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Determining parameters to optimize the pulling force for the luffing jib tower cranes by Taguchi method

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The presented problem consists in optimizing the pulling force of the luffing jib tower cranes, in order to reduce power and save energy by determining reasonable geometrical parameters such as placement of pulley assemblies, position of jib pin, and jib length. To determine the optimal parameters, a mechanical model was developed to calculate the pulling force of the research object. Then, the Taguchi method and Minitab software were applied to evaluate the influence of the parameters. The objective function was the minimum pulling force of the luffing jib. The calculation results show that the position of the pulley assembly used to pull the jib is the most influential factor on the objective function accounting for 81.15%, the less significant factors are the jib length, the pin position of the jib, and the pulley assembly that changes the direction of the load lifting cable. The result of the test presented in the article allowed for determining the rational parameters, and the optimal position of the pulley assemblies on the top of the crane. In the case of the pulley assembly located at the top of the crane, one obtains the optimal height of the crane head $H \approx 0.4L_c$.

1. Introduction

Tower crane is widely used in practice to transport and unload materials, and its specificity consists in large lifting height and large working radius. There are different method of changing the working radius, it is usually done by changing the angle of the jib or by moving the trolley on the horizontal jib. Fig. 1 shows a diagram of the general structure of the luffing jib tower cranes: the crane with the lifting mechanism (8) with the function to raise or lower the load through the pulley assembly (4) and the cable hoist (5). The rotary mechanism on the

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turntable (11) is used to rotate the rotating part about the vertical axis. The pulling mechanism (7) is used to vary the working radius. The jib of the crane is pulled by the mechanism (7) through the pulley (3) and the cable hoist (6). The main structure of the crane consists of the jib (1), the tower body (12), the turntable (11) and the crane head (9). The luffing jib tower cranes gives a large lifting height, but when changing the jib angle it is necessary to pull both the jib and the load. This operation consumes energy and is difficult to perform in the case of a small working radius.

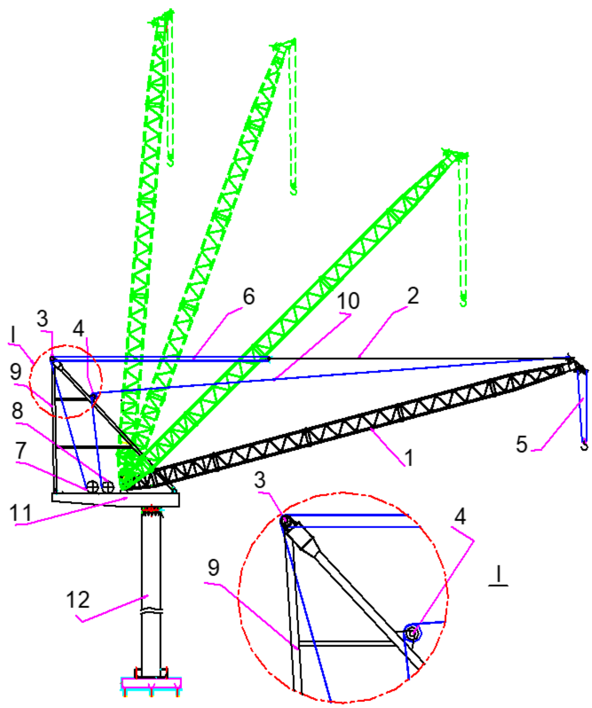

Fig. 1. General structure of luffing jib tower cranes
1 – jib, 2 – cable pulling, 3 – position of the pulley assembly used to pull the jib, 4 – position of the pulley assembly to change the direction of the load lifting cable, 5 – cable hoist for lifting loads, 6 – cable hoist for pulling the jib, 7 – mechanism for changing working radius, 8 – mechanism for lifting the load, 9 – crane head, 10 – lifting cable, 11 – turntable, 12 – tower body

The problem of design, control, use and operation of cranes has been presented in numerous related research publications. The general purpose of the research is to increase productivity, reduce energy and manufacturing costs, and to protect operator's health. The authors in [1] have optimized the installation location of tower cranes on the construction site to build high-rise buildings. The objective function was to minimize costs, and genetic algorithms were used to solve the problem. A research on calculating the influence of tower body deformation for design problems of tower body bracing system, along with calculating the machine foundation reaction and studying the resistance of the hydraulic system to ensure

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safety during jacking and erection have been presented in [2, 3]. The studies in [2, 3] concentrate on the tower body of the crane without studying the working mechanisms such as pulling the jib or lifting the load. In [4], the general structure and optimal design of the supporting system of the tower crane are investigated using MATLAB software and ANSYS software to solve the problem. In order to reduce hanger vibrations and dynamic loads, a physical model of the luffing jib tower cranes has been built to optimize the drive systems for the mechanisms [5]. The authors in [6] search for a globally optimal design solution to provide the best combination of different mechanisms so that dynamic overload values will be greatly reduced at the time of structural design. In [7], one optimizes the geometric parameters related to the size of the luffing mechanism with the objective function of minimizing horizontal deviation of the heavy. In this study, the authors use genetic algorithms to solve the problem.

The authors in [8–13] have focused on the research of optimization of steel structure of the jib, including the method of changing the angle of the jib and the horizontal jib. In [8], one presents optimal design of cross-sections of horizontal jib in tower cranes with stress and displacement constraints. The applied method takes into account the effect achieved for simple models, assuming that only tension and compression in the elements should be considered in the optimization problem. The problem considered in [9] is how to optimize the cross-sectional area of a triangular jib by using the Lagrange multipliers method. The study in [10] presents optimization of the cross-sectional size of the jib based on the Lagrange multiplier method. The authors compared the triangular and trapezoidal cross-sections and the rectangular shape of the tower crane with the aim to achieve the minimum weight and give recommendations on the selection of cross-sections according to different applications. In [11], the model problem and the optimization process of tower crane steel structure were presented based on ANSYS software and finite element method. The studies in [12] and [13] concern optimization of the beam structure to give a more reasonable structure and a more uniform optimal post-stress. Similarly as in [5] and [6], the authors in [14] carried out studies on rationalizing the crane motion trajectory to reduce the structural weight. However, in [14] the problem is investigated under stress and displacement constraints. Reliability-based design optimization is necessary to evaluate failure probabilities in stress and displacement. The authors in [15] analyzed the mechanical model, built a mathematical model, built an optimal algorithm using the penalty function method to optimize the position of the tow point on the port crane and applied tests for a 25 ton crane with a working radius changing from 9.5 m to 33 m. The reduction of the pulling force of the port crane in [15] makes it possible to reduce the crane's mechanical power.

Then, it can be seen that the above studies focus on the issues of rational arrangement of cranes on construction sites and the research on reducing dynamic loads acting on steel structures [1–7]. In [8–14] optimization of steel structures is required according to different methods, which include optimization of the section

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and the structure, whose objective function is to minimize the weight. Studies on the effects of loads have also been published in [2–7] and [16]. The optimization of the parameters in [7] concerns the geometric size related to the pulling mechanism in order to ensure the minimum horizontal movement. The object of the optimization problem in [15] is the port crane, and the penalty function method is applied to solve it.

Advanced optimization methods have many advantages, but the amount of calculations is large and requires using advanced computational hardware and software and applying the techniques mentioned in [17]. The Taguchi method was introduced by Genichi Taguchi, an engineer and statistician (1924–2012), back in nineteen eighties. The method is mainly applied in the parameter design stage based on assessing the influence of factors on the objective function, and at the same time determining the optimal parameters. The Taguchi method is commonly used in many fields such as experimental design, robust design, quality control, economic analysis [18–23]. Taguchi method has the advantages of simplicity, small number of trials and can be both quantitative and qualitative.

In engineering technology, due to its advantages, the Taguchi method has been applied to many technical problems. Research in [18] applied Taguchi method to optimally design a wheeled robot for transporting products to ensure higher adaptability and maximum stability during stair climbing. The authors in [19] identified the most influential factor in the winding process around the winch drum using the Taguchi method. The method was applied in numerical simulation design to assess contact stress in gerotor pumps [20]. In this study, one presented Taguchi techniques as an effective simulation-based strategy to narrow the combinations of geometrical parameters, reduce the solution space, and optimize the number of simulations. In [21] the Taguchi method and finite element method were both used to determine the optimal design parameters for titanium alloy prick hole extrusion. The study confirmed the suitability of the proposed design, allowed the punching die to achieve extrusion and was found perfect in testing using the finite element method. The study in [22] applied the Taguchi method to optimize process parameters for the production of low-strength materials controlled by using mortar size and light aggregates. Using the Taguchi method for the fatigue analysis of the seat right angle mentioned in [23], the authors of the study presented a method to determine the critical design parameters to prolong the product life of components subjected to fatigue loads.

Empirical parameters selection calculations are apparently not very efficient. Therefore, it is necessary to find methods of calculation and design which allow reducing the pulling force for the luffing jib and consequently reducing the power. In this paper, the research object is the luffing jib tower crane, whose structure allows working at a small working radius, possibly even at a zero range. The problem is to optimize the pulling force of the luffing jib, which should contribute to reducing the power and saving energy. It can be achieved by determining the relevant geometrical parameters such as the position of the pulley assemblies, the position of the jib pin and jib length. To determine the optimal parameters, firstly a mechanical model will be set up to calculate the pulling force of the luffing jib tower cranes. Then, the problem will be developed and the Taguchi method and Minitab software will be applied to determine the optimal parameters. Based on the optimal parameters, the problem will be set up and the pulley assemblies will be located on the crane head.

2. Mechanical model determining the pulling force of the jib

Considering the calculation models of Fig. 1 and Fig. 2, it is assumed that influencing factors such as the inclination of the foundation, the friction force at the jib pin are negligible compared to other components. The crane does not move and is fixed to the foundation. According to the working principle of the crane, the jib 1 is lifted and lowered by the mechanism 7. Mechanism 7 through the cable hoist 6 will pull the cable 2, and the pulling force is T_i. The moment due to the pulling force of the jib with the jib pin is balanced with the moment due to the different force components with the jib pin. The pulling force T_i is calculated from this equilibrium condition. The parameters and data in the calculations (shown in Fig. 2) are explained below and in Table 1.

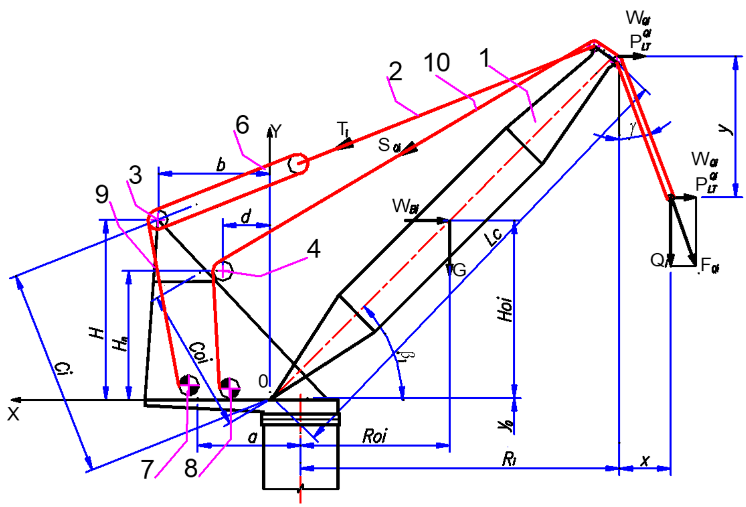

Fig. 2. Calculation model of the pulling force of luffing jib

Parameters shown in Fig. 2:
T_i – pulling force for the jib (N), S_{Q_i} – cable tension lifting the load (N), G – weight of the jib (N), W_{Bi} – wind force on the jib (N), Q_i – weight of the lifting load (N), P_{lt}^{Qi} – centrifugal force of the load (N), W_{Qi} – wind force on the load, F_{Qi} – combined force acting on the rope hoist lifting the load (N), L_c – jib length (m), R_{0i} – distance from the center of rotation of the machine to the center of the jib (m), R_i – working radius (m), β_i – angle of inclination corresponding to the working

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radius (°), a – distance from the jib pin to the center of rotation (m), C_i – lever arm of the pulling force with the jib pin (m), C_{0i} – lever arm of the cable tension lifting the load with the jib pin, (m), x – increased working radius due to the cable bevel (m), γ – bevel of the cable (°), b – horizontal distance from the jib pin to the fixed pulley assembly of the pulling jib (m), d – horizontal distance from the jib pin to the pulley assembly of the lifting cable (m), H – vertical distance from the jib pin to the fixed pulley assembly for the pulling jib (m), H_n – vertical distance from the

Table 1. Explanation of symbols

Symbol	Interpretation of symbols	Unit
T_{1i}	The pulling force of the jib due to the weight of the lifting load and the weight of the jib	N
M_{1i}	The moment due to the weight of the lifting load and the weight of the jib with the jib pin	Nm
T_{2i}	The pulling force due to the wind force on the jib and the load	N
M_{2i}	The moment due to the wind load acting on the jib and the lifting load with the jib pin	Nm
g	Gravitational acceleration, $g = 9.81$ m/s^2	m/s^2
ω	Angular speed of crane during rotation	rad/s
n	Rotational speed of the crane	rpm
T_{3i}	The pulling force of the jib due to the centrifugal force of the load and the centrifugal force of the jib	N
M_{3i}	The moment due to the centrifugal force of the load and the centrifugal force of the jib with the jib pin	Nm
M_{lt}^{G}	The moment due to centrifugal force of the jib mass with the jib pin	Nm
T_{4i}	The pulling force due to dynamic load when lifting and lowering the load	N
M_{4i}	The moment due to the dynamic load when lifting and lowering the load with the jib pin	Nm
ψ	The dynamic load factor	
ξ	The empirical coefficient, according to [16, 24] for the crane, $\xi = 0.3$	
V_L	The lifting speed	m/s
T_{5i}	The pulling force due to the cable tension lifting the load	N
M_{5i}	The moment due to the cable tension lifting the load with the jib pin	Nm
a_Q	The multiplier of the wire rope hoist	
η	The cable hoist efficiency	J
A	The work of the external force	J
$E_1,\ E_2$	The kinetic energy of the system at the beginning and the end of the unstable motion	J
$J_{Qi},\ J_B$	The moment of inertia of the load and the jib with the center of rotation	kgm^2
ω_c	The angular speed of lifting or lowering the jib of the crane	rad/s
t_a	Acceleration time of the crane's jib pulling mechanism	s

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jib pin to the pulley assembly for the lifting cable (m), y – distance from the tip of the jib to the lowest position of the hanger (m), y_0 – distance from the ground to the pin of the jib (m).

The pulling force of the jib T_{1i} (N) due to the weight of the lifting load and the weight of the jib:

$$T_{1i} = \frac{M_{1i}}{C_i}, \tag{1}$$

$$M_{1i} = Q_i\,(R_i + a) + G\,(a + R_{0i}), \tag{2}$$

$$R_i = L_c \cos\beta_i - a, \qquad R_{0i} = \frac{L_c \cos\beta_i}{2} - a, \tag{3}$$

where M_{1i} is shown in Table 1, the parameters (C_i, Q_i, R_i, a, G, R_{0i}, L_c and β_i) are shown in Fig. 2.

The pulling force T_{2i} (N) due to the wind force on the jib and the load:

$$T_{2i} = \frac{M_{2i}}{C_i}, \tag{4}$$

$$M_{2i} = \mathrm{W}_{Bi}\,(a + R_{0i})\tan\beta_i + \mathrm{W}_{Qi}\,(R_i + a)\tan\beta_i, \tag{5}$$

where M_{2i} is shown in Table 1, the parameters W_{Bi} and W_{Qi} are shown in Fig. 2.

The centrifugal force of the load P_{lt}^{Qi} (N), determined when taking into account the obliqueness of the cable during working, can be found from the model (Fig. 2):

$$P_{lt}^{Qi} = \frac{Q_i}{g}\omega^2(R_i + x), \qquad \omega = \frac{\pi n}{30}. \tag{6}$$

Substituting the data into formula (6) and taking $g = 9.81$ m/s^2, after transformation we have:

$$P_{lt}^{Qi} = \frac{Q_i n^2}{900}\,(R_i + y\tan\gamma), \tag{7}$$

where ω and n are shown in Table 1, the parameters P_{lt}^{Qi}, x, y and γ are shown in Fig. 2.

On the other hand, according to the diagram Fig. 2:

$$P_{lt}^{Qi} = Q_i \tan\gamma. \tag{8}$$

From (7) and (8), it is possible to determine:

$$P_{lt}^{Qi} = \frac{Q_i n^2 R_i}{900 - n^2 y}, \qquad y = L_c \sin\beta_i + y_0, \tag{9}$$

where y_0 is shown in Fig. 2.

Assuming that the jib of the crane has a uniform weight distribution, the centrifugal force of the opposite side of the jib through the center of rotation is small, so we can ignore it.

The differential of the jib mass is dm, the differential of the jib length is dl and the differential of the centrifugal force of the jib weight is dP_{lt}^{G}, and the differential of the moment due to centrifugal force of jib weight with jib pin is dM_{lt}^{G}.

$$dm = \frac{G}{gL_c} dl, \tag{10}$$

$$dP_{lt}^{G} = \frac{G}{gL_c} dl\omega^2 \left(l \cos\beta_i - a\right), \tag{11}$$

$$dM_{lt}^{G} = \frac{G}{gL_c} \omega^2 \left(l \cos\beta_i - a\right) l \sin\beta_i dl. \tag{12}$$

The moment due to centrifugal force of the jib mass with the jib pin:

$$M_{lt}^{G} = \int\limits_{a\cos\beta_i}^{Lc} \mathrm{d}M_{lt}^{G} = \int\limits_{a\cos\beta_i}^{Lc} \frac{G}{gL_c} \omega^2 \left(l \cos\beta_i - a\right) l \sin\beta_i \, \mathrm{d}l. \tag{13}$$

Finally, from the above formulas and calculation models, we can obtain the pulling force of the jib T_{3i} (N) due to the centrifugal force of the load and the centrifugal force of jib mass:

$$T_{3i} = \frac{M_{3i}}{C_i}, \tag{14}$$

$$M_{3i} = P_{lt}^{Qi} L_c \sin\beta_i + M_{lt}^{G}, \tag{15}$$

where M_{3i} and M_{lt}^{G} are shown in Table 1, and the parameter P_{lt}^{Qi} is shown in Fig. 2.

The pulling force T_{4i} (N) due to dynamic load when lifting and lowering the load:

$$T_{4i} = \frac{M_{4i}}{C_i}, \tag{16}$$

$$M_{4i} = \psi Q_i L_c \cos\beta_i, \qquad \psi = 1 + \xi V_L, \tag{17}$$

where M_{4i}, ψ, ξ and V_L are shown in Table 1.

The pulling force T_{5i} (N) due to the cable tension lifting the load:

$$T_{5i} = \frac{M_{5i}}{C_i}, \tag{18}$$

$$M_{5i} = \frac{\psi Q_i C_{0i}}{a_Q \eta}, \qquad S_{Qi} = \frac{\psi Q_i}{a_Q \eta}, \tag{19}$$

where ψ, a_Q and η are shown in Table 1, the parameters C_{0i} and S_{Qi} are shown in Fig. 2.

Let us assume that the process of starting or braking is a uniformly variable process with a constant acceleration, and the work done by the external force is equal to the change in kinetic energy of the system. Then, for the process of opening the machine, we have:

$$A = E_2 - E_1\,, \tag{20}$$

$$E_1 = 0, \qquad E_2 = \frac{J_{Qi}\omega_c^2}{2} + \frac{J_B\omega_c^2}{2}\,, \tag{21}$$

$$J_{Qi} = \frac{Q_i}{g}L_c^2, \qquad J_B = \frac{G}{3g}L_c^2\,, \tag{22}$$

where A, E_1, E_2, J_{Qi}, J_B and ω_c are shown in Table 1.

Substituting the above values, we have:

$$A = E_2 = \frac{L_c^2\omega_c^2}{2g}\left(Q_i + \frac{G}{3}\right). \tag{23}$$

The external force is associated with the moment M_{6i} (Nm) due to the inertia force caused by lifting or lowering the jib when the motion is unstable. The jib rotates around the jib pin with a rotation angle of β_i (°). Then, the work done by the external force is:

$$A = M_{6i}\beta_i\,, \qquad \beta_i = \frac{\omega_c t_a}{2}\,, \tag{24}$$

where M_{6i}, t_a are shown in Table 1.

From (23) and (24), the value of the moment with the jib pin due to the inertia force when lifting or lowering the jib is determined as follows:

$$M_{6i}\frac{\omega_c t_a}{2} = \frac{L_c^2\omega_c^2}{2g}\left(Q_i + \frac{G}{3}\right), \tag{25}$$

$$M_{6i} = \left(Q_i + \frac{G}{3}\right)\frac{L_c^2\omega_c}{g t_a}\,. \tag{26}$$

The pulling force T_{6i} (N) due to the inertia force when opening the machine or braking the mechanism for changing working radius:

$$T_{6i} = \frac{M_{6i}}{C_i}\,. \tag{27}$$

The resulting pulling force for the Luffing Jib Tower Cranes:

$$T_i = T_{1i} + T_{2i} + T_{3i} + T_{4i} + T_{5i} + T_{6i}\,. \tag{28}$$

3. Determination of optimal design parameters

3.1. Description of the problem

The pulling force depends on many factors, however, when considering the action of the loads that are known according to the studies in [2–7, 16, 24], we can chose the independent factor of the problem as the load factor through the load moment value, the lifting speed, the mechanism acceleration time, the cable hoist option, etc.

$$\{X_{0i}\} = \left\{M_T, n, \omega_c, V_L, G, \mathrm{W}_{Qi}, \mathrm{W}_{Bi}, a_Q, \eta, t_a, g, y_0\right\}, \tag{29}$$

where M_T is the designed crane loading moment (Nm).

If the load moment $M_T = Q_i R_i$ is intended to be a constant value [2, 24]:

$$L_c = \frac{M_T + aQ_i}{Q_i \cos \beta_i}. \tag{30}$$

For the research object, the design variables are the four geometric factors affecting the pulling force, including:

$$\{X_i\} = \{L_c, a, C_i, C_{0i}\}. \tag{31}$$

The power of the pulling mechanism expended to change the working radius P (kW) [16, 24]:

$$P = \frac{T_i v_p}{1000\eta_J}, \tag{32}$$

where v_p is the speed of the hoist rope pulling the jib (m/s), η_J is the efficiency of the pulling mechanism.

In the case when the value of velocity v_p is determined so as to reduce the power of the motor and save energy, the pulling force T_i must be reduced.

The objective function is the minimum pulling force T_i:

$$\min f(L_c, a, C_i, C_{0i}) = \min T_i. \tag{33}$$

The problem is that optimizing the pulling force should contribute to reducing power and saving energy by determining reasonable geometrical parameters. Among them, C_i and C_{0i} are related such as the position of the pulley assemblies and the position of the jib pin.

$$C_i = C_i(H, b), \quad C_{0i} = C_{0i}(H_n, d), \tag{34}$$

where (H, b) is assumed to locate the position of the pulley assembly used to pull the jib (Fig. 2), and (H_n, d) is assumed to locate the position of the pulley assembly to change the direction of the load lifting cable (Fig. 2).

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3.2. Method of locating pulley assemblies on top of crane

In this section, we set up the equation to locate the pulley assemblies on the top of the crane based on the design variables (L_c, C_i, C_{0i}) that satisfy the objective function (33). The position of the pulley assemblies (3) and (4) is different (see Fig. 1 and Fig. 2). Fig. 3 and Fig. 4 are geometric diagrams that allow one to find the coordinates of the position of pulley assemblies.

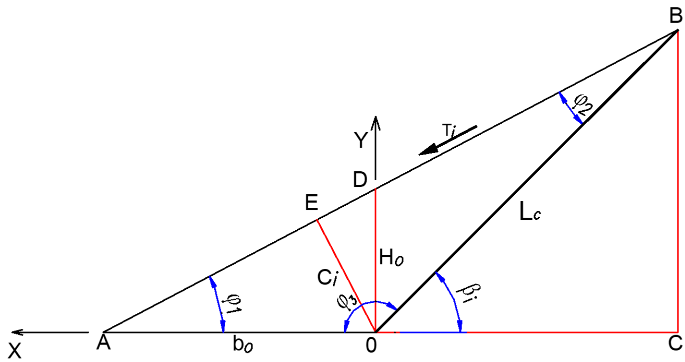

Fig. 3. Model to determine the position of the pulley assembly used to pull the jib

Let us consider the diagram shown in Fig. 3 to locate the pulley assembly to pull the jib with $0B = L_c$, $BC = L_c \sin\beta_i$, $0E = C_i$, $0D = H_0$, $0A = b_0$. The cable and cable hoist pulling the jib of the crane will coincide with the segment AB. The lever arm of T_i with the jib pin is C_i ($OE \perp AB$), the position of the fixed pulley assembly used to pull the jib (No. 3 of Fig. 2) should be on the AD segment.

When angle β_i is large enough:

$$\varphi_2 = \arcsin \frac{C_i}{L_C}, \qquad \varphi_1 = \beta_i - \varphi_2, \tag{35}$$

$$b_0 = \frac{C_i}{\sin \varphi_1}, \qquad H_0 = b_0 \tan \varphi_1 . \tag{36}$$

Taking O as the origin, the equation describing the position of the pulley assembly used to pull the jib takes the form:

$$Y = -\frac{H_0}{b_0} X + H_0, \quad 0 \leqslant Y \leqslant H_0, \quad 0 \leqslant X \leqslant b_0 . \tag{37}$$

The position of the pulley assembly used to pull the jib should be determined through the value (X, Y), satisfying the condition of equation (38):

$$C_i(X, Y) = C_i(b, H). \tag{38}$$

Let us apply the same method of calculation to determine the position of pulley assembly to change the direction of the load lifting cable based on the diagram of

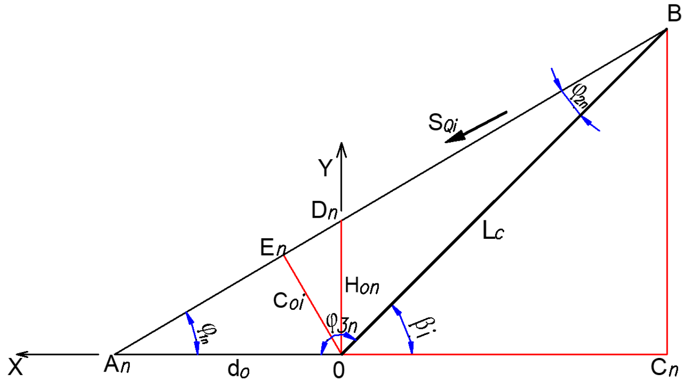

Fig. 4. Model to determine the position of the pulley assembly to change the direction of the load lifting cable

Fig. 4, with $0E_n = C_{0i}$, $0D_n = H_{0n}$, and $0A_n = d_0$. The direction of the cable lifting the load will coincide with the segment A_nB. The lever arm of S_{Qi} with the jib pin is C_{0i} ($OE_n \perp A_nB$). The position of the pulley assembly to change the direction of the lifting cable (No. 4 of Fig. 2) must be on the A_nD_n segment, and this position is determined by the equation (41).

$$\varphi_{2n} = \arcsin \frac{C_{0i}}{L_C}, \qquad \varphi_{1n} = \beta_i - \varphi_{2n}, \tag{39}$$

$$d_0 = \frac{C_{0i}}{\sin \varphi_{1n}}, \qquad H_{0n} = d_0 \tan \varphi_{1n}, \tag{40}$$

$$Y = -\frac{H_{0n}}{d_0} X + H_{0n}, \qquad 0 \leqslant Y \leqslant H_{0n}, \quad 0 \leqslant X \leqslant d_0. \tag{41}$$

The position of the pulley assembly to change the direction of the load lifting cable satisfies the conditions of equation (42):

$$C_{0i}(X, Y) = C_{0i}(d, H_n). \tag{42}$$

4. Optimizing the pulling force of the jib using the Taguchi method

4.1. Taguchi method

The idea of the Taguchi method is to determine the factors in order to achieve the highest efficiency by detecting and eliminating the effect of disturbance as effectively as possible. The design variable that affects the results in two directions, the effect that moves the results closer to the goal, is a useful effect, called the "Signal", the effect that makes the result move away from the goal is called the "Noise". The S/N ratio (Signal-to-Noise ratio) represents the performance indicator

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used to evaluate and select parameters. The parameter is good when S/N is high. The optimal parameters are obtained for the largest S/N.

Minimization problem (Smaller the better):

$$S/N = -10\lg\left(\frac{1}{n}\sum_{u=1}^{n}T_{iu}^{2}\right), \tag{43}$$

where u is the experimental sequence number, n is the number of experiments; T_i is the response value, $\overline{T}_i$ is the mean response value.

$$\overline{T}_i = \frac{1}{n}\sum_{u=1}^{n}T_{iu}\,. \tag{44}$$

Regardless of the type of problem, the goal to be optimized is always to maximize the S/N ratio. The steps of using the Taguchi method include determining the independent factors and the influencing parameters, determining the objective function, choosing the original matrix to conduct the Taguchi method, conducting the method and analyzing the results. The parameters that affect the process are then optimized. These are the controlled parameters that vary according to the Taguchi method level. More levels allow for greater parameter variations. The objective function can take any variable. Depending on the number of parameters affecting the process, an appropriate orthogonal matrix is selected according to the Taguchi method.

Each jib angle of the crane is considered as a test with defined selection parameters. The calculation of the average pulling force is also consistent with the method of calculating the capacity of the mechanism [16, 24]. When using the Taguchi method, it is possible to identify the factors that determine the pulling force of the jib. Quantitatively, it is possible to specifically determine the optimal value as the pulling force corresponding to the reasonable design variable. The problem can be calculated directly or using a supporting software. An optimization problem in engineering sometimes faces some limitations in the result of which the solution found is not the best one, consequently, some problems must be simplified. Therefore, it is found that the method can be applied to the problem of optimizing the pulling force of the crane.

4.2. Example of determining optimal parameters according to Taguchi method

The independent factors in the problem of optimizing the pulling force for the luffing jib tower cranes, including the load moment, the load and the parameters are given as in Table 2. The original design parameters are $L_c = 52$ m, $a = 5.6$ m, $C_i = 14.6$ m, $C_{0i} = 14.7$ m. These factors are given in the first design step and will be determined precisely after determining the influencing parameters in the next step.

$$M_T = \begin{cases} Q_i R_i = 228360 \text{ Nm} & \text{if } 15^\circ \leqslant \beta_i < 70^\circ, \\ Q_i R_i;\ Q_i = 180000 \text{ N} & \text{if } 70^\circ \leqslant \beta_i \leqslant 80^\circ. \end{cases} \tag{45}$$

Table 2. Given factors

Given factors	Symbol		Value	Unit
Rotational speed of the crane	n		0.7	rpm
Lifting speed	V_L		0.5	m/s
Angular speed of lifting or lowering the jib of the crane	ω_c		0.006	rad/s
Acceleration time of the crane's jib pulling mechanism	t_a		6	s
Gravitational acceleration	g		9.81	m/s^2
Weight of the jib	G		182000	N
Wind load on the load	W$_{Qi}$	$\beta_i = 15^\circ$	3125	N
		$\beta_i = 45^\circ$	4375	
		$\beta_i = 70^\circ$	7500	
		$\beta_i = 80^\circ$	7500	
Wind force on the jib	W$_{Bi}$	$\beta_i = 15^\circ$	1686	N
		$\beta_i = 45^\circ$	4598	
		$\beta_i = 70^\circ$	6108	
		$\beta_i = 80^\circ$	6500	
Cable hoist multiplier of hoisting winch	a_Q		4	
Cable hoist efficiency of hoisting winch	η		0.9	
Distance from the ground to the pin of the jib	y_0		30	m

The influencing factors and values include the jib length, the distance from the jib pin to the center of rotation, the lever arm of the pulling force with the jib pin, the lever arm of the cable tension lifting the load with the jib pin, given in Table 3. There are four factors to consider, and the value level is 3. The variable

Table 3. Factors and value levels

Influential factors	Symbol		Value level			Range of change
	Parameter	Encode	1	2	3	
Jib length (m)	L_c	x_1	52	58	64	12
Distance from jib pin to center of rotation (m)	a	x_2	0	5.6	11.2	11.2
Lever arm of the pulling force with the jib pin (m)	C_i	x_3	6	14.6	23.2	17.2
Lever arm of the cable tension lifting the load with the jib pin (m)	C_{0i}	x_4	4	11.7	19.4	15.4

interval of factors is selected on the basis of the required segment modulus and design engineering options.

When the number of factors is 4 and the value level is 3, for Taguchi planning we can choose the orthogonal planning matrix L9 shown in Table 4. The pulling force T_i is determined according to the mechanical model Fig. 2. The S/N ratio represents the efficiency criterion that is the minimum pulling force T_i. This criterion is used to evaluate and select the parameters. The S/N ratio is calculated according to the minimization problem (Smaller the better) by formula (43). The response values calculated at 4 trials corresponding to different arm angles β_i are (Table 4):

Table 4. Planning matrix table and test results

N	x_1 (m)	x_2 (m)	x_3 (m)	x_4 (m)	The response value is the pulling force T_i (N)				$\overline{T}_i$ (N)
					$\beta_i = 15^\circ$	$\beta_i = 45^\circ$	$\beta_i = 70^\circ$	$\beta_i = 80^\circ$	
1	52	0	6	4	1609757	1448433	156453	863174	1371475
2	52	5.6	14.6	11.7	712283	667988	670148	381941	608090
3	52	11.2	23.2	19.4	489464	486478	438912	257552	418101
4	58	0	14.6	19.4	712277	643173	777114	455079	646910
5	58	5.6	23.2	4	462955	426223	448668	246024	395967
6	58	11.2	6	11.7	1924297	1857447	1800379	1016865	1649747
7	64	0	23.2	11.7	466894	416791	516370	292364	423104
8	64	5.6	6	19.4	1904097	1752303	2060963	1194873	1728059
9	64	11.2	14.6	4	809462	763507	782543	426640	695538

Using Minitab software, in DOE we create and analyze the Taguchi design according to the S/N ratio. Based on the S/N ratio in Table 5 and Fig. 6, for each factor we can choose the value with the largest S/N. This level of value gives a reasonable result corresponding to the design variable.

Table 5. Results of assessing the influence of factors

Factor level	Influence factor				Result
	x_1	x_2	x_3	x_4	
1	–116.9	–117.2	–123.9	–117.2	Rational parameters: $x_1 = 52$ m, $x_2 = 0$ m $x_3 = 23.2$ m, $x_4 = 4$ m From equation (28) and (44), the average value of the optimal objective function is: $\overline{T}_i = 354691$ N
2	–117.5	–117.5	–116.2	–117.5	
3	–118	–117.9	–112.3	–117.8	
Mean	–117.47	–117.53	–117.47	–117.5	
Max	–116.9	–117.2	–112.3	–117.2	
Max – Mean	0.57	0.33	5.17	0.3	
% influence factor	8.90	5.24	81.15	4.71	
Delta	1.1	0.7	11.6	0.6	
Rank	2	4	1	3	

The average value of $\overline{SN}$ at the factors affecting the goal is:

$$\overline{SN} = \frac{1}{m}\sum_{j=1}^{m}\left(\frac{S}{N}\right)_m, \tag{46}$$

where, m is the number of experiments for a set of parameters.

The percentage influence of each factor is:

$$\%\, X_i = 100\%\frac{\max\frac{S}{N} - \overline{SN}}{\sum\limits_{i=1}^{n}\left(\max\frac{S}{N} - \overline{SN}\right)}, \tag{47}$$

where, $\max\frac{S}{N}$ is the maximum ratio of $\frac{S}{N}$ for each factor in the value.

Regarding the ranking of influencing parameters according to the S/N ratio, that is (x_3, x_1, x_2, x_4), the most influential factor is $x_3 = C_i$ which accounts for 82.15%, the least influential factor is $x_4 = C_{0i}$ accounting for 5.24%. If ranked according to response value, the factor $x_2 = a$ has the least influence. Fig. 5 illustrates an assessment of the influence of factors on the objective function. It can be seen that when x_1 increases, the value of the pulling force increases, for $x_2 < 5.6$ m the value of the objective function increases rapidly and when $x_2 \geqslant 5.6$ m the objective function increases insignificantly; with x_3 increasing, the target value decreases, and when x_4 increases, the objective function value increases.

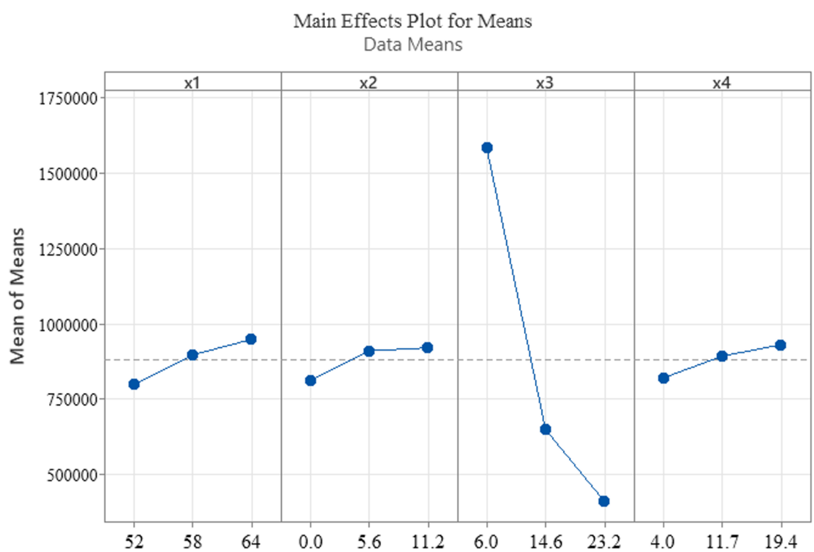

Fig. 5. Main effects plot for means

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The original design parameters are $x_1 = 52$ m, $x_2 = 5.6$ m, $x_3 = 14.6$ m, $x_4 = 14.7$ m. Using formula (28), we calculate each angle of inclination β_i, then we obtain the average pulling force $\overline{T}_i = 593303$ N. Fig. 6 shows the main factors affecting the S/N ratio. Because large S/N is good, we find the corresponding optimal values (Table 5): $x_1 = 52$ m, $x_2 = 0$ m, $x_3 = 23.2$ m, $x_4 = 4$ m, $\overline{T}_i = 354691$ N. The comparison results show that the average pulling force is reduced by 42.2% compared to the original design. The graph in Fig. 7 is a comparison of specific results when optimizing at different jib angles.

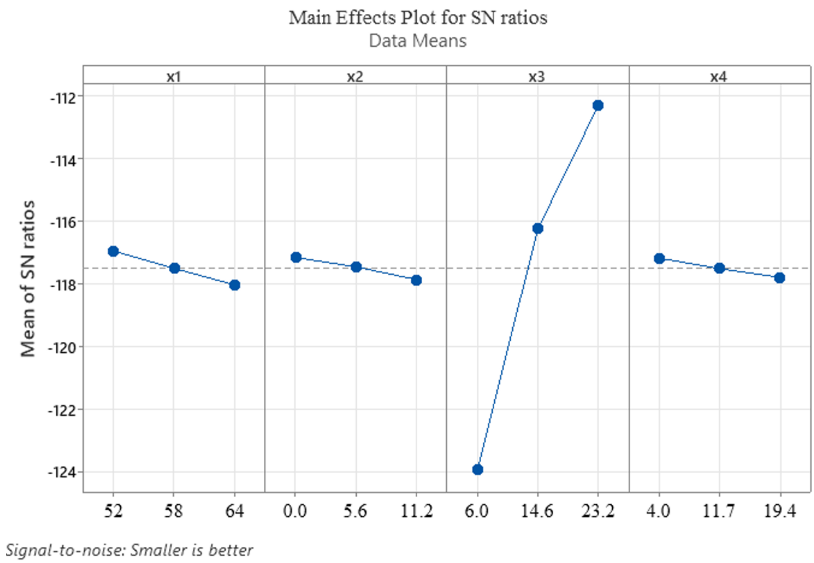

Fig. 6. Main effects plot for S/N ratios

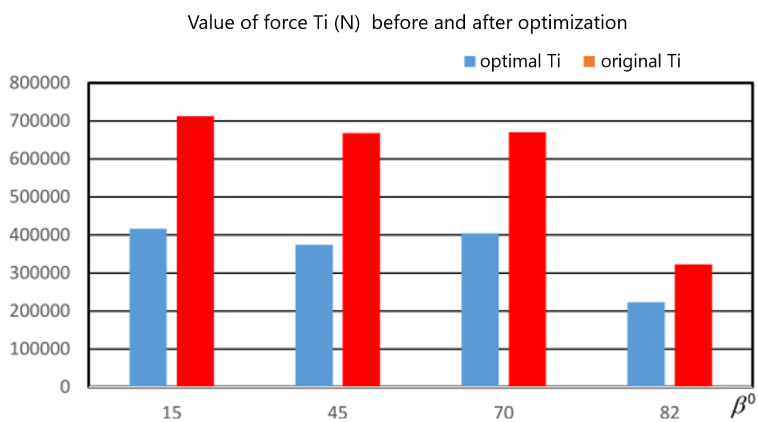

Fig. 7. Calculation results before and after optimization

4.3. Optimizing the position of the pulley assemblies on the top of the crane

Taking into account the results obtained according to the Taguchi method with the use of Minitab software, we determine the optimal parameters to locate the pulleys on the top of the crane so that the pulling force is minimal.

Let us substitute the optimal parameters $x_1 = L_c$, $x_2 = a$, $x_3 = C_i$, $x_4 = C_{0i}$ into equations from (35) to (37), tested at $\beta_1 = 30°$, $\beta_2 = 45°$, $\beta_3 = 70°$, $\beta_4 = 80°$. The optimal position of the fixed pulley assembly on the top of the crane for pulling the jib is determined by equation (48).

$$\begin{aligned} Y_1 &= -0.036X + 23, \\ Y_2 &= -0.32877X + 24, \\ Y_3 &= -0.94955X + 32, \\ Y_4 &= -1.3855X + 40. \end{aligned} \tag{48}$$

Using the resulting graph method for Fig. 8, we can determine the optimal value domain in the vicinity of coordinates $X = b = 13$ m; $Y = H = 21$ m. In case the pulley assembly is located at the top of the crane, we have the optimal height of the crane head $H \approx 0.4L_c$.

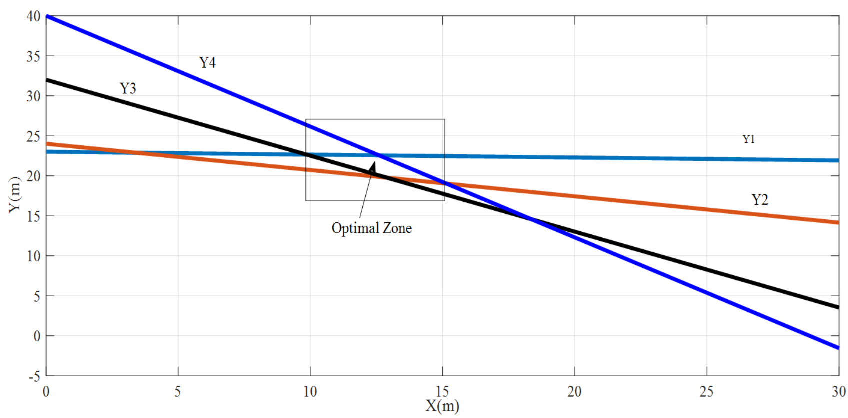

Fig. 8. Determining the optimal position of the pulley assembly used to pull the jib

With the same method, we substitute the optimal parameters into the equations from (39) to (41). The optimal position of the pulley assembly to change the direction of the lifting cable on the top of the crane is determined by equation (49).

$$\begin{aligned} Y_1 &= -0.4781X + 4.43, \\ Y_2 &= -0.8553X + 5.26, \\ Y_3 &= -2.19545X + 9.66, \\ Y_4 &= -3.8813X + 16.03. \end{aligned} \tag{49}$$

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According to the calculation results in section 4.2, this factor does not significantly affect the value of the pulling force. Using the resulting graph method for Fig. 9, we can determine the optimal value domain in the vicinity of the coordinates $X = d = 3.5$ m; $Y = H_n = 3$ m.

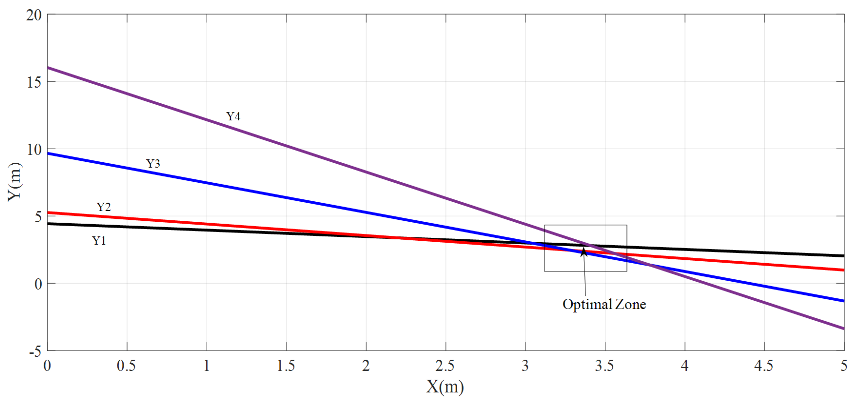

Fig. 9. Determining the optimal position of the pulley assembly to change the direction of the load lifting cable

The calculation results are inferred from the reasonable parameters, so the parameters given in Table 2 should be satisfied and the objective function response is the minimum pulling force.

5. Conclusion

In the article, we have studied the method of optimizing the pulling force to change the working radius, determining reasonable geometric parameters such as the position of the pulley assemblies, the position of the jib pin, and the length of the jib. The research object was the luffing jib tower cranes, whose structure allows working at a small working radius, possibly at even zero range. The content of the article provided a mechanical model to calculate the pulling force to change the working radius and resolve the problem. The Taguchi method and Minitab software are used for the optimal design, in which the number of factors is 4 and the value level is 3, and the orthogonal planning matrix is L9. The experiment was conducted in turn at 4 levels of tilt angle of the jib. Calculation results show that the position of the pulley assembly used to pull the jib is the factor that has the most significant influence on the objective function, it accounts for 81.15%, followed by the length of the jib, the position of the jib pin, and the position of the pulley assembly to change the direction of the load lifting cable. The results of comparison show that the average pulling force is reduced by 42.2% compared to the original design. The test results determine the position of the pulley assemblies on the top of the crane. In the case, the pulley assembly is located at the top of the crane, we have the optimal height of the crane head $H \approx 0.4L_c$. One of the issues that need to be improved in the next studies is to determine the general law of the applied forces corresponding to the reach to ensure the minimum change in the pulling force. This is also the basis for the control problem. In addition, the random factors acting on the crane affecting the objective function in Taguchi analysis is also a problem to be solved in the future.

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