Calculation of the Required Power of Electric Motors for Overhead Crane Movement Mechanisms Using the Statistical Method

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Abstract. The aim of this study is to determine the functional dependence for the calculation of the required power of the electric motor of the traveling mechanism of the overhead traveling crane on the load capacity and span. The set aim is achieved by solving the following problems: figuring out how to calculate the power of the electric motor of the overhead traveling crane mechanism by the traditional methodology; selecting constants known from the initial data and variables that require additional calculation; collecting and arranging statistical data - values of variables for overhead cranes of different spans and load capacities; determining regression equations based on statistical data - functions of two variables from the load capacity and span; substituting the equations for the power of the overhead traveling crane mechanism from the load capacity and span. In the process of solving the set tasks, it was found that the greatest difficulty is the determination of static resistance and crane mass. Traditional calculation of these parameters requires the use of additional reference data and making design decisions of high responsibility. When searching for regression equations, the cubic model is-used, which provides high accuracy and does not overload the equation with summands. The most important result is the derivation of the final expression for determining the motor power as a function of two variables - load capacity and span. The importance of the obtained results is that the proposed method of calculation significantly reduces the time for the selection of the electric motor in the design of a new crane, because there is no need to calculate or select additional parameters that are included in the traditional calculation. The proposed regularity is easier to integrate into computer-aided design systems. Since the calculation was based on statistical data of cranes manufactured and successfully operating, the probability of erroneous calculation is practically excluded.
***Keywords*:** overhead crane, traveling mechanism, automation, calculation, electric motor power.

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Calculul puterii necesare a motoarelor electrice ale mecanismelor de deplasare a macaralelor de pod prin metoda statistică

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Rezumat. Scopul studiului este de a determina dependența funcțională pentru calcularea puterii motorului electric a mecanismului de deplasare al unui pod rulant în dependența de încărcare și deschidere. Scopul stabilit se realizează prin rezolvarea următoarelor probleme: analiza ecuației de calcul a puterii motorului electric a mecanismului de deplasare al ruloului rulant prin metoda tradițională; identificarea constantelor cunoscute din datele sursă și a variabilelor care necesită calcul suplimentar; colectarea și organizarea datelor statistice - valori variabile pentru podurile rulante de diferite deschideri și capacități de ridicare; determinarea ecuațiilor de regresie pe baza datelor statistice - funcții a două variabile din capacitatea de încărcare și span; înlocuirea ecuațiilor de regresie în ecuația originală în loc de variabile. În procesul de rezolvare a problemelor, s-a constatat că cea mai mare dificultate este în determinarea rezistenței statice și a masei macaralei. Calculul tradițional al acestor parametri necesită utilizarea unor date de referință suplimentare și adoptarea unor decizii de proiectare de înaltă responsabilitate. La căutarea ecuațiilor de regresie, a fost folosit un model cubic, care oferă o precizie ridicată și nu supraîncărcă ecuația cu termeni. Cel mai semnificativ rezultat este obținerea unei expresii finale pentru determinarea puterii motorului electric în funcție de două variabile - capacitatea de sarcină și deschiderea. Semnificația rezultatelor obținute este că metoda de calcul propusă reduce semnificativ timpul de alegere a unui motor electric la proiectarea unei noi macarale, deoarece nu este nevoie să se calculeze sau să selecteze parametri suplimentari care sunt incluși în calculul tradițional. Modelul propus este mai ușor de integrat în sistemele de proiectare asistată de computer. Deoarece calculul se bazează pe date statistice de la macarale fabricate și care operează cu succes, probabilitatea unui calcul eronat este practic eliminată.
***Cuvinte-cheie*:** pod rulant, mecanism de deplasare, automatizare, calcul, putere motorului electric.

Расчет необходимой мощности электродвигателей механизмов движения мостовых кранов статистическим методом

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Аннотация. Целью исследования является определение функциональной зависимости для расчёта мощности электродвигателя механизма передвижения мостового крана от грузоподъёмности и пролёта. Поставленная цель достигается за счёт решения таких задач: анализ уравнения для расчёта мощности электродвигателя механизма передвижения мостового крана по традиционной методике; выделение констант, известных из исходных данных и переменных, требующих дополнительного расчёта; сбор и упорядочивание статистических данных – значений переменных для мостовых кранов разных пролётов и грузоподъемностей; определение уравнений регрессии на основании статистических данных – функций двух переменных от грузоподъемности и пролёта; подстановка уравнений регрессии в исходное уравнение вместо переменных. В процессе решения поставленных задач было установлено, что наибольшую сложность вызывает определение статического сопротивления и массы крана. Традиционный расчёт этих параметров требует использования дополнительных справочных данных и принятия конструкторских решений высокой ответственности. При поиске уравнений регрессии использована кубическая модель, которая обеспечивает высокую точность и не перегружает уравнение слагаемыми. Наиболее существенным результатом является получение итогового выражения для определения мощности электродвигателя как функции двух переменных – грузоподъемности и пролёта. Значимость полученных результатов состоит в том, что предложенный метод расчёта значительно сокращает время на выбор электродвигателя при проектировании нового крана, поскольку нет необходимости рассчитывать или выбирать дополнительные параметры, которые входят в традиционный расчёт. Предложенную закономерность проще интегрировать в системы автоматизированного проектирования. Поскольку расчёт опирается на статистические данные изготовленных и успешно действующих кранов вероятность ошибочного расчёта практически исключается.

Ключевые слова: мостовой кран, механизм передвижения, автоматизация, расчёт, мощность двигателя.

INTRODUCTION

Scientific schools have created a significant theoretical foundation for the design of hoisting machine mechanisms and confirmed their strength, stability, endurance and other indicators of safe operation. In addition, enterprises have accumulated a large amount of data on the actual designs of manufactured and operating machines.

However, approaches to designing new machines in most cases still involve a large amount of manual work.

The traditional calculation of the drive power of a crane movement mechanism requires determining the resistance to static movement of the crane. In turn, this parameter depends on specific indicators - wheel diameter, journal diameter, and the coefficient of the flange, which can vary depending on the type of current supply, as well as the shape of the rail and wheel surfaces (friction resistance).

That is, at this stage we need to make numerous design decisions, which is very costly. To solve this problem, we propose to change the approach using mathematical methods of statistics.

This will automate the calculation process and reduce the design time, which is relevant today.

RESEARCH OVERVIEW

The calculation of the travel mechanism is described in detail in a number of reference books and standards, including papers. The relevance of the topic is also confirmed by numerous recent studies in the field of overhead crane traveling mechanisms and the improvement of the calculation of force factors acting during their operation.

Article [1] presents the calculation of engine power for the transmission of electric vehicles. The paper takes into account various forces that have an impact during movement: rolling resistance, aerodynamic resistance, active resistance when climbing a slope, and dynamic forces in transient modes. The calculation is used to select the power of the motor, converter, and battery and converter capacity. The advantage of this study is that the resulting model can be used for analysis of various performance parameters, such as acceleration time, vehicle range, and battery charge level. The disadvantages include the need to include numerous parameters of a real vehicle and driving conditions in the model.

Paper [2] calculates the drive power of an amphibious bus, including calculations of wind resistance, rolling resistance, force acting on a slope, torque, and electricity consumption calculation. In this case, the use of asynchronous motors is envisaged. As in other studies, a standard set of input data is used here to calculate and select the drive power equipment

Paper [3] investigates the drive of a semi-automatic machine for sawing metal parts. The value of the drive power required to equip the machine, depending on the cross-sectional area of the workpiece, was determined. The disadvantage of the article is that the results of the study are supposed to be used for the production of future structures, but they have not been generalised, so there may be difficulties in their practical use.

Paper [4] considers the problem of determining the motor power when using a frequency converter and a constantly changing speed. Although the study considers pump and fan drives, it is also relevant for crane drives because they also operate at intermediate speeds for a long time and are mostly frequency-controlled. The difficulty in effectively applying this approach to crane mechanisms is that the proposed methods for improving drive efficiency are based on efficiency coefficients, which for liquid and gas pumps vary over a much wider range.

Paper [5] investigates a new and simplified method for estimating the parameters of an equivalent circuit of induction motors based on data from a manufacturer's catalogue. The method demonstrates high convergence and calculation speed. The disadvantage of the study is the possible limitation of the application of the proposed method due to the fact, that there may be significant deviations from the standard parameters in real motors, or the reference data may not have sufficient initial parameters for calculation.

Paper [6] proposes a method for rapid sizing of electrical machine components that uses maximum torque, maximum speed, maximum power, and efficiency as input parameters. All unknown parameters are determined using a pre-created motor database using statistical interpolation methods. The method is fast and accurate enough to be used in the automated design of automotive transmissions. A disadvantage is the lack of coverage of the issue of selecting specific design features of some engines that may not be available in the database, such as the need to equip additional engine cooling systems.

The paper [7] investigates power losses, analyses and evaluates the test results of induction motors powered by a pulse-width modulated (PWM) converter. The article proposes new methods for determining harmonic losses by means of calculations and measurements, which have been proven to be appropriate for different voltage harmonics. The main drawback of the study is that it focuses only on certain pulsation frequencies, which may not fully cover the entire harmonic loss picture. The impact of variable operating conditions and different types of loads on harmonic losses was not taken into account.

Article [8] investigates the dynamic loads that occur in the rope systems of overhead crane lifting mechanisms during start-up and braking, and their reduction. Methods of mathematical modeling of the dynamics of loading processes and optimisation of the modes of movement of drive mechanisms are used to reduce dynamic loads. Integral optimisation criteria and methods of calculus of variations are used. The main disadvantage of the study is that the presented dynamic models make it difficult to take into account the complexity of real crane mechanical systems when implementing the results on existing machines.

Paper [9] proposes a scheme for using a curved section of the crane track based on the Bézier curve to solve the problems of rail damage and wheel jamming during crane turning. The parameters of the Bézier curve are optimised using a heuristic global optimisation algorithm with multiple starting points, and the trajectory of the outer rail is selected using Hermite interpolation. The disadvantage of this study is that it does not take into account possible mechanical wear and deformation that may occur during long-term operation.

Article [10] investigates an automatic positioning system for an overhead crane based on RFID technology to improve the accuracy and efficiency of work. The braking process of the crane movement mechanism is analysed, the braking distance and braking time are calculated under different conditions. The dependence between the braking distance and braking time under normal braking conditions, as well as the dynamics of the braking distance during emergency braking, were obtained. The main drawback of the study is the possibility of underestimating the impact of external factors, such as weather conditions or electromagnetic interference, on the operation of the RFID system in real-world conditions.

In [11], a mechanical system of a pump and motor with power recovery was developed using theoretical analysis and calculations, and numerical modeling of the workflow was carried out. The methods of loading the system, the efficiency of energy recovery, and simulated typical operating conditions. The strategy for managing the minimum system overflow by adjusting the hydraulic motor displacement is analysed. The main drawback of the study is the lack of coverage of the issue of potential wear of elements during long-term operation in real conditions and the corresponding change in their parameters, which may affect the results of simulating system efficiency.

Paper [12] considers the development of an optimal energy consumption strategy for trains based on a model of a hybrid train powertrain. The optimal points of energy consumption characteristics in different modes of movement are investigated, and a map of the optimal integrated train efficiency is generated based on the analysis of energy flows in different modes of operation. The simulation of optimising the overall train efficiency was performed in MATLAB/Simulink software. A disadvantage of the study is the lack of certainty regarding the applicability of the proposed methods for different conditions of real tachograms, as well as the need for preliminary calculation of traffic modes.

Paper [13] proposes a system for the rapid selection of electric drive elements for buses based on a polynomial function that takes into account the characteristics of the route (or traffic cycle) and the technical characteristics of the traction system components. The proposed approach makes it possible to quickly assemble electric traction devices based on driving conditions at the initial stage of vehicle design. The calculation system is implemented as computer software in combination with a database of motors, inverters, drive train systems and batteries. It remains unclear how the subsequent confirmation of the calculations is performed and at what stage it should be performed, since it is noted that the system also performs functions during the operation of the created drive on a real vehicle.

Article [14] discusses trends in the development of electric drive technologies for passenger electric and hybrid vehicles. It discusses promising materials and technologies for power electronics and electric motors, as well as the challenges and opportunities for creating even more efficient designs to meet the needs of next-generation vehicles. Some innovative drive and motor designs are discussed that have the potential to achieve the technical goals. The disadvantage of this publication is that the study does not sufficiently address the design of advanced drives. Taking into account the increasing complexity of drive systems, the calculation becomes more complicated, which requires new approaches to the selection of their elements.

The paper [15] investigates the problem of selecting the power of an electric motor for electric vehicles in order to ensure optimal performance and reduce costs. The paper presents a new approach to predicting the correct rated motor power based on a comparison with four different methods, which allows for a balance between cost and performance. The main drawback of the study is the difficulty of taking into account possible variations in operating conditions that may affect the accuracy of motor power selection.

Paper [16] presents a methodology for the parametric design of a drive system for electric vehicles. The proposed methodology determines the nominal values and maximum capabilities of the powertrain based on the characteristics of the vehicle: engine power, torque, speed and battery capacity, maximum speed and acceleration for the most difficult road conditions. The methodology is discussed with the example of a specific car model, and the performance of the cars is evaluated through simulations with two different driving cycles. This is another limitation of the study, as the simulation results may not take into account potential technical parameters when scaling the methodology to other models of electric vehicles.

Paper [17] discusses the issues of calculating the optimal size of energy sources and strategies for hybrid electric vehicles with fuel cells. The sizing algorithm begins with calculating the power demand, which is determined by the mechanical characteristics of the vehicle. The calculation is based on the instantaneous speed of the selected driving cycle and the current instantaneous road slope, after which the algorithm proceeds to determine the mechanical power required by the engine. The main drawback of the study is the potential underestimation of the impact of long-term component degradation and operating conditions on system efficiency. The study does not take into account the possible difficulties in scaling the methodology to other types of hybrid vehicles with different technical characteristics.

In many works the calculations are based on the study of crane dynamics. In [18], a model describing the oscillations of a spherical pendulum is elaborated.

The paper [19] considers the dynamic model of a rotating tower crane with a saddle-shaped boom and a mobile trolley with a load on a flexible suspension. The equations of motion of this mechanical system, which are Lagrange equations, are constructed. The tangential and radial vibrations of the load are investigated. The main attention is paid to the dynamic processes associated with low-frequency oscillations caused by the load on the flexible suspension and the drive of the crane slewing mechanism. Dynamic processes in the links of the crane steel structure have not been studied until now.

The issue of studying the residual life and damage of a crane in operation is considered in [20]. The issue of studying effective stresses is also covered in [21]. However, this approach allows to assess only the effects of load perception. To reduce the amount of fatigue damage, it is necessary to study dynamic processes, since electric drives that affect the effective stresses determine the rate of damage accumulation.

In the article [22], a mathematical model is considered, which describes the multi-axial movement of the machine during movement over bumps. The generalized coordinates are: vertical and angular displacement of the center of gravity in the lateral plane. In the process of modeling, it was found that sufficient damping of oscillations is achieved much earlier than with extreme aperiodic motion.

Models based on LaGrange equations for tower cranes are described in the works by Loveikin et al. In [23], the mechanism of sweep of the load of a tower crane is considered. The movement of the load trolley, which is moved along the horizontal saddle jib, is optimized. Optimization is performed using the Euler-Poisson equation with the integral criterion. Thus, dynamic loads and energy losses are reduced.

Thus, from the literature review, it can be seen that the issue of improving the calculation in terms of determining the drive power remains insufficiently studied. In particular, many studies are devoted to the mechanisms of movement of pneumatic wheeled vehicles - electric vehicles. Automation and improvement of the calculation of the required power for crane movement mechanisms are not covered, so this issue requires further study.

AIM AND OBJECTIVES

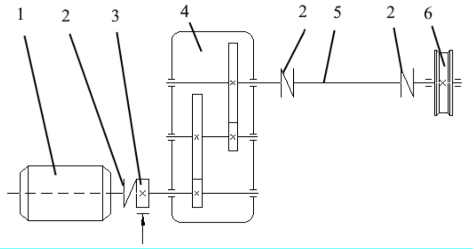
The aim and objectives of the study are to form a functional dependence for determining the power of the electric motor of the overhead crane movement mechanism on the lifting capacity and span, which will automate the selection of electric motors.

To achieve this goal, it is necessary to solve the following tasks: to analyse the equation for calculating the power of the electric motor of the overhead crane movement mechanism using the traditional method; to identify the constants known from the initial data and the variables that require additional calculation; to collect and organize statistical data - the values of variables for overhead cranes of different spans and lifting capacities; to determine the regression equation based on statistical data - functions of two variables on lifting capacity and span; and to obtain the end.

The subject of the study is how to determine the required power of the electric motor of the traveling mechanism.

RESEARCH RESULTS

To achieve this goal, it is necessary to consider the structure of a typical traveling mechanism and determine its components and parameters that need to be known for calculation. Fig. 1 shows a diagram of an overhead crane travel mechanism. With a separate drive, each side of the crane is equipped with such a mechanism. The mechanism consists of an electric motor with a spur gearbox, an intermediate shaft, clutches, wheel and a brake.



1 – electric motor, 2 – clutches, 3 – brake, 4 - spur gearbox, 5 – intermediate shaft, 6 - wheel

Fig. 1. Kinematic diagram of the crane movement mechanism.

Since two such mechanisms are installed on the crane, the power of each is determined by the formula:

$$N_c = 0,6 \cdot N \qquad (1)$$

where: N is the power required to move the crane at a given speed, determined taking into account inertial loads:

$$N = \frac{W_0 \cdot V_K}{1000 \cdot 60 \cdot \varphi_{av} \cdot \eta_m}, \qquad (2)$$

where: φ_{av} is the motor overload factor (assigned in advance);

η_m - efficiency of the movement mechanism;

V_K - crane travel speed.

W_0 - resistance to movement of a loaded crane, taking into account inertial forces:

$$W_0 = W + 1,2 \cdot \left(m_к + m_Q\right) \cdot a \qquad (3)$$

where: a - average acceleration of the crane during acceleration;

1.2 - coefficient taking into account the inertia of rotating masses.

W – is the resistance to movement of a crane with a rated load due to friction forces and track slope, determined by the formula:

$$W = \left(Q + G_K\right) \cdot \left(\frac{f \cdot d_{sh} + 2 \cdot \mu}{D_W} \cdot k_r + \alpha\right), \qquad (4)$$

where: Q is the nominal weight of the load to be lifted by the crane;

G_K is the tare weight of the crane, which is selected according to reference tables or nomograms;

f is the sliding friction coefficient of roller bearings;

D_W is the diameter of the running wheel, which is selected depending on the wheel pressure and the speed of movement;

d_{sh} - diameter of the wheel shaft:

$$d_ц = (0,2...0,25) \cdot D_{х.к} \qquad (5)$$

μ is the coefficient of friction of the running wheel on the rail, which is selected from the reference tables depending on the diameter of the running wheel and the type of rail;

k_r is a coefficient that takes into account the additional friction resistance of the running wheel flanges against the head rails

α - track inclination.

Thus, it can be seen that to determine the resistance to crane movement, it is necessary to make a number of crucial decisions that are usually made manually: determine the diameter of the wheel of the future crane, the diameter of its trunnion, determine the weight of the crane, and select a number of table coefficients of friction and additional resistance.

In accordance with the proposed approach, the statistical data were collected by the authors based on projects completed by the Kharkiv Crane Machinery Plant of previously calculated and designed cranes of various spans and lifting capacities (Table 1). Table 1 shows the values of the travel resistance in N for overhead cranes of a number of lifting capacities (from 15 t to 50 t) and spans (from 19.5 m to 34.5 m). These data are considered experimental and will be considered as the measured values. Then it is possible to apply the methods of mathematical statistics.

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Table 1

Traveling resistance data for cranes with different lifting capacities and spans.

Span, m / load, t	19,5	22,5	25,5	28,5	31,5	34,5
15	5550	5980	6400	6900	7420	8390
20	6190	6480	6850	7410	7820	8860
30	6580	6910	7370	7900	8420	9440
50	9860	10210	11000	11480	12040	12870

To test the null hypothesis of a normal distribution of the experimental data, we studied the distribution of residuals – the deviations of the experimental results from the theoretical values of the mathematical model. There are 24 data in Table 1, and it is inappropriate to use the Pearson test with this amount of data, since we can only build a histogram on 4 or 5 segments. Therefore, we used the Kolmogorov test. The mathematical expectation of the residuals *r*i (residuals mean) is m_R=-2.5769×10-12, and their standard deviation is s_R=72.3167. The values $x_i=(r_i-m_R)/s_R$ were calculated, and the Kolmogorov test was performed for them to compare them with the standard normal distribution.

In fig. 2 shows the empirical cumulative distribution function of xi values and the theoretical standard normal cumulative distribution function. The Kolmogorov statistic is 0.09798, which is less than the critical value of 0.3229. Therefore, at the significance level of α=0.01, there is no reason to reject the null hypothesis at the 1% significance level.

Based on the table data, we will look for a function of two variables that can be used to calculate the resistance to movement for arbitrary values of load capacity and span.

For obtaining function constructed mathematical model y in the form of a polynomial of optimal degree:

$$\begin{aligned} z_W &= b_1 + b_2 \cdot x + b_3 \cdot y + b_4 \cdot x^2 + \\ &+ b_5 \cdot x \cdot y + b_6 \cdot y^2 + b_7 \cdot x^3 + \\ &+ b_8 \cdot x^2 \cdot y + b_9 \cdot x \cdot y^2 + b_{10} \cdot y^3 + \ldots \end{aligned} \tag{6}$$

where: x - span, m,
y - load capacity, tons.

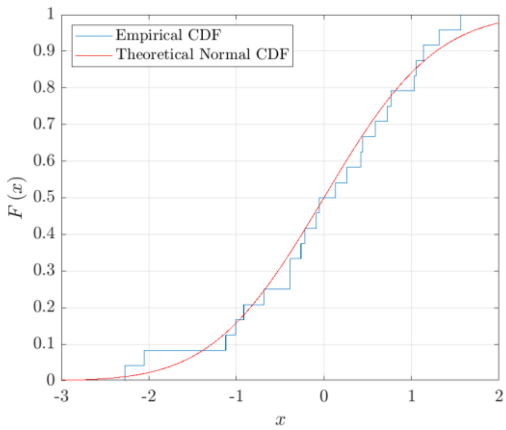

Fig.2. Empirical and theoretical standard normal cumulative distribution function for movement resistance.

To apply the methods of mathematical statistics, the following hypotheses must be assumed: independence of the measured data, absence of systematic errors and omissions, randomness of the influencing factors, absence of an influencing factor that is predominant or decisive, distribution of values in accordance with the normal law, and equal accuracy of all the measured data. Because of applying the appropriate assumption, we can use the least squares method to find the coefficients of the functional dependence equation.

According to the least squares method, the coefficients b_k are choosen so that the sum of the squared deviations of the experimental data is the minimum value z_{ij} from theoretical $z(x_i, y_j)$ acceptance.

From the point of view of linear algebra, the vector of best coefficients is an orthogonal projection of the column vector of experimental measurements z of length n onto an m-dimensional Euclidean space, which is a linear envelope of basis vectors formed from functions 1, x, y, etc. The number of basis functions m must not exceed the number of measurements n. In our case, the vector column of experimental data z of length 24 is as follows:

$$z = \begin{pmatrix} 5550 \; 6190 \; 6580 \; 9860 \; 5980 \\ 6420 \; \ldots \; 9440 \; 12870 \end{pmatrix}^T \tag{7}$$

The first three basis vectors are formed from the basis functions:

$$1 = (1\;1\;1\;1\;1\;1\;\ldots\;1\;1)^T \tag{8}$$

$$x = \begin{pmatrix} 19{,}5 \; 19{,}5 \; 19{,}5 \; 19{,}5 \; 22{,}5 \\ 22{,}5 \; \ldots \; 34{,}5 \; 34{,}5 \end{pmatrix}^T \tag{9}$$

$$y = (15\;20\;30\;50\;15\;20\;\ldots\;30\;50)^T \tag{10}$$

and the subsequent ones are constructed from different products (or powers) of the functions

1, x, y by elementwise multiplication (or exponentiation) of the corresponding elements of the first three basis vectors. In the constructed polynomial model, the number of basis functions is $m < n$, and the corresponding basis vectors are linearly independent. The matrix Ψ, constructed from these column vectors, will have full column rank m:

$$\Psi = (1\ x\ y\ ...) . \qquad (11)$$

Model coefficients:

$$b = (b_1\ b_2 ... b_m)^T \qquad (12)$$

is found by solving the orthogonal projection problem:

$$b = (\Psi^T \cdot \Psi)^{-1} \cdot \Psi^T \cdot z \qquad (13)$$

Assuming that the scatter of the experimental data is normal, we use Student's t-test to find the confidence integrals for the model parameters b_k.

If the generalised variance of the experimental data *D* is known, then the adequacy of the model can be assessed by comparing it with the sample variance of the model, which is calculated as follows:

$$D_m = L_{\min} / f \ , \qquad (14)$$

Where: $L_{\min}$ is the minimum value of the likelihood function:

$$L_{\min} = \|z - \Psi \cdot b\|^2 ; \qquad (15)$$

f is the number of degrees of freedom equal to the difference between the number of data and the number of basis functions:

$$f = n - m . \qquad (16)$$

When adding new basis functions to the model, the numerator of the sample variances $L_{\min}$, but the denominator also decreases f. Therefore, the sample variance can be either decrease or increase. If the addition of a new basis function to the model reduces its variance significantly, then it makes sense to include the new basis function, and if not, then it does not.

Now we can evaluate the adequacy of the model using the Fisher's F-criterion: If the ratio of the sample variance with the $m+1$ basis function D_{m+1} to the sample variance with m basis functions D_m is less than the quantile of the Fisher F-distribution at a given significance level α:

$$D_{m+1} / D_m \le F_\alpha \cdot (n - m - 1, n - m) , \qquad (17)$$

then the new, (m +1)-th basis function makes sense to take into account, and if not, then not.

To simplify the calculations, we added the terms to the model not one by one, but in groups: linear, quadratic, etc. For a linear model ($m = 3$), the sample variance $D_3 = 1{,}9964e+5$, and for the quadratic ($m = 6$): $D_6 = 3{,}2063e+4$. Their ratio $D_3 / D_6 = 0{,}1606$, and the quantile F of the Fisher distribution for $f_1 = 24 - 6 = 18$; $f_2 = 24 - 3 = 21$ at the significance level $\alpha = 0{,}01$ is equal to $F_{0{,}01}(18,\ 21) = 0{,}3273$. Since $0{,}1606 < 0{,}3273$, the reduction in sample variance when taking into account quadratic terms is significant, so quadratic terms should be taken into account.

Then we add four more cubic terms: these are the basis functions $x^3,\ x^2 \cdot y,\ x \cdot y^2,\ y^3$.

The sample variance of the cubic model $D_{10} = 8{,}5917e+3$, and its ratio to the sample variance quadratic model $D_{10} / D_6 = 0{,}2680$. Corresponding *F*-distribution quantile $F_{0{,}01}(14,\ 18) = 0{,}2680$. Here, too, we have $0{,}2680 < 0{,}2812$, so the cubic terms should also be taken into account.

Now let's add four 4th degree terms: $x^4,\ x^3 \cdot y,\ x^2 \cdot y^2,\ x \cdot y^3$. The term with y^4 cannot be added, because we have only four different values of y_i; and this addition will be i linearly dependent з with the previous ones: $1,\ y,\ y^2, y^3$. For this model sample variance $D_{14} = 2{,}8268e+3$; its ratio to the sample variance of the cubic model $D_{14} / D_{10} = 0{,}3290$.

Quantile $F_{0{,}01}(10,\ 14) = 0{,}2174$. Here, $0{,}3290 > 0{,}2174$; that is, the decrease in the sample variance is insignificant. Therefore, the 4th degree terms can be ignored and the cubic model can be used.

At the significance level $\alpha = 0{,}01$, the parameters of the cubic model and their confidence levels were found intervals. They are shown in Table 2.

Table 2

Model parameters and confidence intervals for them.

Parameter	Lower boundary Parameter	Value	Upper limit
b_1	-17984,376	-5151,473	7681,429
b_2	-655,971	743,255	2142,481
b_3	108,196	437,013	765,830
b_4	-82,619	-30,861	20,896
b_5	-6,360	5,381	17,123
b_6	-26,757	-17,513	-8,271
b_7	-0,14	0,496	1,13
b_8	-0,3	-0,113	0,08
b_9	-0,07	0,021	0,11
b_{10}	0,12	0,211	0,31

In Fig. 3. is shown surface of dependence of resistance of motion on the span and load capacity. The red asterisks show the experimental data.

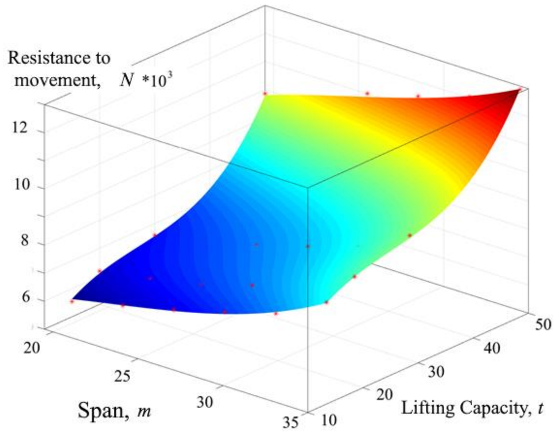

Fig.3. Graph of the approximation of the function of two variables of crane movement resistance.

Thus, the experimental data were approximated by a cubic model. At the initial stage, it is difficult to predict the shape of the curve that will best describe the relationship we are looking for. In this case, the calculation shows that the cubic function has sufficient accuracy, and the fourth degree function does not significantly increase the accuracy.

Thus, based on statistical data, a functional dependence was obtained that makes it possible to calculate the resistance to movement of an overhead crane of any lifting capacity and span within a given range without making design decisions about the wheel diameter and selecting numerous coefficients. The calculation requires only the values of the crane's lifting capacity and span, which are contained in the original data for any crane.

This determines the static component of the travel resistance. To determine the dynamic component, you need to know the weight of the crane. The most accurate determination of the crane's weight occurs only at the final design stage, when the design of all its main elements is known. In practice, in order to obtain data on the weight of the crane to be manufactured, data from completed projects is used at the initial stages of design. The difficulty lies in the fact that the parameters of the crane to be manufactured may not match the parameters of already-manufactured cranes. Therefore, based on completed projects, it is necessary to perform an average, which requires manual work and responsible decision-making. The solution to this problem and a way to improve accuracy is also the application of the statistical method. The weight of a crane depends mainly on the following factors: operating mode group, span and lifting capacity.

Let's find a numerical example of the weight of overhead cranes for the A6-A8 heavy-duty groups. Most overhead cranes, especially technological and special cranes, operate in these modes. Using the data from completed projects (we will consider them experimental data), for each of these three groups, we will write down the weight of cranes in the table in accordance with the span and lifting capacity (Table 3).

Table 3

Experimental data on the weight of overhead cranes depending on the span and lifting capacity for the A6-A8 operating mode groups.

Span, m / load, t	10	14	18	22	26	30
5	14,99	16,88	20,62	25,46	30,36	33,79
10	19,04	21,81	24,65	27,64	31,96	38,08
15	25,39	28,19	31,35	35,49	40,7	48,76
20	27,73	30,17	33,26	37,09	42,28	49,12
30	37,31	41,5	47,57	54,45	60,81	69,93
50	50,89	59,45	63,83	68,52	75,42	83,59

The study of the null hypothesis of a normal distribution of the experimental data was conducted for the weight data as well (Table 3). Here, we have 36 experimental points, the mathematical expectation of the residuals is m_R=2.9606×10-15, and their standard deviation is s_R=1.7327. Fig. 4 shows the empirical distribution function of these data, normalized by the formula $x_i=(r_i-m_R)/s_R$, and the standard normal distribution function. Here, the Kolmogorov statistic is 0.1101, which is less than the critical value of 0.2653. Therefore, at the significance level of α=0.01, there is no reason to reject the null hypothesis, and the distribution of the experimental results can be considered normal (fig. 4).

For these data, we will construct a mathematical model in the form of an optimal degree polynomial according to formula (6). Then the weight of the crane will be found by the expression:

$$\begin{aligned} z_{mk} &= b_1 + b_2 \cdot x + b_3 \cdot y + b_4 \cdot x^2 + \\ &+ b_5 \cdot x \cdot y + + b_6 \cdot y^2 + b_7 \cdot x^3 + \\ &+ b_8 \cdot x^2 \cdot y + b_9 \cdot x \cdot y^2 + b_{10} \cdot y^3 + \ldots \end{aligned} \tag{18}$$

For all groups of operating modes, we added terms to the model not one at a time, but in groups: linear, quadratic, etc. Table 4 shows the sample variances for different mathematical models and different groups of operating modes and their comparison. considered normal.

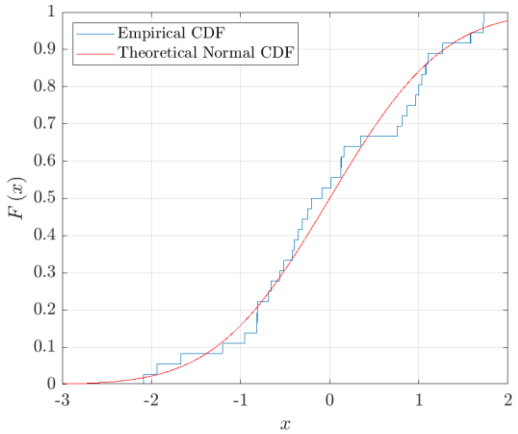

Fig. 4. Empirical and theoretical standard normal cumulative distribution function for crane weight.

Let us explain the filling of Table 3. To simplify the calculations, we added the terms to the model not one by one, but in groups: linear, quadratic, etc. For the linear model (*m*=3), the sample variance is D_3=9,3593, and for the quadratic model (*m*=6): D_6=5,2323. Their ratio D_6/D_3=0.5591, and the quantile of the Fisher's *F*-distribution for f_1=36-6=30; f_2=36-3=33 at the level of significance α=0,01 equals $F_{0,01}(30,33)$=0,4246. Despite the fact that 0,5591>0,4246, we still accept the quadratic model.

Table 4

Experimental data on the weight of overhead cranes depending on the span and lifting capacity for the A6-A8 operating mode groups.

Mode of operation	A6-A8
A model for finding functional dependence:	
linear: m=3; f=36–3=33	D_3=9.3593
quadratic: m=6; f=36–6=30	D_6=5.2323
comparison of the quadratic model with the linear one: $F_{0.01}(30,33)$=0.4246	D_6/D_3=0.5591; 0.5591>0.4246; but we still accept the quadratic model
cubic: m=10; f=36–10=26	D_{10}=4.0417
Comparison of the cubic model with the quadratic model: $F_{0.01}(26,30)$=0.3996	D_{10}/D_6=0.7725; 0.7725>0.3996; but still accept the cubic model
4th degree: m=15; f=36–15=21	D_{14}=3.4154
comparison of the 4th degree model with the cubic model: $F_{0.01}(21,26)$=0.3610	D_{14}/D_{10}=0.8450; 0.8450>0.3610; Reject the 4th degree model

Next, we add four more cubic terms: the basis functions x^3, $x^2\ y$, $x\ y^2$ and y^3. The sample variance of the cubic model is D_{10}=4,0417, and its ratio to the sample variance of the quadratic model is D_{10}/D_6=0.7725. The corresponding quantile of the Fisher F-distribution is $F_{0.01}(26,30)$=0,3996. In this case, 0,7725>0,3996, but we will still take into account the cubic terms.

Table 5

Parameters of the cubic model and confidence intervals for them.

b_1	0,660≤10,927≤21,193
b_2	-0,881≤0,732≤2,345
b_3	-0,898≤-0,494≤-0,091
b_4	-0,118≤-0,035≤0,048
b_5	0,019≤0,042≤0,065
b_6	0,036≤0,0488≤0,0619
b_7	-0,0002≤0,001≤0,002
b_8	-0,0007≤-0,0002≤0,0002
b_9	-0,0005≤-0,0003≤0
b_{10}	-0,0007≤-0,0006≤-0,0004

Now let's add five 4th degree terms: x^4; $x^3\ y$; $x^2\ y^2$, $x\ y^3$ and y^4. For this model, the sample variance is D_{14}=3,4154; its ratio to the sample variance of the cubic model is D_{14}/D_{10}=0,8450. The quantile $F_{0,01}(21,26)$=0,3610. Here, 0,8450>0,3610; that is, the gap is constantly increasing and the decrease in the sample variance is insignificant. Therefore, we can disregard the 4th degree terms and limit ourselves to the cubic model.

Table 5 shows the calculated parameters of the cubic model and their confidence intervals for cranes of all operating mode groups.

Figure 3 shows the weight versus span and lifting capacity surfaces for different crane operating mode groups A6-A8. The red asterisks indicate the experimental data, and the translucent surfaces show the upper and lower confidence limits. The figures show that the proposed cubic model approximates the experimental data quite accurately, and all experimental points fall within the confidence limits.

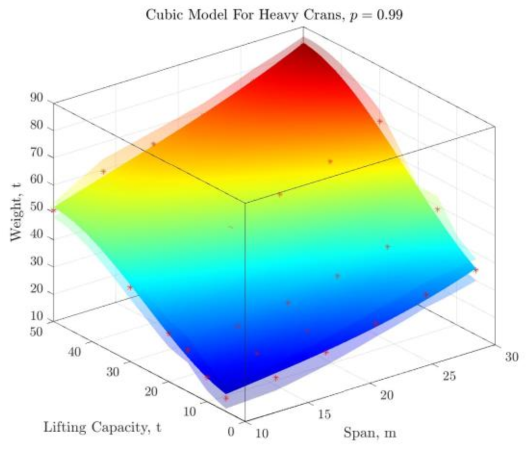

Fig. 5. Graph of the approximation of the function of two variables of crane weight.

Thus, by combining the values of static travel resistance (6) and crane weight (18) in formula (3) with the coefficients from tables 2 and 5, respectively, we can find the value of total travel resistance for formula 2, which will be as follows:

$$W_0^z = z_W + 1,2 \cdot \left(z_{mk} + m_Q \right) \cdot a \qquad (19)$$

To find the required electric motor power, substitute the value calculated by formula 19 into formula 2 and then into formula 1 (for a separate drive). In this case, to determine the power of the electric motor, we use only the general parameters of the crane - lifting capacity, span, travel speed and a few other constants, which is a very convenient way. Numerical checks using the formulas confirmed the convergence of the calculation results and the actual values of electric motor power in real projects.

DISCUSSION

The conducted literature review on the topic of the article confirms the high relevance of the considered topic of improving the methods of selection and calculation of electric drives for traveling mechanisms. However, the authors' attention is more focused on automotive vehicles. The closest research is the work [6]. The authors of this paper also aim to accelerate the selection of electric drive parameters, but the attention is focused on the elements of the electric motor of an automotive vehicle, which significantly distinguishes this study. At the same time, the idea of using statistical data based on existing designs has shown high efficiency and calculation accuracy.

In this study, it is proposed to select an existing motor from the catalog based on the calculated value of the power requirement, which is traditional for the design of hoisting machine mechanisms. As can be seen from the final formula 19, static traveling resistance and crane weight are determined by a statistical method. The actual distribution of values of these parameters depending on the lifting capacity and span of the crane shows that the most appropriate approximation model is the cubic model. The numerical data given in Tables 1 and 3 allow us to construct actual surfaces for approximating functions and to determine confidence intervals for their coefficients. The minimum value of the actual parameters necessary for the possibility of carrying out the study is given in the paper. This allowed us to justify the most appropriate form of approximating functions and to present the calculation procedure for practical use of the proposed method. As can be seen from the graphs of Fig. 2 and Fig. 3, the tabular data are located very close to the calculated surfaces, which can be explained by their small number. Obviously, in practical use, more statistical data can be taken into account, and the accuracy of the calculation will slightly decrease. The main significance of the proposed calculation method is that it significantly reduces the time for motor selection, and it is easier to integrate it into computer-aided design systems due to the absence of the need to select parameters in manual mode.

CONCLUSIONS

In the article, the functional dependencies for determination of the power of the electric motor of the traveling mechanism of the overhead traveling crane - static resistance to movement and weight of the crane—as well as the final dependence of the total resistance to movement on the load capacity and span are formed.

The equation for calculating the power of the electric motor of the traveling mechanism of the overhead traveling crane by the traditional method has been analysed. Constants known from the initial data and variables requiring additional calculation are determined. An example of collection and systematisation of statistical data - values of variables for overhead cranes of different spans and load capacity is given. On the basis of systematised data on actual parameters regression equations - coefficients and confidence intervals of functions of two variables from load capacity and span are defined. It is established that the most suitable model for searching for approximating functions is cubic. The obtained results and the possibility of practical application of the proposed method are analysed.

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