Advances in Science and Technology Research Journal

Advances in Science and Technology Research Journal, 2025, 19(9), 129–142
https://doi.org/10.12913/22998624/207149
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Received: 2025.04.30
Accepted: 2025.07.15
Published: 2025.08.01

Designing a connecting pin for a gantry crane by integrating the weighted sum method with the Taguchi method

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ABSTRACT

The paper includes studying the method of calculating joint design and then building a multi-objective optimization problem model using the weighted sum and Taguchi methods. The research object is the joint between the gantry crane leg and the beam. Due to the characteristics of the connection bearing large and changing loads, research is necessary to improve safety, longevity, and reliability. The experimental problem model has four design variables and four value levels. The study uses the orthogonal matrix L16 to calculate the response values for each objective function at each stage. To apply the weighted sum method, the study selects the objective function weights for equivalent stress, contact stress, and fatigue strength, transforming the multi-objective problem into a single-objective problem. The test results identified a new set of parameters meeting the goals. The fatigue criterion was reduced by 8.9%, and the fatigue safety factor increased from 1.36 to 1.39. The equivalent stress was decreased by 9%, and the safety factor increased from 2.58 to 2.84. Contact stress was reduced by 37%, and the safety factor increased from 1.29 to 2. Combining these two methods not only solves problems in engineering but can also solve many problems in different fields.

Keywords: gantry crane, pin connection, multi-objective optimization, Taguchi method, weighted sum method.

INTRODUCTION

Pin connection is a widely used mechanical structure, especially in lifting machines such as cranes and gantry cranes. The example in Figure 1 is the trolley of a heavy-duty shipbuilding gantry crane we designed. Beam system 1 is designed with a gantry crane leg 2 by joint 3. Joint 3 has the structure shown in Figure 2, designed as a hinge joint, consisting of two halves connected through a pin shaft. Thanks to this joint, it is possible to compensate for deviations due to manufacturing, temperature deformation, or equipment installation. Due to the characteristics of the connection bearing large and changing loads, research is necessary to improve safety, longevity, and reliability.

When researching cranes and lifting equipment, there is a certain focus on optimizing engine power and the crane's beam. Typically, the authors in [1] optimized the position of the crane's pulling point to reduce capacity. The authors in [2] and [3] designed the crane beam based on aspects such as assessing the impact of local stability and optimizing the beam cross-section according to the goal of minimum weight. Studies on pin connections on crane structures have also been presented in [4] and [5]. The analysis in [4] determined the stress in pin connections on cranes at the tensile part. The research aims to collect design information through testing and analysis using the finite element method, thereby providing design recommendations. The authors in [5] described and proposed the design of the details and components of the lifting beam, especially the pin connections and connecting components. In addition, the design of mechanical details related to materials, connections, durability, and fatigue failure conditions has been presented in [6], which is the basis for performing specific application problems. The joint between the gantry crane leg and the beam is subjected to compression, only bears a moment in one direction, and has variable load effects. This is the research object in this paper.

The rules stated in standards [7–9] have been used more and more widely in many countries around the world. In particular, the regulations in [7] are used for the design, manufacture, and inspection of lifting equipment. The purpose of the rules specified in [7] is to prescribe the loads and load combinations to be taken into account when designing lifting appliances, and to establish conditions for strength and stability, and for failure. EN 1993-1-1 is the basic design rule for steel structures with material thickness greater than 3 mm, covering general issues, materials, structural strength, and general calculation methods [8]. The provisions in standard EN 1993-1-8 provide design methods for the design of predominantly statically loaded connections using steels with yield strengths from 235 N/mm^2 to 460 N/mm^2 [9]. The pin connections and the connection part, the calculation of the contact stress between the two surfaces, have also been instructed in [9]. The authors in [10] identified two main aspects that needed to be revised in [9], namely the ability to design the pin as a shear bolt and verify the strength of the pin. Based on a thorough review of the literature [9] and experimental tests, parameter surveys, the study proposed a design approach to address the above two issues. Many authors have studied the calculation model of pin stress, stress on the contact surface between the pin and the hole, as well as software suitable for design solutions [11, 12]. The study in [13] showed that the stress concentration factor depends on the tolerance between the pin and the assembly parts. By finite element analysis, the contact model in the paper can reduce the stress concentration factor by up to 18%. It can be said that the above studies are the basis for designing connections in mechanical engineering in general and on gantry cranes in particular.

Mathematical models of optimization problems often include design variables, constraints, objective functions, and problem-solving methods [14–20]. Depending on the problem, these factors are different. In studies [15–18], the optimal shaft design variables were geometric parameters, the ratio between the shaft and the hole, and the constraint conditions were stress or local stress, and displacement. The parameters of the gear train were the design variables in the study [19]. The objective function can be single-objective or multi-objective. In [14–21], the objective functions were single-objective, and in [23–25], they were multi-objective. The objective function for the problems in [15, 16, 18] was minimum weight, in [17] was minimum local stress, and in [20] was minimum force. Multi-objective functions have two or more objectives; for example, in [24], the multi-objective function was profit maximization and travel time minimization. In [25], the objective functions were for different load cases. The above studies, most of which use numerical methods to solve the problem.

An optimization problem in engineering often has to accept some limitations, such as the found solution is not necessarily the best, and some issues have to be simplified [14]. In studies [3, 19–22], the Taguchi method was used to design optimal applications for engineering problems such as optimizing overhead crane beams,

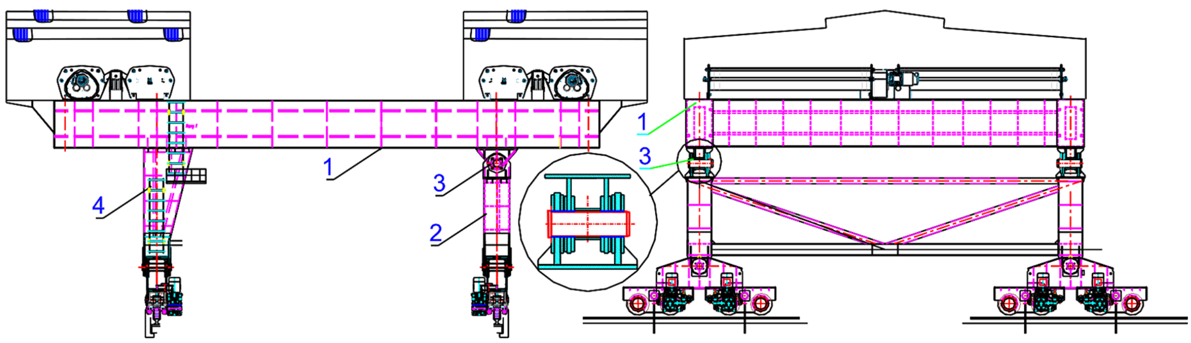

Fig. 1. The trolley of a heavy-duty shipbuilding gantry crane: 1 – beam system, 2 – gantry crane legs with joints, 3 – linked joints, 4 – rigidly linked gantry crane legs

optimizing production processes, and optimizing crane pulling force. The Taguchi method has the advantage of being simple and can be quantitative or qualitative. These studies also show that this method has the disadvantage of having discrete data, being unable to provide constraints, and being difficult to solve multi-objective problems [19–22]. A simple way to perform multi-objective optimization is to use the weighted sum method. The weighted sum method combines multiple objective functions by adding them together, along with a weight for each function. This is a multi-objective optimization method widely used in many different fields [23–26]. In engineering, the study in [25] used the weighted sum method for multi-objective optimization of a structure that bears multiple loads under different working conditions. The results show that the severity of the load case is less affected by different weighted summation functions. It can be seen that the above studies do not mention the combination of these two methods. The combination of the Taguchi and weighted summation methods will be a suitable idea for solving a multi-objective optimization problem in engineering. An application example for the problem is the pin connection on a gantry crane (Figure 1).

The research object in this paper is the connection joint between the gantry crane leg and the beam, as described in Figures 1 and 2. Based on the analysis of the above studies, the basis for the study of calculating the design of this joint is the studies in [6–20]. Due to the application characteristics, the design of this joint must consider many factors, so it is necessary to study and optimize the parameters to improve safety, service life, and reliability. In this study, the geometrical and material parameters of the joint will be optimized to increase the safety factor. Therefore, in this study, the design variables are the parameters of shaft and hole relationship, shaft diameter, and strength of manufacturing materials. The multi-objective functions are equivalent stress, minimum contact stress, and minimum fatigue failure criterion.

Therefore, this paper will present the design basis of the joint between the gantry crane leg and the beam, then build a multi-objective optimization problem model using the weighted sum method and the Taguchi method. The initial data set for the multi-objective optimization design problem is the joint on the 500-ton trolley of the gantry crane serving the 1,500-ton shipbuilding. The research results will give the optimal parameters such as pin diameter, material yield limit of connecting parts, size correlation ratios, and other geometric parameters. Minitab software is a tool to support numerical experiments. The effectiveness of combining these two methods will be demonstrated by solving a specific problem of pin connection on a gantry crane.

RESEARCH METHODS

Research based on joint design

The structure of the connection between the gantry crane leg and the gantry crane beam, as well as the force diagram, is given in Figure 2. The forces acting on the joint connection halves include the compressive force F and the moment M determined according to [7]; these values vary from the minimum value (F_{min}, M_{min}) at no load to the maximum value (F_{max}, M_{max}); the values (M_a, F_a) represent the average values of the moment and force acting on the joint; (M_m, F_m) represent the amplitude values of the moment and force acting on the joint.

Convert the maximum force acting on the contact surfaces of the upper half joint a and the lower half joint b according to the following formula:

$$F_{12}^{max} = \frac{F_a + F_m}{2} + \frac{M_a + M_m}{A} \quad (1)$$

where: A, B – the distances of the contact segments corresponding to the upper and lower half-joint surfaces shown in Figure 2; (M_a, F_a) – the average values of the moment and force acting on the joint; (M_m, F_m) – the amplitude values of the moment and force acting on the joint; F_{22}^{max} – the maximum force acting on the contact surface of the pin with the lower half of the joint; F_{12}^{max} – the maximum force acting on the contact surface of the pin with the upper half of the joint.

The pin of the joint has a solid structure, circular cross-section, and the dangerous cross-section at location 5 is subjected to bending moment and shear force. At this local point, there is a normal stress due to the bending moment and a shear stress due to the shear force. Combining

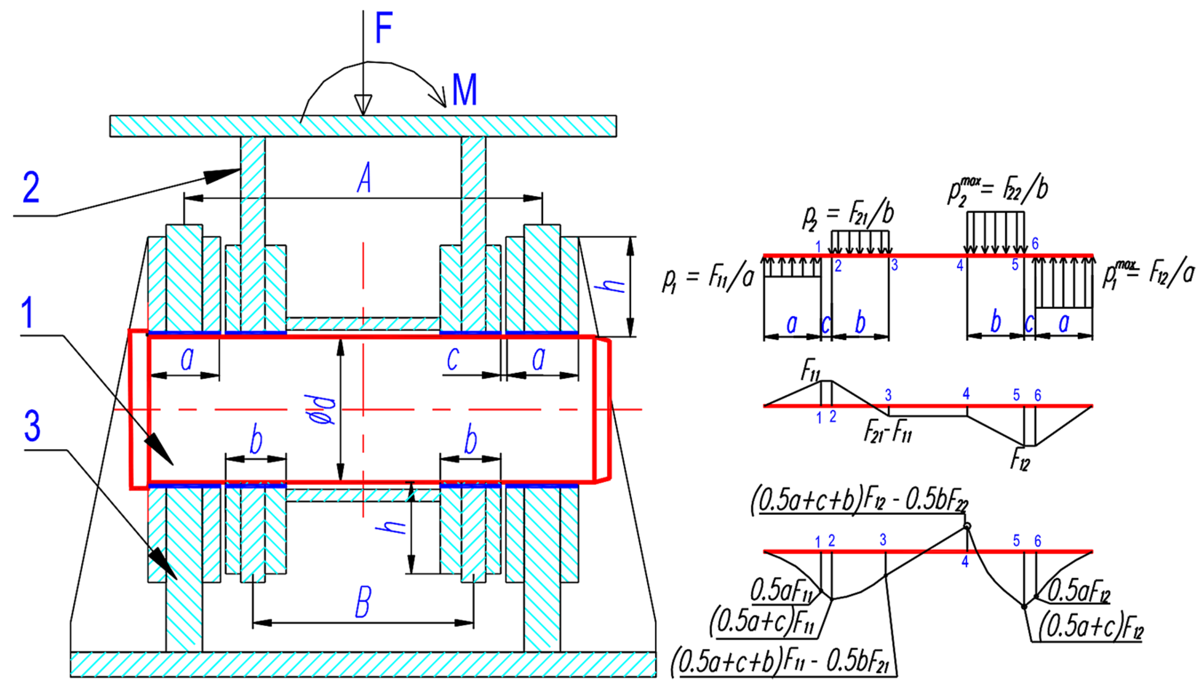

Fig. 2. Joint and force diagram acting on the pin connection: 1 – pin shaft; 2 – upper half joint; 3 – lower half joint

these stresses according to the theory of strength of materials, von Mises stress for the rotating circular solid pin corresponds [6]. Formulas (2), (3), (4) are the basis for calculating stresses for static strength and fatigue failure conditions.

Average stress $\sigma_a^{'}$:

$$\sigma_a^{'} = (\sigma_a^2 + 3\tau_a^2)^{\frac{1}{2}} = = \left[\left(\frac{32K_f M_{ia}}{\pi d^3}\right)^2 + 3\left(\frac{4K_{fs}Q_{ia}}{\pi d^2}\right)^2\right]^{1/2} \quad (2)$$

Stress amplitude $\sigma_m^{'}$:

$$\sigma_m^{'} = (\sigma_m^2 + 3\tau_m^2)^{\frac{1}{2}} = = \left[\left(\frac{32K_f M_{im}}{\pi d^3}\right)^2 + 3\left(\frac{4K_{fs}Q_{im}}{\pi d^2}\right)^2\right]^{1/2} \quad (3)$$

Maximum stress $\sigma_{\max}^{'}$:

$$\sigma_{max}^{'[(\sigma_m+\sigma_a)^2+3(\tau_m+\tau_a)^2]^{1/2}} = = \left[\left(\frac{32K_f M_{im}}{\pi d^3} + \frac{32K_f M_{ia}}{\pi d^3}\right)^2 + 3\left(\frac{4K_{fs}Q_{im}}{\pi d^2} + \frac{4K_{fs}Q_{ia}}{\pi d^2}\right)^2\right]^{1/2} \quad (4)$$

where: σ_a, σ_m – the average stress values and stress amplitude values due to bending moment; τ_a, τ_m x – the average stress values and stress amplitude values due to shear force; K_f, K_{fs} – the fatigue stress concentration coefficients corresponding to the stress; d – the pin diameter; M_{ia}, M_{im} – the average values of bending moment and bending moment amplitude; Q_{ia}, Q_{im} – the average values of shear force and shear force amplitude.

Fatigue failure criteria using Goodman curve [6] by equation (5):

$$\frac{1}{n} = \frac{\sigma_a^{'}}{S_e} + \frac{\sigma_m^{'}}{S_{ut}} \quad (5)$$

where: n – the design safety factor due to fatigue; S_e – the allowable strength at the local location, which depends on the shape and conditions of use; S_{ut} – the tensile strength of the pin material.Substitute formulas (2), (3) into (5) to get equation (6):

$$\frac{1}{n} = \frac{1}{S_e}\left[\left(\frac{32K_f M_{ia}}{\pi d^3}\right)^2 + 3\left(\frac{4K_{fs}Q_{ia}}{\pi d^2}\right)^2\right]^{\frac{1}{2}} + + \frac{1}{S_{ut}}\left[\left(\frac{32K_f M_{im}}{\pi d^3}\right)^2 + 3\left(\frac{4K_{fs}Q_{im}}{\pi d^2}\right)^2\right]^{1/2} \quad (6)$$

According to the diagram in Figure 2, the maximum shear force and bending moment are concentrated at location 5. The average values and amplitudes of shear force and moment at position 5 are as follows:

$$Q_{5a} = \frac{F_a}{2} + \frac{M_a}{A}; Q_{5m} = \frac{F_m}{2} + \frac{M_m}{A} \quad (7)$$

$$M_{5a} = (0.5a + c)\left(\frac{F_a}{2} + \frac{M_a}{A}\right); M_{5m} = = (0.5a + c)\left(\frac{F_m}{2} + \frac{M_m}{A}\right) \quad (8)$$

where: a, c – the contact length and clearance are shown in Figure 2.

Let the minimum ratio between the contact length a and the distance A of the lower half joint be $\Psi_1 = a / A$, formula (8) is transformed:

$$M_{5a} = (0.5\psi_1 A + c)\left(\frac{F_a}{2} + \frac{M_a}{A}\right); M_{5m} = = (0.5\psi_1 A + c)\left(\frac{F_m}{2} + \frac{M_m}{A}\right) \quad (9)$$

The maximum moment and shear force are at position 5, so the functions representing the design parameters according to the maximum stress σ'_{max} and fatigue failure criteria are rewritten as formula (10):

$$\begin{cases} f_1(X) = \frac{1}{S_e}\left[\left(\frac{32K_f M_{5a}}{\pi d^3}\right)^2 + 3\left(\frac{4K_{fs}Q_{5a}}{\pi d^2}\right)^2\right]^{\frac{1}{2}} + \\ + \frac{1}{S_{ut}}\left[\left(\frac{32K_f M_{5m}}{\pi d^3}\right)^2 + 3\left(\frac{4K_{fs}Q_{5m}}{\pi d^2}\right)^2\right]^{1/2} + \\ f_2(X) = \left[\begin{matrix}\left(\frac{32K_f M_{5m}}{\pi d^3} + \frac{32K_f M_{5a}}{\pi d^3}\right)^2 \\ +3\left(\frac{4K_{fs}Q_{5m}}{\pi d^2} + \frac{4K_{fs}Q_{5a}}{\pi d^2}\right)^2\end{matrix}\right]^{1/2} \end{cases} \quad (10)$$

Functions $f_1(X)$, and $f_2(X)$ are the objective functions for fatigue failure criteria and static strength conditions. In addition, these functions are subject to fatigue criterion constraints, values less than the allowable stress. Stress on contact length b, determined according to [9]:

$$\sigma_{txb} = 0.591\sqrt{\frac{EF_{22}^{max}(D-d)}{d^2 b}} \quad (11)$$

where: b – the length of the contact section of the upper half joint with the pin (mm); D– the hole diameter (mm); E – the elastic modulus of steel (N/mm²); F_{12}^{max} – the force acting on the contact section (N); f_h – the allowable contact stress (N/mm²).

Let the minimum ratio between the contact length b and the distance B be $\Psi_2 = b / B$, and the ratio between the hole and pin diameter is $\xi = D / d$. Local position 5 on the contact length has the largest equivalent stress; the contact condition is $\sigma_{eq} \leq f_h$. Substitute the material data as $v = 0.3$, and $E = 2.1*10^5$ (N/mm²), and from formula (11) we have formula (12):

$$\sigma_{txb} = 0.591\sqrt{\frac{EF_{22}^{max}(\xi - 1)}{d\psi_2 B}}; \xi \leq \frac{f_h^2 \psi_2 B d_{min}}{0.35 EF_{22}^{max}} + 1 \quad (12)$$

where: f_h – the allowable contact stress (N/mm²) $fh = 2.5\ fy\ \gamma_{M6}$; γ_{M6} – the safety factor related to the conditions of use [8]; f_y – the yield strength of the hole material (with a value smaller than the pin material) (N/mm²); d_{min} – minimum diameter of the pin (mm).

The stress on the contact length a, determined according to [9]:

$$\sigma_{txa} = 0.591\sqrt{\frac{EF_{12}^{max}(\xi - 1)}{d\psi_1 A}} \quad (13)$$

The functions represented by the design parameters are the maximum contact stresses, after transformation from formulas (12) and (13):

$$\begin{cases} f_3(X) = f_h = 0.591\sqrt{\frac{EF_{22}^{max}(\xi - 1)}{d\psi_2 B}} \\ f_4(X) = 0.591\sqrt{\frac{EF_{12}^{max}\left(\frac{f_h^2 \psi_2 B d_{min}}{0.35 EF_{22}^{max}} - 1\right)}{d\psi_1 A}} \end{cases} \quad (14)$$

Functions $f_3(X)$ and $f_4(X)$ are the objective functions for the contact stress of the upper and lower half joints. In addition, these functions are also subject to constraints on allowable contact stress conditions.

For longitudinal dimensions of the pin, to ensure the structural conditions as shown in Figure 2, the dimensions must satisfy the formula (15):

$$f_5(X) = A(\psi_1 - 1) + 2(\psi_2 B + c) < 0 \quad (15)$$

The $f_5(X)$ function determines the structural constraints of the joints on the gantry crane. The joint supports have the parameters described in Figure 3. Minimum width is $W \geq D + 2e$, dimensions e and h meet the requirements in [9].

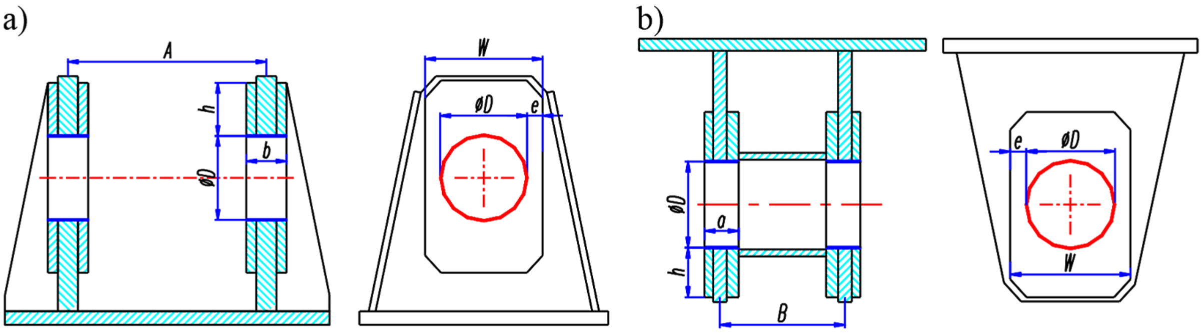

Fig. 3. Joint support: (a) joint supports of the lower half joint, (b) joint supports of the upper half joint

Required joint support size for the upper half of the joint [9]:

$$h \geq \frac{F_{22}^{\max}\gamma_{Mo}}{2bf_y} + \frac{2d}{3}; e \geq \frac{F_{22}^{\max}\gamma_{Mo}}{2bf_y} + \frac{d}{3} \tag{16}$$

Required joint support size for the lower half of the joint [9]:

$$h \geq \frac{F_{12}^{\max}\gamma_{Mo}}{2af_y} + \frac{2d}{3}; e \geq \frac{F_{12}^{\max}\gamma_{Mo}}{2af_y} + \frac{d}{3} \tag{17}$$

where: γ_{M0} – a coefficient determined according to [8].

Optimization problem model using the weighted sum method and the Taguchi method

The purpose of the problem is to optimally design the joint structure as shown in Figure 2. The design variables *X* are chosen as aspect ratios $\Psi_1 = a / A$, $\Psi_2 = b / B$, the yield strength of the material for the support is f_y, the minimum diameter of the pin is *d*. The given parameters X_0 include the corresponding fatigue stress concentration factors for normal and shear stresses, the mean values, load force and moment amplitudes, bearing spacing, and pin material. The joint structure is as shown in Figure 2, because the pin is smooth, so $K_f = K_{fs} = 1$. The structure of the joint consists of the pin and the half joints, which must ensure the conditions of durability, service life, and structural requirements. The optimization problem is set with the goal of selecting the optimal multi-objective parameters, including the objective functions of equivalent stress and contact stress, and the criterion of fatigue failure is the smallest. The optimization problem using the weighted sum method and the Taguchi method in this study is depicted in Figure 4. Mathematically, a multi-objective optimization problem is as follows.

- Independent factors X_0:

$$X_0 = \{A, B, K_f, K_{fs}, F_a, F_m, M_a, M_m \ldots\} \tag{18}$$

- Design variables X:

$$X = \{\psi_1, \psi_2, f_y, d\} \tag{19}$$

- Objective function *f (X)*:

$$f(X) = \min\left(f_1(X), f_2(X), f_3(X), f_4(X)\right) \tag{20}$$

- Constraints:

$$\begin{cases} \psi_{1\min} \leq \psi_1 \leq \psi_{1\max}; \psi_1 > 0 \\ \psi_{2\min} \leq \psi_2 \leq \psi_{2\max}; \psi_2 > 0 \\ f_y = \{f_{y1}, f_{y2}, f_{y3} \ldots\} \\ d_{\min} \leq d \leq d_{\max}; d > 0 \\ f_1(X) \leq 1; f_2(X) \leq \dfrac{S_{ut}}{k} \\ f_4(X) \leq f_3(X); f_5(X) < 0 \end{cases} \tag{21}$$

where: Ψ_{1min}, Ψ_{1max}, Ψ_{2min}, Ψ_{2max} – the largest and smallest ratios it is determined according to the structural conditions of the joint; f_{yi} – strength data of the material intended for manufacture; d_{min}, d_{max} – the largest and smallest diameter of the pin.

The weighted sum multi-objective optimization method is a method of converting multiple objectives into a single scalar objective function. We assign a weight coefficient to each objective function and then sum all the contributing factors to obtain the overall objective function. By using the weighted sum method, each objective is assigned a weight to distinguish its relative importance in the synthesis of the overall objective function [23], [26].

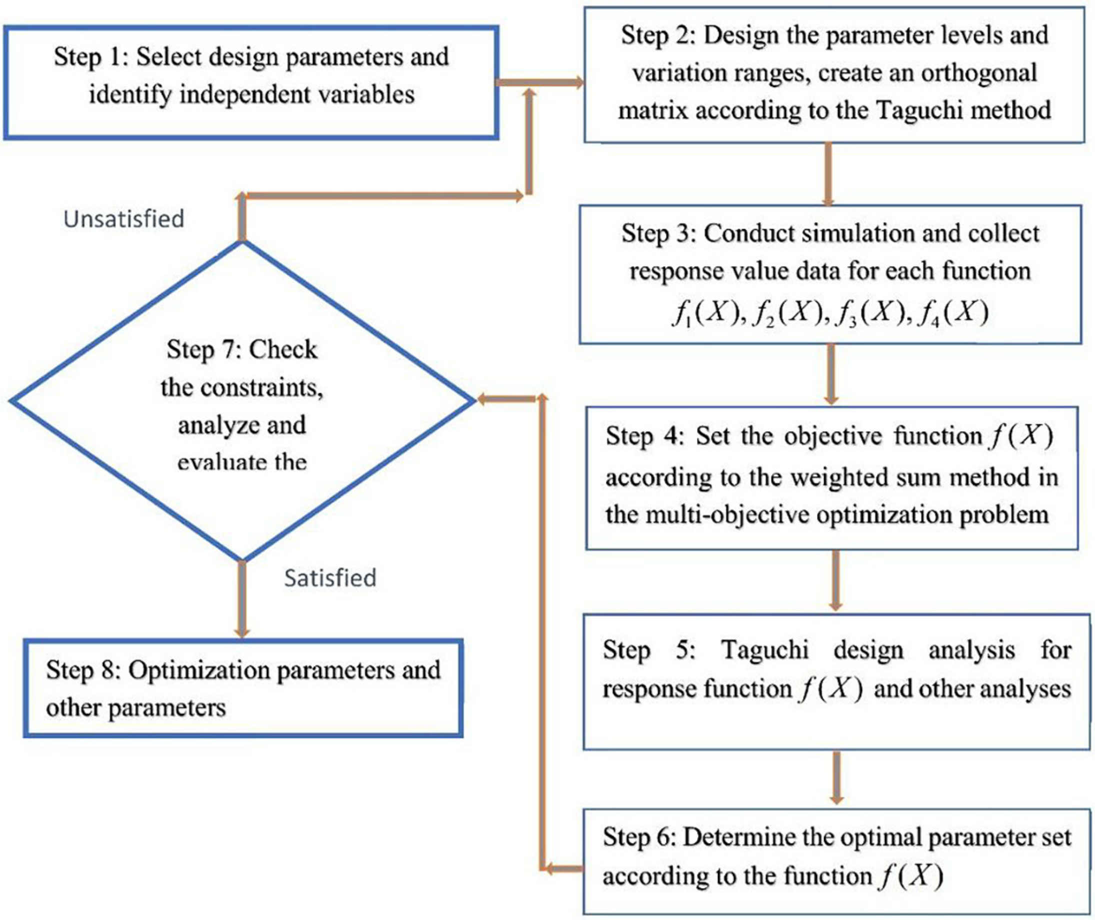

Fig. 4. Research method

$$\min F(X) = \frac{a_1 f_1(X)}{a_{10}} + \frac{a_2 f_2(X)}{a_{20}} + \frac{a_3 f_3(X)}{a_{30}} + \frac{a_4 f_4(X)}{a_{40}} \quad (22)$$

where: a_i – the weights of the objective functions $\Sigma a_i = 1$; a_{i0} – scalar parameters to be equated.

The problem now becomes optimizing an objective *f(X)*, which is to find the value that minimizes the function. In this study, the Taguchi method is used to optimize the function *f(X)* by evaluating the influencing factors to achieve the highest efficiency by detecting and eliminating the maximum influence of disturbances. A design variable affects the result in two directions; the effect that makes the result closer to the target is a useful signal, abbreviated as "Signal", the effect that makes the result farther from the target is "Noise". For the problem of minimizing the response function described by (22), the S/N ratio is as follows [19-22]:

$$S/N = -10\lg\left(\frac{1}{m}\sum_{k=1}^{m} F_{ik}^2\right) \quad (23)$$

where: k – the test sequence number, m – the number of tests, F_{ik} – the response value according to function (22).

Building a problem model using the Taguchi method will include determining the given factors (18) and influencing variables (19), and the objective function (22). Then select the original matrix to conduct the Taguchi method, carry out the method according to the selected matrix, and evaluate the results. The parameter is reasonable when the S/N ratio is the largest value. Minitab software is used to support calculations using the Taguchi method.

Numerical testing

This study applies the weighted sum optimization method and combines the Taguchi method to improve the existing design. The initial data set is based on the 500-ton trolley design of the 1,500-ton shipbuilding gantry crane designed by the Institute of Electromechanical Science and Technology, Vietnam. The gantry crane has been

Table 1. Given factors

No	Parameter	Symbol	Value	Unit
1	Distance between the two lower half joint supports	A	480	mm
2	Distance between the two upper half joint supports	B	280	mm
3	Average value of the force acting on the joint	F_a	1360000	N
4	Amplitude value of the force acting on the joint	F_m	663000	N
5	Average value of the moment acting on the joint	M_a	762000000	Nmm
6	Amplitude value of the moment acting on the joint	M_m	762000000	Nmm
7	Fatigue stress concentration factor with normal stress	K_f	1	
8	Fatigue stress concentration factor with shear stress	K_{fs}	1	
9	Allowable strength at the local location of the pin	S_e	236	N/mm²
10	Tensile strength of the pin	S_{ut}	690	N/mm²
11	Minimum safety factor according to the static strength condition of the pin	k	1.5	
12	The gap between the two halves of the joint	c	5	mm
13	Poison Coefficient	υ	0.3	
14	Modulus of elasticity of steel	E	210000	N/mm²

Table 2. Design variables and value levels

Design variables	Symbol	Value level				Variation range
		1	2	3	4	
Ratio $\Psi_1 = a / A$	x_1	0.1	0.14	0.18	0.22	0.12
Ratio $\Psi_2 = b / B$	x_2	0.3	0.35	0.4	0.45	0.15
Material yield strength for joint supports f_y (N/mm²)	x_3	240	290	340	390	150
Minimum pin diameter d (mm)	x_4	210	220	230	240	30

calculated and designed according to the regulations in [7]. Given the optimization problem parameters as in Table 1, the designed gantry crane has $x_1 = \Psi_1 = a/A = 0.22$, $x_2 = \Psi_2 = b/B = 0.35$, $x_3 = f_y = 390$ N/mm², $x_4 = d = 230$ mm.

The optimization problem determines four design parameters to optimize the multi-objective relationship between four objective functions: minimum stress and best fatigue failure criterion. The constraints (24) are given based on the structure of the designed gantry joint and common manufacturing materials.

$$\begin{cases} 0.1 \le x_1 \le 0.22; \Delta x_1 = 0.12 \\ 0.3 \le x_2 \le 0.45; \Delta x_2 = 0.15 \\ 240 \le x_3 \le 390; \Delta x_3 = 150 \\ 210 \le x_4 \le 240; \Delta x_4 = 30 \end{cases} \quad (24)$$

where: x_i, Δx_i – the design variables and ranges.

The study uses the L16 orthogonal matrix according to the Taguchi method for four design variables and four value levels. Table 2 is the design variables and value levels. Table 3 is the response value from formulas (10) and (14) of the orthogonal matrix test.

RESULTS AND DISCUSSION

Research results

In this design, the importance level related to contact stress and equivalent stress is considered to have the same role; these values are considered the maximum value in the stress cycle. In this numerical test, the optimal design weights for these objective functions are the same $a_2 = a_3 = a_4 = 0.3$, $\Sigma a_i = 1$, so the weights related to the fatigue strength criteria should be optimized $a_1 = 0.1$. The scalar conversion parameter a_{i0} is determined based on Table 3:

$$\begin{cases} a_{10} = \max f_1(X) = 1 \\ a_{20} = \max f_2(X) = 339 \\ a_{30} = \max f_3(X) = 975 \\ a_{40} = \max f_4(X) = 1270 \end{cases} \quad (25)$$

Table 3. Orthogonal matrix and response values of objective functions

No	x_1	x_2	x_3	x_4	$f_1(X)$	$f_2(X)$	$f_2(X)$	$f_4(X)$
1	0.1	0.3	240	210	0.73	248	600	638
2	0.1	0.35	290	220	0.66	223	725	833
3	0.1	0.4	340	230	0.6	202	850	1044
4	0.1	0.45	390	240	0.54	183	974	1270
5	0.14	0.3	290	230	0.65	220	725	652
6	0.14	0.35	240	240	0.59	200	600	1012
7	0.14	0.4	390	210	0.81	275	975	1012
8	0.14	0.45	340	220	0.72	245	850	936
9	0.18	0.3	340	240	0.65	218	850	674
10	0.18	0.35	390	230	0.72	243	975	835
11	0.18	0.4	240	220	0.8	271	600	550
12	0.18	0.45	290	210	0.9	305	725	704
13	0.22	0.3	390	220	0.88	300	975	700
14	0.22	0.35	340	210	1	339	850	659
15	0.22	0.4	290	240	0.71	240	725	600
16	0.22	0.45	240	230	0.79	267	600	527

Table 4. Objective function design of the weighted sum optimization method

No	x_1	x_2	x_3	x_4	$\frac{f_1(X)}{a_{10}}$	$\frac{f_2(X)}{a_{20}}$	$\frac{f_3(X)}{a_{30}}$	$\frac{f_4(X)}{a_{40}}$	F(X)
1	0.1	0.3	240	210	0.73	0.73	0.615	0.502	0.628
2	0.1	0.35	290	220	0.66	0.66	0.744	0.656	0.683
3	0.1	0.4	340	230	0.6	0.6	0.872	0.822	0.747
4	0.1	0.45	390	240	0.54	0.54	0.999	1	0.816
5	0.14	0.3	290	230	0.65	0.65	0.744	0.513	0.637
6	0.14	0.35	240	240	0.59	0.59	0.615	0.797	0.66
7	0.14	0.4	390	210	0.81	0.81	1	0.797	0.863
8	0.14	0.45	340	220	0.72	0.72	0.872	0.737	0.771
9	0.18	0.3	340	240	0.65	0.64	0.872	0.531	0.679
10	0.18	0.35	390	230	0.72	0.72	1	0.657	0.784
11	0.18	0.4	240	220	0.8	0.8	0.615	0.433	0.634
12	0.18	0.45	290	210	0.9	0.9	0.744	0.554	0.749
13	0.22	0.3	390	220	0.88	0.88	1	0.551	0.819
14	0.22	0.35	340	210	1	1	0.872	0.519	0.817
15	0.22	0.4	290	240	0.71	0.71	0.744	0.472	0.648
16	0.22	0.45	240	230	0.79	0.79	0.615	0.415	0.624

Applying the weights a_i, and the convergence parameters a_{i0} to calculate the function *F(X)*according to (22) using the orthogonal matrix L16, we get the calculation results shown in Table 4.

Taguchi analysis with response function *F(X)* is performed using Minitab software. The response function *F(X)* has a small value, which is good. The analysis results according to the S/N ratio are shown in Figure 5. Table 5 is the result of response value analysis according to S/N ratio, Table 6 is the result of mean value analysis according to each level of each variable, Table 7 is the variance analysis of design variables. The results showed that the most influential factor was the yield limit of the joint structure material $x_3 = f_y$, followed by the minimum pin diameter $x_4 = d$, then $x_2 = \Psi_2$ and $x_1 = \Psi_1$. In addition, the regression analysis results of the *F(X)* function compared to

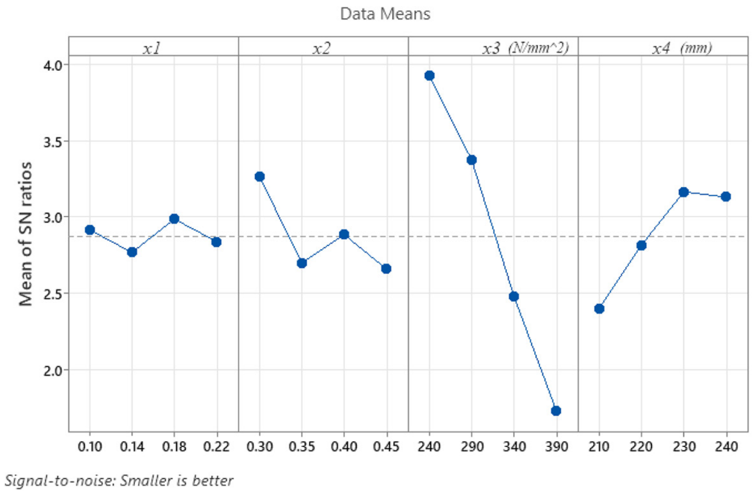

Fig. 5. Main effects plot for SN ratios

Table 5. Response table for signal-to-noise ratios

Level	x_1	x_2	x_3	x_4
1	2.914	3.267	3.925	2.395
2	2.766	2.697	3.376	2.813
3	2.984	2.882	2.477	3.164
4	2.837	2.655	1.723	3.129
Delta	0.219	0.611	2.202	0.769
Rank	4	3	1	2

Table 6. Response table for means

Level	x_1	x_2	x_3	x_4
1	0.7184	0.6905	0.6366	0.7644
2	0.7328	0.7361	0.6794	0.7270
3	0.7117	0.7232	0.7536	0.6981
4	0.7272	0.7402	0.8205	0.7005
Delta	0.0212	0.0497	0.1840	0.0663
Rank	4	3	1	2

x_3 are shown in Figure 6, and Figure 7 is a graph representing the relationship *F(X)*, and x_3, x_4.

For the response function using the weighted sum method, the regression function determined by Minitab software is expressed by function (26). Function (26) will allow to approximate the function F(X) by an analytical formula.

$$F(X) = 0.720 + 0.013\,x_1 + 0.272\,x_2 + + 0.001252\,x_3 - 0.002205\,x_4 \quad (26)$$

The analysis results showed that the best parameter set was $x_1 = \Psi_1 = 0.18$; $x_2 = \Psi_2 = 0.3$; $x_3 = f_y = 240$ N/mm²; $x_4 = d_{min} = 230$ mm. This parameter set gives the largest S/N. The response values $f_1(X) = 0.72$; $f_2(X) = 243$ N/mm²; $f_3(X) = 600$ N/mm²; $f_4(X) = 476$ N/mm² satisfy the constraint conditions (21), (27). Table 8 shows other geometrical parameters of the joint from which the optimum values were determined.

$$\begin{cases} f_1(X) = 0.72 \le 1 \\ f_2(X) = 243 \le \dfrac{S_{ut}}{k} = 460N/mm^2 \\ f_4(X) = 476 \le f_3(X) = 600N/mm^2 \end{cases} \quad (27)$$

DISCUSSION

Combining the weighted sum method and the Taguchi method for the design of gantry crane joints has given specific results as reasonable parameters in Table 8. The higher the mechanical properties of the joint support material, the more negative the impact on the objective function. According to the general rule in single-objective problems, the large pin diameter will give the smallest stress; however, if we consider the multi-objective problem in combination, the above rule is only true when 210 mm ≤ d ≤ 230 mm. All values obtained are reduced compared to the original design and increase the safety factor when comparing the design variables and response values before and after optimization (Table 9). The fatigue strength index decreased by 8.9%, increasing the fatigue strength safety factor from 1.26 to 1.39. The maximum equivalent stress of the pin is reduced by 9%, which increases the safety factor from 2.58 to 2.84. In particular, contact stresses

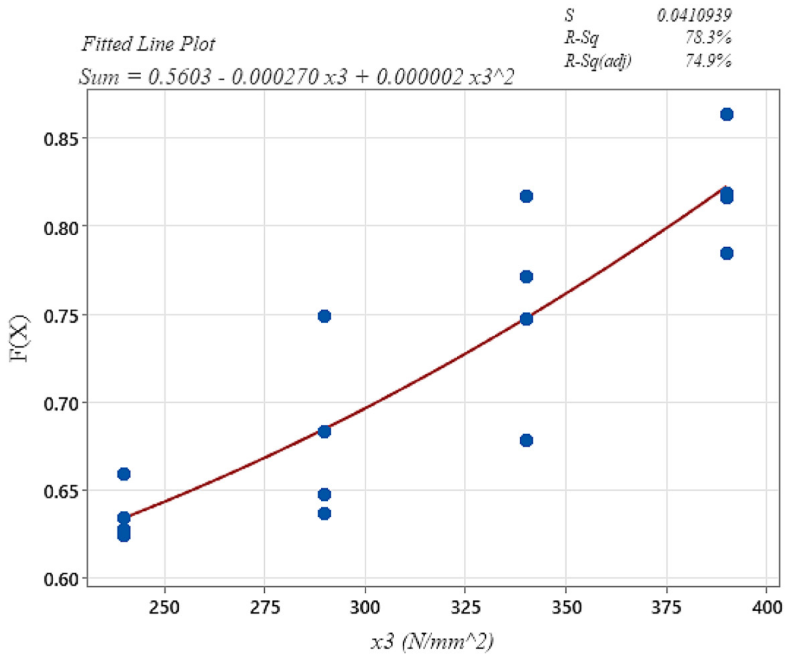

Fig 6. Regression analysis graph of *F(X)* the function with x_3

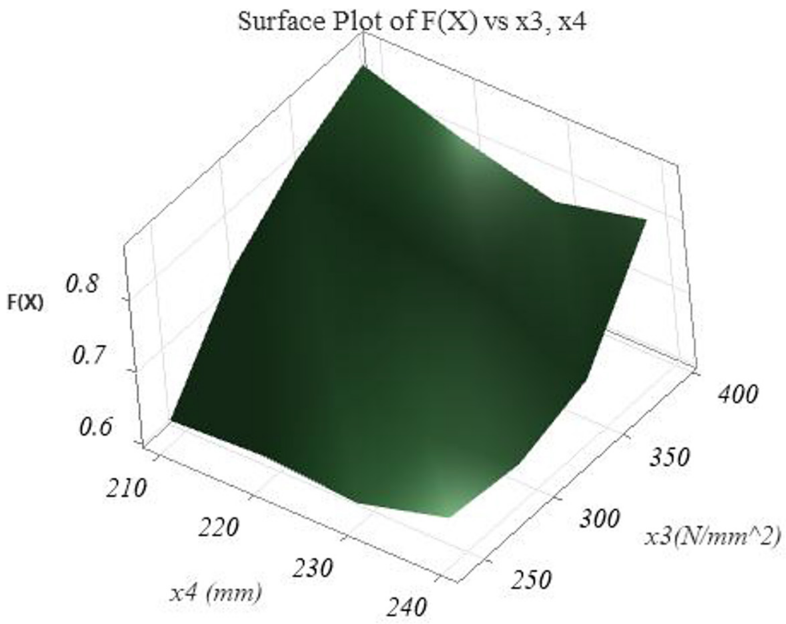

Fig 7. Graph representing the relationship *F(X)* with x_3, x_4

Table 7. Analysis of variance

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Regression	4	0.091860	0.022965	27.76	0.000
x_1	1	0.000005	0.000005	0.01	0.938
x_2	1	0.003707	0.003707	4.48	0.058
x_3	1	0.078422	0.078422	94.80	0.000
x_4	1	0.009726	0.009726	11.76	0.006
Error	11	0.009099	0.000827		
Total	15	0.100959			

Table 8. Parameters of the joint structure

No	Parameters	Value	Unit
1	Ratio $\Psi_1 = a / A$	0.18	
2	Ratio $\Psi_2 = b / B$	0.3	
3	Material yield strength for joint supports f_y	240	N/mm^2
4	Minimum pin diameter d_{min}	230	mm
5	Contact length *a*	86	mm
6	Contact length *b*	84	mm
7	Hole diameter *D*	233	mm
8	Minimum dimension h for the joint supports of the upper half joint	313	mm
9	Minimum dimension *h* for the joint supports of the lower half joint	254	mm
10	Minimum dimension *e* for the joint supports of the upper half joint	237	mm
11	Minimum dimension *e* for the joint supports of the lower half joint	178	mm

Table 9. Comparison of results before and after optimization

Comparison of stress and fatigue criteria								
Design parameters and response values	Ψ_1	Ψ_2	f_y N/mm^2	*d* mm	$\frac{1}{n}$	σ'_{max} N/mm^2	σ_{txb} N/mm^2	σ_{txa} N/mm^2
Value of initial design options	0.22	0.35	390	230	0.79	267	974	755
Optimal solution value	0.18	0.3	240	230	0.72	243	600	476
Rate change %	-18%	-14.3%	-38%	0 %	-8.9%	-9%	-38%	-37%
Compare safety factors								
Design parameters and response values	Ψ_1	Ψ_2	f_y N/mm^2	*d* mm	*n*	$k = \frac{S_{ut}}{\sigma'_{max}}$	$k_{tx} = \frac{f_h}{\sigma_{txb}}$	$k_{tx} = \frac{f_h}{\sigma_{txa}}$
Initial parameters and safety factors	0.22	0.35	390	230	1.26	2.58	1	1.29
Optimal parameters and safety factor	0.18	0.3	240	230	1.39	2.84	1.62	2

are reduced by 37% and 38%. In case the joint support structures have materials with a yield limit of 390 N/mm^2, the safety factor at the contact section is 1.62, and the safety factor at the contact section increases from 1.29 to 2.

Solving multi-objective problems is complicated. The combination of the weighted sum method and the Taguchi method in this study, as shown in Figure 4, has achieved the desired results. The Taguchi method has the advantage of being simple, but it is difficult to solve multi-objective problems. The weighted sum method solves the multi-objective optimization problem by combining multiple objective functions by adding them together, along with the importance weights for each function. This combination allows for the promotion of the effectiveness and advantages of each method. It allows for the expansion of the survey and can evaluate the influence of each objective function on design parameters in different technical conditions. Combining these two methods not only solves problems in engineering but can also solve many problems in different fields.

CONCLUSIONS

The article has studied the reasonable design calculation of the connection between the gantry crane legs and the beam to improve safety, service life, and reliability. Based on analysis and inheritance of related studies, the paper presents the basis of joint design and a multi-objective optimization problem model. The optimization method is a combination of the Taguchi method and the weighted sum method. This is a new idea suitable for solving a multi-objective optimization problem applied to this joint engineering design.

The experimental problem model has four design variables and four value levels. The study uses the orthogonal matrix L16 of the Taguchi method to calculate the response values for each objective function. Applying the sum-of-numbers method, the study chooses the weight of the objective function of equivalent stress and contact stress as 0.3, and the criterion of fatigue strength as 0.1, thus transforming the multi-objective problem into a single-objective problem. The study used Minitab software to evaluate the influence of parameters on the objective function, determine the regression function, and analyze variance. The results found the optimal set of parameters. The test results of the new set of parameters were smaller than the original design parameters, but the safety factors for equivalent stress, contact stress, and fatigue failure criteria were higher. Specifically, the fatigue criterion decreased by 8.9%, and the fatigue safety factor increased from 1.36 to 1.39; the equivalent stress decreased by 9%, and the safety factor increased from 2.58 to 2.84; the contact stress decreased by 37%, and the safety factor increased from 1.29 to 2.

The combination of the weighted sum method and the Taguchi method allows for the promotion of the effectiveness and advantages of each method. The combination of the two methods not only solves problems in engineering but can also solve many problems in different fields.

Acknowledgement

The authors would like to thank the Hanoi University of Civil Engineering (HUCE) for allowing them to conduct this research. The author would also like to thank the "Science - Technology Institute for Electric and Mechanical Engineering (STIEME), Viet Nam" for providing experimental data and applying research results into practice.

REFERENCES

1. Jiao Q., Qin Y., Han Y., Gu J. Modeling and Optimization of Pulling Point Position of Luffing Jib on Portal Crane. Mathematical Problems in Engineering, 2021. https://doi.org/10.1155/2021/4627257
2. Duong T.G., Ha T.P. Study on reasonable geometrical parameters of box-shaped crane steel structure taking into account the influence of local stability conditions. Journal of Science and Technology in Civil Engineering, 2014; 8(4): 36–43. https://stce.huce.edu.vn/index.php/vn/article/view/581/345
3. Duong T.G. Reasonable Design Method of Box Crane Girder by Taguchi Method. Journal of Applied Engineering Science, 2024; 22(1): 100–112. https://doi.org/10.5937/jaes0-45536
4. Hoshii T., Harada T. Strength Evaluation of Pin Joints for Crane Structure under Tensile Loads, Engineering Review, 1987; 20(3). https://www.ihi.co.jp/blkhnd/pdf/3.pdf
5. Ricker D.T. Design and Construction of Lifting Beams, Sengineering Journal, American Institute of Steel Construction, 1991; 149–158.
6. Budynas R.G., Nisbett J.K. Shigley's Mechanical Engineering Design. 10th edition, McGraw-Hill, 2020.
7. FEM 1.001: Rules for the Design of Hoisting Appliances (3rd Edition Revised 1998.10.01).
8. EN 1993-1-1: Eurocode 3: Design of steel structures – Part 1-1: General rules and rules for buildings (2005).
9. EN 1993-1-8: Eurocode 3: Design of steel structures – Part 1-8: Design of joints (2005).
10. Conde J., Da Silva L.S., Tankova T., Simões R., Abecasis T. Design of pin connections between steel members, Journal of Constructional Steel Research, 2022; 201: 107752. https://doi.org/10.1016/j.jcsr.2022.107752
11. Li Y., Huang R., Zhao S., Wang J. Contact pressure analysis of pin-loaded lug with clearance. Advances in Mechanical Engineering, 2022; 14(6): 1–15. https://doi.org/10.1177/16878132221107475
12. Bernardin P., Lasova V., Sedlacek F. Strength analysis of pin connections using computer-aided systems, MM Science Journal, 2017. https://doi.org/10.10.17973/mmsj.2017_03_2016103
13. Pedersen N.L. Stress concentration and optimal design of pinned connections. The Journal of Strain Analysis for Engineering Design, 2019; 54(2): 95–104. https://doi.org/10.1177/0309324719842766
14. Rao R.V., Savsani V.J. Mechanical design optimization using advanced optimization techniques. Springer, 2012.
15. Vinod, N.P. Optimization of Shaft Dimensions: A Comparative Study Using Computational Method and Finite Element Analysis. International Journal of Intelligent Systems and Applications in Engineering, 2024; 12(22s): 1049–1054. https://ijisae.org/index.php/IJISAE/article/view/6610
16. Choi J. W., Han S. H., Lee K. H. Structural analysis and optimization of an automotive propeller shaft. Advances in Mechanical Engineering. 2021; 13(10). https://doi.org/10.1177/16878140211053173
17. Pedersen, N. L. Aspects of stress in optimal shaft shoulder fillet. Journal of Strain Analysis for Engineering Design. 2018; 53(5): 285–294. https://doi.org/10.1177/0309324718763514
18. Duong T. G., Nguyen V.T., Nguyen T.D. Optimizing the weight of the two-level gear train in the personal

rescue winch. Archive of Mechanical Engineering, 2021; 68(3): 271–286. https://doi.org/10.24425/ame.2021.138393

19. Chen H.J., Lin H.C., Tang C.W. Application of the Taguchi Method for Optimizing the Process Parameters of Producing Controlled Low-Strength Materials by Using Dimension Stone Sludge and Lightweight Aggregates. Sustainability, 2021; 13: 5576. https://doi.org/10.3390/su13105576
20. Duong T. G. Determining parameters to optimize the pulling force for the Luffing Jib Tower Cranes by Taguchi method. Archive of Mechanical Engineering, 2023; 70(3): 387–407. https://doi.org/10.24425/ame.2023.146845
21. Duong T. G. Application of the Taguchi method to determine optimized parameters for designing brake of hand winch. EUREKA: Physics and Engineering, 2023; 6: 137–148. https://doi.org/10.21303/2461-4262.2023.002956
22. Duong T.G. Study to Determine the Effect of Blade Distance and Chain Speed on the Productivity of Trench Excavators Using Taguchi Method. Advances in Science and Technology Research Journal 2023; 17(4): 139–149. https://doi.org/10.12913/22998624/169427
23. Marler R. T., Arora J. S. The weighted sum method for multi-objective optimization: new insights, Struct Multidisc Optim, 2010; 41: 853–862. https://doi.org/10.1007/s00158-009-0460-7
24. Chagas J. B.C., Wagner M. A weighted-sum method for solving the bi-objective traveling thief problem. Computers & Operations Research, 2022; 138. https://doi.org/10.1016/j.cor.2021.105560
25. Li Y., Yang Q., Chang T., Qin T., Wu F. Multi-load cases topological optimization by weighted sum method based on load case severity degree and ideality. Advances in Mechanical Engineering, 2014; 12(8). https://doi.org/10.1177/1687814020947510
26. Stanimirovic I. P., Zlatanovic, M L., Petkovic M. D. On the linear weighted sum method for multi-objective optimization. Facta Acta Univ, 2011: 26(4): 49–63.