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Optimization of Design and Power Characteristics of Hydraulically-Driven Three-Section Loader Cranes

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Abstract

The research objective consists in working out an optimization mathematic model and the method of two-objective selection of the basic design and power characteristics of hydraulically-driven three-section loader cranes aimed at increasing their performance properties and competitiveness. Based on the optimization calculations and comparing their results with the characteristics of the existing three-section loader-crane the authors determined the feasibility of using the optimization approaches to defining basic structural dimensions of the hydraulic drive kinematics and parameters at the pre-project stage of designing the mobile machine manipulation system. The pre-project optimization provides a comprehensive definition of the best configuration of a large number of basic design parameters – the characteristic structural dimensions of the manipulation system metal structure (lenghts and overall dimensions of the sections cross-sectional area, linkage dimensions for hydraulic drive) and the hydraulic drive characteristics (operating pressure and hydraulic fluid consumption). This approach makes it possible to initially set the most important parameters of the structure to be further designed and to obtain the highest values of the technical characteristics of the designed loader crane.

Keywords: Mobile machine; Loader crane; Two-objective optimization; Weight of metal structure; Pumping unit power

1. Introduction

Hydraulic loader cranes installed in mobile transportation and technological machines of various application due to their versatile character became wide-spread for a substantial range of basic and auxiliary technological operations including weight handling, pick-and-place and storing operations in different spheres of economics, such as industrial production, construction, gas and oil recovery, freight operations, forestry, agriculture etc. [1-3]. Same countries as South Korea, Japan, China, Germany, Italy, the USA, Austria, Russia and others are widely represented in the world market of loader cranes [2, 4].

Due to the considerable diversity of the performed technological operations, the kinematic schemes of loader cranes are also diverse. The crane carrying iron may consist by 3 to 12 series of movable sections which pairwise form lower kinematic pairs of the V-th class: rotating and rectilinear pairs [2, 5]. Rotating pairs are based on the hinge joints and provide the rotational movement along the section longitudinal axis or the rotational relative movement of the adjoining sections. Rectilinear pairs are based on prismatic joints and provide the telescoping of a series of kinematic chain elements. Regardless of the number of elements in the kinematic chain, the special movement of the load handling device and the payload itself correspond to one type of the coordinates system - polar and spherical coordinates system [12]. Therefore, the simplest kinematic scheme consisting of three pivotally connected elements is the minimum requirement for obtaining the spherical configuration of the loader crane functional area. Inclusion of the additional elements into the kinematic scheme is necessary only for increasing the limiting dimension of the spherical functional area.

The three-section loader cranes are currently usefully employed as all-purpose lifting machines in conditions when for the technological operations of load handling a comparatively small functional zone with the radius of from 3 to 5 meters is necessary. Due to their structural simplicity and the minimum number of hydraulic motors which gear the crane sections they have high reliability targets and they are rather cost-efficient [5, 11]. This kinematic scheme is characteristic of the manipulation systems of such mobile machines as Tadano TM-20 (South Korea) [13], Barco 295ML (Belgium) [3], AST-4-A (Russia) [11] and others (Fig. 1).

At present the design optimization for the hydraulically driven loader cranes including the three-section ones is a promising area for increasing the effectiveness of this type of handling machines and an efficient tool for developing the cargo capacity reserves for the already known constructions [3, 5].

For the majority of structural elements of the hydraulically driven loader cranes in mobile machines several (two and more) quality parameters appear significant. These parameters as a rule express mass and energy characteristics of the constructions and systems to be optimized [6]. The computation results [7, 8] show that the single-optimization of mobile machines manipulation systems construction with the same design intent leads to inconsistent optimal values of the required parameters. Therefore, the task of optimal design

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must be set and solved as a task of a multi-objective optimization with reference to all the relevant quality parameters and the degree of their priority for the reliable and energy-efficient functioning of loader cranes.

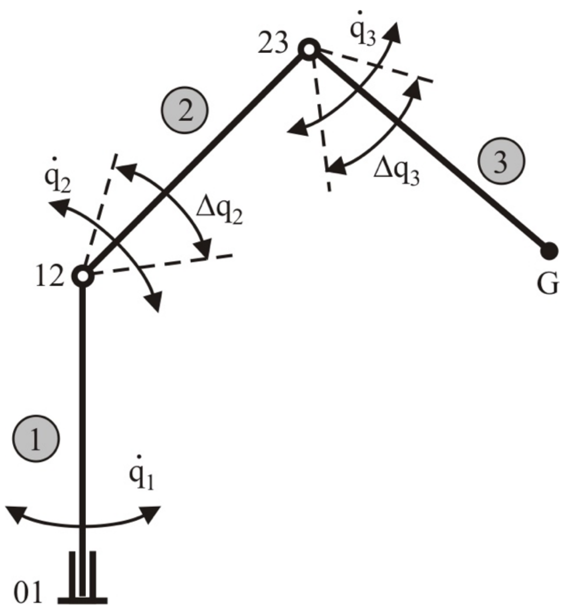

Fig. 1. Kinematic scheme of the three-section loader crane based on three rotational kinematic pairs of the V-th class [3, 11]: 1 – column; 2 – boom; 3 – arm; 01, 12, 23 – hinge joints linking sections 0 and 1, 1 and 2, 2 and 3 correspondingly

In the present study in order to construct the function of the multi-objective optimization of loader cranes construction and power characteristics, the previously developed [8] additive function was implemented

$$F(\{x\},\{z\}) = \left\langle \sum_k w_k \left\{ 1 - \left[\frac{F_k(\{x\}_k^{opt},\{z\}_k^{opt})}{F_k(\{x\},\{z\})} \right]^{s_{extr,k}} \right\}^2 \right\rangle^{0,5} \to \min, \quad (1)$$

where w_k represents weighting factors characterizing the degree of importance of the k-th quality parameter from a design perspective ($\sum w_k = 1$); $F_k(\{x\},\{z\})$ represents single-objective optimization function in terms of the k-th quality parameter; $F_k(\{x\}_k^{opt},\{z\}_k^{opt})$ represents objective function value in the optimal point in a single-objective optimization in terms of the k-th quality parameter; $s_{extr,k}$ represents extremality indicator of the k-th quality parameter (in search for the minimum $s_{extr,k}$ =1, in search for the maximum $s_{extr,k}$ = -1). As required by the multi-objective optimization theory [9, 10], several summands in equation (1) are non-dimensional and standardized variables, the values of which in the feasible space of the sought vectors $\{x\}$ belong to the interval $[0;1)$. The objective function (1) expresses the measure of proximity of the optimum point trend in the variable parameters spacing $\{x\}$ to the spacing of optimum points of all single-objective functions F_k taken into account in the spacing of the same parameters. Therefore, the optimal vector $\{x\}^{opt}$ expresses such spacing of the objective function optimum point (1), which is characterized by the minimal sum of distances to the objective functions F_k optimum points. In other words, the parameters of optimal vectors $\{x\}^{opt}$ and $\{z\}^{opt}$ characterize such a construction to be designed which to the maximum extent reflects the quality parameters of this construction taken into account and in the best way possible provides their balance.

The computational efficiency of checking a number of quality parameters in a multi-objective optimization in the form of the objective function (1) is conditioned by the fact that in this case a number of monotypic computations are required for several different combinations of weight coefficients when the algorithm is unchanged. It has a significant importance for designing versatile software intended for automatization of designing the mobile transportation and technological machines.

The search for the optimum alte th of lifting cranes within the framework of Hamiltonian mechanics are also known [19, 20], though they have not come into widespread acceptance due to the necessity of applying the special mathematical apparatus. To analyze the kinematics and dynamics of loader cranes, the research methods of robotic manipulators are also applicable. These methods are based on the matrix analysis [12, 14, 15, 16, 21], fuzzy logic [22], spatial operators [23], Li Group [24], robots operating area [25] and others. The magnitude of forces and moments of forces in the characteristic sections obtained by means of the dynamic analysis is the basis for calculating the stress-strain state of the loader cranes metal structure. As a rule, simplified dependences obtained within the framework of Mechanics of Materials are used in the algorithm of the optimal construction design [26]. After the termination of the optimization process numerical methods are used, for example the Finite Element Method in order to get the ultimate assessment of strength and stiffness of the loader crane optimum alternative and to update the stress patterns in the stress concentration zones [16, 27].

2. Numerical Scheme

As a rule, the optimization of the loader crane is carried out at the stage of the full-scale engineering development. However, it is feasible to use the optimization approaches at the earlier pre-project stage – at the stage of developing the technical specification for the manipulation system of the mobile transportation and technological machine. At this stage, proceeding from the technical specification, the limited set of structural requirements was obtained. These requirements include the type of the loader crane kinematic scheme, characteristic dimensions of the operating area (minimal $R_{\min}$ and maximal $R_{\max}$ outreach of the boom, upper $Y_{\max}$ and lower $Y_{\min}$ limit positions of the load-handling device), rated lifting capacity for the maximum outreach G_n, the sections conveying speed etc. This approach allows to factor the optimum values of the most significant quantitative characteristics in the construction to be designed. These characteristics determine the realization of the highest values of the quality parameters of the designed loader crane.

2.1. Modeling the construction scheme of the loader crane

The operating area of the three-section loader crane (Fig. 1) is expressed by the spherical polar coordinate system. Fig. 2

features the vertical plane section of the operating area. The lifting boom configuration corresponding to the lower limit position of the characteristic point G (the attachment point of the load-handling device) is characterized by position I, and the configuration corresponding to the upper limit position – by position II. The plane figure bounded by the closed curve $abcda$, determines the geometric locus of the G point in any possible combination of mutual arrangements of the lifting boom adjacent sections 2 and 3 when the loader crane is at work. Curves ab and cd are circular arcs with the radius L_3 centered at joint 23, curve bc is a circular arc with the radius

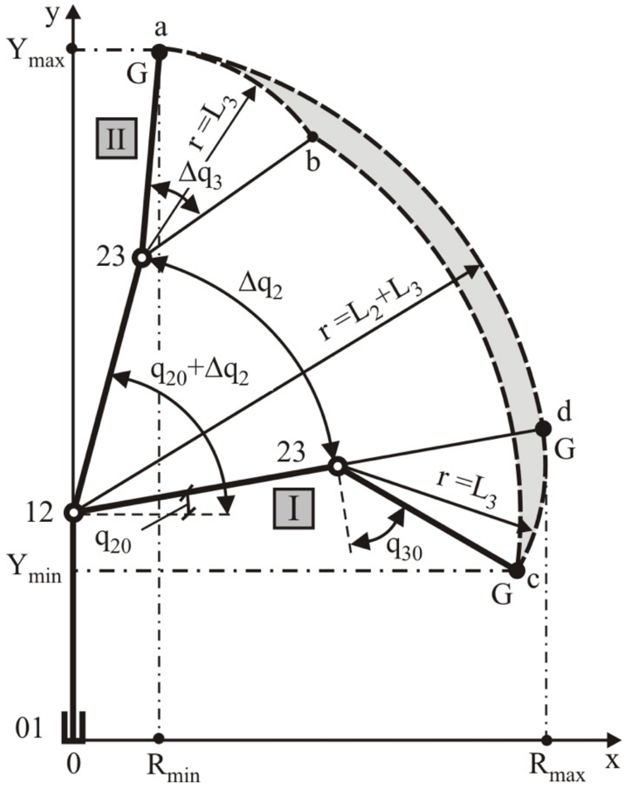

Fig. 2. The operating area of the three-section loader crane in Cartesian coordinates: I - the manipulator lower limit position; II - the manipulator upper limit position

$$r_{bc} = \sqrt{L_2^2 + L_3^2 + L_2 L_3 \sin q_{30}} \tag{2}$$

centered at joint 12, curve adc is a circular arc of the radius

$$r_{abc} = L_2 + L_3 \tag{3}$$

centered at joint 12. The maximum outreach of the boom characteristic G point is

$$R_{max} = (L_2 + L_3)\cos q_{20}, \tag{4}$$

minimum –

$$R_{min} = L_2 \cos(q_{20} + \Delta q_2) + \\ + L_3 \sin(q_{20} + q_{30} + \Delta q_2 + \Delta q_3). \tag{5}$$

The upper limit position of the G point in relation to the boom horizontal plane (of joint 01) is

$$Y_{max} = L_1 + L_2 \sin(q_{20} + \Delta q_2) - \\ - L_3 \cos(q_{20} + q_{30} + \Delta q_2 + \Delta q_3) \tag{6}$$

the lower limit position –

$$Y_{min} = L_1 + L_2 \sin q_{20} - L_3 \cos(q_{20} + q_{30}). \tag{7}$$

The analysis of the provided geometric correlations shows that the characteristic dimensions of the operation area stated in the technical specification of the loader crane ($R_{max}, R_{min}, Y_{max}, Y_{min}$) can be obtained in case of different combinations of 7 design values – section lengths L_1, L_2, L_3, angular coordinates of the sections starting positions q_{20}, q_{30} and the maximum turning angles $\Delta q_2, \Delta q_3$.

In the process of designing the three-section loader cranes 4 variants for arranging the crane metal structure depending on the arrangement of hydraulic cylinders power mechanisms of the sections movement are used (Fig. 3). The rotation of the crane column 1 is exercised by the pivoted hydraulic engine in relation to joint 01 having the vertical axis of rotation. The boom swing of crane 2 is exercised by the the hydraulic cylinder hc1 in relation to joint 12, having the horizontal axis of rotation. The rotation of the arm of crane 3 is exercised by the hydraulic cylinder hc2 in relation to joint 23, having the horizontal axis of rotation. In the process of installing the power hydraulic cylinders outside the operating area in the construction of sections 2 and 3, lever 4 is included (sections 2a and 3a are featured in Fig. 3 correspondingly).

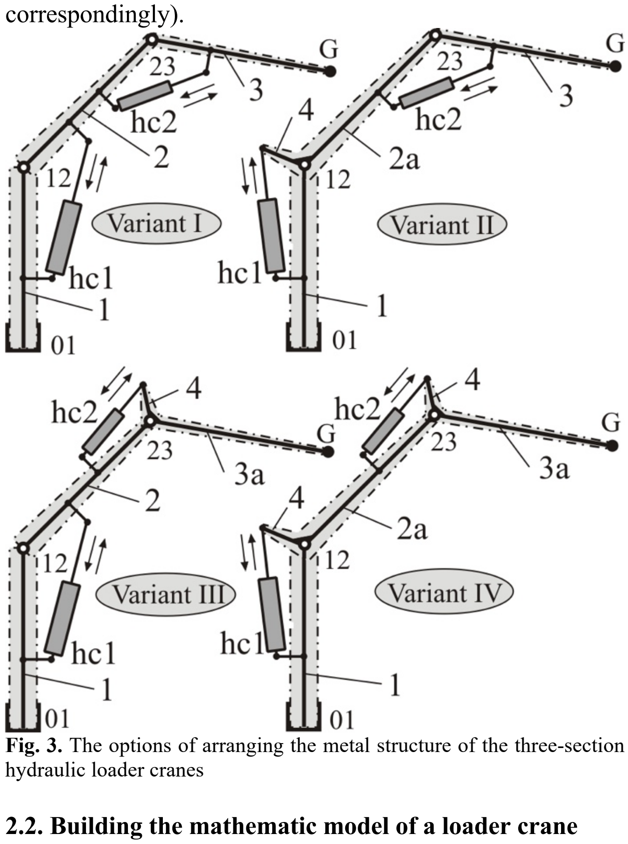

Fig. 3. The options of arranging the metal structure of the three-section hydraulic loader cranes

2.2. Building the mathematic model of a loader crane

As the sequence of actions accomplished in the process of building the mathematic model and the optimization process for any of the four options of the loader crane metal structure arrangement (Fig. 3), is the same, it will be further exemplified by variant III. It corresponds to the kinematic scheme of AST-4-A mobile machine for welding of long distance gas and oil pipelines [11]. Fig. 4 features the computation scheme of the crane according to variant III with the indication of all characteristic structural dimensions used in the process of building the mathematic model.

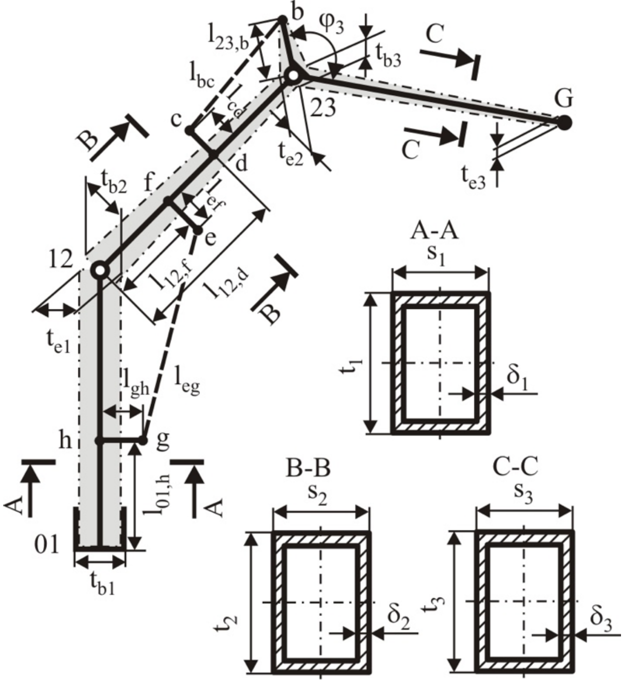

Fig. 4. Computational scheme of the three-section hydraulically driven loader crane (variant III)

Changes in the process of search for the optimal solution of the attachment dimensions determining the fixturing points of hydraulic cylinders to the sections metal structures make it possible to control:

- the magnitude of internal forces in the characteristic sections (it influences the cross-section dimensions and the loader crane mass M_{ms});
- the required traction efforts and linear speeds of rods movement (it influences the operating pressure p_{nom}, hydraulic fluid consumption Q_{nom} and the hydraulic drive power N_{pu}).

The analysis of the three-section hydraulic loader crane composition shows (Fig. 4) that the metal structure mass and the hydraulic pump assembly power are determined by the combination of the following parameters: section lengths (L_1, L_2, L_3), the hydraulic cylinders attachment dimensions ($l_{01,h}$, $l_{12,f}$, $l_{12,d}$, l_{gh}, l_{ef}, l_{cd}), cross sections overall dimensions (height t_{b1}, t_{e1}, t_h, t_{b2}, t_{e2}, t_f, t_d, t_{b3}, t_{e3}, t_b; width s_1, s_2, s_3), widths of sections walls (δ_1, δ_2, δ_3), angle of lever (ϕ_3), sections motion ($\Delta q_2, \Delta q_3$) and sections starting position (q_{20}, q_{30}). In total – 31 variable values for each variant.

The quality parameter of the loader crane metal structure mass M_{ms} is determined by the sum of separate sections masses with reference to their structural design (with the control lever or without it) and power hydraulic cylinders. For variant III under consideration it is determined by the following dependence:

$$
\begin{aligned}
M_{ms,III} = {} & M_{s1} + M_{s2} + M_{s3} + M_{hc1} + M_{hc2} = \\
& = \rho L_1 \delta_1 (t_{b1} + t_{e1} + 2s_1 - 4\delta_1) + \\
& + \rho \delta_2 \{ s_2 [L_2 + t_{b2} + t_{e2} + \sqrt{(t_f - t_{b2})^2 + l_{12,f}^2} + \\
& + \sqrt{(t_f - t_d)^2 + (l_{12,d} - l_{12,f})^2} + \\
& + \sqrt{(t_d - t_{e2})^2 + (L_2 - l_{12,d})^2}] + (t_{b2} + t_f) l_{12,f} + \\
& + (t_f + t_d)(l_{12,d} - l_{12,f}) + (t_d + t_{e2})(L_2 - l_{12,d}) \} + \\
& \rho \delta_3 \{ s_3 [L_3 + t_{b3} + t_{e3} + l_{23,b} + \sqrt{(t_{b3} - t_b)^2 + l_{23,b}^2} + \\
& + \sqrt{(t_{b3} - t_{e3})^2 + L_3^2}] + (t_b + t_{b3}) l_{23,b} + (t_{b3} + t_{e3}) L_3 \} + \\
& + \pi \rho k_{hc} \Delta l_{hc1} [(D_{hc1} + 2s_{hc1})^2 - (1 - k_d) D_{hc1}^2] / 4 + \\
& \pi \rho k_{hc} \Delta l_{hc2} [(D_{hc2} + 2s_{hc2})^2 - (1 - k_d) D_{hc2}^2] / 4 ,
\end{aligned} \tag{8}
$$

where M_{si} represents mass of the i-th section; M_{hci} represents mass of the i-th hydraulic cylinder; ρ represents density of section material; k_d represents diameter coefficient of the hydraulic cylinder rod [28]; k_{hc} represents coefficient with reference to the masses of the hydraulic cylinder auxiliary components; Δl_{hci} represents rod travel of the i-th hydraulic cylinder; D_{hci}, s_{hci} are diameter and wall thickness of the i-th hydraulic cylinder, respectively.

The second quality parameter, i.e. the loader crane pumping unit power N_{pu}, is defined as the peak power necessary for sections 2 and 3 admissible cooperative motion. For variant III under consideration it is determined by the following dependence:

$$
N_{pu,III} = 0{,}25\pi\, p_{nom} [D_{hc1}^2 v_{hc1,\max}(\dot{q}_2) + D_{hc2}^2 v_{hc2,\max}(\dot{q}_3)], \tag{9}
$$

where $v_{hci,\max}(\dot{q}_i)$ represents maximal linear speed of the i-th hydraulic cylinder rod providing the necessary angular turning rate of the i-th section $\dot{q}_i$. The analysis of equation (9) shows the pumping unit power is determined by the combination of three parameters - $p_{nom}, D_{hc1}, D_{hc2}$.

2.3. Setting the problem of a two-objective optimization of a loader crane

Parameters belonging to dependences (8) and (9) for calculating the quality parameters M_{msi} and N_{pu} can be viewed as controlled parameters of the loader crane metal structure optimization. Vector $\{x\}$ to be further optimized must be formed on the basis of these parameters. For variant III of the metal structure composition it will be written as:

$$
\begin{aligned}
\{x\}_{III}^{\blacklozenge} = \{ & L_1\ L_2\ L_3\ q_{20}\ q_{30}\ \Delta q_2\ \Delta q_3\ p_{nom}\ D_{hc1}\ D_{hc2} \\
& s_1\ s_2\ s_3\ \delta_1\ \delta_2\ \delta_3\ l_{01,h}\ l_{gh}\ t_{b1}\ t_{e1}\ t_{b2}\ t_{e2}\ t_{b3}\ t_{e3}\ l_{12,f} \\
& l_{ef}\ l_{12,d}\ l_{cd}\ l_{23,b}\ t_h\ t_f\ t_d\ \phi_3\ t_b \} = \{x_1\ x_2\ x_3 \ldots x_{34}\} .
\end{aligned} \tag{10}
$$

To construct the function of a two-objective optimization the authors employed the additive function (1) developed in [8] which is adjusted to:

$$F(\{x\}_{III},\{z\}_{III}) = \left\{ w_1 \left[1 - \frac{F_{M,III}(\{x\}_{M,III}^{opt},\{z\}_{M,III}^{opt})}{F_{M,III}(\{x\}_{III},\{z\}_{III})} \right]^2 + \right.$$

$$\left. + w_2 \left[1 - \frac{F_{N,III}(\{x\}_{N,III}^{opt},\{z\}_{N,III}^{opt})}{F_{N,III}(\{x\}_{III},\{z\}_{III})} \right]^2 \right\}^{0,5} \to \min \,, \qquad (11)$$

In equation (11) the following conventional signs are employed: $F_{M,III}, F_{N,III}$ stand for the function of single-objective optimization of the quality parameters M_{ms} and N_{pu} for variant III; $\{x\}_{M,III}^{opt}$ $(\{x\}_{N,III}^{opt})$, $\{z\}_{N,III}^{opt}$ $(\{z\}_{N,III}^{opt})$ stand for the vectors of controlled and uncontrolled parameters in the quality parameter optimum point M_{ms} (N_{pu}) in the process of single-objective optimization; $F_{M,III}(\{x\}_{M,III}^{opt},\{z\}_{M,III}^{opt})$, $F_{N,III}(\{x\}_{N,III}^{opt},\{z\}_{N,III}^{opt})$ stand for the objective functions values in the optimum point in the process of single-objective optimization.

When the objective function (11) is employed, the two-objective optimal design of the loader crane metal structure undergoes two distinct stages.

Stage 1: the single-objective optimization of functions $F_{k,III}$ for all k with reference to all the quality parameters and finding the values of vectors for controlled $\{x\}_{k,III}^{opt}$ and uncontrolled $\{z\}_{k,III}^{opt}$ parameters in the optimum point of the k-th objective function $F_{k,III}$, as well as the objective functions values in the optimum point $F_{k,III}(\{x\}_{k,III}^{opt},\{z\}_{k,III}^{opt})$.

Stage 2: two-objective optimization of the objective function (11), with finding the optimum vectors $\{x\}_{III}^{opt}$ and $\{z\}_{III}^{opt}$.

The parameters of the optimum vectors $\{x\}_{III}^{opt}$ and $\{z\}_{III}^{opt}$ characterize such a structure of the loader crane being designed which reflects the quality parameters M_{msi} and N_{pu} to the maximum extent and most accurately provides their combination.

Finding the objective function minimum (11), as well as vectors $\{x\}_{III}^{opt}$ and $\{z\}_{III}^{opt}$ in the optimum point must be accomplished with due account of the set of constraints in terms of inequalities, expressing the conditions for arranging sections and hydraulic cylinders, the admissible combination of the cross section overall dimensions, the required dimensions of the crane operating area, the selection of the manufactured pumps and cylinders dimension types, the required traction effort, the hydraulic cylinders power and running smoothness, the strength of the loader crane metal structure characteristic sections and a number of others.

The particular combination of the conditions mentioned above as well as their mathematic expression are determined by the specific option of the metal structure composition of the three-section hydraulic loader crane (Fig. 3). For variant III under consideration the constraints set for the optimization objective includes 99 constraints in terms of inequalities and is written as follows:

- general conditions of sections arrangement:

$$\begin{gathered}
L_1 - l_{01,h} \ge 0\,; \quad L_2 - l_{12,d} \ge 0\,; \quad l_{12,d} - l_{12,f} \ge 0\,; \\
L_3/3 - l_{23,b} \ge 0\,; \quad l_{23,b} - 0{,}7t_{b3} \ge 0\,; \quad t_{b3} - t_b \ge 0\,; \\
t_b - t_{e3}/3 \ge 0\,; \quad l_{ef} - 0{,}7t_f \ge 0\,; \quad l_{cd} - 0{,}7t_d \ge 0\,; \\
l_{gh} - 0{,}7t_h \ge 0\,; \quad t_f - t_{b2} - (t_{e2} - t_{b2})l_{12,f}/L_2 \ge 0\,; \\
t_h - t_{b1} - (t_{e1} - t_{b1})l_{01,h}/L_1 \ge 0\,; \quad t_{e1} - t_{b2} \ge 0\,; \\
t_d - t_{b2} - (t_{e2} - t_{b2})l_{12,d}/L_2 \ge 0\,; \quad t_{e2} - t_{b3} \ge 0\,; \\
t_{b1} - t_{e1} \ge 0\,; \quad t_{b2} - t_{e2} \ge 0\,; \quad t_{b3} - t_{e3} \ge 0\,; \\
\pi/6 - |q_{20}| \ge 0\,; \quad \pi/6 - |q_{30}| \ge 0\,; \\
\pi/2 - q_{20} - \Delta q_2 \ge 0\,; \quad \pi/2 - q_{30} - \Delta q_3 \ge 0\,; \\
\phi_3 - \pi/4 \ge 0\,; \quad 5\pi/6 - \phi_3 \ge 0\,,
\end{gathered} \qquad (12)$$

- conditions for hydraulic cylinders sections arrangement:

$$\Big\langle l_{12,e}^2 + l_{12,g}^2 + 2l_{12,e}l_{12,g}\sin\{q_{20} - arctg(l_{ef}/l_{12,f}) -$$

$$- arctg[l_{gh}/(L_1 - l_{01,h})]\}\Big\rangle^{0,5} - \Delta l_{hc1} \ge 0\,; \qquad (13)$$

$$\Big\langle l_{23,b}^2 + l_{23,c}^2 + 2l_{23,b}l_{23,c}\sin\{q_{30} + \Delta q_3 + \phi_3 +$$

$$+ arctg[l_{cd}/(L_2 - l_{12,d})]\} + \Big\rangle^{0,5} - \Delta l_{hc2} \ge 0\,, \qquad (14)$$

- the admissible combination of cross sections overall dimensions (as exemplified by section 2):

$$\begin{gathered}
s_2/t_{b2} - \xi_s^{\min} \ge 0\,; \quad \xi_s^{\max} - s_2/t_{b2} \ge 0\,; \\
s_2/t_{e2} - \xi_s^{\min} \ge 0\,; \quad \xi_s^{\max} - s_2/t_{e2} \ge 0\,; \\
s_2/t_f - \xi_s^{\min} \ge 0\,; \quad \xi_s^{\max} - s_2/t_f \ge 0\,; \\
s_2/t_d - \xi_s^{\min} \ge 0\,; \quad \xi_s^{\max} - s_2/t_d \ge 0\,; \\
s_2/s_1 - \zeta_s^{\min}\,; \quad \zeta_s^{\max} - s_2/s_3\,; \quad s_3/s_2 - \zeta_s^{\min}\,; \\
s_2 - 3\delta_2 \ge 0\,; \quad \psi_{st} - t_f/t_{b2} \ge 0\,; \\
\psi_{st} - t_f/t_{e2} \ge 0\,; \quad \psi_{st} - t_d/t_{b2} \ge 0\,; \\
\psi_{st} - t_d/t_{e2} \ge 0\,; \quad \delta_2 - \delta_{\min} \ge 0\,,
\end{gathered} \qquad (15)$$

- conditions for the required dimensions of the manipulation system operating area:

$$(L_2 + L_3)\cos q_{20} - R_{\max} \ge 0\,; \qquad (16)$$

$$Y_{\min} - L_1 - L_2\sin q_{20} + L_3\cos(q_{20} + q_{30}) \ge 0\,; \qquad (17)$$

$$R_{\min} - L_2\cos(q_{20} + \Delta q_2) -$$

$$-L_3 \sin(q_{20} + q_{30} + \Delta q_2 + \Delta q_3) \geq 0 \quad (18)$$

$$L_1 + L_2 \sin(q_{20} + \Delta q_2) -$$
$$-L_3 \cos(q_{20} + q_{30} + \Delta q_2 + \Delta q_3) - Y_{\max} \geq 0 \quad (19)$$

- the possibility of adjusting the commercial positive displacement pump of the hydraulic drive:

$$p_{nom} - p_{p,\min} \geq 0\,;\ \ p_{p,\max} - p_{nom} \geq 0\,; \quad (20)$$

$$Q_{p,\max} - \pi(D_{hc1}^2 v_{hc1,\max} + D_{hc2}^2 v_{hc2,\max})/4 \geq 0\,, \quad (21)$$

- the conditions for the required traction effort, power, running smoothness and the structural strength of the power hydraulic cylinders:

$$\pi D_{hci}^2 p_{nom}/4 - U_{hci}^{\max} \geq 0\,; \quad D_{hci} - \Delta l_{hci}/18 \geq 0\,; \quad (22)$$

$$\pi\, p_{nom} D_{hc2}^2 v_{hc2,\max}/4 - L_3(G_n + gM_{s3})\dot{q}_3 \geq 0\,; \quad (23)$$

$$\pi\, p_{nom} D_{hc1}^2 v_{hc1,\max}/4 - [L_2(G_n + gM_{s3} + gM_{s2}/2) +$$
$$+ L_3(G_n + gM_{s3}/2)]\dot{q}_2 \geq 0\,; \quad (24)$$

$$\delta_{hc1} - \delta_{hc,\min} \geq 0\,; \quad \delta_{hc2} - \delta_{hc,\min} \geq 0\,,$$
(25)

- the possibility of adjusting the commercial hydraulic cylinder:

$$p_{hc,\max} - p_{nom} \geq 0\,; \quad D_{hc,\max} - D_{hci} \geq 0\,;$$
$$\Delta L_{hc,\max} - \Delta l_{hci} \geq 0\,; \quad V_{hc,\max} - v_{hci,\max}(\dot{q}_i) \geq 0\,; \quad (26)$$
$$U_{hc,\max} - U_{hci}^{\max} \geq 0\,,$$

- the conditions for the bending strength of the manipulation system characteristic sections:

$$[\sigma] - 6t_{b1} M_{01}^{\max}/[s_1 t_{b1}^3 - (s_1 - 2\delta_1)(t_{b1} - 2\delta_1)^3] \geq 0\,; \quad (27)$$

$$[\sigma] - 6t_h M_h^{\max}/[s_1 t_h^3 - (s_1 - 2\delta_1)(t_h - 2\delta_1)^3] \geq 0\,; \quad (28)$$

$$[\sigma] - 6t_d M_d^{\max}/[s_2 t_d^3 - (s_2 - 2\delta_2)(t_d - 2\delta_2)^3] \geq 0\,; \quad (29)$$

$$[\sigma] - 6t_f M_f^{\max}/[s_2 t_f^3 - (s_2 - 2\delta_2)(t_f - 2\delta_2)^3] \geq 0\,; \quad (30)$$

$$[\sigma] - 6t_{b3} M_{23}^{\max}/[s_3 t_{b3}^3 - (s_3 - 2\delta_3)(t_{b3} - 2\delta_3)^3] \geq 0\,, \quad (31)$$

- the conditions for the characteristic sections strength when shear forces are at work:

$$[\tau] - 0{,}75\zeta_{f3} G_n/[\delta_3(t_{e3} + s_3 - 2\delta_3)] \geq 0\,; \quad (32)$$

$$[\tau] - 0{,}75 Q_b^{\max}/[\delta_3(t_b + s_3 - 2\delta_3)] \geq 0\,; \quad (33)$$

$$[\tau] - 0{,}75 Q_{23}^{\max}/[\delta_3(t_{e3} + s_3 - 2\delta_3)] \geq 0\,; \quad (34)$$

$$[\tau] - 0{,}75 Q_{23}^{\max}/[\delta_2(t_{e2} + s_2 - 2\delta_2)] \geq 0\,; \quad (35)$$

$$[\tau] - 0{,}75 Q_f^{\max}/[\delta_2(t_f + s_2 - 2\delta_2)] \geq 0\,; \quad (36)$$

$$[\tau] - 0{,}75 Q_{12}^{\max}/[\delta_2(t_{b2} + s_2 - 2\delta_2)] \geq 0\,; \quad (37)$$

$$[\tau] - 0{,}75 Q_{12}^{\max}/[\delta_1(t_{e1} + s_1 - 2\delta_1)] \geq 0\,, \quad (38)$$

- the conditions for characteristic sections in section 1 when the axial force is at work:

$$[\sigma] - 0{,}5N_{01}/[\delta_1(t_{b1} + s_1 - 2\delta_1)] \geq 0\,; \quad (39)$$

$$[\sigma] - 0{,}5N_{12}/[\delta_1(t_{e1} + s_1 - 2\delta_1)] \geq 0\,, \quad (40)$$

- the conditions for section 1 stability under longitudinal compressive loads with the off-center bending:

$$2\phi_e[\sigma]\delta_1(t_{b1} + s_1 - 2\delta_1) - N_{01} \geq 0\,, \quad (41)$$

where $\delta_{\min}$ represents minimum allowable thickness of the section wall; $\zeta_s^{\min}, \zeta_s^{\max}$ represents minimal and maximal width ratio of the cross area of the adjoining sections; $\xi_s^{\min}, \xi_s^{\max}$ represents minimal and maximal width and height ratio of the cross section of the adjoining sections; $[\sigma], [\tau]$ represents admissible normal and shear stresses for the sections material; ϕ_e represents stability coefficient under compression and shear; ψ_{st} represents coefficient of admissible amplification of the section cross area in the hydraulic cylinder attachment point; $D_{hc,\max}$, $\Delta L_{hc,\max}$, $U_{hc,\max}$, $V_{hc,\max}$ represents maximal diameter, stroke, traction force and rod speed of commercial hydraulic cylinders; $p_{p,\min}$ ($p_{p,\max}$), $Q_{p,\max}$ represents minimal (maximal) exit pressure and maximal pump output flow of commercial pumps; $U_{hci}^{\max}$ represents peak pulling capacity developed by the i-th hydraulic cylinder. The maximum values of the internal force factors $M_j^{\max}, Q_j^{\max}, N_j$ in the j-th characteristic sections which we used while calculating the left parts of the above-mentioned constraints can be determined on the basis of dependences obtained in [11] for the three-section loader crane of AST-4-A mobile machine.

The goals of a single-objective optimization of the quality parameters M_{ms} and N_{pu}, the solutions of which are necessary for forming the two-objective optimization function (11) are represented as minimization of the following objective functions based on equations (8) and (9):

$$F_{M,III}(\{x\}_{M,III},\{z\}_{M,III}) \equiv$$
$$\equiv M_{ms,III}(\{x\}_{M,III},\{z\}_{M,III}) \to \min\,; \qquad (42)$$

$$F_{N,III}(\{x\}_{N,III},\{z\}_{N,III}) \equiv$$
$$\equiv N_{pu,III}(\{x\}_{N,III},\{z\}_{N,III}) \to \min \qquad (43)$$

and finding the vectors of controlled parameters $\{x\}^{opt}_{M,III}$ and $\{x\}^{opt}_{N,III}$ in their optimum point. The structure of vectors $\{x\}_{M,III}$, $\{x\}_{N,III}$ and the constraint set used in the process of minimizing the objective functions data coincide with the vector structure for the controlled parameters $\{x\}_{III}$ and the two-objective optimization constraints.

3. Optimization findings and their analysis

The suggested methods of design optimization for the three-section loader cranes was tested applied to optimizing the loader crane of AST-4-A mobile energy transforming machine (Fig. 5) [11]. The rated lifting capacity for the maximum outreach of the crane is G_n=750 kg, maximal outreach of the boom is $R_{\max}$ = 5.8 m. The results of its single- and two-objective optimization are shown in Fig. 6.
rnative of the construction being designed which is characterized by the $\{x\}^{opt}$ vector, in the parameters feasible space characterized by the $\{x\}$ vector, calls for introducing the constraints system of various nature [6, 9]. For hydraulically driven loader cranes the specified constraints must include [5, 17]:

- structural constraints (the admissible combination of overall dimensions of the crane sections cross section area, conditions for the required dimensions of the operating area);
- installation constraints (the conditions for arranging the sections and the hydraulic power cylinders of the sections movement);
- driving gear constraints (the possibility of selecting the commercial volumetric hydraulic pumps and power hydraulic cylinders, conditions for the required traction effort, power supply, running smoothness and the structural strength of the power hydraulic cylinders);
- strength constraints (conditions for bending strength of the characteristic sections when the axial and shear forces are at work);
- deformational constraints (conditions for the sections overall and local stability under longitudinal compressive loads with the off-center bending).

Fig. 5. The loader crane of AST-4-A mobile energy transforming machine

The computation of the strength and deformation constraints is of the greatest complexity as it calls for the dynamic and strength analysis of the loader crane operation [3]. The d'Alambert's principle within the framework of Newtonian mechanics, according to which the active forces are counterbalanced by the inertia and resistance forces, is the most wide-spread method of designing the motion equation for the crane sections and calculating stresses in their characteristic sections [11, 18]. The approaches to the study of dynamics and streng

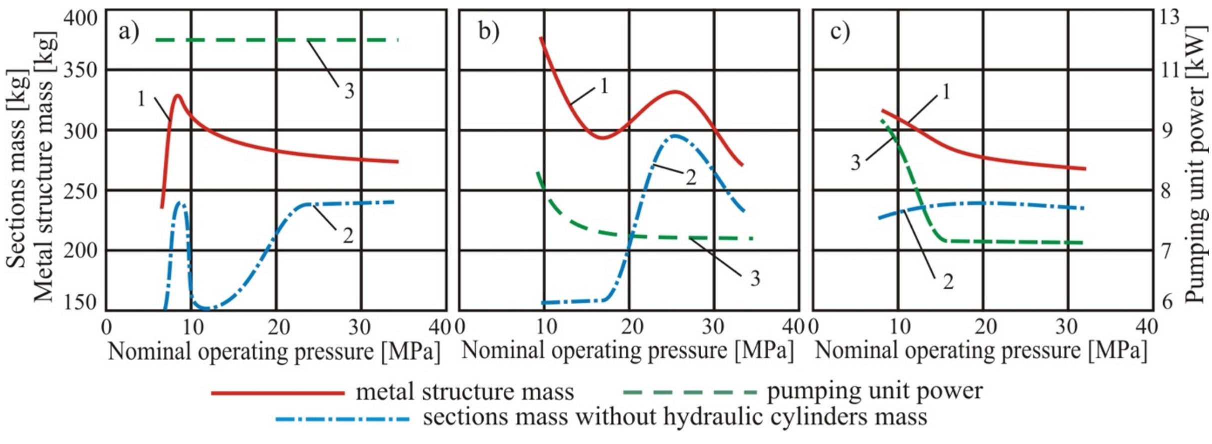

Fig. 6. Dependence of the optimized loader crane from the operating pressure in the hydraulic system: a – metal structure mass optimization; b - the hydraulic drive power optimization; c - multi-objective optimization

The findings obtained show that the single-objective optimization which is effective for minimizing one quality parameter of the loader crane results in substantial degradation of another quality parameter (Fig. 6, a, b). For instance, within the range of the hydraulic system operating pressure of > 16 MPa the single-objective optimization of the pumping unit power provides its minimum value of ~7,1 kW, whereas in the process of the single-objective optimization of the metal structure mass it has the value of ~10,3 kW, i.e 1.45 higher. The same situation is characteristic of the single-objective optimization of the metal structure mass optimization: in the process of single-objective optimization of the pumping unit power the metal structure mass appears 10…20 % higher than its minimum value. Fig. 6 also features the curve of the metal structure sections mass without reference to the hydraulic motors. For low operating pressures in the hydraulic system (p_{nom} < 16 MPa) the hydraulic motors contribution to the metal structure total mass is rather high amounting to 50%. For p_{nom} > 16 MPa the dominating contribution belongs to the sections mass, whereas the hydraulic motors contribution amounts to ~15 %.

Against this background one can easily see the effectiveness of the two-objective optimization which makes it possible to eliminate the main drawback of single-objective approaches. Fig. 6 shows that on the basis of two-objective optimization it is possible to obtain the optimal loader crane construction which will simultaneously provide the required mass of the metal structure M_{ms} and the pumping unit power N_{pu} equal to their corresponding minimum values obtained after the single-objective optimization. Fig. 7 shows that the complex-valued objective function (11) suggested for the two-objective optimization has a favorable structure from the mathematic point of view, as it provides the stability of optimization results within the wide range of interdependent correlation between weight factors w_1 and w_2.

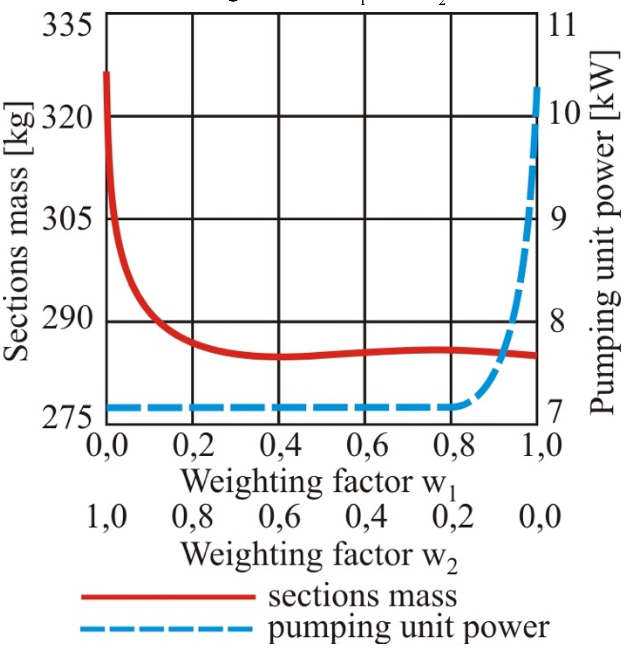

Fig. 7. The influence of weighting factors on the results of multi-objective optimization

The loader crane metal structure dimensions included in the controlled parameters vectors $\{x\}^{opt}_{M,III}$, $\{x\}^{opt}_{N,III}$ and $\{x\}^{opt}_{III}$, are partially unequal which conditions the specific external difference of the optimal constructions. For illustrative purposes Fig. 8 shows (at a scale of 1:75) the appearance of the existing loader crane of AST-4-A mobile machine as compared to its optimal configurations in the process of single- and two-objective optimization. The overall dimensions of the cross sections as well as the hydraulic cylinder diameters appear considerably larger for the existing loader crane than for any of the optimal configurations. Therefore, the crane has a larger mass of its metal structure and larger power of a pumping unit. The sections dimensions in the optimum configurations are rather similar, however, there is a significant discrepancy in the hydraulic cylinders attachment dimensions. As compared to the existing structure of the loader crane in AST-4-A mobile machine, the optimum configuration obtained as a result of a two-objective optimization is characterized by the significantly improved quality parameters: the metal structure mass amounts to 285 kg vs 454 kg of the existing construction, the pumping unit power of the hydraulic drive is 7.1 kW vs. 7.9 kW.

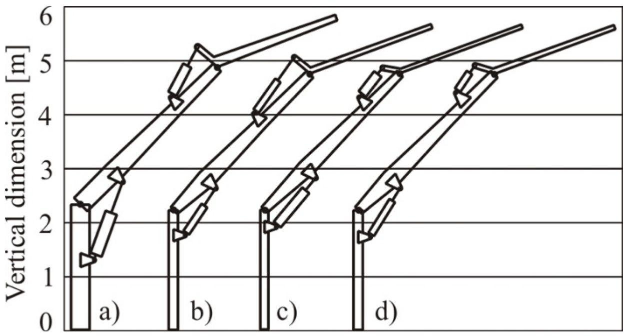

Fig. 8. Optimization results: a - initial construction (Fig. 5); b - metal structure mass optimization; c - hydraulic drive power optimization; d – multi-objective optimization for $w_1 = w_2 = 0.5$

4. Conclusion

The feasibility of implementing optimization approaches to determining the main construction dimensions of the kinematic scheme and the hydraulic drive parameters at the pre-project stage of a loader crane design was ascertained. Under this approach there appears the opportunity to factor the optimum values of the most important qualitative characteristics in the construction to be further designed. These characteristics determine the realization of the highest possible quality parameters in the designed loader crane. For the hydraulically driven three-section loader cranes it is practical to focus on such quality parameters as their own mass and the pumping unit power which must be minimized in the pre-project optimization. In this case the dual goal (technical and economic) is accomplished: low operating and maintenance costs are provided for the designed loader crane due to the drive energy efficiency and the metal structure material intensity. Optimization provides a comprehensive assessment of the optimum combination of a large number of basic design parameters which include the metal structure dimensions (lengths and cross sections dimensions and attachment dimensions for hydraulic motors) and the hydraulic drive characteristics (operating pressure and hydraulic fluid consumption). The suggested approach to optimal designing the loader cranes of mobile transportation and technological machines allows to effectively reveal the reserves of increasing the quality parameters of the designed and operated constructions providing high values of their economical and energy efficiency.

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