy PID controller and the transfer function of the plant.

The authors of [2] proposed an autotuning method that consists in estimating the key points of transients in a relay identification experiment and then applying the proposed analytical dependencies to find the optimal set of PID controller tuning parameters.

In [3], the authors proposed a method for reducing load oscillations on a flexible suspension using proportional-differential (PD) and PID controllers. The PD controller affects the dynamics of crane movement, and the PID controller affects the movement of the load. The criteria of the integral of the absolute and quadratic error were used to tune the PID controller. The particle swarm method (PSO) was used to find the optimal coefficients of the controllers.

In [4], the authors gains the PSO method to tune the PID-PD controller. The time criterion function was used for control. The main task of the criterion used is to track the time spent on lifting and lowering the load and re-regulation. The PID controller is used for positioning the overshoot crane, and the PD controller is used to control the oscillations of the load.

In [5], the authors developed a method for reducing load fluctuations using a PID controller, which was tuned using a oscillations PSO algorithm and a simulation annealing algorithm (SA).

The authors of [6] proposed the Ziegler-Nichols method to tune the PID controller. The PID controller controls the position of a crane that transports load with maximal speed.

In [7], the authors proposed a method of automatic PID controller tuning using the method of differential evolution to determine the optimal values of the controller gains that satisfy the stability criterion of Kharitonov polynomials.

In [8], the authors proposed a method for tuning a PID controller using fuzzy logic, genetic algorithms, and the bee algorithm. The research proposes replacing the PID controller with a fuzzy logic controller. PID controllers tuned using genetic and bee algorithms are used to control linear horizontal motion.

In [9], the authors propose an intelligent hybrid structure used for the online tuning of a PID controller. It is based on two adaptive neural networks, both with built-in Chebyshev and orthogonal polynomials. The first neural network with an endocrine factor is used to approximate control signals that are introduced in the form of stimuli from the environment. These, in turn, are used to adapt the second neural network, whose main task is to generate control signals.

The authors of [10] proposed an intelligent crane control system with communication delays that can learn from past mistakes and use genetic algorithms to adjust the PID controller. The new parameters of the PID controller are memorized and if a similar delay occurs in the future, the parameters are read from the memory.

In the analyzed sources, the authors concentrate on solving a narrow range of problems of moving a crane with a load with the elimination of load oscillations. However, they did not consider the factors of the duration of elimination of pendulum oscillations of the load, the constraints imposed on the crane drive mechanisms, the variability of the parameters of the dynamic system, etc. In this work, all these factors are addressed.

MATERIALS AND METHODS

To conduct the research, a two-axis dynamic model was used, as shown in Figure 1. The model (Fig. 1) has been tested for research tasks and for controlling pendulum oscillations of a load on a flexible suspension [11-13].

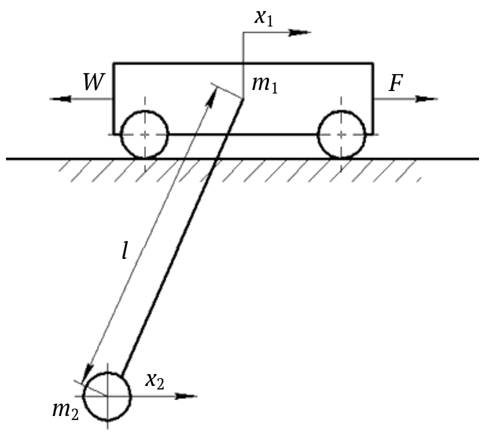

Figure 1. A dynamic model of the "crane-load" movement system
Source: compiled by the authors

The dynamic model shown in Figure 1, is described by the mathematical model (1):

$$\begin{cases} F - W = m_1\ddot{x}_1 + m_2\ddot{x}_2; \\ x_1 = x_2 + \frac{l}{g}\ddot{x}_2, \end{cases} \quad (1)$$

where x_1 and x_2 – generalized coordinates of the combined masses of the crane and the load respectively; m_1

and m_2 – total mass of the crane and load respectively; F – total driving or braking force of the crane drive; W – the force of static resistance to crane movement ($W=(m_1+m_2)g\cdot 0{,}01$); l – length of flexible load suspension; g – free-fall acceleration.

Differentiating the second equation of system (1), the following is true:

$$\dot{x}_1 = \dot{x}_2 + \frac{l}{g}\dddot{x}_2. \quad (2)$$

Differentiating the resulting expression (2), the following is true:

$$\ddot{x}_1 = \ddot{x}_2 + \frac{l}{g}\overset{IV}{x}_2. \quad (3)$$

Substituting (3) into the first equation of system (1), the result is as follows:

$$F - W = m_1\left(\ddot{x}_2 + \frac{l}{g}\overset{IV}{x}_2\right) + m_2\ddot{x}_2 = m_1\ddot{x}_2 + +m_1\frac{l}{g}\overset{IV}{x}_2 + m_2\ddot{x}_2 = (m_1+m_2)\ddot{x}_2 + m_1\frac{l}{g}\overset{IV}{x}_2. \quad (4)$$

Simplify equation (4):

$$F - W = (m_1+m_2)\ddot{x}_2 + m_1\frac{l}{g}\overset{IV}{x}_2. \quad (5)$$

Dividing the result by m_1, the output is:

$$\frac{F-W}{m_1}\frac{g}{l} = \frac{m_1+m_2}{m_1}\frac{g}{l}\ddot{x}_2 + \overset{IV}{x}_2. \quad (6)$$

Let introduce some denotions:

$$\frac{F\text{-}W}{m_1} = u, \frac{m_1+m_2}{m_1}\frac{g}{l} = \Omega^2, \frac{g}{l} = \Omega_0^2,\ \dot{x}_2 = v_2, \quad (7)$$

where u – crane acceleration, v_2 – load velocity on flexible suspension; Ω and Ω_0 – "crane-load" system for the movable and mathematical pendulum.

The following is true:

$$u\Omega_0^2 = \dot{v}_2\Omega^2 + \dddot{v}_2. \quad (8)$$

The 3rd-order equation (8) is represented as three first-order equations (9):

$$\begin{cases} \dot{v}_2 = a_2; \\ \dot{a}_2 = b_2; \\ b_{\dot{2}} = u\Omega_0^2 - a_2\Omega^2. \end{cases} \quad (9)$$

where a_2 – load acceleration; b_2 – load jerk (rate of acceleration change).

The constraints associated with the overload capacity of the crane movement drive can be represented as follows:

$$u_{min} \leqslant u \leqslant u_{max}, \quad (10)$$

where u_{min} and u_{max} – maximum and minimum possible values of crane acceleration.

To ensure the elimination of oscillations of the load on a flexible suspension and the crane's reaching a steady-state speed, a PID controller will be used [14-16]. It's generally accepted mathematical model is as follows:

$$\begin{cases} u_0 = K_P e + K_I\int_0^t e dt + K_D\frac{de}{dt}; \\ e = V - v_2, \end{cases} \quad (11)$$

where e – load velocity regulation error; K_P, K_I, and K_D – gains of the proportional, integral, and differential components of the PID controller; V – constant movement speed of the crane and load on a flexible suspension; u_0 – control function (signal from the PID output). Given the constraints imposed by (10), the following is obtained:

$$u = \begin{cases} u_0, \text{if } u_{min} \le u \le u_{max}; \\ u_{max}, \text{if } u_0 \ge u_{max}; \\ u_{min}, \text{if } u_0 \le u_{min}. \end{cases} \quad (12)$$

Thus, it is necessary to find the values K_P, K_I, and K_D, that would ensure the meeting of the control problem, namely, the output of the trolley to a steady-state speed at which the acceleration and load jerk will be equal to zero. As such, it is possible to note:

$$\begin{cases} \lim\limits_{t\to\infty} v_2 = V; \\ \lim\limits_{t\to\infty} a_2 = 0; \\ \lim\limits_{t\to\infty} b_2 = 0. \end{cases} \quad (13)$$

However, for practical purposes, the expression $t\to\infty$ does not meet the conditions of crane operation. Therefore, the expressions (13) are replaced by the following:

$$\begin{cases} v_2(T) = V \pm \Delta v; \\ a_2(T) = 0 \pm \Delta a; \\ b_2(T) = 0 \pm \Delta b, \end{cases} \quad (14)$$

where Δv, Δa, Δb – permissible deviations of the speed, acceleration, and load jerk from the set values (13) at a given time T; T – duration of the controlled crane movement mode.

For variables m_2 and l coefficients K_P, K_I and K_D of the PID controller will be different. Therefore, the problem of tuning the PID controller for different combinations of m_2 and l was solved many times. In each such problem, conditions (14) were set. The solution to each problem was obtained using the ME-D-PSO particle swarm method [17]. A swarm of 30 particles was used for the solution and 200 calculation iterations were performed. In each problem, the values of m_2 and l were varied within certain limits that correspond to the conditions of crane operation practice: the mass of the load was varied from 500 kg to 20,000 kg in increments of 500 kg; the length of the flexible load suspension was varied from 2 to 12 meters in increments of 1 meter. From the data obtained as a result of the calculation, a data array is created that includes 451 sets of gains K_R, K_I, and K_D of the PID controller.

RESULTS AND DISCUSSION

In the operation of cranes, such ratios are not common. Therefore, the PID controller needs to be set in such a way as to obtain results regardless of the load mass and the length of the flexible suspension.

For example, the case of applying this methodology to the parameters m_2=18700 kg ang l=11.5 m was considered.

The algorithm diagram for finding the PID controller coefficients is shown in Figure 2.

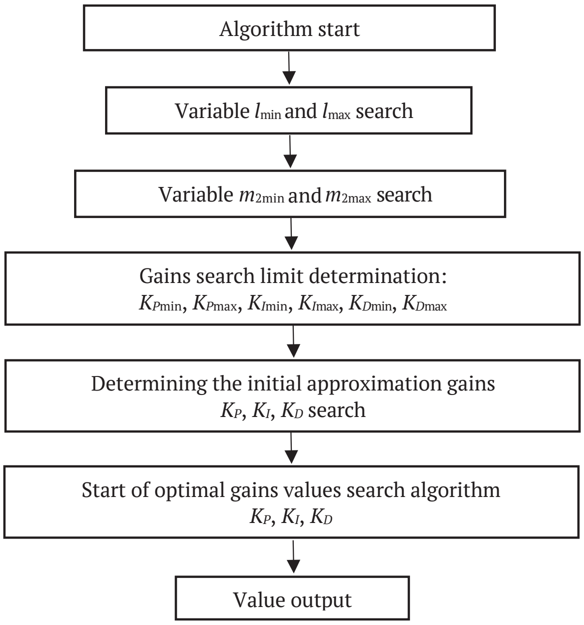

Figure 2. Diagram of searching for PID controller gains

Source: compiled by the authors

First, the tabulated values of the length of the flexible suspension and the mass of the load are found, for which the following inequalities are valid:

$$l_{min} \leq l_r \leq l_{max};\ m_{2min} \leq m_{2r} \leq m_{2max}, \tag{15}$$

where m_{2r} and l_r – the actual mass of the load and the length of the flexible suspension (certain values for the crane movement cycle with a load on a flexible suspension); l_{min} and l_{max} – the minimum and maximum tabulated values of the length of the flexible suspension for which the inequalities are valid (15); m_{2min} and m_{2max} – the minimum and maximum tabulated values of the cargo weight for which the inequalities are valid (15).

Next, the coefficients K_P, K_I, K_D are determined for four parameter combinations: l_{min}, m_{2min}; l_{min}, m_{2max}; l_{max}, m_{2min}; l_{max}, m_{2max} (Table 1).

Table 1. Values of the gains K_P, K_I, K_D for all possible combinations l_{min}, m_{2min}; l_{min}, m_{2max}; l_{max}, m_{2min}; l_{max}, m_{2max}

Coefficients Parameters	K_P	K_I	K_D
l_{min}, m_{2min}	K_{P1} = 4.023	K_{I1} = 6.491	K_{D1} = 19.756
l_{min}, m_{2max}	K_{P2} = 3.854	K_{I2} = 6.305	K_{D2} = 19.114
l_{max}, m_{2min}	K_{P3} = 2.962	K_{I3} = 4.541	K_{D3} = 14.899
l_{max}, m_{2max}	K_{P4} = 3.815	K_{I4} = 5.918	K_{D4} = 19.365

Source: compiled by the authors

Next, using the created array of gains K_P, K_I, K_D define the boundaries of their search. Using the array, the already calculated gains for the crane parameters are selected, which are one step less and one step more than the desired parameters.

For the example of calculation under consideration, using Table 1, the following search boundaries are obtained: $K_{Pmin}=K_{P3}$, $K_{Pmax}=K_{P1}$, $K_{Imin}=K_{I3}$, $K_{Imax}=K_{I1}$, $K_{Dmin}=K_{D3}$, $K_{Dmax}=K_{D1}$. For the option considered in this case: l_{min}=11 m, l_{max}=12 m, m_{2min}=18500 kg, m_{2max}=19000 kg.

The next step is to calculate initial gains values for K_P, K_I, K_D (first iteration) from which the search for values for the "crane-load" system parameter variant will begin $m_2=m_{2r}$, $l=l_r$. These initial values of the gains are determined as an arithmetic average for all 4 options (Table 1). For example, for the proportional component of the K_R, the following is true:

$$K_{P.0} = \frac{K_{P1}+K_{P2}+K_{P3}+K_{P4}}{4}, \tag{16}$$

where $K_{P.0}$ – initial gains value K_P, from which the search for value K_P is started. Initial values for gains $K_{I.0}$ and $K_{D.0}$ are determined similarly.

The following algorithm is used to find the values of the gains K_P, K_I, K_D for set values m_2 and l, based on the modified particle swarm method ME-D-PSO [10]. In this case, the optimization criterion is the following expression obtained based on conditions (14):

$$\underset{K_P,K_I,K_D}{argmin}(V - v_2(T))^2 + (a_2(T))^2 + (b_2(T))^2. \quad (17)$$

The algorithm presented in the flowchart (Figure 2) allows to quickly obtain the values of the coefficients K_P, K_I, K_D.

To analyze the dynamics of motion control of the dynamic system, three options were chosen: at the lowest values of the load mass and the length of the flexible suspension (l=2 m, m_2=500 kg) – the first option; at the highest values (l=11,5 m, m_2=18700 kg) – the second option and for (l=12 m, m_2=20000 kg) – third variant. The values of the gains used in the calculation of these parameters are shown in Table 2.

Table 2. Gains value K_P, K_I, K_D for three variants used to analyze the efficiency of the algorithm

Coefficients Parameters	K_P	K_I	K_D
l = 2 m, m_2 = 500 kg	2. 027	5. 706	12. 449
l = 11,5 m, m_2 =18700 kg	3. 348	5. 250	16. 707
l = 12 m, m_2 = 20000 kg	2. 553	3. 924	12. 883

Source: compiled by the authors based on research

Figure 3 shows the phase profile of load oscillations during crane acceleration to the steady-state speed V. As can be seen from Figure 3a, the maximum load deflection is 0.26 m, and in Figures 3b and 3c, it is 0.98 m, which is explained by the longer length of the flexible suspension. In addition, the shapes of the phase trajectories are also different, as can be seen in Figure 4. They are caused by the form of change in the driving force: for the first case, it changes smoothly, not reaching the minimum limit (minimum allowable control).

Figure 4 shows plots of changes in the driving and braking forces of the crane drive.

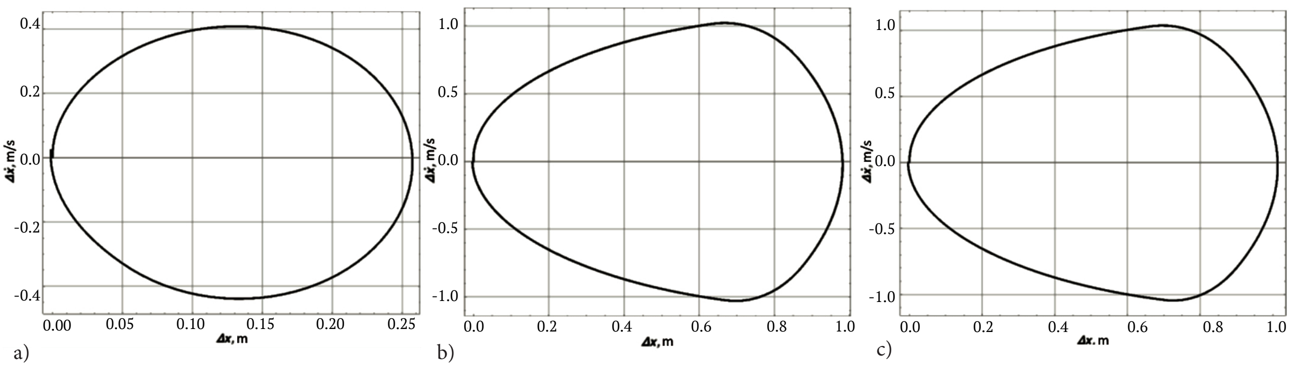

Figure 3. The phase profile of load pendulum oscillations during crane acceleration: a) l=2 m, m_2=500 kg, b) l=11.5 m, m_2=18700 kg c) l=12 m, m_2=20000 kg

Source: compiled by the authors

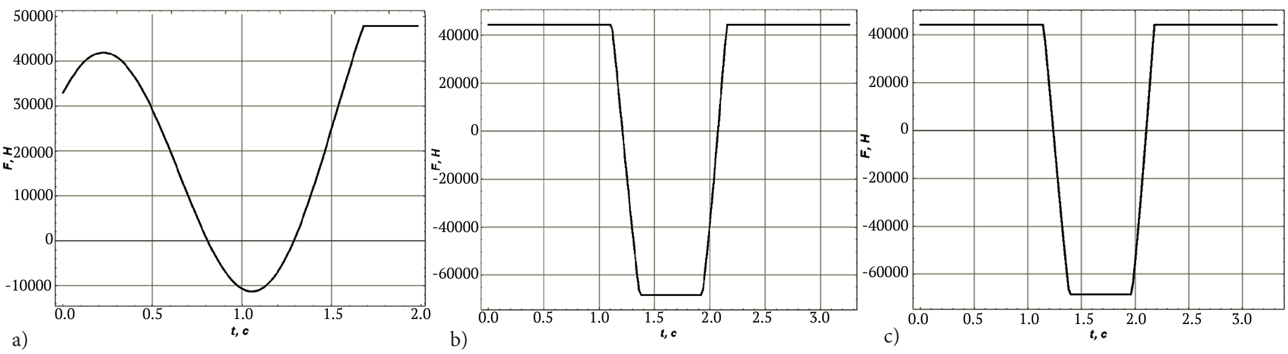

Figure 4. Plots of changes in crane drive force: a) l=2 m, m_2=500 kg, b) l=11.5 m, m_2=18700 kg c) l=12 m, m_2=20000 kg

Source: compiled by the authors

As can be seen from Figure 4, the duration of the movement of the dynamic system for the first variant is 2 s, and for the second 3.5 s. This is since with an increase in the length of the flexible suspension, the period of oscillation of the load increases, so more time is needed to eliminate oscillations in the second case.

Figure 4 demonstrates the force in both cases having a familiar character. The decrease in force for the first case

(Figure 4a) is smooth and starts at 0.4 s. The force reaches its minimum value (-11000 N) at 1.1 s. Figure 4c shows that the change in force occurs abruptly from 1.4 s to 2 s. The force reaches its minimum value (-68000 N) from 1.3 s to 1.9 s. Figure 4b is similar to Figure 4c, but in the case of Figure 4c, the change in force occurs from 1.2 s to 1.8 s. The minimum value of the force, as in the case of Figure 4c (-68000 N).

Here are also plots of changes in crane speed (Figure 5).

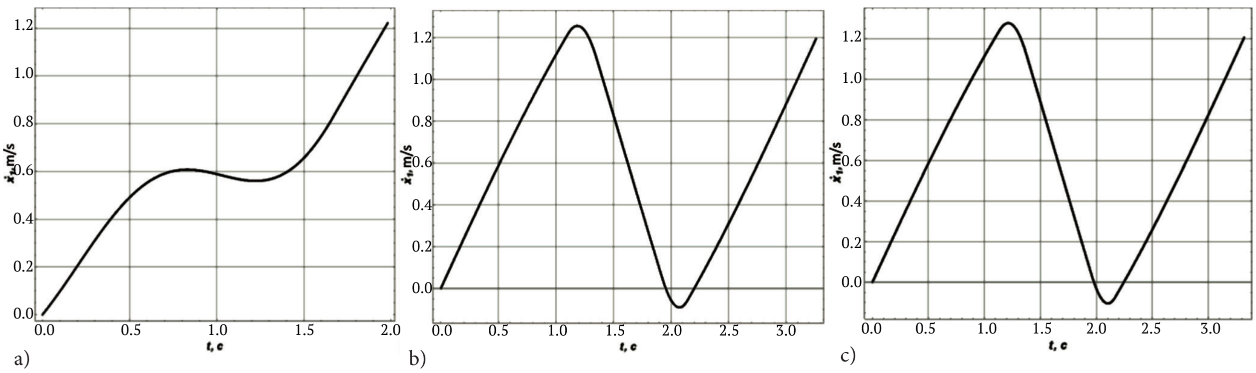

Figure 5. Plots of crane speed changes: a) l=2 m, m_2=500 kg, b) l=11.5 m, m_2=18700 kg c) l=12 m, m_2=20000 kg
Source: compiled by the authors

For the second variant (Fig. 5c), the maximum velocity of 1.22 m/s is reached at 1.2 s, after reaching the peak, the velocity decreases from 1.23 s to 3.4 s. Moreover, Figure 5c shows that for a short period, the velocity becomes negative, which is a negative factor that should be avoided. Figure 5a shows a slight (from 0.6 m/s to 0.55 m/s) decrease in velocity in the period from 0.6 s to 1.4 s, the maximum velocity, in this case, reaches 1.21 m/s, which corresponds to the numerical value of V used in the calculations. As can be seen from Figures 5c and b, they are very similar to each other, the difference between them is that in Figure 5b the velocity value becomes negative for a shorter period from 1.9 s to 2.3 s, and the maximum velocity of 1.21 m/s is reached at 1.2 s. The graphs of the crane drive power change are shown in Figure 6. The graph shown in Figure 6a shows that the minimum power value is 8000 W, i.e., for a certain period, the crane drive motor will operate in generator mode. The maximum power value is 58000 W, which is achieved at 1.98 s. The plot in Figure 6c shows that the minimum power value is much higher (-75000 W), it is reached at 1.4 s, and the maximum value is the same as for the first option and is reached at 1.25 s. The graph in Figure 6b shows that the maximum power value is 58 kW, reached at 1.1 s, and the minimum value is -70 kW, reached at 1.35s.

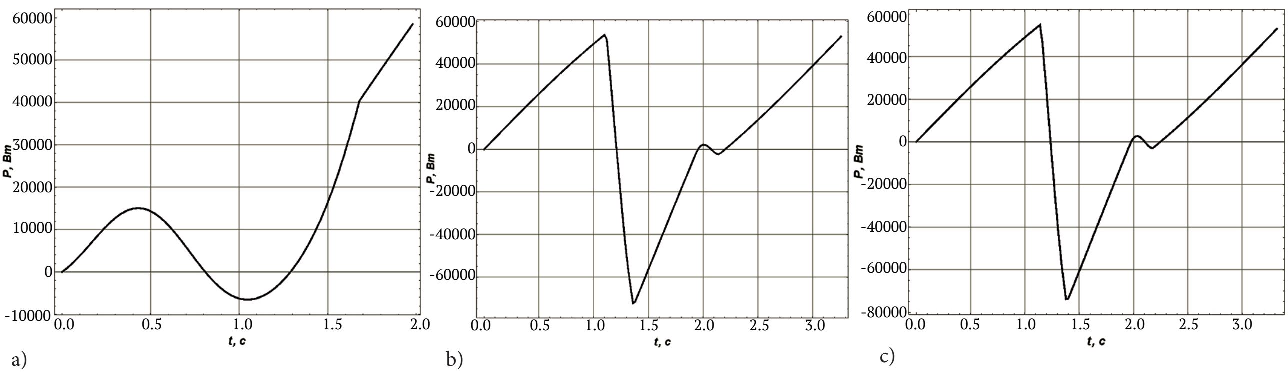

Figure 6. Crane drive power plot: a) l=2 m, m_2=500 kg, b) l=11.5 m, m_2=18700 kg c) l=12 m, m_2=20000 kg
Source: compiled by the authors

Thus, to implement the obtained control laws, regenerative braking should be provided in the drive of the crane movement mechanism. By comparing the methods and results of this research with well-known studies, in particular [3; 4], a commonality of using modifications of the PSO method [18-21] can be noted. Thus, it can be considered an effective tool for solving similar problems. These works also use mathematical models that consider pendulum effects (oscillations of a load on a flexible suspension). However, this research considers the overload capacity of the drive of the crane movement mechanism, which corresponds to the consideration of a crane as an electromechanical system. Consequently, the statement part of the problem is wider than those presented in [3; 4].

The calculation of the controller parameters in [3; 4] was performed for a limited set of system parameters (length of the flexible suspension, the mass of the load, etc.). However, in this research, an algorithm for finding the optimal PID controller gains for a wide range of system parameters that correspond to those encountered in crane operation practice was obtained. This is undoubtedly an advantage of this research over studies [3; 4]. Some authors [9; 10] used artificial neural networks to tune PID controllers, the obvious disadvantage of which is the need to process a large amount of data, which significantly slows down the process of tuning the PID controller and increases the requirements for computing. In the case of [10], the tuning process requires a constant connection to the Internet. The use of such tuning methods increases the flexibility of the parameters during tuning but reduces the autonomy. In studies [6; 7], the authors proposed methods for autotuning PID controllers; the models used in the works are based on an analytical model derived from the transient characteristics of the control object. In calculations, analytical models require the use of several assumptions, for example, the nonlinearity of the object is not considered, and differentiation errors are not considered. The calculations do not address system constraints, which leads to an error in system identification and a decrease in the quality of optimization. The research partially considers the features of this system. In [2], the author used the fuzzy logic method to tune the PID controller, which is based on several sets and variables that are formed by a base (set) of rules formed by a human expert. The rules used may be contradictory or incomplete. The fuzzy logic method used in this research is effective only for a narrow range of system parameters (drive power of the crane movement mechanism, load mass, and length of the flexible suspension). This does not involve changing the microcontroller and other means (sensors, controlled drives, etc.), so in terms of software and hardware, such approaches can be equally effective for different crane parameters. In this research, the author did not consider changes in the parameters (load mass, suspension length, etc.) of the dynamic system. In [8], the authors use several different methods to tune PID controllers to realize optimal control (genetic algorithm, bee algorithm, which belongs to the class of swarm intelligence methods, and fuzzy logic method). The genetic algorithm was used to tune the PID controllers responsible for the crane movement. The linear movement of the crane is provided by PID controllers configured using the bee and genetic algorithms, and the positioning of the load is provided by PID controllers configured using fuzzy logic. Using other parameters requires recalculation of the controller settings each time. This places stringent requirements on the computing capabilities of the crane motion control system, which, of course, makes it more expensive than similar systems. The algorithm developed by the authors and presented in this article makes it possible to find the optimal values of the PID controller settings using insignificant computing resources.

CONCLUSIONS

The problem of tuning the gains of a PID controller that contrals the movement of the crane-load system is formulated. Using the ME-D-PSO method, solutions to the problem were obtained for sets of changes in system parameters: load mass – from 500 kg to 20,000 kg in increments of 500 kg; length of flexible load suspension from 2 to 12 meters in increments of 1 meter.

An algorithm for calculating the values of the coefficients of the PID controller is developed. The first step of the algorithm is to determine the upper and lower limits of the parameters m_2 and l for which the PID-controller gains K_P, K_I, K_D are searched. Next, an initial approximation of the gains values is determined, and the values of the gains are determined using the modified particle swarm method ME-D-PSO.

The dynamics of movement of the crane-load system are analyzed for two variants of the load mass and the length of the flexible load suspension (l=2 m, m_2=500 kg; l=12 m, m_2=20000 kg and l=11.5 m, m_2=18700 kg). As can be seen from the results of the study, the second and third options are similar. The duration of the movement at the minimum parameters is 2 s, and at the maximum – 3.5 s, which is caused by a longer period of pendulum oscillations of the load. The maximum amplitude of load deflection for the first variant is 0.26 m, and for the second and third – 0.98 m. The maximum power for all the considered options is the same – 58 kW. In all the cases considered, the pendulum oscillations of the load are eliminated during the crane acceleration.

Further research in this area should address other constraints on the kinematic, energy, and dynamic characteristics of the system's motion, as well as other criteria for tuning the PID controller (RMS value of the driving force, the amplitude of pendulum oscillations, etc.).

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Розробка алгоритму налаштування ПІД-регулятора руху системи «кран-вантаж»

Анотація. Однією із основних проблем при роботі вантажопідйомних кранів є коливання вантажу на гнучкому підвісі. Одним із способів усунення коливань вантажу на гнучкому підвісі є застосування пропорційно-інтегрально-диференціального регулятора, що виконує формування керуючого сигналу руху крана. Однак, для виконання цієї задачі він повинен бути відповідним чином налаштованим. Стандартні підходи до налаштування ПІД-регулятора, які поширені у практиці інженерних розрахунків, не дозволяють вирішити цю задачу, і саме тому її можна розглядати як науково-прикладну. Основною метою роботи була розробка алгоритму налаштування пропорційно-інтегрально-диференціального регулятора. Для цього виконано постановку задачі, яка включає: математичну модель динамічної системи, обмеження на перевантажувальну здатність приводу крану та функцію регулювання, умови досягнення усталеної швидкості руху крана та усунення маятникових коливань вантажу на гнучкому підвісі. Користуючись модифікованим методом рою часток ME-D-PSO визначено коефіцієнти пропорційно-інтегрально-диференціального регулятора для широкого діапазону значень маси вантажу та довжини гнучкого підвісу. На основі отриманих значень коефіцієнтів представлено алгоритм, який дозволяє розраховувати значення коефіцієнтів для будь-яких значень маси вантажу та довжини підвісу. Проведено аналіз динаміки руху системи «кран-вантаж» для найменших і найбільших обраних параметрів та для випадку, який отриманий на основі застосування розробленого алгоритму. Практичне застосування розробленого алгоритму дозволить отримати оптимальні значення пропорційно-інтегрально-диференціального регулятора, що в свою чергу усуває коливання вантажу на гнучкому підвісі при роботі крану, що в свою чергу підвищує безпеку експлуатації кранів, довговічність конструкції, збільшує продуктивність роботи крана

Ключові слова: математична модель; обмеження; маятникові коливання вантажу; метод рою часточок; продуктивність