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Dynamic Processes of Loader Cranes Manipulators with Excessive Backlashes and Elastic Damping in Their Hinges

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Abstract

This research is aimed at developing a mathematical model and the methodology for computer simulation of hydraulically driven manipulators of mobile cranes having excessive backlashes in cylindrical joints. The authors proposed a structural design and considered the mechanism of reducing the additional impact load in the hinges by means of elastic damping of oscillatory processes. This method allows estimating the degree of influence of the backlash and stiffness of the elastic damper on the change in the quantitative characteristics of the dynamic loading of the manipulators metalwork and the motion parameters of the transported load. While in operation, the excessive backlashes may cause an increase in the level of dynamic loading of manipulators up to 2 times or more. However, the rational choice of the elastic dampers stiffness allows an effective solution to this problem to the point of complete elimination of the additional impact load.

Keywords

backlash, cylindrical hinge, dynamic loading, elastic damping, loader crane

1 Introduction

At present, hydraulic manipulators are widely used as operating equipment for mobile transportation and production machines of various purposes [1]. They are used to perform a wide range of lifting, transportation, transfer, unloading and storage operations [2]. Mobile machines equipped with manipulators have shown their effectiveness in a large number of economic sectors - industrial production, construction, gas and oil production, freight transportation, timber industry, agriculture, etc. Therefore, a variety of hydraulic manipulator designs that are significantly different in their technical characteristics (nominal carrying capacity, dimensions of the operating area, rate of individual movements, etc.) and kinematic schemes are available at the world market [3]. As for the countries manufacturing this equipment, one may mention such countries as South Korea, Japan, China, Germany, Italy, the USA, Austria, Russia, etc.

The load bearing steelwork of the manipulator can consist of 3 to 12 consequently connected mobile links. The links in pairs form the kinematic pairs of the V class - rotational and translational [2]. Rotational pairs are accomplished on the basis of cylindrical hinges. They provide the rotational motion relative to the link longitudinal axis or the turning relative motion of neighboring links. Structurally cylindrical hinges [4] are lugs made of thick-walled sheet metal, which are fixed on the surface of neighboring links metal structures. The lugs have coaxial holes for mounting a hinge pin in them. This ensures the formation of a cylindrical hinge joint [5]. Steel, bronze or plastic bushes can be additionally installed in the lug holes to reduce friction losses [6].

2 Problem statement and ways of its solution

Despite the simplicity of this design and the wide distribution of such cylindrical hinge joints in manipulators, they have a significant drawback. During operation, the clearance between the hinge pins and the surfaces of the lug holes increases. This results from the frictional wear and impact collapse of their contact surfaces [3, 7].

As the operational experience [3] shows, the presence of increased clearances in the cylindrical hinges of manipulators has dangerous consequences for their reliable operation.

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The metal structure of manipulators develops impact stresses and additional dynamic stresses due to the almost instantaneous change in the contact points of the hinge pin when the lugs perform the required movements [8, 9]. This leads to a two-fold (or even more) increase in the stress level in the crane's metal structure and a twenty-fold increase in the acceleration of the load movement [4]. Also, the kinematic accuracy of the manipulator and the accuracy of positioning of its working member are deteriorated. The resulting significant inertial forces of the unscheduled direction can be one of the main causes of cracks formation in the welding seams and the basic material of the manipulator metal structure. The use of intermediate antifriction bushes [6], polymer coatings [10] or different hardening methods [11] is able to slow but not eliminate the development of the abovementioned negative processes.

Therefore, Russian regulatory documents in the field of safe operation of load-lifting machines establish the maximum permissible increase in the diameter of holes in the hinges relative to their initial dimensions [3]. It is 2 mm at the rated hole diameter of up to 50 mm, 3 mm at the rated diameter of 50 to 100 mm and 4 mm at the rated diameter of more than 100 mm.

At present, the influence of excessive backlashes in sections hinge joints on the working processes and dynamics of the loader cranes has been studied very little. Earlier in [12], a kinematic model of a cylindrical hinge with backlashes was developed. Backlashes were modeled by the introduction of additional degrees of freedom in the joint. However, it can't be used to analyze dynamic processes and dynamic stress-strain state of the metal structure of the hydraulic loader cranes with excessive backlashes in sections hinge joints. For the first time, the problem of modeling a stress-strain state in hinges with excessive backlashes was considered in [13, 14].

The mechanism for the formation of additional dynamic load of the manipulators metal structure with the increased clearance in the cylindrical hinge is shown in Fig. 1. With the course of time, clearances appear in the cylindrical hinges: δ_{w1} - between the surface of the hole of lug 2 and the surface of hinge pin 1; δ_{w2} - between the surface of the hole in lug 3 and the surface of hinge pin 1. With the above clearances available, mutual skewing in the vertical planes in which the neighboring links 4 and 5 move relative to the original neutral position 0-0 will occur. The skew range of link 5 relative to link 4 characterizes the extreme positions I-I and II-II. The extreme position I-I is conditioned by the contact of lugs 3 with hinge pin 1

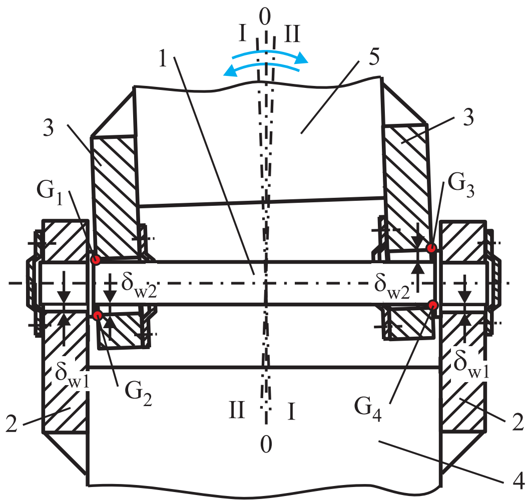

Fig. 1 The scheme of possible displacements of the hinge pin in the presence of increased clearances in the cylindrical hinge: 1 – hinge pin; 2, 3 – lugs of neighboring links; 4, 5 – neighboring links

at the extreme points of G_1 and G_4. Extreme position II-II is due to the contact at the extreme points of G_2 and G_3.

With the relative rotational movement of neighboring links 4 and 5 (under conditions of natural swinging of the transported load), there will be an alternating, almost instantaneous change in the pairs of contact points of lugs 3 with hinge pin 1. The change in pairs of contact points will be accompanied by an impact and a pulse increase in the level of the dynamic state of stress of the manipulator metal structure.

In order to reduce the dynamic impact loads acting on the transported load and the metal structure of manipulators, special damping structures with elastic cushioning elements for the hinges of adjacent sections of manipulators were proposed in [15-17]. Tension-compression springs [15], rings [16] (Fig. 2), V-shaped segments [17] or arc elements [3] can be used as elastic elements. The mode of functioning is based on the braking effect the elastic cushioning elements (rings) have on the movement of hinge pin 2 due to the appearance of an elastic resistance force. Its value increases in proportion to the displacement of the hinge pin from the equilibrium position, and the direction is strictly opposite to the direction of displacement. The hinge pin is made elongated by means of shank 3. Metal bush 4 is mounted on the shank. It freely slides over the surface of the shank. Shock absorbers are installed between the bush and the body of damper 5. The internal cavity of the device is filled with a plastic lubricant.

In the initial period of the manipulator operation, clearances δ_{w1} and δ_{w2} are almost zero. Shank 3 of hinge pin 2

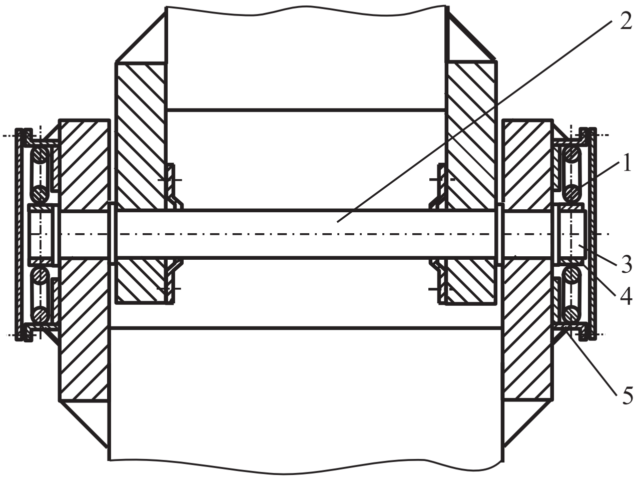

Fig. 2 Hinge structure with an elastic damper:
1 – elastic damping elements (rings); 2 –hinge pin;
3 – shank; 4 – metal bush; 5 – damper body

rotates freely inside bush 4 and exerts no pressure on it. As clearances δ_{w1} and δ_{w2} are formed, the manipulator's operation begins to be accompanied by a skew of neighboring links and impact forces. If any of the neighboring links is skewed relative to the initial neutral position 0-0 (Fig. 1), the relative shank displacement (Fig. 3) occurs. In Fig. 3, the initial neutral position of the shank is shown by fine lines with the center of the cross section at the point O_0. The displaced position due to the skew up to the extreme position I-I is marked by the main lines with the center of the displaced section at the point O_1. Thus, when the links are skewed, the shank cross-section is displaced along the line O_0-O_1. There is a deformation of the ring elastic elements. Some of these elements, located in the direction

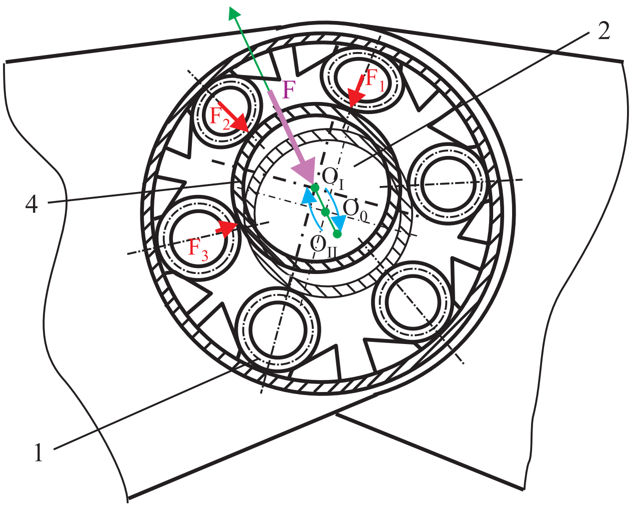

Fig. 3 Elastic damper operation scheme:
1 – elastic damping elements (rings);
2 – hinge pin; 4 – metal bush

of the shank displacement along the line O_0-O_1, undergo compression. The other part of them, located in the opposite direction of the shank displacement, does not undergo deformation. Elastic forces F_1, F_2, F_3 arise in all deformed circular elastic elements (Fig. 3).

Stresses appearing in individual elements F_i are proportional to their stiffness and compression. They sum up and create a joint resistance force F. It is directed against the shank displacement along the line O_0-O_1 and is transmitted to the hinge pin, exerting a braking effect on it.

At the transition of the manipulator links from the extreme position I-I to the extreme position II-II, the shank cross-section is displaced along the line O_I-O_0-O_{II}. At the same time, the resistance force F occurs similarly, which inhibits the shank displacement along the line O_I-O_0-O_{II}. The automatic occurrence of the braking force F causes a decrease in impact acceleration, additional impact stresses and dynamic stresses in the metal structure of the manipulator.

During the operation of the damper, it is possible to wear the mating surfaces of the metal bush 4 and the shank 3 of the hinge pin 2. However, the backlash that occurs between these surfaces does not affect the performance of the elastic damper. The presence of a gap can only lead to some delay in the reaction of the damper to the mutual skewing of neighboring links of the manipulator. During operation, when carrying out repairs of the metalwork of the manipulator, it is necessary to replace worn bushes.

3 Mathematical model of a hinge and a damper joint operation

A mathematical model of the joint operation of a hinge with an increased clearance and an elastic damper was constructed by introducing additional possible displacements s and imposing constraints on them related to the clearance size δ_w (Fig. 4). These constraints are realized by means of elastic and viscous elements. The influence of the damper is taken into account only by means of elastic elements.

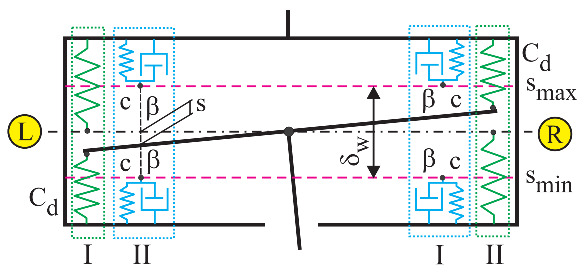

Fig. 4 Calculation scheme for the hinge with an increased clearance and an elastic damper: I – damper; II – hinge restraints

The force value R, which acts on the hinge pin from the clearance limiting elements and the damper, is determined as follows:

$$R(s,\dot{s}) = \begin{cases} c(s_{\max} - s) - \beta\dot{s} - k_{hp}C_d s, & s \geq s_{\max}; \\ c(s_{\min} - s) - \beta\dot{s} - k_{hp}C_d s, & s \leq s_{\min}; \\ -k_{hp}C_d s, & \text{else}, \end{cases} \quad (1)$$

$$C_d = c_e \left\{ \sum_{n=1}^{n=n_1} \cos\left[(n-1)\Delta\alpha + \alpha_0\right] + \sum_{n=1}^{n=n_2} \cos(n\Delta\alpha - \alpha_0) \right\},$$

where c, β are stiffness and viscosity factors of hinge restraints; C_d is the damper stiffness factor; $s_{\min}$, $s_{\max}$ are the clearance-dependent restraints δ_w; $k_{hp} = 1 + 2l_{hp} / b_{hp}$ is the factor of the shank length of the hinge pin l_{hp}; b_{hp} is the hinge pin length; c_e is the stiffness factor of the damping element; $\Delta\alpha$ is the angular spacing of the neighboring damping elements; α_0 is the angle between the direction of the hinge pin displacement and the axis of the closest damping element; $n_e = n_1 + n_2$ is the number of damping elements; $s, \dot{s}$ are the generalized coordinate and the velocity corresponding to the appended possible displacement s.

The correctness of the used system of conditions for including/excluding the limiting elements from the contact is confirmed by the study [18]. This model can be used if n additional possible displacements are required to model the hinge clearance. In this case stresses $R_1, R_2, \ldots, R_n$ will be responsible for the compliance with the imposed kinematic constraints. Each of them corresponds to the i-th appended possible displacement and is determined from the dependence similar to Eq. (1):

$$R_i = f(s_1, s_2, \ldots, s_n, \dot{s}_1, \dot{s}_2, \ldots, \dot{s}_n).$$

The procedure of introducing additional possible displacements s is illustrated by Fig. 5. A three-section manipulator was taken as an example. Its metal structure consists of three links, connected to each other and to the base by means of three cylindrical hinges. Hinges 12 and 23 allow the boom (link 2) and the handle (link 3) to rotate in one vertical plane in the direction of q_2 and q_3, respectively. Hinge 01 allows the links 2 and 3 to rotate around the vertical axis of the column (of link 1) in the direction q_1. Hinge 3G serves for installing the load-handling device of the crane. When an increased clearance δ_w is not available, hinge 3G allows the swinging of the transported cargo only in a vertical plane in the direction of path q_4. Due to the formation of a clearance δ_w in hinge 3G, the crane load-handling device gets the opportunity of further movement. The projection of this displacement on the vertical plane is characterized by the value q_{4v}, and on the horizontal plane by the value q_{4h}. Thus, it is necessary to introduce two additional possible displacements $s_1 \equiv q_{4h}$ and $s_2 \equiv q_{4v}$ into the hinge.

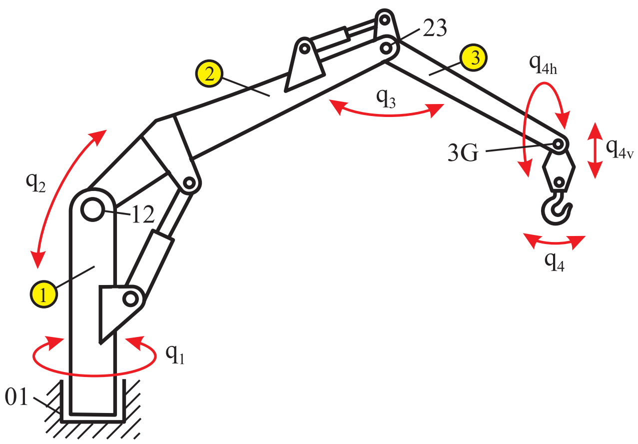

Fig. 5 The computation scheme for the three-section loader crane with an increased clearance in hinge 3G

The generalized coordinates $s_1 \equiv q_{4h}$ and $s_2 \equiv q_{4v}$ and the limiting generalized stresses R_1 and R_2 corresponding to them are connected to the central displacements of the R (right) and L (left) frontal sections of the hinge pin in the holes of lug u_L, u_R and displacement velocities $\dot{u}_L, \dot{u}_R$ the following dependences:

$$u_L = q_{4v} - b_{hp}\sin(q_{4h}); \quad \dot{u}_L = \dot{q}_{4v} - \dot{q}_{4h}b_{hp};$$

$$u_R = q_{4v} + b_{hp}\sin(q_{4h}); \quad \dot{u}_R = \dot{q}_{4v} + \dot{q}_{4h}b_{hp};$$

$$R_1 = F_R + F_L; \quad R_2 = (F_R - F_L)b_{hp};$$

$$F_L = \begin{cases} c(u_{L_{\max}} - u_L) - \beta\dot{u}_L - C_d k_{hp} u_L, & u_L \geq u_{L_{\max}}; \\ c(u_{L_{\min}} - u_L) - \beta\dot{u}_L - C_d k_{hp} u_L, & u_L \leq u_{L_{\min}}; \\ -C_d k_{hp} u_L, & \text{else}, \end{cases}$$

$$F_R = \begin{cases} c(u_{R_{\max}} - u_R) - \beta\dot{u}_R - C_d k_{hp} u_R, & u_R \geq u_{R_{\max}}; \\ c(u_{R_{\min}} - u_R) - \beta\dot{u}_R - C_d k_{hp} u_R, & u_R \leq u_{R_{\min}}; \\ -C_d k_{hp} u_R, & \text{else}. \end{cases}$$

4 Computer computation findings and their analysis

The effect of elastic dampers on the dynamic processes occurring in the metal structure of hydraulic manipulators was considered as exemplified by the rotation of the boom of the mobile machine AST-4-A. The kinematic diagram of the manipulator of this machine is shown in Fig. 5. The rotation was effected from quiescent state with angular acceleration of $\ddot{q}_1 = 0.15$ rad/s². The design configuration of the loader crane was characterized by the following links arrangements: the boom and the handle tilts relative to the horizontal plane were equal to +30 °C and −30 °C, respectively. The weight of the transported cargo is 7.5 kN. The size of the clearance was considered in the interval of δ_w from 0 to 2 mm.

To accomplish the necessary dynamic calculations, the software package KBCrane Dynamics [3] was used. It includes previously developed mathematical models and universal methods for dynamic analysis and determination of the stress-strain state of metal structures of mobile machines manipulators [8, 9]. KBCrane Dynamics allows you to solve the following problems:

1. solution of direct, inverse and hybrid tasks of the dynamics of the manipulator;
2. accounting for the action on the manipulator of various power factors;
3. numerical integration of the equations of motion of the manipulator links by various methods;
4. determination of the forces transmitted through hinge joints;
5. determination of the speeds and accelerations of the manipulator links;
6. the calculation of dynamic stresses in the manipulator links when moving the load.

The processor code is written in the C++ programming language in the MS Visual Studio Express 2013 development environment. The pre/postprocessor code is written in the C# programming language in the SharpDevelop development environment. OpenGL from OpenTK is used as a graphic library.

The authors studied the influence of the clearance size δ_w (depending on the damper stiffness C_d) on the quantitative characteristics of the time laws of the following values:

- central displacement of the frontal section of the hinge pin from the initial position $u_R(\tau)$;
- central acceleration of the frontal section of the hinge pin $a(\tau)$;
- dynamic stress in the most loaded section of the boom $\sigma_2(\tau)$;
- dynamic stress in the most loaded section of the handle $\sigma_3(\tau)$.

Comparison of these results with the results of the above analysis for the case of absence of a damper (C_d = 0) [4] made it possible to reveal the regularities of dynamic processes in cylindrical hinges equipped with elastic dampers.

As an example, Fig. 6 shows the graphs of the time laws for stresses $\sigma_2(\tau)$ for the clearance in the hinge δ_w = 0.4 mm at the rated weight of the transported load and the three characteristic values of the damper stiffness. The quality of graphs $\sigma_3(\tau)$ and $a(\tau)$ is similar to graphs $\sigma_2(\tau)$. Fig. 7 shows the influence of the clearance size δ_w and the damper stiffness C_d on the main quantitative parameters of the dynamic processes in the manipulator boom. The following parameters are shown: $\sigma_{2\max}$ (C_d = 0) - maximum dynamic stress in the absence of a damper; $\sigma_{2\max}$, $\sigma_{2\min}$ - the maximum and minimum dynamic stresses of the impact load cycle in the presence of a damper; $\sigma_{2,st}$ - the stress in the stationary mode (in the absence of a clearance); $R_{\sigma 2} = \sigma_{2\max} - \sigma_{2\min}$ - the range of dynamic stresses.

The damper has a significant effect on the quantitative and qualitative characteristics of the dynamic processes in the manipulator metal structure. This impact is ambiguous. Depending on the rigidity value C_d, the damper can have both a positive and a negative effect on the dynamic stress condition of the metal structure. With small clearances present in the hinge (in the interval of $\delta_w \in [0;\delta_{th}]$), the maximum dynamic stresses occurring as a result of the "hinge pin – lug" impact are slightly higher than in the absence of a damper. Stable damping is observed when the clearance threshold value δ_{th} is exceeded. It is inversely proportional to the stiffness of the damper. The higher is the stiffness of C_d, the smaller is δ_{th}.

This is due to the fact that for small clearances (small displacements of the hinge pin), the forces of elastic resistance of damping elements are insignificant, and therefore, ineffective for braking.

The damper is not only a shock absorber of dynamic processes in the metal structure, but also a restraint for the greatest possible wear of the lug hole when the

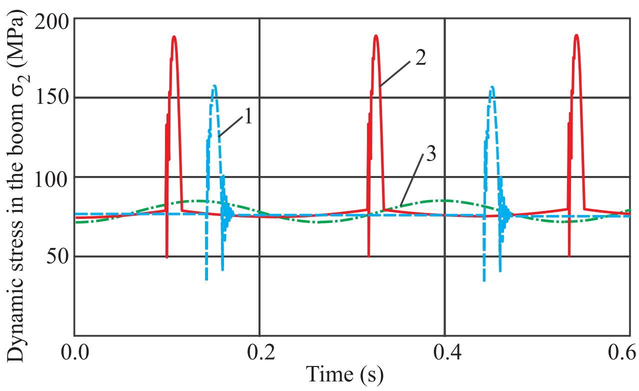

Fig. 6 Time based graph of stresses σ_2 for δ_w = 0.4 mm:
1 - C_d = 0; 2 - C_d = 5 MN/m; 3 - C_d = 20 MN/m

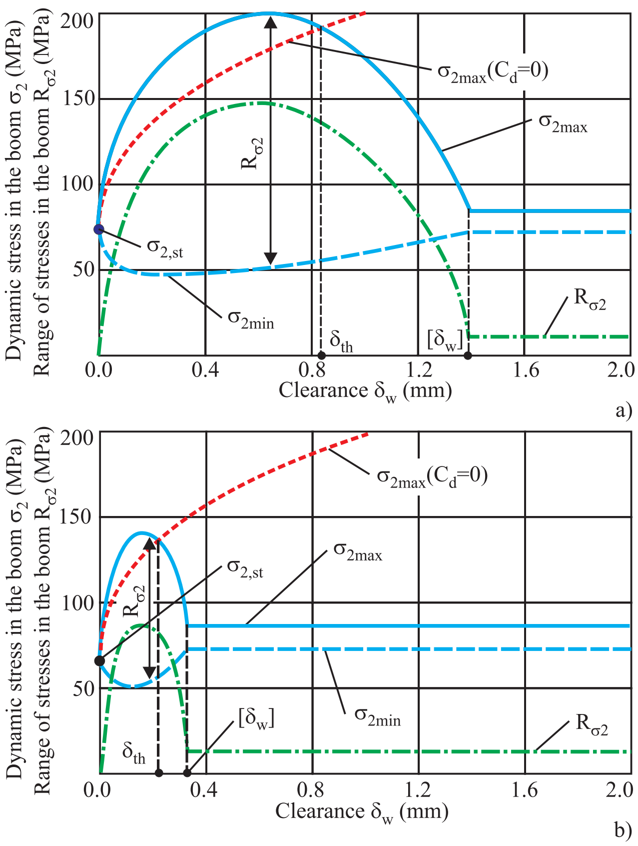

Fig. 7 The influence of clearance δ_w and the damper stiffness C_d on the basic qualitative parameters of dynamic processes occurring in the manipulator boom: a) $C_d = 5$ MN/m; b) $C_d = 20$ MN/m

manipulator is in operation. The magnitude of the greatest possible limit of wear is determined by the clearance $[\delta_w]$, because it is a minimum clearance at which no "hinge pin - lug" impact occurs. Elastic damping elements during their deformation process completely absorb the kinetic energy of the hinge pin during its displacement from its original position in the range of from 0 to $u_{R(L)} = [\delta_w]$. Thus, these elements do not allow the hinge pin to deviate from the reference position by more than the value of $[\delta_w]$. The limiting value of the clearance $[\delta_w]$ is inversely proportional to the damper stiffness. The higher is the stiffness of C_d, the smaller is $[\delta_w]$. This fact is also illustrated by Fig. 8. It shows the dependence of the dynamic coefficient $k_{dyn} = \sigma_{2\max} / \sigma_{2,st}$ on the clearance present in the cylindrical hinge for different values of the damper stiffness.

Thus, when selecting the damper's stiffness (that is, when selecting the dimensions and number of elastic damping elements), it is possible to control the rate of hinge wear during the operation of the manipulators. This makes it possible to eliminate an increase in the clearance values $[\delta_w]$ above the normalized values [3]. When repairing

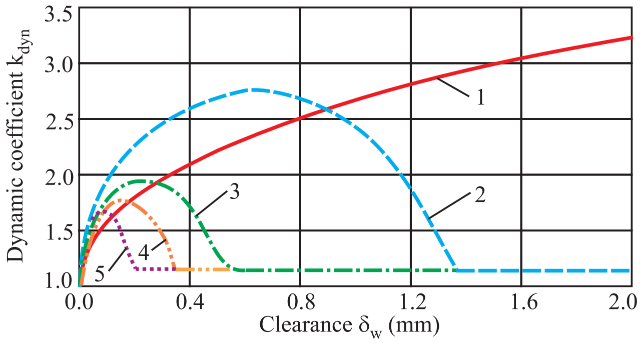

Fig. 8 Dependence of the dynamic coefficient on the clearance in the cylindrical hinge with different values of the damper stiffness: 1 – $C_d = 0$; 2 – $C_d = 5$ MN/m; 3 – $C_d = 10$ MN/m; 4 – $C_d = 20$ MN/m; 5 – $C_d = 40$ MN/m

the operated manipulators it is advisable to install elastic dampers. When selecting the damper stiffness, it is necessary to choose such a value of C_d, so that the value of the clearance limit $[\delta_w]$ can be smaller than the value of the clearance already formed in the hinge. This will significantly improve the dynamic state of the manipulators metal structures due to the elimination of the "hinge pin - lug" impacts. At the same time, the mode of operation $\delta_w > [\delta_w]$ is realized. It is characterized by the minimum values of the maximum dynamic stresses $\sigma_{\max}$, the range of dynamic stresses R_σ and the dynamic factor k_{dvn} (Figs. 7, 8). The value of k_{dvn} is minimal. It is several times smaller than the dynamic factor in the absence of a device damper (curve 1, Fig. 8), and the dynamic factor in the presence of a damper when the clearances are $\delta_w < [\delta_w]$.

5 Conclusion

1. Dampers of the considered design having elastic cushioning elements may provide:
   - a significant reduction in the additional impact loads that occur when the backlash is selected in the hinge with an increased clearance (the dynamic factor with equal clearance values can be reduced up to 3 times);
   - greater safety in the process of transporting fragile and explosive loads;
   - slower growth of increased clearances in the cylindrical hinges of the manipulators due to the braking of the approaching contact surfaces ("hinge pin - lug") before the impact;
   - limiting the clearance in the hinge by a specified value, which will not be exceeded during the lifetime of the manipulator, regardless of the intensity of its loading.

2. The technical and economic result from the use of dampers of the considered design with elastic cushioning elements is associated with an increase in reliability indicators and a coefficient of technical use of manipulators of mobile transportation and production machines and a reduction in financial expenses during their operation.
   The proposed method is the first experience of modeling the dynamic loading of the metal structure of the hydraulic crane-manipulators in the presence of increased gaps in the hinges of links and elastic damping in the hinges. It is advisable to use it for the following design tasks:
   - determination of the upper and lower limits of the range of maximum stresses arising in the metal structure of the crane in the presence of clearances in the cylindrical hinges and the absence of elastic damping;
   - determination of stress, shock acceleration and other characteristics of the dynamic loading of the metal structure of the crane-manipulator on the size of the clearance in the cylindrical hinges;
   - evaluation of the performance of elastic dampers in hinges and the magnitude of the reduction of dynamic stresses in the metal structure of the crane;
   - selection of the necessary rigidity of the elastic damper, based on the required reduction of the maximum values of dynamic stresses in the metal structure of the crane;
   - limitations of the size of the clearance in the hinge joints of the adjacent links of the manipulator, forming in the operation of the crane.

3. Further development of the theoretical studies presented in this article are experimental studies of the behavior of elastic dampers in full-scale operation conditions of real manipulators. The purpose of the planned experimental studies is to confirm the reliability of the developed mathematical model of an elastic damper and to quantify its influence on improving the stress-strain state of metal structures of manipulators. The experimental research program provides for video-fixing of the process of moving the cargo by a manipulator having a hinge with increased clearance, in the presence of an elastic damper and in its absence. The technique of video-fixing and analysis of its results in relation to the manipulators of mobile machines was previously developed by the authors in [19].

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