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DESIGN AND ANALYSIS OF THE OVERHEAD CRANE WITH SIX VERTICAL COLUMNS

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Abstract. This study delves into the comprehensive examination of an overhead crane's construction, focusing on its frame, columns, and beams, while considering factors such as strength, stability, and stiffness. Through an in-depth review in the design domain, it is proposed a specific structural configuration for the overhead crane. This design comprises six vertical columns supporting two longitudinal beams equipped with tracks for the trolley's longitudinal movement. Additionally, cross beams featuring winches are mounted on the trolley's cross beams. The crane's columns are securely mounted on a foundation, and struts are employed to attach the crane to the load-bearing wall of the building, ensuring longitudinal and transverse stability. The inclusion of cross truss structures with longitudinal struts further enhances the crane's overall stiffness, with additional vertical struts provided to augment the left side's longitudinal stiffness.

The study also encompasses the analysis of the crane frame's construction, complete with the development of an appropriate calculation scheme and the computation of static reactions in supports. Further calculations involve determining the cross-sections of vertical columns and longitudinal beams, ensuring compliance with strength, rigidity, and stability requirements. The selected cross-section for the columns, in the form of a square profile pipe (100×100×3 mm), is meticulously chosen to meet these criteria. Simulation modeling of load scenarios on the crane frame elements in SolidWorks software validates their strength, stiffness, and stability.

Mathematical models and calculations provided the optimal parameters and characteristics of each crane component, ensuring a superior level of safety and operational efficiency. These results provide valuable insights for future research in mechanical engineering and the design of industrial mechanisms.

Key words: overhead crane; structural analysis; design optimization; strength; industrial applications; simulation modeling.

Introduction

Overhead cranes play an important role in the industrial and construction industries, providing efficient lifting and moving of large loads. However, creating optimal overhead crane designs requires a comprehensive approach that encompasses both technical and engineering aspects.

This article deals with the process of development and analysis of the structure of a overhead crane made with six vertical columns. The proposed design of the crane includes carefully designed elements such as support columns, girders, trusses and carriage, which have a significant effect on the strength, stiffness and stability of the structure.

A literature review was conducted, functional features were analyzed, and structural solutions were developed, which made it possible to establish the optimal design of the overhead crane.

The presented research results can be of significant practical interest to engineers and designers involved in the development of overhead cranes and similar equipment.

Problem statement

The problem addressed in this article revolves around the design and optimization of overhead cranes, which is an important equipment in industrial settings for lifting and transporting heavy loads. These cranes must meet stringent requirements for strength, stability, efficiency, and safety to ensure smooth operations and minimize downtime. However, designing such complex systems involves numerous challenges and considerations.

Review of modern information sources on the subject of the paper

Overhead cranes are an important component in the industrial and construction industries, providing efficient lifting and moving of large loads. The process of designing and optimizing overhead crane structures is a complex task that requires a comprehensive approach and consideration of various technical and engineering aspects [1, 2]. To achieve optimal solutions in the design of overhead cranes, it is important to familiarize with the results of previous research and development in this field. In modern industry, cranes play an important role in providing efficient transportation of various materials and products.

The works [3, 4] consider the analysis of crane dynamics taking into account the geometry of the load and the suspension cable system, and consider the dynamic analysis and reliable control of a ship crane with a multi-cable anti-sway system. The work [5] is devoted to the design and built-in application of NMRS for a overhead crane on the example of a specific case. Paper [6] provides a valuable contribution to this field by investigating a crane system designed to transport galvanized coils. The authors analyzed various aspects of the operation of the crane system, including elements, theoretical calculations, as well as the creation of three-dimensional models. The results of the study indicate the possibility of successful production of a crane designed by the needs of industrial production.

Article [7] studies the influence of eccentric loads on the dynamics and oscillations of industrial cranes that transport heterogeneous loads. The authors developed a dynamic model of overhead cranes carrying heterogeneous loads and proposed control methods for the efficient management of such cranes. The study confirms the need to consider eccentricity in the analysis and management of industrial cranes with heterogeneous loads. The obtained results can be useful for the development and optimization of heavy cargo transportation systems in industrial conditions. Article [8] considers the method of assessing the stability of the mobile crane control system based on the values of safety indicators. The authors conduct analyses using a mathematical model and a model built in an integrated CAD/CAE environment to determine the deviations of the center of mass of the crane, reactions of the support system, stabilizing and overturning moments, as well as the values of safety indicators for the corresponding trajectories of the working elements of the crane. The study proposes a new approach to evaluate the effectiveness of the crane control system based on the values of the safety indicators for different load conditions and different load movement trajectories. In the article [9], a visual control scheme for overhead crane systems is proposed. This scheme uses a high-level decision engine, called a supervisor, to select the most suitable controller from a family of candidates based on real monitoring signals. The results of simulations and experiments confirm the stability, accurate estimation of cargo mass, and convenient selection of parameters of the proposed visual control scheme.

Paper [10] examines different models and control methods for overhead cranes operating in 2D and 3D spaces. The characteristics of various models, including models with rigid and flexible connections, are studied, and the control methods developed for such models are considered. Numerical experiments are conducted to evaluate and compare the effectiveness of different methods.

The development of a control method for overhead crane systems using an observer and disturbance feedback using the Disturbance Feedback Control technique is described in paper [11]. A method for assessing the technical condition of overhead crane equipment, in particular, equipment that operates in conditions that promote corrosion, is considered in work [12]. The method includes the use of non-destructive methods of assessing the technical condition and numerical simulations of the operation of cranes to identify possible problems and find ways to solve them.

Based on this comprehensive literature review, the design aspect of the study presents a practical overhead crane design comprising six vertical columns.

Objectives and problems of research

The main objective of this research is to address the challenges associated with the design and optimization of overhead cranes, which play a crucial role in industrial environments for the lifting and transportation of heavy loads. These cranes are integral to various industrial processes and must adhere to strict standards regarding strength, stability, efficiency, and safety to facilitate seamless operations while minimizing downtime. The design process for these complex systems requires careful consideration of various factors.

Main material presentation

1. General view of the crane.

Based on the conducted review of information, a proprietary structural design of the overhead crane is proposed in Fig. 1, consisting of six vertical columns 1. I-Beams 2 and 6 with rails for longitudinal movement of carriage 9 are installed on the columns. Rails for lateral carriage movement 10 with a winch are placed on the I-Beams 8 of carriage 9.

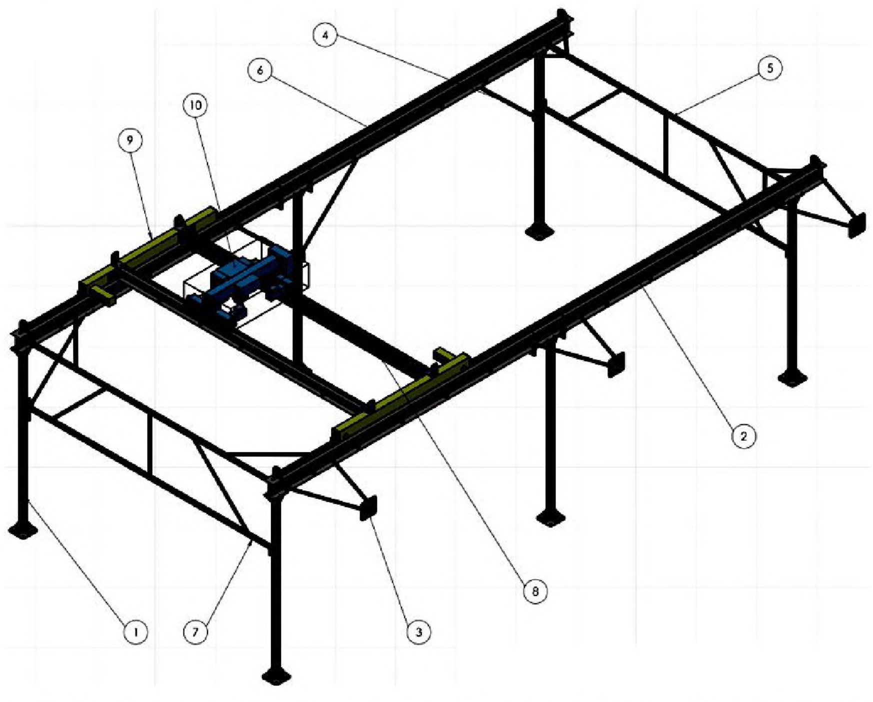

Fig. 1. General layout of the overhead crane: 1 – strut; 2, 6 – longitudinal I-Beams; 3 – horizontal strut; 4 – vertical brace; 5, 7 – trusses; 8 – transverse I-Beam; 9 – carriage of longitudinal movement; 10 – carriage of transverse movement

The struts 1 of the crane are mounted on the foundation. To ensure the longitudinal and transverse stability of the metal structure of the crane, its fastening to the load-bearing wall of the building using struts 3 is provided. Transverse truss structures 5 and 7 with longitudinal struts are installed between the two longitudinal I-Beams, which also increase the longitudinal and transverse stiffness of the overhead crane structure. To increase the longitudinal stiffness of the left side of the crane, which is not connected to the load-bearing wall, vertical braces 4 are provided in the design.

1.1. Design features of individual elements of the stationary frame of the crane.

The general view of the crane column is shown in Fig. 2. It consists of upper 3 and lower 1 plates, pipe 2, and stiffeners 4. Plate 1 is installed on a specially prepared foundation. A vertical support pipe 2 is welded to it, which is additionally fixed with eight stiffeners 4. In the upper part, a plate 3 is attached to the pipe 2, which is additionally fixed with four stiffeners 4. A longitudinal I-Beam is installed on plate 3 during the installation of the general structure of the crane. The cross-section of the column is a square profile pipe 100×100×3 mm. The thickness of the upper plate and stiffeners is 10 mm, and the lower plate is 20 mm.

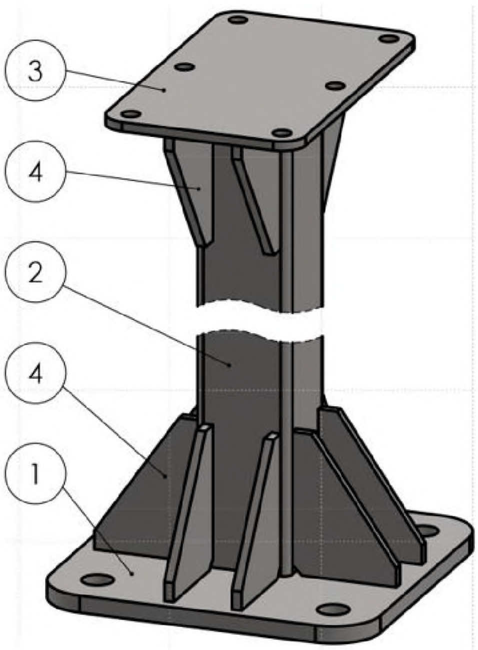

Fig. 2. General view of a vertical support column: 1 – lower plate of the column; 2 – column pipe; 3 – upper plate; 4 – stiffeners

The longitudinal I-Beam (Fig. 3) is welded: the horizontal upper and lower shelves are welded to the vertical plate. A rail is welded on the upper plane of the I-Beam shelf, along which the longitudinal movement carriage moves. The movement of the carriage is limited on both sides of the rail by special stoppers. The general assembly of the I-Beam consists of three parts, the two outermost of which are equal in length with a total dimension of about 6 m, and the other part is a connecting part – about 700 mm long. Stiffeners are welded along the entire length of the I-Beam with a certain step on both sides of the vertical plate. The lower plates of all I-Beams (two long and one short) have holes for installation and connection to the upper support plates of the vertical columns. The corresponding sections of the longitudinal I-Beam are connected using 20 mm thick connecting plates with bolted connections.

To ensure both transverse and longitudinal stiffness of the overhead crane structure, the use of two transverse trusses with longitudinal bracing is provided (Fig. 4).

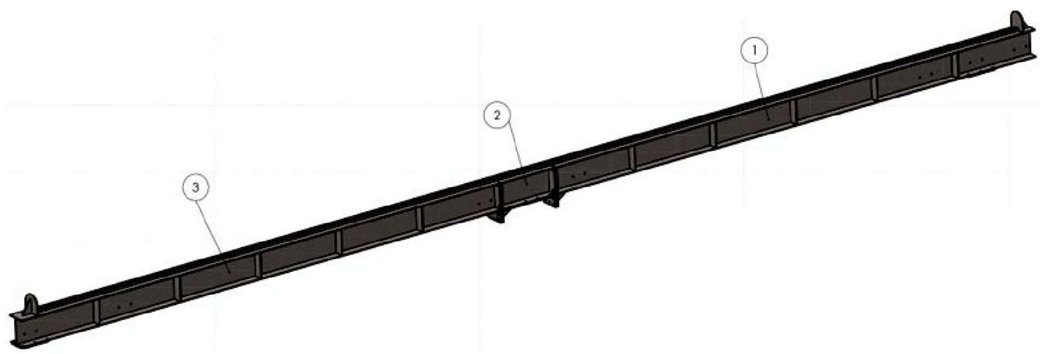

Fig. 3. General view of the longitudinal I-Beam: 1, 3 – long sections (6 m); 2 – short section (700 mm)

The trusses are solid (welded) truss structures made from profile pipes. Connective plates are welded to the corresponding pipes of the trusses on the lateral ends, through which the trusses are attached to the vertical support columns and longitudinal I-Beams. Considering the precisely specified placement of the columns and the projections of the longitudinal I-Beams, the transverse trusses on both sides of the crane structurally differ. One truss is designed with one pair of braces, while the other truss is designed with two pairs of braces (Fig. 1, 4).

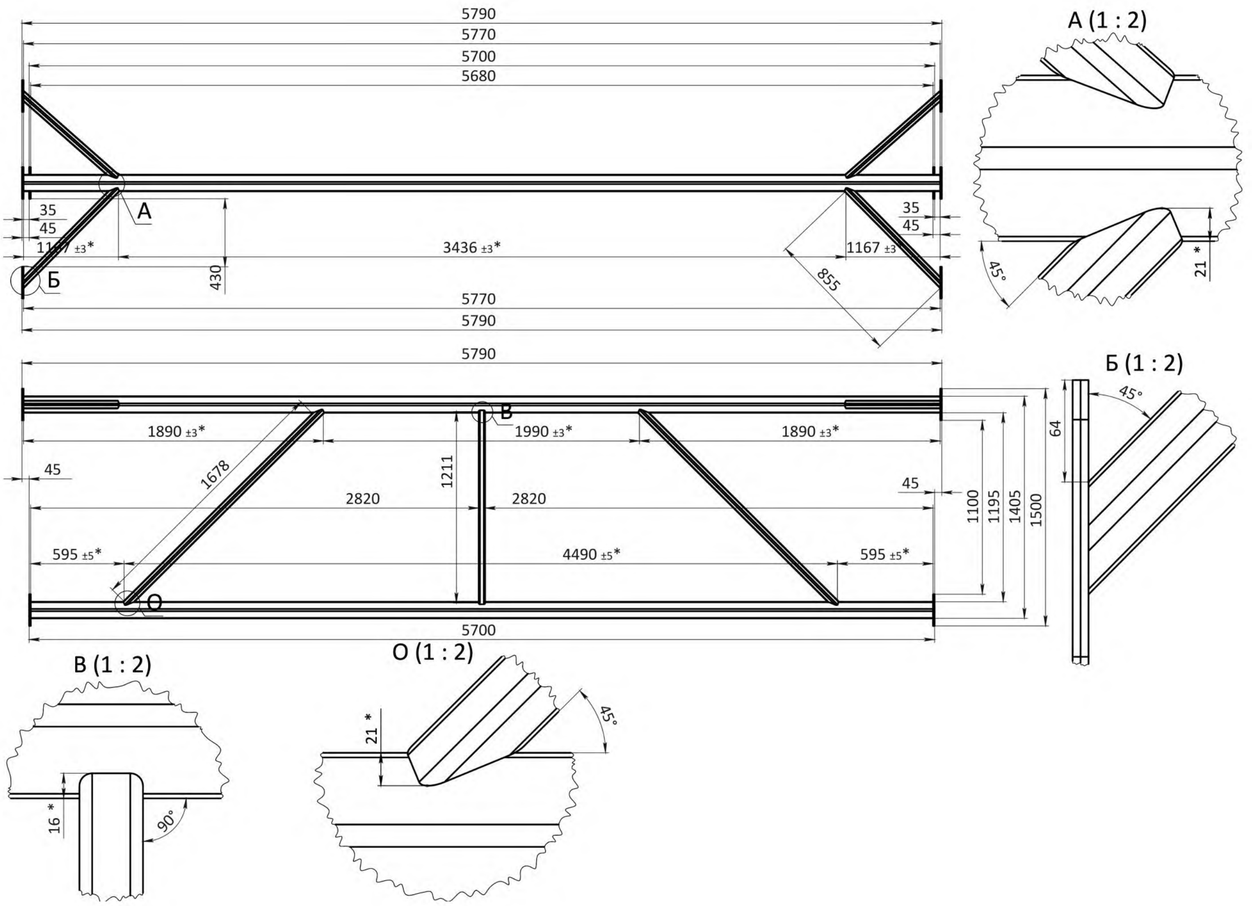

Fig. 4. General view of the transverse truss

1.2. Carriage structure for longitudinal movement

The designed layout of the carriage for longitudinal movement is presented in Fig. 5. The carriage, supported by two wheels (5), is installed on the rails of the longitudinal I-Beams. One of the wheels serves purely as a support, while the other is driven by the shaft of the motor-reducer (13, 22). The carriage body is of a welded box type. Bumpers (rubber buffers) are installed on the ends of the carriage.

1.3. Structure of the cargo carriage with winch

The overall layout of the cargo carriage with winch is shown in Fig. 6. The winch drum (11), on which the rope (13) is wound, is mounted via bearing supports (10) on the frame (3) of the carriage. The rotation of the drum (11), which simultaneously raises or lowers the hook (12), is achieved through a chain coupling (9) and a cylindrical reducer (6) driven by the motor shaft (5). To immobilize the winch drive in unforeseen situations, a disc friction brake (7) is used. The frame (3) with the winch is installed on two mobile (wheel) carriages (1), which are powered by electric motors (4).

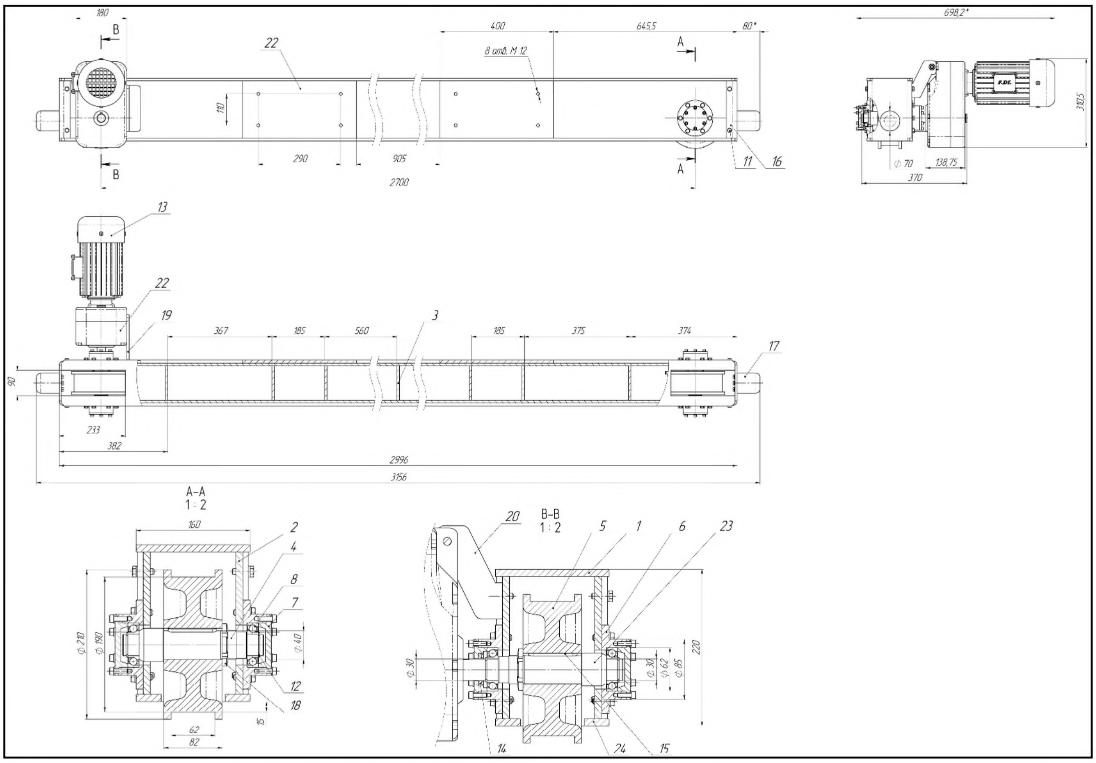

Fig. 5. General view of the longitudinal carriage

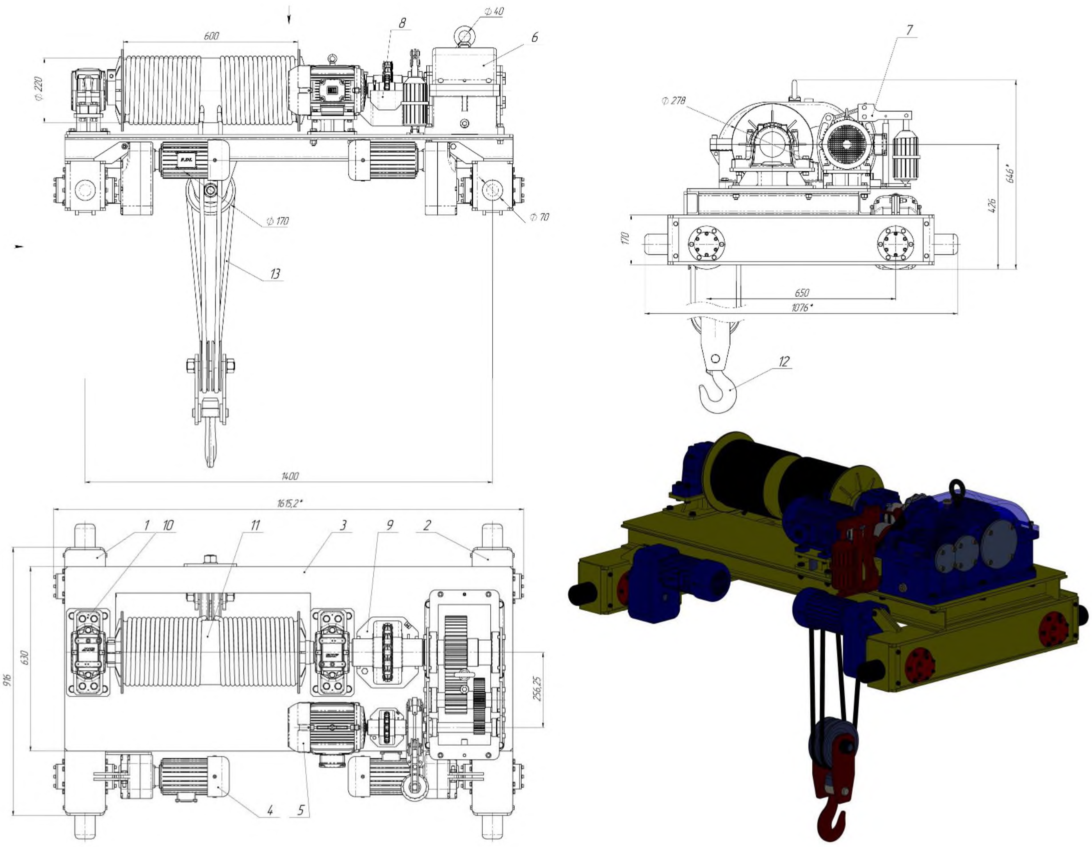

Fig. 6. General view of the cargo carriage with winch

2. Analysis of the crane frame construction and calculation of static reactions at supports

The crane consists of a fixed frame and carriages for longitudinal and transverse movements (Fig. 8). A single active force P acts on the beams of the transverse carriage, which includes the weight of the load and the weight of the carriage itself ($m = 2000kg$):

$$P = m \cdot g = 2000 \cdot 9{,}81 = 1{,}961 \cdot 10^4 \ N .$$

The transverse carriage moves along the longitudinal carriage on four wheels. In further calculations, we assume that each wheel bears one-fourth of the force P:

$$RK = RN = RO = RQ = \frac{P}{4} = \frac{1{,}961 \cdot 10^4}{4} = 4{,}903 \cdot 10^3 \ N .$$

In the extreme (left or right) positions of the transverse carriage, the vertical forces from the wheels K, N, O, Q are transferred to the supports E, H or X, W, respectively.

Let's consider the case of left positioning of the carriage:

$$RE = RK + RN = 4{,}903 \cdot 10^3 + 4{,}903 \cdot 10^3 = 9{,}807 \cdot 10^3 \ N ;$$

$$RH = RO + RQ = 4{,}903 \cdot 10^3 + 4{,}903 \cdot 10^3 = 9{,}807 \cdot 10^3 \ N .$$

Let's consider the case of left positioning of the carriage: E, H relative to the wheels B, C of the longitudinal carriage, we can determine the respective reactions:

$$RB = RE = RC = RH = 9{,}807 \cdot 10^3 \ N .$$

To find the maximum axial load on the vertical columns, we express the equilibrium equations:

$$RA(a) = \frac{RC \cdot a + RB \cdot (a + 2700[mm])}{6000[mm]} ;$$

$$RD(a) = \frac{RC \cdot (6000[mm] - a) + RB \cdot (3300[mm] - a)}{6000[mm]} ,$$

where $a = CD = 0 \ldots 3300 \ mm$ (see Fig. 8).

If the reaction at support A reaches its maximum value, it is logical that the reaction at support D will have its minimum value. Let's solve analytically and graphically the maximization problem of the expression $RA(a)$ and the minimization problem of the expression $RD(a)$ in MathCad (Fig. 7).

$a := 0mm$

Given

$0mm < a < 3300mm$

$a := Maximize(RA, a)$

$a = 3300mm$

$a := 0mm$

Given

$0mm < a < 3300mm$

$a := Maximize(RD, a)$

$a = 3300mm$

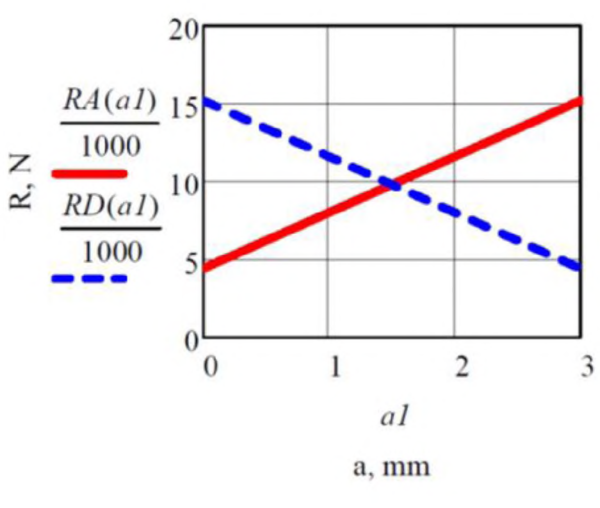

Fig. 7. Analytical and graphical solution of the maximization problem of the expression $RA(a)$ and the minimization problem of the expression $RD(a)$ in MathCad

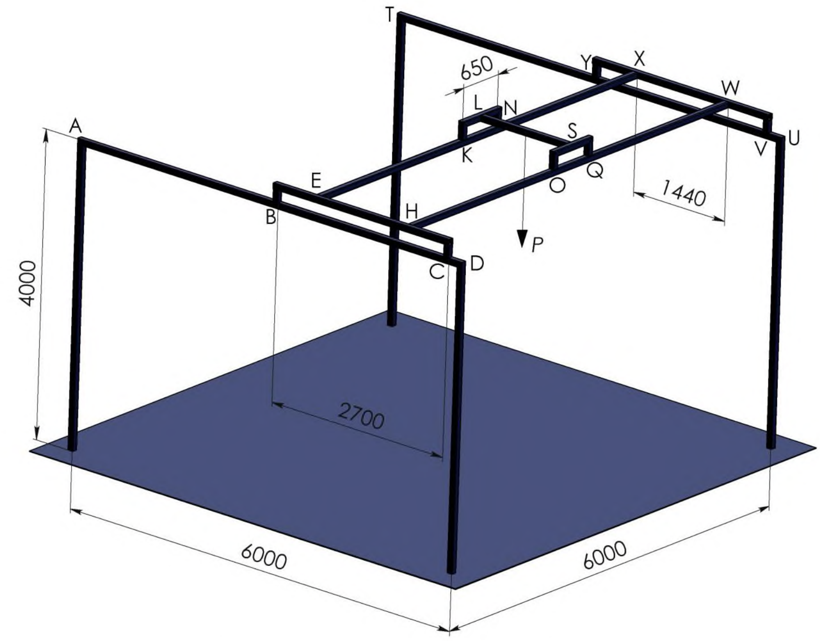

Fig. 8. Calculation scheme of the overhead crane frame

In both cases, the same values for the distance *a* were obtained. Let"s determine the maximum and minimum reactions and perform a check of the conducted calculations:

$$RA_{\max} = RA\left(3300[mm]\right) = \frac{9{,}807\cdot 10^3\cdot 3300 + \\ +9.807\cdot 10^3\left(3300+2700\right)}{6000} = 15200{,}308\ N;$$

$$RD\left(3300[mm]\right) = \frac{9{,}807\cdot 10^3\cdot\left(6000-3300\right) - \\ -9.807\cdot 10^3\left(3300-3300\right)}{6000} = 4412{,}992\ N;$$

$$RA\left(3300[mm]\right) + RD\left(3300[mm]\right) - RB - RC =$$

$$= 15200{,}308 + 4412{,}992 - 9{,}807\cdot 10^3 - 9{,}807\cdot 10^3 = 0\ N.$$

Taking into account the equality of zero of the last expression, we can conclude about the correctness of the calculations conducted.

2.1. Calculation of the cross-section of the vertical columns considering strength, stiffness, and stability conditions

The primary load on the crane columns is axial compressive force from the weight of the structure. In this case, the columns work under compression. Before proceeding to the direct calculation of the transverse section of the column, it is necessary to establish the allowable stress for its material (steel AISI 201 (12X15GN9ND), AISI 304 (08X18H10)):

– Material yield strength:

$$\sigma_t = 310\ MPa;$$

– Normative safety factor:

$$n_{st} = 1{,}6;$$

– Allowable stress for compression:

$$\sigma_{\max} = \frac{\sigma_t}{n_{st}} = \frac{310\cdot 10^6}{1{,}6} = 1{,}938\cdot 10^8\ Pa.$$

Additionally, in further calculations, it is necessary to consider the possibility of short-term overloading of the columns due to dynamic forces. For this purpose, we introduce a corrective coefficient for dynamic overload, which, in general, is determined by the formula:

$$k_d = 1 + \frac{a_v}{g} = 1 + \frac{9,807}{9,807} = 2,$$

when the maximum value of the vertical acceleration of the load is $a_v = 9,807 \cdot 10^3 \frac{mm}{s^2}$.

We will determine the minimum required cross-sectional area of the column from the following condition:

$$A_k = \frac{k_d \cdot RA_{\max}}{\sigma_{\max}} = \frac{2 \cdot 15200,308}{1,938} = 156,909\ mm^2.$$

Taking the cross-sectional area of the vertical column in the form of a square profile tube and considering its minimum required area, according to the assortment of profiled metal products, we select the tube minimum possible parameters $25 \times 25 \times 2\ mm$, the cross-sectional area of the tube is:

$$A_k = 177\ mm^2.$$

The absolute deformation of the column, i. e., its shortening under compression, within the elasticity of the column, will be calculated using Hooke's law, according to this the length of the undeformed pipe will be

$$L_k = 4000\ mm,$$

the modulus of elasticity (Young's modulus) of the pipe material is

$$E = 2 \cdot 10^5\ MPa,$$

and the absolute axial deformation of the pipe is

$$\Delta L = \frac{RA_{\max} \cdot L_k}{E \cdot A_k} = \frac{15200,308 \cdot 4000}{2 \cdot 10^5 \cdot 177} = 1,718\ mm.$$

Let's verify the vertical column for stability, considering that the stress at which the column is liable to lose stability may be significantly less than the material's yield strength, from which it is made.

The dimensions of the cross-section of the column are selected from the stability condition according to the method of successive approximations. For this, in the first approximation, we will assume the following value $\varphi = 0,15$ for the coefficient of longitudinal flexure.

From the stability condition, we determine the required cross-section area of the column:

$$A_k = \frac{k_d \cdot RA_{\max}}{\varphi \cdot \sigma_{\max}} = \frac{2 \cdot 15200,308}{0,15 \cdot 1,938 \cdot 10^8} = 1,046 \cdot 10^3\ mm^2.$$

The necessary dimensions of the pipe to ensure the calculated cross-sectional area of the column are $100 \times 100 \times 3\ mm$, the cross-sectional area of which is:

$$A_k = 1140\ mm^2.$$

The minimum moment of inertia of the column's cross-section is determined according to the assortment:

$$I_{\min} = 177\ \text{cm}^4 = 177 \cdot 10^4\ mm^4.$$

Let's calculate the minimum radius of inertia of the column's cross-section:

$$i_{\min} = \sqrt{\frac{I_{\min}}{A_k}} = \sqrt{\frac{177 \cdot 10^4}{1140}} = 39,403\ mm.$$

Let's find the flexibility of the column, taking into account the coefficient $\mu_{kr} = 2$, which takes into account column joint type:

$$\lambda = \frac{\mu_{kr} \cdot L_k}{i_{\min}} = \frac{2 \cdot 4000}{39,403} = 203,028.$$

The obtained value of the column's flexibility, according to reference data, allows us to find the corresponding coefficient of longitudinal flexure:

$$\varphi = 0{,}15 .$$

Next, check of the conducted calculations is performed.

Let's determine the actual compressive stress:

$$\sigma_d = \frac{k_d \cdot RA_{\max}}{A_k} = \frac{2 \cdot 15200{,}308}{1140} = 2{,}667 \cdot 10^7 \ Pa ,$$

allowable compressive stress of the column from the stability condition:

$$\sigma_{st} = \varphi \cdot \sigma_{\max} = 0{,}15 \cdot 1{,}938 \cdot 10^7 = 2{,}667 \cdot 10^7 \ Pa ,$$

underloading coefficient of the column:

$$\xi = \frac{\sigma_{st} - \sigma_d}{\sigma_{st}} \cdot 100 = \frac{2{,}906 \cdot 10^7 - 2{,}667 \cdot 10^7}{2{,}906 \cdot 10^7} \cdot 100 = 8{,}24 .$$

Considering that the calculated underloading coefficient of the column is 8.24 %, there is no need to further recalculate its cross-sectional area.

Using Euler's formula, we will determine the critical force F_{kr} that the support can withstand without losing stability, and the stability factor n_{st} :

$$F_{kr} = \frac{\pi^2 \cdot E \cdot I_{\min}}{\left(\mu_{kr} \cdot L_k\right)^2} = \frac{\pi^2 \cdot 2 \cdot 10^5 \cdot 177 \cdot 10^4}{\left(2 \cdot 4000\right)^2} = 5{,}459 \cdot 10^4 \ N ,$$

$$n_{st} = \frac{F_{kr}}{k_d \cdot RA_{\max}} \frac{5{,}459 \cdot 10^4}{2 \cdot 15200{,}308} = 1{,}796 .$$

The obtained stability factor value falls within the regulatory range of 1.5 to 3.

Taking into account the selected column cross-section, let's recalculate the value of its absolute axial deformation:

$$\Delta L = \frac{RA_{\max} \cdot L_k}{E \cdot A_k} = \frac{15200{,}308 \cdot 4000}{2 \cdot 10^5 \cdot 1140} = 0{,}267 \ mm .$$

2.2. Calculation of the cross-section of longitudinal beams for strength and stiffness

The primary load on the longitudinal beams of the crane is the vertical bending force from the weight of the structure. In this case, the beams work under bending.

To determine the minimum required axial elastic section modulus of the beam section, it is necessary to calculate the maximum bending moment. Knowing the maximum forces acting on the beam from the longitudinal motion carriage side, we find the maximum value of the bending moment distribution function along the length of the beam.

The reactions in the vertical columns (supports) of the beam are determined by the expressions:

$$RA(a) = \frac{RC \cdot a + RB \cdot \left(a + 2700[mm]\right)}{6000[mm]} ;$$

$$RD(a) = \frac{RC \cdot \left(6000[mm] - a\right) + RB \cdot \left(3300[mm] - a\right)}{6000[mm]} .$$

Let's determine the distribution function of the bending moment along the length of the beam for its different sections:

for section DC:

$$M_{b1}(a,x) = RD(a) \cdot x ,$$

for section CB:

$$M_{b2}(a,x) = RD(a) \cdot x - RC \cdot (x - a) ,$$

and for section BA:

$$M_{b3}(a,x) = RD(a)\cdot x - RC\cdot(x-a) - RB\cdot(x-a-2700).$$

Let's solve the problem of maximizing the distribution function of the bending moment along the length of the beam for its different sections in the MathCad software:

– section DC:

$$a := 0mm \quad x := 0$$
$$Given$$
$$0mm < a < 3300mm$$
$$0 < x < a$$
$$ax1 := Maximize(Mзг1, a, x)$$
$$ax1 := \begin{pmatrix} 2325 \\ 2325 \end{pmatrix} mm$$
$$Mb1(ax1_0, ax1_1) = 1.767\times10^4 N\cdot m$$

– section CB:

$$a := 0mm \quad x := 0$$
$$Given$$
$$0mm < a < 3300mm$$
$$0 < x < a + 2700mm$$
$$ax2 := Maximize(Mзг2, a, x)$$
$$ax2 := \begin{pmatrix} 2325 \\ 2325 \end{pmatrix} mm$$
$$Mb2(ax2_0, ax2_1) = 1.767\times10^4 N\cdot m$$

– section BA:

$$a := 0mm \quad x := 0$$
$$Given$$
$$0mm < a < 3300mm$$
$$0 + 2700 < x < a + 6000mm$$
$$ax3 := Maximize(Mb3, a, x)$$
$$ax3 := \begin{pmatrix} 975 \\ 3675 \end{pmatrix} mm$$
$$Mb3(ax3_0, ax3_1) = 1.767\times10^4 N\cdot m.$$

Based on the results of the calculations, we can conclude that for any positioning of the longitudinal motion carriage relative to the vertical support columns, the maximum bending moment value remains the same and, considering the dynamic overload factor, will be equal:

$$M_b = k_d\cdot\max\left(M_{b1}\left(ax_{1_0}, ax_{1_1}\right), M_{b2}\left(ax_{2_0}, ax_{2_1}\right), M_{b3}\left(ax_{3_0}, ax_{3_1}\right)\right) =$$
$$= 2\cdot\max\left(1,767\cdot10^4, 1,767\cdot10^4, 1,767\cdot10^4\right) = 35341,4\ N\cdot m.$$

Let's determine the minimum required axial section modulus of the beam section based on its strength condition under bending:

$$W_0 = \frac{M_b}{\sigma_{\max}} = \frac{35341,4\cdot10^3}{1,938\cdot10^2} = 1,824\cdot10^5\ mm^3.$$

We assume the beam section to be a welded I-Beam with the following parameters:
flange width:

$$b = 210 \text{ мм};$$

overall height:

$$h = 300 \text{ мм}.$$

For the calculated value of the axial section modulus of the I-Beam, we will determine the minimum required thickness, of the sheet from which the I-Beam will be made. First, we calculate the areas of the flanges and vertical flanges:

$$A_1(a) = A_2(a) = b \cdot a; \qquad A_3(a) = (h - 2 \cdot a) \cdot a.$$

Next, we find the corresponding central moments of inertia of the flanges and vertical flanges:

$$Ix_1(a) = Ix_2(a) = \frac{h \cdot a^3}{12}; \qquad Ix_3(a) = \frac{a \cdot (h - 2 \cdot a)^3}{12}.$$

Now, we determine the total moment of inertia of the beam about the bending axis Ox:

$$Ix_1(a) = Ix_2(a) = \frac{h \cdot a^3}{12}; \qquad Ix_3(a) = \frac{a \cdot (h - 2 \cdot a)^3}{12};$$

$$Ix(a) = Ix_1(a) + \left(\frac{h-a}{2}\right)^2 \cdot A_1(a) + Ix_2(a) + \left(\frac{h-a}{2}\right)^2 \cdot A_1(a) + Ix_3(a).$$

Calculate the axial section modulus of the beam:

$$Wx(a) = \frac{Ix(a)}{\frac{h}{2}}.$$

Substituting the calculated value Wo into the obtained equation, we solve the equation for the value of a in the MathCad program:

$$Given$$

$$Wx(a) = Wo$$

$$a := Find(a) = 2.391 \times 10^{-3} m$$

Thus, the minimum required thickness of the sheet from which the I-Beam should be made is:

$$a = 2{,}391 \cdot 10^{-3}\ m; \qquad a_1 = 2{,}391\ mm.$$

Let's perform a check of the selected beam section to ensure stiffness, for acceptable deflections, using the method of initial parameters. We will find the linear displacement (deflection) of the beam using the equation of the elastic line, assuming that the beam is rigidly fixed at both ends (the deflection and the angle of rotation at both ends are zero). Let's down the functions of the maximum angles of rotation and deflections in the two most dangerous positions of the longitudinal movement of the carriage:
position 1:

$$a = 2375\ mm;$$

$$RA(a) = \frac{RC \cdot a + RB \cdot (a + 2700)}{6000} =$$

$$= \frac{9{,}807 \cdot 10^3 \cdot 2375 + 9{,}807 \cdot 10^3 \cdot (2375 + 2700)}{6000} = 1{,}218 \cdot 10^4\ N;$$

$$w(x_1) = \frac{RA(a) \cdot x_1^3}{6 \cdot E \cdot Ix(a_1)} - \frac{RC \cdot (x_1 - 925)^3}{6 \cdot E \cdot Ix(a_1)} - \frac{RB \cdot (x_1 - 3625)^3}{6 \cdot E \cdot Ix(a_1)};$$

$$w_1 = \frac{RA(a) \cdot 925^3}{6 \cdot E \cdot Ix(a_1)} = \frac{1{,}218 \cdot 10^4 \cdot 925^3}{6 \cdot 2 \cdot 10^5 \cdot 2{,}736 \cdot 10^{10}} = 0{,}00029\ m;$$

$$w_2 = \frac{RA(a)\cdot 3625^3}{6\cdot E\cdot Ix(a_1)} - \frac{RC\cdot(3625-925)^3}{6\cdot E\cdot Ix(a_1)} =$$

$$= \frac{1,218\cdot 10^4\cdot 3625^3}{6\cdot 2\cdot 10^5\cdot 2,736\cdot 10^3} - \frac{9,807\cdot 10^3\cdot(3625-925)^3}{6\cdot 2\cdot 10^5\cdot 2,736\cdot 10^3} = 0,012\ m;$$

position 2:

$$a = 3300\ mm$$

$$RA(a) = \frac{RC\cdot a + RB\cdot(a+2700)}{6000} =$$

$$= \frac{9,807\cdot 10^3\cdot 3300 + 9,807\cdot 10^3\cdot(3300+2700)}{6000} = 1,52\cdot 10^4\ N;$$

$$w(x_1) = \frac{RA(a)\cdot x_1^3}{6\cdot E\cdot Ix(a)} - \frac{RC\cdot(x_1-925)^3}{6\cdot E\cdot Ix(a)} - \frac{RB\cdot(x_1-3625)^3}{6\cdot E\cdot Ix(a)};$$

$$w_1 = \frac{RA(a)\cdot 2700^3}{6\cdot E\cdot Ix(a_1)} = \frac{1,52\cdot 10^4\cdot 2700^3}{6\cdot 2\cdot 10^5\cdot 2,736\cdot 10^3} = 9,112\cdot 10^{-3}\ m;$$

$$w_2 = \frac{RA(a)\cdot 3625^3}{6\cdot E\cdot Ix(a_1)} - \frac{RC\cdot(3625-925)^3}{6\cdot E\cdot Ix(a_1)} =$$

$$= \frac{1,52\cdot 10^4\cdot 3625^3}{6\cdot 2\cdot 10^5\cdot 2,736\cdot 10^3} - \frac{9,807\cdot 10^3\cdot(3625-925)^3}{6\cdot 2\cdot 10^5\cdot 2,736\cdot 10^3} = 0,016\ m.$$

According to the results of the calculations, we can see that the deflection of the beams of longitudinal movement exceeds the maximum permissible. In this case, it is necessary to increase the stiffness of the beams by thickening the sheet from which they are made. Let'’s accept:

$$a_1 = 10\ mm.$$

As a result, we get:

position 1:

$$a = 2375\ mm;$$

$$RA(a) = \frac{RC\cdot a + RB\cdot(a+2700)}{6000} =$$

$$= \frac{9,807\cdot 10^3\cdot 2375 + 9,807\cdot 10^3\cdot(2375+2700)}{6000} = 1,218\cdot 10^4\ N;$$

$$w(x_1) = \frac{RA(a)\cdot x_1^3}{6\cdot E\cdot Ix(a_1)} - \frac{RC\cdot(x_1-925)^3}{6\cdot E\cdot Ix(a_1)} - \frac{RB\cdot(x_1-3625)^3}{6\cdot E\cdot Ix(a_1)};$$

$$w_1 = \frac{RA(a)\cdot 925^3}{6\cdot E\cdot Ix(a_1)} = \frac{1,218\cdot 10^4\cdot 925^3}{6\cdot 2\cdot 10^5\cdot 1,066\cdot 10^{11}} = 0,00008\ m;$$

$$w_2 = \frac{RA(a)\cdot 3625^3}{6\cdot E\cdot Ix(a_1)} - \frac{RC\cdot(3625-925)^3}{6\cdot E\cdot Ix(a_1)} =$$

$$= \frac{1,218\cdot 10^4\cdot 3625^3}{6\cdot 2\cdot 10^5\cdot 1,066\cdot 10^{11}} - \frac{9,807\cdot 10^3\cdot(3625-925)^3}{6\cdot 2\cdot 10^5\cdot 1,066\cdot 10^{11}} = 3,024\cdot 10^{-3}\ m;$$

position 2:

$$a = 3300\ mm;$$

$$RA(a) = \frac{RC\cdot a + RB\cdot(a+2700)}{6000} =$$

$$= \frac{9,807\cdot 10^3\cdot 3300 + 9,807\cdot 10^3\cdot(3300+2700)}{6000} = 1,52\cdot 10^4\ N;$$

$$w(x_1)=\frac{RA(a)\cdot x_1^{\ 3}}{6\cdot E\cdot Ix(a)}-\frac{RC\cdot(x_1-925)^3}{6\cdot E\cdot Ix(a)}-\frac{RB\cdot(x_1-3625)^3}{6\cdot E\cdot Ix(a)};$$

$$w_1=\frac{RA(a)\cdot 2700^3}{6\cdot E\cdot Ix(a_1)}=\frac{1{,}52\cdot 10^4\cdot 2700^3}{6\cdot 2\cdot 10^5\cdot 1{,}066\cdot 10^{11}}=2{,}338\cdot 10^{-3}\ m;$$

$$w_2=\frac{RA(a)\cdot 3625^3}{6\cdot E\cdot Ix(a_1)}-\frac{RC\cdot(3625-925)^3}{6\cdot E\cdot Ix(a_1)}=$$

$$=\frac{1{,}52\cdot 10^4\cdot 3625^3}{6\cdot 2\cdot 10^5\cdot 1{,}066\cdot 10^{11}}-\frac{9{,}807\cdot 10^3\cdot(3625-925)^3}{6\cdot 2\cdot 10^5\cdot 1{,}066\cdot 10^{11}}=4{,}149\cdot 10^{-3}\ m.$$

According to the results of the calculations, we can see that the deflection of the beams of longitudinal movement does not exceed the recommended 5 mm.

2.3. Simulation modeling of the load of overhead crane frame elements in the solidworks

In order to analyze the correctness of the performed strength and stability calculations of the vertical support columns and longitudinal I-Beams of the crane, simulation of the load of solid-state models of the corresponding elements of the crane was carried out in the SolidWorks software product. The results of simulation modeling, presented in the form of a plot of the distribution of equivalent stresses in the material of the longitudinal I-Beam and a plot of the deflections of the I-Beam, are presented in Fig. 9. The maximum stress is 132 MPa at the yield point of the I-Beam material of 206 MPa (Fig. 9, *a*). The maximum deflection of the I-Beam is 1 mm (Fig. 9, *b*).

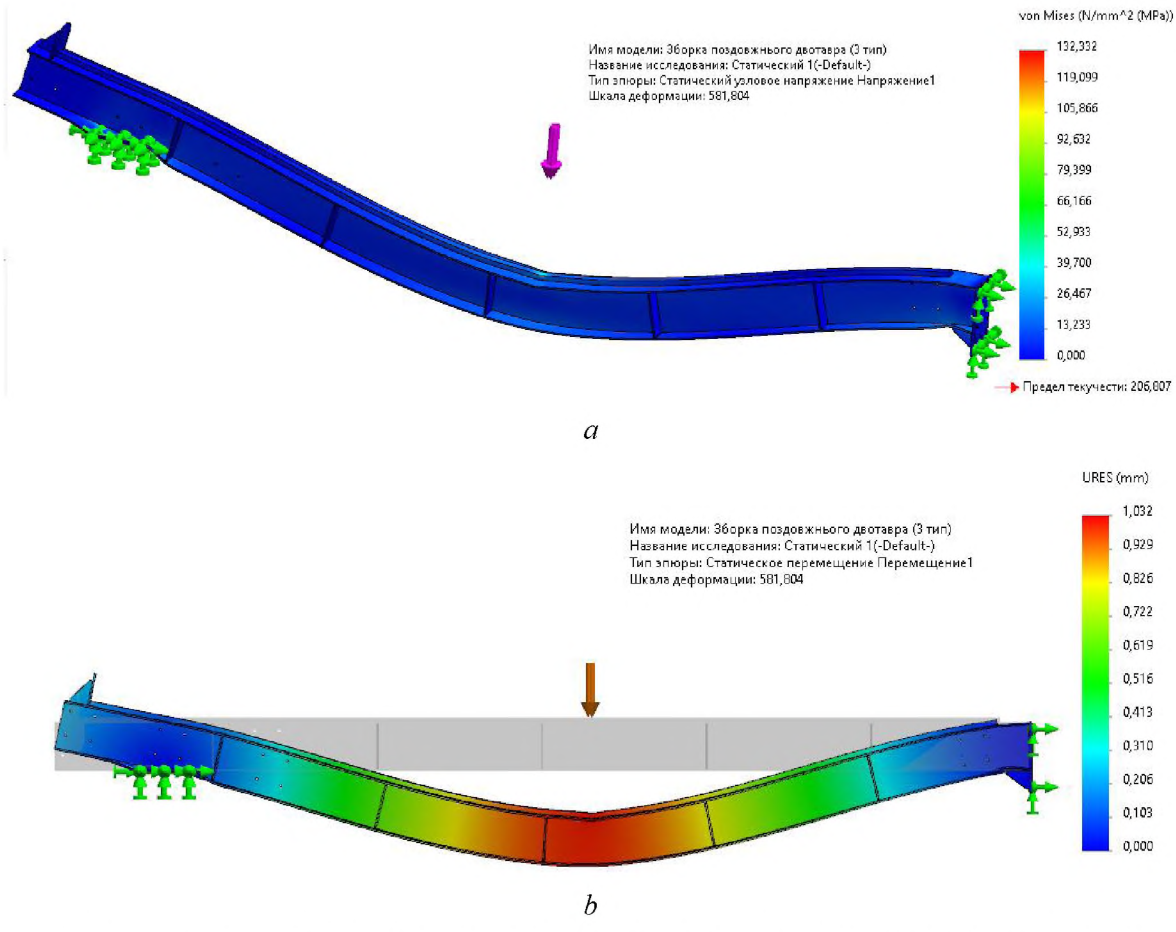

Fig. 9. The results of simulation of the load of the longitudinal I-Beam of the overhead crane obtained in the SolidWorks software product: a – diagram of stress distribution in the I-Beam material; b – diagram of deflections of the I-Beam

Fig. 10 shows the diagram of column compression under the action of vertical (axial) force, which is accompanied by a maximum axial deformation (column compression) of 0.5 mm; compressive stress plot with a maximum value of 28 MPa. Based on the results of column stability modeling, it was established that the stability margin factor is equal to 1.9.

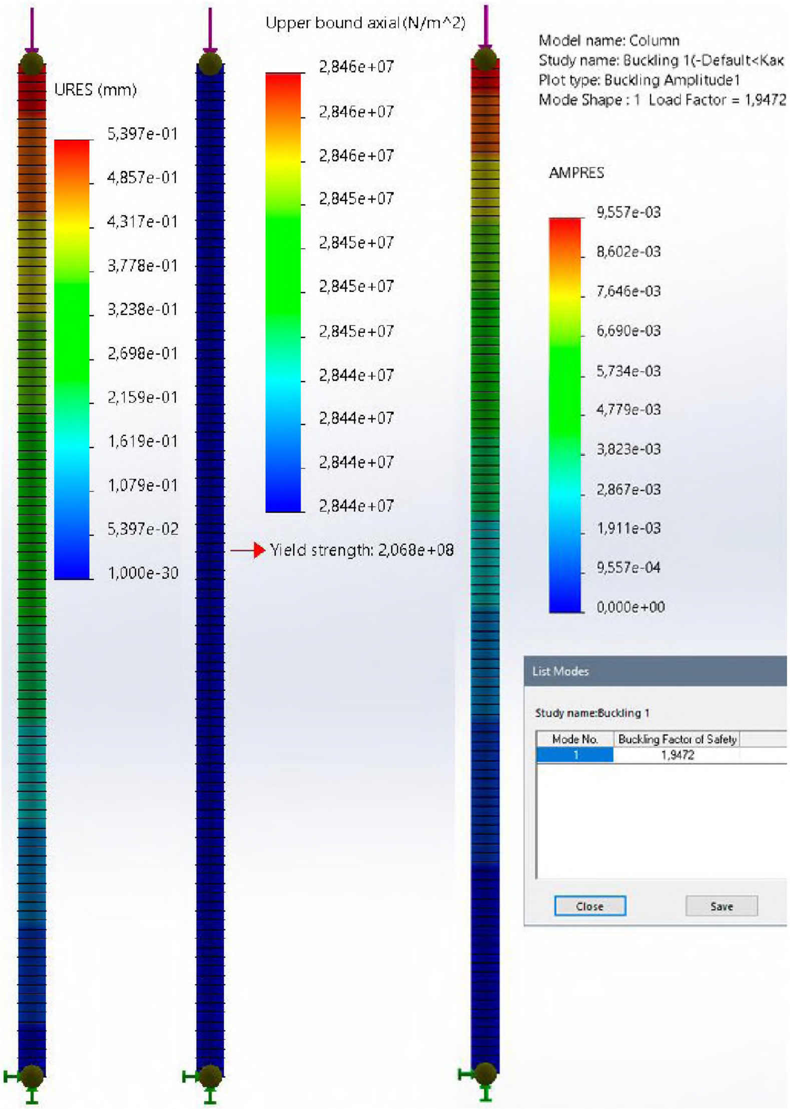

Fig. 10. Results of simulated modeling of the axial (longitudinal) load of the column: left graph – deformation (compression) of the column; middle plot – compressive stress; the right plot is a plot of the resulting amplitude, which reflects the margin of stability of the column

Conclusions

As a result of the analysis, the overhead crane design is provided, in particular the calculation of its frame, columns and beams, taking into account the important factors of strength, stability and stiffness.

On the basis of the conducted informational review, a proper design of the overhead crane, consisting of six vertical columns, is proposed. I-Beams with rails for the longitudinal movement of the carriage are installed on the columns. Rails for the transverse movement of the carriage with winch are placed on the transverse I-Beams of the carriage. Crane columns are mounted on the foundation. To ensure the longitudinal and transverse stability of the metal structure of the crane, its attachment to the load-bearing wall of the building with the help of braces is provided. Transverse truss structures with longitudinal braces are installed between the two longitudinal I-Beams, which also increase the longitudinal and transverse rigidity of the overhead crane structure. To increase the longitudinal stiffness of the left side of the crane (which is not connected to the load-bearing wall), vertical braces are provided in the design.

The design features of individual elements of the stationary frame of the crane are considered: vertical support column, longitudinal I-Beam, transverse truss, etc. The compositional solution of the carriage of longitudinal movement was developed and its functional features were analyzed. The proposed layout of the cargo carriage with winch.

The structure of the crane frame was analyzed, the corresponding calculation scheme was developed, and the static reactions in the supports were calculated. The calculation of the cross-section of the vertical columns was carried out based on the conditions of strength, stiffness and stability. The cross-section of the columns in the form of a square profile pipe 100×100×3 mm was chosen. The calculation of the cross-section of the longitudinal beams from the conditions of strength and stiffness was performed.

Simulation modeling of the load of overhead crane frame elements was carried out in the SolidWorks software product, and their strength, stiffness and stability were substantiated.

With the help of mathematical models and calculations, we managed to determine the optimal parameters and characteristics of each component part of the crane, ensuring a high level of safety and efficiency of its operation. The obtained results can be used for further research in the field of mechanical engineering and design of industrial mechanisms.

Further development prospects are to improve the design of the crane, taking into account new technologies and materials, which will allow to increase its load, increase productivity and reduce energy consumption.

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Please, place here the title and number of the project, in terms of which the presented results were obtained and Acknowledgements to the organizations and/or people contributing to or sponsoring the project (at the discretion of the author).

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