NEW STRUCTURE OF THE GANTRY CRANES LEVEL LUFFING JIB SYSTEM

НОВА КОНСТРУКЦИЯ НА СИСТЕМАТА ЗА ИЗМЕНЕНИЕ НА ОБСЕГА НА ПОРТАЛНИТЕ КРАНОВЕ

MSc. (Eng). Bargazov E.[1], Assoc. Prof. PhD. Uzunov T.[1], Prof. PhD. Alipiev O.[1],
Principal assistant. PhD. Antonov S.[1], Assoc. Prof. PhD. Bortyakov D.[2]
University of Ruse, Ruse, Bulgaria[1]
Peter the Great St. Petersburg Polytechnic University, St. Petersburg, Russia[2]
E-mail: ebargazov@uni-ruse.bg, tuzunov@uni-ruse.bg, oalipiev@uni-ruse.bg,
santonov@uni-ruse.bg, bortyakov@ratte.ru

Abstract: *A new construction of the gantry cranes level luffing jib system has been proposed. Its important operating characteristics has been studied. An optimization mathematical model of these structure is built. The parameters, optimization criteria and their limitations are defined. The optimization of construction is done using the Pareto optimization procedure in MATLAB. All parts and assembled units of the gantry cranes level luffing jib system have undergone optimization in practice. The results of the optimization study of the new design are compared with those of the traditional structure.*

KEYWORDS: NEW STRUCTURE, GANTRY CRANES, LEVEL-LUFFING JIB SYSTEM, OPTIMIZATION MATHEMATICAL MODEL, PARETO OPTIMIZATION PROCEDURE, MATLAB.

1. Introduction

Gantry cranes provide a substantial part of the cargo flow of the economy in each country. The moving assemblies belonging to the level-luffing jib system, which determine to a large extent the performance of these cranes, have the largest mobile mass in crane structure. This is sufficient reason to have a effort for improvement and therefore a new design of this system has been created.

The aim of the report is to present the operation parameters research of the new structure of gantry cranes level-luffing jib system. It is made using a new complex of Pareto multi-criteria optimization programs.

2. Preconditions and means for resolving the problem

We have applied a system approach to conducting the research. The principles of the system approach are limited to the following:

1. Purposefulness - Have to define the purpose of the work and to ensure efficiency in the operation of the machine;
2. Relativity - Each element should be considered ***as part of the whole*** in the studies and calculations;
3. Modeling - The possibility to build a model of the machine under study.

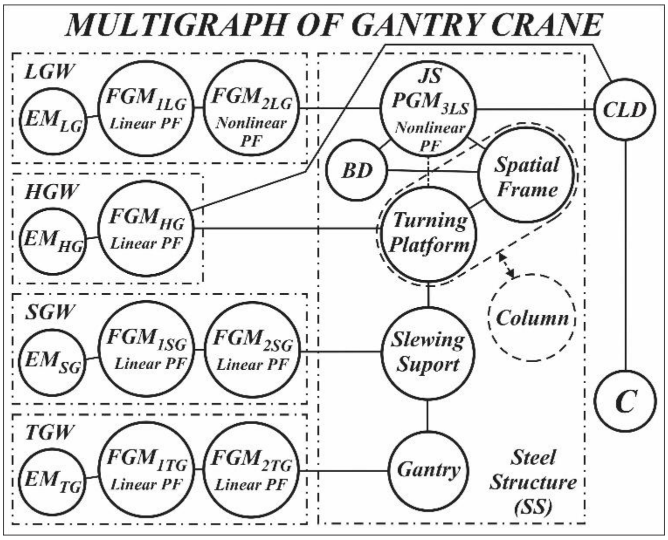

Fig. 1. Multigraph of gantry crane

Structural interaction of all systems and their elements of the gantry crane is indicated in Fig. 1. The structure of the gantry crane is shown in Fig. 2. It is composed of four main systems with their drive winches. The systems and devices favoring the motions at the respective degrees of mobility of the gantry crane are also presented. The gantry crane has four degrees of mobility. They are determine by the four drive winches operation. The gantry crane luffing gear winches in two variants are shown in Fig. 3. The first is of crane ZUBR in Fig. 3 a) and the second is of crane KIROVEC in Fig. 3 b). The symbols and denominations of the crane systems and elements in Fig. 1, Fig. 2 and Fig. 3. are written and clarified in Table 1.

Table 1. Symbols and denominations in Fig. 1, Fig. 2 and Fig. 3

№	SYMBOL	DENOMINATION
1.	EM_{LG}, EM_{HG}, EM_{SG}, EM_{TG}	*Power aggregates of crane gear winches (electric motor, clutch, brake)*
2.	FGM_{1LG}, FGM_{HG}, FGM_{1SG}, FGM_{2SG}, FGM_{1TG}, FGM_{2TG}	*Function gear mechanisms with linear position function (reduction units, open spur gearing) in the respective crane gear winches*
3.	FGM_{2LG}, JS (FPM_{3LG}), BD	*Function gear mechanisms or function position mechanisms with nonlinear position function (rack and pinion, two planar four-bar linkages) in the respective crane gear winches or systems*
4.	LGW, HGW, SGW, TGW	*Crane gear winches accordingly for operating radius changing, hoisting of cargo, turning platform, crane traveling and steel structure (foundation) above which, they are mounted*
5.	LJS, HS, CSS, TS	*Crane systems accordingly for operating radius changing, hoisting of cargo, turning platform, crane traveling and gantry crane steel structure adjacent part, which is driven or manipulated by crane gear winches*
6.	SS, CLD, C	*Steel structure of the gantry crane, Cargo-lifting device, Cargo*

The level-luffing jib system shown in Fig. 4 and Fig. 5 consists of the following subsystems and their elements:

A) Jib system (JS) comprising: jib - 4, jib arm – 3 and guy - 5;

B) Balancing device (BD) consisting of: tie-bar - 7, rocker arm – 9 and counterweight – 10;

C) Driving mechanism of system (LGM) with: winch - 11 and rack - 8.

The hoisting system is shown in Fig. 4 and Fig. 5 by cargo – 1, wire ropes – 2 and hoisting gear winch – 6.

The lower axle support point O_2 of the guy is stationary assembled to the level-luffing jib system base, which can be a spatial farm, frame or column, traditionally. Such a traditional construction is thoroughly studied and researched in [1] and is shown therein in Fig. 6 and in Fig. 2 of present report. The new changing in the proposed and studied construction of the level-luffing jib system is that point O_2 on Fig. 1 now is moved and attached to the rocker arm - 9 of the counterweight - 10.

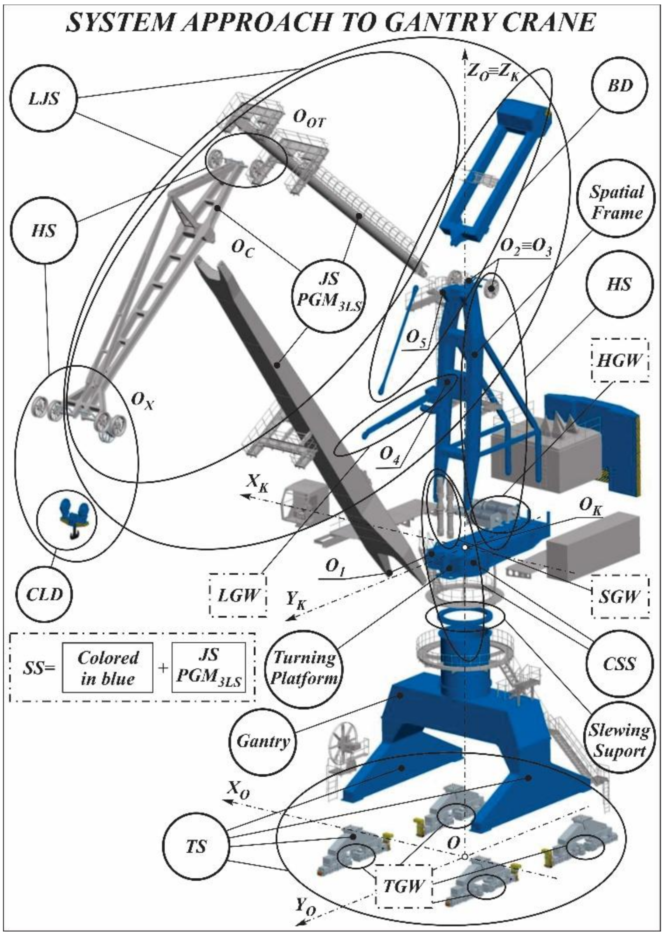

Fig. 2. System approach to gantry crane
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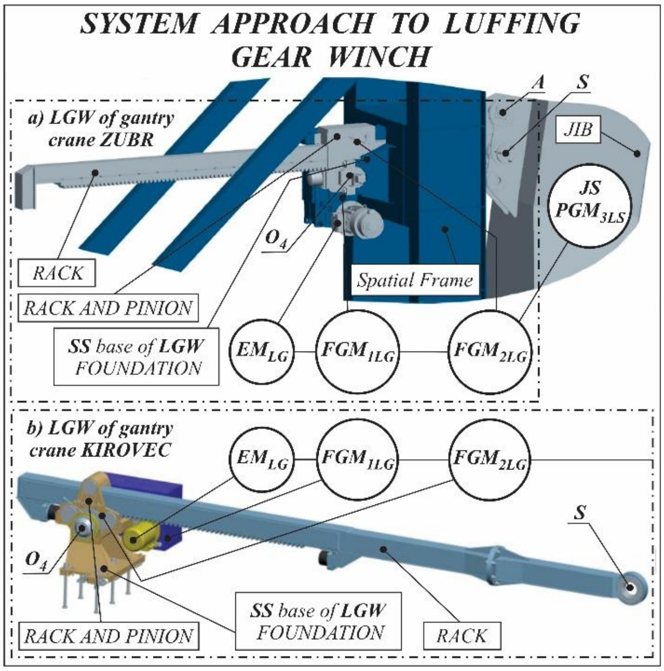

Fig. 3. Luffing gear winches in two variants:
a) of crane ZUBR and b) of crane KIROVEC.
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Let us recall the structure of a level-luffing jib system with a profiled jib arm. This old construction have purpose to achieve an unloading of the jib from bending practically, and reach straight cargo trajectory, by changing the lengths of the rear arm of the jib arm – l_{2X} and the guy - l_{OT}.

The change in the new construction is achieved by moving of point O_2 onto the rocker arm - 8 of the counterweight - 10. This time are changed the length of the guy - l_{OT} again, and also the polar radius from point O_1 to point O_2 - d_{OT} and its corresponding polar angle - η_{OT}. The two four-bar linkages JS and BD from the traditional construction of the level-luffing jib system, shown on Fig. 6 in [1] and researched there, which is now widespread, are transformed into the six-bar linkage structure of the new design of level-luffing jib system, shown in Fig. 4, Fig. 5 and Fig. 6 in present report in this way.

We aim to achieve a positive effect in performance characteristics by this change of structure.

3. Solution to the problem

An new optimization model of the gantry cranes level-luffing jib systems based on the universal methodology [3, 8, 11] is built. It corresponds to the Pareto optimization procedure explained in Fig. 4, Fig. 5 and Fig. 6 in [1], but the new things of structure are shown in Fig. 4, Fig. 5 and Fig. 6 in present report. These new optimization model of the gantry cranes level-luffing jib systems was created in MATLAB [4, 6]. Experimental studies of the new gantry cranes level-luffing jib systems were done using this model.

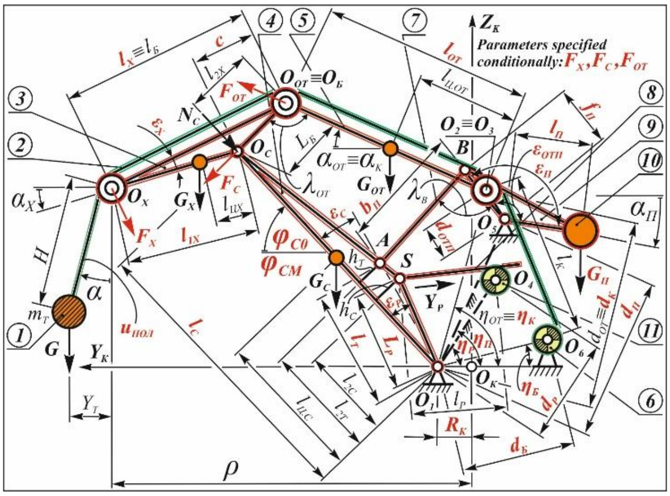

Fig. 4. Optimization mathematical model schema when the hoisting wire ropes pass through the guy

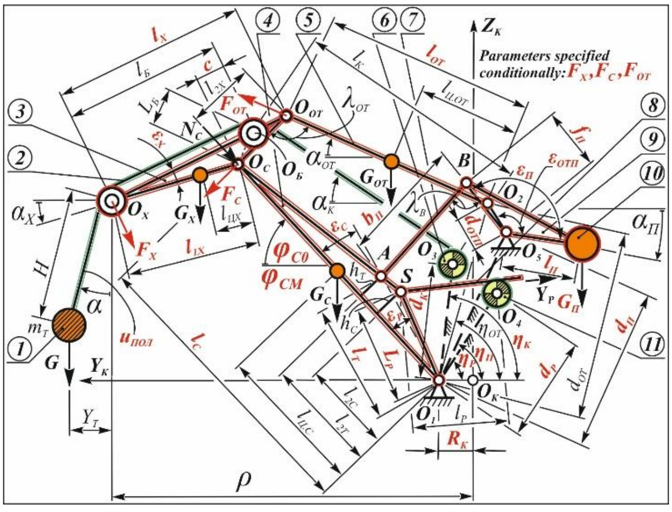

Fig. 5. Optimization mathematical model schema when the hoisting ropes pass arbitrary between the jib and the guy

The Pareto optimization procedure uses the interaction of the 30 parameters characterizing units and parts of the gantry cranes level-luffing jib system shown in red in in Fig. 4 and Fig. 5, which are involved in the formation of 20 criteria defining the quality of its work [5, 7, 10]. Parameters are graphically depicted in Fig. 4 and Fig. 5, and their names are shown below in the next paragraph text. The adequacy of the optimization model has been verified by setting a minimum range of parameters variations adjusted to a specific construction, where the result is similar to previous studies of this type of level-luffing jib system [2].

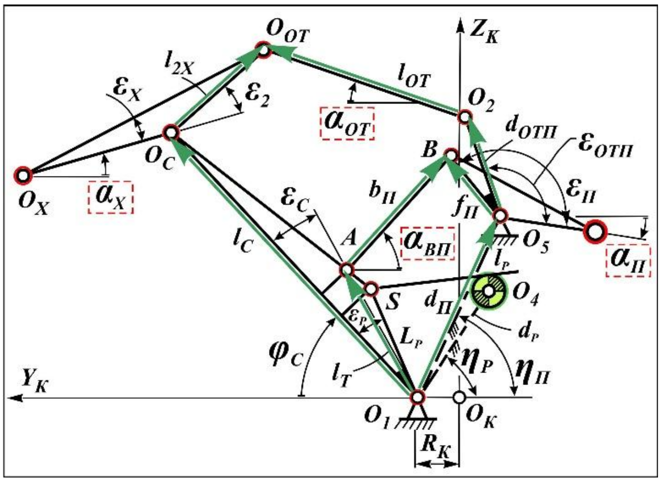

Fig. 6. Schema of the closed vector loop method

The kinematic study in the optimization procedure is carried out by the method of closed vector loop [9] and its adequacy is checked by a graphical method. The first step of kinematic research is to solve the problem of the position of linkages of mechanism. So that, we compile the output system of the equations for analysis, which connects the unchangeable and the variable parameters of the mechanism. This system is easiest and most convenient to compose by the method of projection the closed vector contours on coordinate axes.

The simbols in next four formulas correspond with Fig. 4, Fig. 5 and Fig. 6 and their denomination are shown in next paragraph.

We introduce vectors associated with the linkages of mechanism, organized into two closed vector loops as shown in Fig. 6. We compose the following two vector equations for them:

$$(1)\quad \begin{cases} \overrightarrow{O_1O_C}+\overrightarrow{O_CO_{OT}}=\overrightarrow{O_1O_5}+\overrightarrow{O_5O_2}+\overrightarrow{O_2O_{OT}}\ ; \\ \overrightarrow{O_1A}+\overrightarrow{AB}=\overrightarrow{O_1O_5}+\overrightarrow{O_5B}\ . \end{cases}$$

We project the vector equations of this system on the coordinate axes and we get the following system (2) of four non-linear equations for the unknown four angles: $\boldsymbol{\alpha_X}$, $\boldsymbol{\alpha}_{\text{OT}}$, $\boldsymbol{\alpha}_{\Pi}$ and $\boldsymbol{\alpha}_{\text{B}\Pi}$. The angles are shown in Fig.6.

$$(2)\quad \begin{cases} l_C\cos\varphi_C-l_{2X}\cos(\boldsymbol{\alpha_X}+\varepsilon_2)=-d_{\Pi}\cos\eta_{\Pi}+ \\ +d_{\text{OT}\Pi}\cos(\pi-\varepsilon_{\text{OT}\Pi}+\boldsymbol{\alpha}_{\Pi})+l_{\text{OT}}\cos\boldsymbol{\alpha}_{\text{OT}}; \\ l_C\sin\varphi_C+l_{2X}\sin(\boldsymbol{\alpha_X}+\varepsilon_2)=d_{\Pi}\sin\eta_{\Pi}+ \\ +d_{\text{OT}\Pi}\sin(\pi-\varepsilon_{\text{OT}\Pi}+\boldsymbol{\alpha}_{\Pi})+l_{\text{OT}}\sin\boldsymbol{\alpha}_{\text{OT}}\ ; \\ l_{\text{T}}\cos(\varphi_C+\varepsilon_{\text{C}})-b_{\Pi}\cos\boldsymbol{\alpha}_{\text{B}\Pi}= \\ =-d_{\Pi}\cos\eta_{\Pi}+f_{\Pi}\cos(\pi-\varepsilon_{\Pi}+\boldsymbol{\alpha}_{\Pi})\ ; \\ l_{\text{T}}\sin(\varphi_C+\varepsilon_{\text{C}})+b_{\Pi}\sin\boldsymbol{\alpha}_{\text{B}\Pi}= \\ =d_{\Pi}\sin\eta_{\Pi}+f_{\Pi}\sin(\pi-\varepsilon_{\Pi}+\boldsymbol{\alpha}_{\Pi})\ . \end{cases}$$

The angle φ_C we treat as known and it is the generalized coordinate of this mechanism. It determines a degree of mobility of a six-linkages mechanism.

As a result of solving this system (2) we get the values of the four angles: $\boldsymbol{\alpha_X}$, $\boldsymbol{\alpha}_{\text{OT}}$, $\boldsymbol{\alpha}_{\Pi}$ and $\boldsymbol{\alpha}_{\text{B}\Pi}$., so that the position of the linkages is already known.

The second stage of the mechanism analysis is the determination of first order kinematic transmission functions. To do this, we differentiate the system (2) on the time. We obtain the following system (3) of four linear equations:

$$(3)\quad \begin{cases} -l_C\sin\varphi_C+[l_{2X}\sin(\alpha_X+\varepsilon_2)]\boldsymbol{U_X}= \\ =-[d_{\text{OT}\Pi}\sin(\pi-\varepsilon_{\text{OT}\Pi}+\alpha_{\Pi})]\boldsymbol{U}_{\Pi}-[l_{\text{OT}}\sin\alpha_{\text{OT}}]\boldsymbol{U}_{\text{OT}}\ ; \\ l_C\cos\varphi_C+[l_{2X}\cos(\alpha_X+\varepsilon_2)]\boldsymbol{U_X}= \\ =[d_{\text{OT}\Pi}\cos(\pi-\varepsilon_{\text{OT}\Pi}+\alpha_{\Pi})]\boldsymbol{U}_{\Pi}+[l_{\text{OT}}\cos\alpha_{\text{OT}}]\boldsymbol{U}_{\text{OT}}\ ; \\ -l_{\text{T}}\sin(\varphi_C+\varepsilon_{\text{C}})+[b_{\Pi}\sin\alpha_{\text{B}\Pi}]\boldsymbol{U}_{\text{B}\Pi}= \\ =-[f_{\Pi}\sin(\pi-\varepsilon_{\Pi}+\alpha_{\Pi})]\boldsymbol{U}_{\Pi}\ ; \\ l_{\text{T}}\cos(\varphi_C+\varepsilon_{\text{C}})+[b_{\Pi}\cos\alpha_{\text{B}\Pi}]\boldsymbol{U}_{\text{B}\Pi}= \\ =[f_{\Pi}\cos(\pi-\varepsilon_{\Pi}+\alpha_{\Pi})]\boldsymbol{U}_{\Pi}\ . \end{cases}$$

In this system, kinematic transmission functions $\boldsymbol{U_X}$, $\boldsymbol{U}_{\text{OT}}$, $\boldsymbol{U}_{\Pi}$ and $\boldsymbol{U}_{\text{B}\Pi}$ of first order are unknown. After solving the system we get the values of these kinematic transmission functions.

The third stage of the mechanism's analysis is the determination of the second-order kinematic transmission functions. To do this, we differentiate the system (3) by the time. We get the following system again from four linear equations:

$$(4)\quad \begin{cases} -l_C\cos\varphi_C+[l_{2X}\cos(\alpha_X+\varepsilon_2)]U_X^2+ \\ +[l_{2X}\sin(\alpha_X+\varepsilon_2)]\boldsymbol{dU_X/d\varphi_C}= \\ =-[d_{\text{OT}\Pi}\cos(\pi-\varepsilon_{\text{OT}\Pi}+\alpha_{\Pi})]U_{\Pi}^2-[l_{\text{OT}}\cos\alpha_{\text{OT}}]U_{\text{OT}}^2+ \\ -[d_{\text{OT}\Pi}\sin(\pi-\varepsilon_{\text{OT}\Pi}+\alpha_{\Pi})]\boldsymbol{dU_{\Pi}/d\varphi_C}+ \\ -[l_{\text{OT}}\sin\alpha_{\text{OT}}]\boldsymbol{dU_{\text{OT}}/d\varphi_C}\ ; \\ -l_C\sin\varphi_C-[l_{2X}\sin(\alpha_X+\varepsilon_2)]U_X^2+ \\ +[l_{2X}\cos(\alpha_X+\varepsilon_2)]\boldsymbol{dU_X/d\varphi_C}= \\ =-[d_{\text{OT}\Pi}\sin(\pi-\varepsilon_{\text{OT}\Pi}+\alpha_{\Pi})]U_{\Pi}^2-[l_{\text{OT}}\sin\alpha_{\text{OT}}]U_{\text{OT}}^2+ \\ +[d_{\text{OT}\Pi}\cos(\pi-\varepsilon_{\text{OT}\Pi}+\alpha_{\Pi})]\boldsymbol{dU_{\Pi}/d\varphi_C}+ \\ +[l_{\text{OT}}\cos\alpha_{\text{OT}}]\boldsymbol{dU_{\text{OT}}/d\varphi_C}\ ; \\ -l_{\text{T}}\cos(\varphi_C+\varepsilon_{\text{C}})+[b_{\Pi}\cos\alpha_{\text{B}\Pi}]U_{\text{B}\Pi}^2+ \\ +[b_{\Pi}\sin\alpha_{\text{B}\Pi}]\boldsymbol{dU_{\text{B}\Pi}/d\varphi_C}= \\ =-[f_{\Pi}\cos(\pi-\varepsilon_{\Pi}+\alpha_{\Pi})]U_{\Pi}^2+ \\ -[f_{\Pi}\sin(\pi-\varepsilon_{\Pi}+\alpha_{\Pi})]\boldsymbol{dU_{\Pi}/d\varphi_C}\ ; \\ -l_{\text{T}}\sin(\varphi_C+\varepsilon_{\text{C}})-[b_{\Pi}\sin\alpha_{\text{B}\Pi}]U_{\text{B}\Pi}^2+ \\ +[b_{\Pi}\cos\alpha_{\text{B}\Pi}]\boldsymbol{dU_{\text{B}\Pi}/d\varphi_C}= \\ =-[f_{\Pi}\sin(\pi-\varepsilon_{\Pi}+\alpha_{\Pi})]U_{\Pi}^2+ \\ +[f_{\Pi}\cos(\pi-\varepsilon_{\Pi}+\alpha_{\Pi})]\boldsymbol{dU_{\Pi}/d\varphi_C}\ . \end{cases}$$

In this system, second-order kinematic transmission functions $\boldsymbol{dU_X/d\varphi_C}$, $\boldsymbol{dU_{\text{OT}}/d\varphi_C}$, $\boldsymbol{dU_{\Pi}/d\varphi_C}$ and $\boldsymbol{dU_{\text{B}\Pi}/d\varphi_C}$ are unknown. As a result of solving system (4) we get their values.

Now, after determining the first and second order kinematic transmission functions, we can easily determine the angular velocities and angular accelerations of the all linkages of the mechanism, as well as the speeds and accelerations of points from these linkages.

4. Results and discussion

Two program complexes have been compiled for the purposes of the study. They correspond to the two design schemas studied, which are shown in Fig. 4 and Fig. 5. Results are obtained for both shown structures.

A third type of level luffing jib system structure exists when hoisting wire ropes pass through the boom, but it is constructively inconvenient and not addressed in this report. The results shown below refer only to Fig. 4.

In the optimization procedures, is taking into account that the force in the guy acts on the counterweight rocker arm, and when the hoisting ropes pass along the guy, hoisting wire ropes act on the counterweight rocker arm too in point O_2.

The experimental optimization study was carried out with a cargo of 160 kN, and results are pictured on the graphs in Fig. 7, Fig. 8 and Fig. 9. Specified conditional forces are needed to close the circle of "force - inter sectional dimensions - mass" needed for the study. The forces are shown in Fig. 4 and Fig. 5.

The parameters shown in Fig. 4 in red, of selected optimal solution are described by the names and the values obtained after the optimization as follows:

1. The coordinate of jib base point O_1 along coordinate axis $O_K Y_K - R_K$ = 1.920 m;
2. The polar radius of guy base point O_2 in regard to point O_5 - $d_{OT\Pi}$ = 0,5650 m;
3. The polar angle of distance $d_{OT\Pi}$ to guy base point O_2 in regard to the rear arm of the counterweight l_{Π} - $\varepsilon_{OT\Pi}$ = 2,6472 rad;
4. The polar radius of axis in point O_3 wire roupe deviating pulleys on the guy base point $O_2 \equiv O_3$ these points are coincident - d_K = it changes by moving of point O_2;
5. The polar angle of distance d_K to point $O_2 \equiv O_3$ wire roupe deviating pulleys on the guy base point $O_2 - \eta_K$ = it changes by moving of point O_2;
6. The length of jib - l_C = 27.693 m;
7. The length of jib arm - l_X = 16.048 m;
8. The length of jib arm front part - l_{IX} = 11.511 m;
9. The angle in jib arm front part between axles $O_X O_C$ and $O_X O_{OT}$ - ε_X = 0.0592 rad;
10. The length of guy - l_{OT} = 23.548 m;
11. The coordinate of wire roupe deviating pulleys along the length of jib arm - c = according to l_X, l_{IX} and ε_X;
12. The distance from point O_1 to point A - l_T = 6.751 m;
13. The angle between the longitudinal axis of jib $O_1 C$ and $O_1 A$ - ε_C = 0.1553 rad;
14. The polar radius of the swing counterweight axis point O_5 in regard to point O_1 - d_{Π} = 11.792 m;
15. The polar angle of distance d_{Π} to point O_5 in regard to the negative direction of the axis $O_K Y_K - \eta_{\Pi}$ = 1.0604 rad;
16. The length of counterweight arm - l_{Π} = 3.518 m;
17. The angle on the swing counterweight axis in point O_5 - ε_{Π} = 2.5998 rad;
18. The length of rocker arm front arm - f_{Π} = 2.185 m;
19. The length of tie-bar - b_{Π} = 8.652 m;
20. The distance from point O_1 to S - L_P = 6.011 m;
21. The angle between the longitudinal axis of jib $O_1 C$ and $O_1 S$ - ε_P = 0.1503 rad;
22. The polar radius of axis in point O_4 of LGM in regard to point O_1 - d_P = 6.552 m;
23. The polar angle of distance d_P to point O_4 in regard to the negative direction of the axis $O_K Y_K$ η_P = 0.9089 rad;
24. The maximum bending force of jib arm, specified conditionally - F_{XMax} = 189 kN;
25. The maximum bending force of jib, specified conditionally - F_{CMax} = 53 kN;
26. The maximum tensing force of guy, specified conditionally - F_{OTMax} = 608 kN;
27. Mass of counterweight - m_{Π} = 20.993 t;
28. The jib inclination angle, which corresponding to maximum operating radius - φ_{C0} = 0.8127 rad;
29. The jib inclination angle, which corresponding to minimum operating radius - φ_{CM} = 1.4184 rad;
30. The cargo block and tackle reduces factor - $u_{\Pi O\Pi}$ = 1;

The dependencies of the forces acting on the level luffing jib system elements for a selected prototype of gantry crane and the results for the forces of the optimized new structure according to Fig. 4, are shown in Fig. 7. It can be seen that the values of the forces in the new optimized structure are significantly smaller than the existing gantry crane level luffing jib system. This is a very good indicator. In this way, the total mass of the level luffing jib system structure can be reduced. This is followed by other positive effects. When reducing the total mass of the level luffing jib system, the mass of the gantry

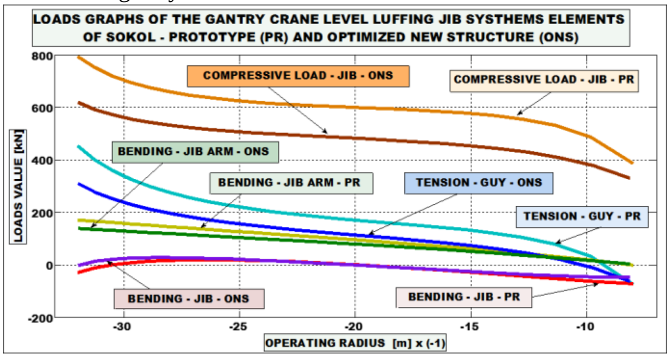

Fig. 7. Dependances of the forces acting on level luffing jib system elements

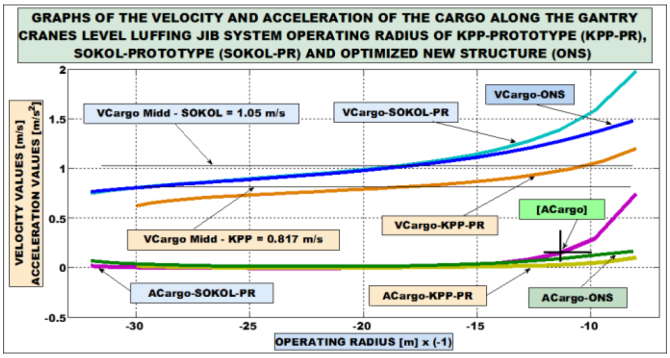

Fig. 8. Graphs of the cargo velocity and acceleration

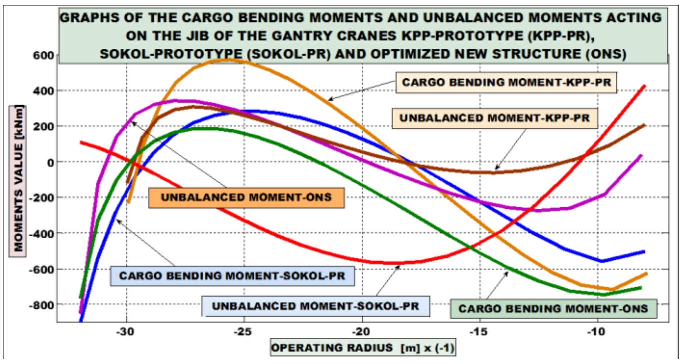

Fig. 9. Graphs of the moments acting on the jib

crane rotating part is reduced. It is possible to reduce the power of the electric motor in the crane slewing system also. When the mass of moving elements decreases, the forces of inertia acting on them and other elements are also reduced.

The cargo velocity and acceleration graphs for two models of gantry cranes selected for prototypes and the graphs of cargo velocity and acceleration for the new structure are shown in Fig. 8. The graphs show that for the new structure on minimum operating radius, significantly lower velocity and acceleration values have been achieved. Decreasing speed is beneficial as the diapason of speed variation along the operating radius is reduced. This makes the movement of the cargo evenly. The acceleration reduction at minimum operating radius is also beneficial because it limits the possibility of cargo pendulum. This makes crane and cranedriver work calm and precise, which is essential for quality and productive work.

The graphs of the cargo unbalanced moments and jib unbalanced moments for the two model of gantry cranes selected for prototypes and the same graphics for the optimized new structure of level luffing jib systems are shown in Fig. 9. It is seen that the values of these criteria are commensurate between the prototypes and the new construction.

In the optimization studies to date the power of the electric motor has not been reduced.

The deviation of the cargo from the horizontal trajectory is also maintained within half a meter. When working with both complexes of programs feels the influence of equidistant and non-equidistant trajectories of the top of the jib arm point O_X and the cargo. These trajectories are reached by the passing of the lifting ropes through the level luffing jib system.

5. Conclusion

5.1. The systematic approach to research, analysis and synthesis of gantry cranes level luffing jib system has significant positive aspects. It allows that the level luffing jib system to be viewed in conjunction with other systems of the crane, load system of the crane and human systems. The system approach creates a clear and universal point of view for the denominations and interactions between the separate systems of the crane and between the elements in the systems themselves.

5.2. No additional elements have been added to the gantry crane level luffing jib system when constructing the new construction. Only the unit in point O_2 of the guy mast has been changed, this node being mounted on the counterweight jib arm. This way we have done, point O_2 movable. This implies that the designs of the counterweight and of the guy moves so that all the gantry crane level luffing jib system elements can be moved reciprocately. The construction must ensure that the hoisting ropes when pass through the guy and after that lead them to a machine room, point O_6 in Fig. 4 where the hoisting winches are located.

5.3. The study of the new construction has shown that can achieve better, lower values of the acceleration of the cargo along the operating radius, especially at minimum operating radius, while preserving the other basic parameters.

5.4. Trajectories at the top of the jib arm point O_X and the cargo can be controversial criteria.

5.5. Improved criteria values are obtained when the location of point O_2 is in the second and third quadrants of a coordinate system centered in point O_5 and a positive direction of the abscissa is in the $l_П$ direction.

5.6. The forces values in the new optimized structure are significantly smaller round about 15…25 %, than the existing gantry crane level luffing jib system. Then come the positive effects caused by the lower loading forces of the gantry crane level luffing jib system elements - the smaller masses of the elements and the smaller inertial forces in the work of both systems, luffing sysytem and crane slewing system.

As a future work to improve and develop of this new structure of gantry cranes level luffing jib systems, it may be envisaged to develop the optimization procedure in order to improve other criteria defining the quality of the level luffing jib system work. It is a good idea to make a design improvement of the unit in point O_2, which now is mounted on the counterbalance jib arm, taking into account the specificity of the new structure and the passage of the hoisting wire ropes through the level luffing jib system. It is possible after improving to check with the help of optimization procedure the performance of the level luffing jib system in different operating modes and typical technological cycles.

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