

Increasing operating life of hinge lugs in manipulators of mobile transportation and production machines

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Abstract. The article deals with the issues of increasing the reliability of hinge lugs in manipulators of mobile transportation and production machines (loader cranes). At present, the actual operating life of hinges in heavily loaded cranes amounts to not more than 3000 operating hours, with the estimated value being 6000...8000 operating hours. The paper provides technical solutions of how to increase the operating life of hinge lugs. The first way consists in providing crack resistance in bimetallic compositions with specified crack resistance which allows increasing the operating life of hinge lugs by up to 50%. The second way involves equipping hinge joints with damper devices having elastic and dissipative elements. The damper devices allow reducing or completely eliminating the impact loads which occur due to the presence of a backlash in a worn-out hinge. The rational values of damper device elasticity and viscosity coefficients, which allow attaining the maximal reduction of impact loads, were determined. To prove the effectiveness of the proposed technical solutions, a mathematical modeling approach to the dynamic processes in loader cranes was employed

1. Introduction

Hydraulically driven multi-boom loader cranes have come into common use as operating devices in mobile transportation and production machines (including mining machines). A trouble spot of such loader cranes is hinge joints. After being in operation for up to 3000 operating hours they come out of action despite their estimated operating time being 6000...8000 operating hours [1]. It is caused by both frictional wear and the influence of increased impact loads, the occurrence of which is conditioned by the backlash in the worn out hinge joints [1, 2, 3]. Thus, increasing the reliability of hinge joints in loader cranes is an acute scientific and technical issue.

There are two ways of increasing the operating life of loader crane lugs with gaps in hinge joints. The first approach consists in increasing their durability by means of implementing new design concepts (changing the lug construction) [2, 3]. It should be noted that the design concepts worked out for manipulating industrial robots are not suitable for loader cranes based on mobile machines due to their peculiar control, manufacture accuracy and size factor [3]. A considerable number of research works are also conditioned by working out technological methods of increasing the mechanical properties of the lug material (quenching, cementation etc.) [2, 4, 5, 6]. These methods are mainly directed at reducing the intensity of the frictional wear rather than at counteracting the impact loads acting on an operating hinge having a backlash. The second approach involves reduction or complete elimination of impact loads which occur in a hinge joint due to backlash [3, 7].



2. Modeling technique of the manipulator dynamics

Eliminating the above-mentioned causes of decreasing the operating life of hinge pins is impossible without the elaborate study of the manipulator dynamics. The main algorithms for solving dynamics problems within the framework of these studies are the Recursive Newton-Euler Algorithm (RNEA), the Composite Rigid Body Algorithm (CRBA) and the Articulated Body Algorithm (ABA) [8, 9, 10].

The RNEA is implemented in the processor unit to solve the inverse dynamics problem. The algorithm has two main phases: the forward recursion (calculation of link velocities and accelerations in order from the base to the end effector) and the inverse recursion (calculation of loads transferred through joints in order from the end effector to the base) [9, 11].

The vector of unknown joint accelerations $\{\ddot{q}\}$ in the forward dynamics problem is obtained from the solution of the system of motion equations [9]:

$$\{\tau\} = [H]\{\ddot{q}\} + \{C\},$$

where the vector of joint actuator forces/torques $\{\tau\}$ is given; the joint space inertia matrix $[H]$ is calculated using CRBA [8, 9]; the joint space bias force vector $\{C\}$ is found using RNEA.

The CRBA is selected over ABA for solving the forward dynamics problem in consequence of a smaller number of computational operations required for the systems with small numbers of degrees of freedom [8, 9, 11, 12].

When it is required to take into account the elasticity of the loader crane sections, the equations based on Lagrange multipliers are employed [13, 14]:

$$[M_q]\{\ddot{q}\} + [C_q]^T\{\lambda\} = \{Q\},$$

where $\{q\}$ stands for the generalized coordinate vector (including the coordinates determining both section movement of the rigid body and its elastic deformations[14]), $[M_q]$ – mass matrix, $[C_q]$ – constraint equation matrix (Jacobian matrix); λ – Lagrange multipliers, $\{Q\}$ – the resultant vector of active and inertial forces [13].

3. Increasing crack resistance of hinge lugs

With the purpose of increasing the operating life, a hinge lug design was proposed (fig. 1). The method proposed consists in thinning hinge lugs crossbars in the zones bearing against the openings by no less than $\delta_g = (2/3 \dots 3/4)\delta_l$ or $2\delta_r = (2/3 \dots 3/4)\delta_l$ (where δ_l stands for hinge lugs nominal thickness), and then in building them up to the nominal thickness by the material with the impact viscosity coefficient higher than the impact viscosity coefficient of the base material. Reduction of the fatigue-cracking rate is achieved by the plane stress condition around the cracks apexes.

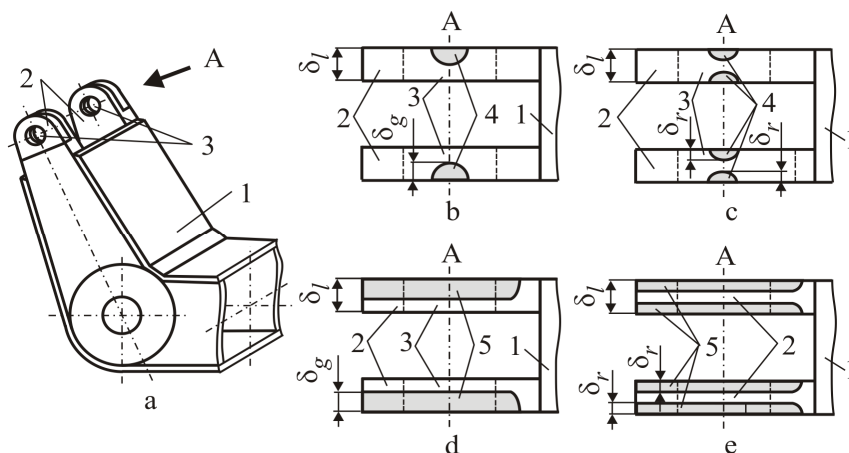


Figure 1. Improved hinge lugs: a – manipulator's section; b, d – grooves proximal to the profile plane; c, e – grooves proximal to both profile planes; 1 – load crane boom section; 2 – hinge lug; 3 – hinge lug opening; 4 – grooves together with the built-up material; 5 – the layer of built-up material

A numerical expression was offered to select the appropriate grade of the built-up material for increasing the operating life of a hinge lug:

$$K_{1c} = K_{1c} \sqrt{\frac{\varepsilon_f E K_{1c}^2}{72(1 - \varphi_p) \delta_l \sigma_T^3}},$$

where K_{1c} stands for the critical coefficient of the material stress intensity in response to the plane deformation; ε_f – the material true failure strain; σ_T , E – yield stress and elasticity modulus; φ_p – groove depth coefficient: $\varphi_p = (2/3 \dots 3/4) \delta_l$.

The dependence for estimating the T_p operating life of its original, and modernized design is written as:

$$T_p = \tau_1 + \frac{1}{C\nu} \int_{a_{th}}^{\varphi[a]} \frac{K_c - K_{max}}{\Delta K^n} da,$$

where τ_1 stands for the duration of the fatigue crack nucleation stage; a_{th} – the fatigue crack threshold length; $[a]$ – the maximum permissible length of the fatigue crack in a hinge lug crossbar; φ – safety coefficient for the maximum permissible crack length; ν – the frequency of the alternate operating load; a – the crack length; K_c – the critical stress intensity factor for the hinge lug material; K_{max} – the stress intensity factor at the crack apex for the maximum cyclic stress; ΔK – stress intensity range at the crack apex; C , n – the empirical factors in the Forman equation for predicting the crack growth rate characterizing the material properties of a hinge lug.

The computation of the operating life of a hinge lug for the three-section hydraulic loader crane AST-4-A shows that the effectiveness of the method proposed consists in improving the operating life coefficient of a hinge lug which amounts to $k = 1,47$.

4. Resistance to the impact forces in hinge units with a backlash

In the perfect cylindrical hinge without a backlash, there is only one degree of freedom q , a.k.a. the main coordinate of the manipulation system. The backlash allows the connected sections to get freely displaced and bent towards one another. Therefore when the boom moves, there is an alternate, almost instantaneous change in the paired attachment points of a hinge pin and hinge lugs accompanied by an impact. To control this effect, a damper device design of a hinge joint of a loader crane was offered (fig. 2). It allows one to reduce (up to the complete elimination) the impact loads caused by the hinge backlash. They are springs or elastic and dissipative elements connecting the adjoining boom sections.

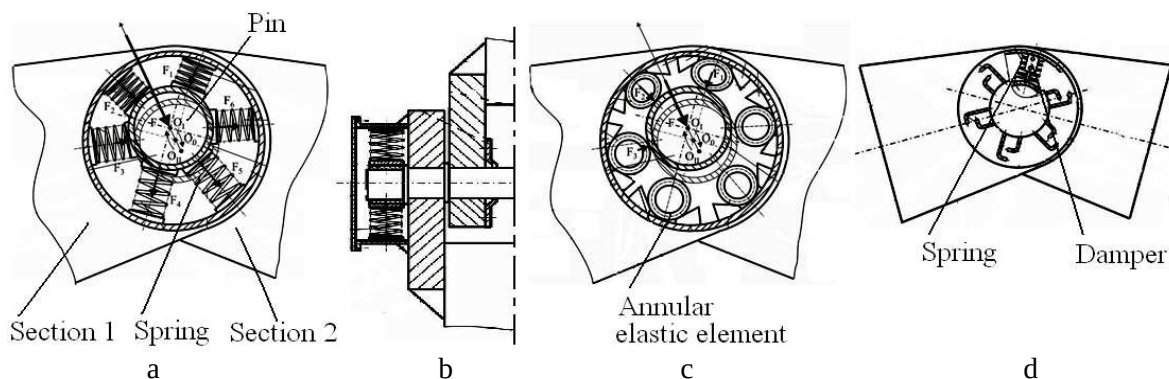


Figure 2. Hinge unit damper devices: a – with springs; b – with springs (side view); c – with annular elastic elements; d – with springs and hydraulic dampers

To design a backlash, the additional linear s_1 and rotational s_2 degree of freedom was introduced to the loader crane hinge joint (fig. 3). Accordingly, the computation algorithm [9, 10] is modified. The system of fictional instantaneous sections is introduced between the backward section of the loader

crane and the successive hinge. The additional degrees of freedom are connected with the abovementioned fictional instantaneous sections z_i [3].

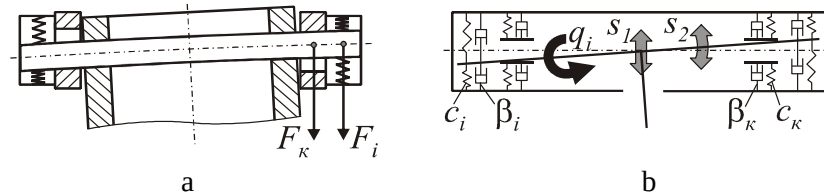


Figure 3. The model of free play hinge with a damper device: a – the hinge operating condition; b – computation scheme

Resistance forces F_1 and F_2 , applied by damping elements are computed using the following formulae:

$$\begin{aligned} F_1 &= -2(c_1 u_1 + \beta_1 \dot{u}_1); \quad F_2 = -2(c_2 u_2 + \beta_2 \dot{u}_2); \\ u_1 &= s_1 - 0,5b \sin s_2; \quad \dot{u}_1 = \dot{s}_1 - 0,5b \dot{s}_2 \cos s_2; \\ u_2 &= s_1 + 0,5b \sin s_2; \quad \dot{u}_2 = \dot{s}_1 + 0,5b \dot{s}_2 \cos s_2; \end{aligned}$$

where u_i , $\dot{u}_i$ stand for the vertical displacement and velocities of the left and right faces of a hinge pin (correspondingly); s_i , $\dot{s}_i$ – generalized coordinates and velocities; b – the distance between the damper device centers; β_i , c_i – reduced damping and stiffness coefficients of the damping elements.

The computation results for the three-section hydraulic loader crane AST-4-A (fig. 4) show that increasing stiffness C and (or) viscosity β of the damper device results in reducing (by up to 40...50 %) the stress in the boom sections and load acceleration. At the same time, the oscillation frequency of a hinge pin increases. At relatively low stiffness factors (up to 5 MPa/m), the dynamic stress increase (by 10...20 %) is possible. The damper device stiffness influences only the oscillatory amplitude, whereas the viscosity increase also results in lagging of the first collision of a hinge pin with the lug surface.

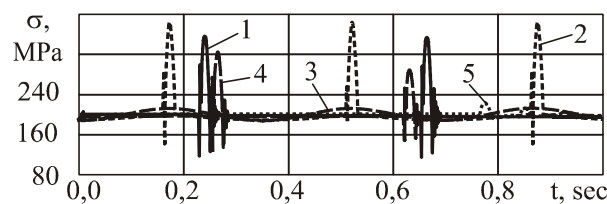


Figure 4. The influence of the damper device parameters on the stress in the manipulator section: 1 – $C=0$, $\beta=0$; 2 – $C=5$ MN/m, $\beta=0$; 3 – $C=20$ MN/m, $\beta=0$; 4 – $C=0$, $\beta=100$ kN·s/m; 5 – $C=0$, $\beta=1000$ kN·s/m

The further study of this effect revealed that the impact stress values depend on the hinge backlash value δ_w . When the backlash exceeds the limit value $[\delta_w]$, the dynamic forces plummet and tend to zero. Depending on the selected values for the damper device stiffness and viscosity coefficients, two operating modes are possible:

1. The impact mode when the contact of a hinge pin with the lugs surface occurs.
2. The non-impact mode when there is no impact contact and no further hinge wear is available.

The damper device has a positive influence on the crane loading when the backlash value exceeds the threshold one δ_{th} . To secure the non-impact mode, it is necessary to fulfil the condition of $\delta_w > [\delta_w]$. The value $[\delta_w]$ increases together with the load mass increase and decreases, when the stiffness and viscosity values grow.

The dependences obtained make it possible to rationally select the damper device parameters in the design process (fig. 5).

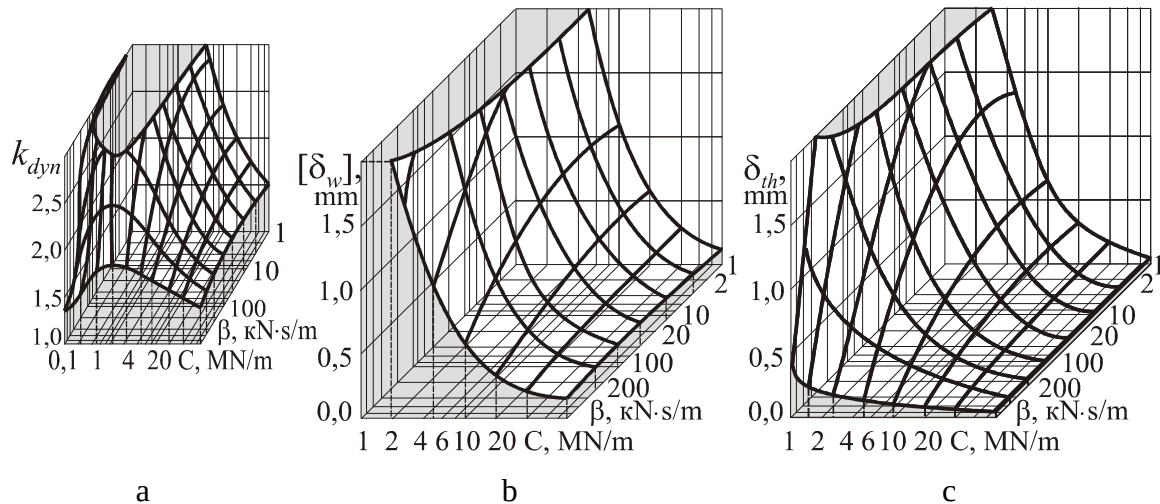


Figure 5. The influence of the damper device stiffness C and viscosity β on the loader crane dynamics: a – dynamic response factor; b – maximum backlash; c – threshold backlash

5. Conclusions and Future Work

The scientific and technical solutions found make it possible to increase the reliability of the loader cranes hinge units.

The bimetallic compositions in the hinge lugs with the specified characteristics of crack resistance allow increasing the operating life of a lug by up to 50%. As a part of the study, dependences for selecting the crack resistance of the built-up material were proposed.

Using the mathematical model worked out for the hinge joints with the excessive backlash caused by the service-induced damage, regular patterns of the dynamic stress and strain state in the loader crane boom were found. The conditions of the effective overload reduction and decreasing the boom sections lateral oscillations based on using elastic and dissipative damping devices were justified. Depending on the damper device stiffness and viscosity coefficients implemented in the design process, two operating modes are possible – the impact mode in which there is an impact contact of a hinge pin in the process of its oscillatory displacement within the backlash present in the hinge with the lug surface; and the non-impact mode in which no impact contact is available. The non-impact mode corresponds to the lowest level of dynamic forces and acceleration impulse which the displaced load undergoes. This mode implementation is connected with increasing the damper device stiffness and viscosity factors, whereby the most significant contribution belongs to the elastic resistance forces which amount to ~85 % of the general reduction of dynamic stress. The viscosity influence is not available within a wide range of values and begins to have a visible effect only at high values (for the MC under study – over 50 kN·s/m).

Further studies will be devoted to summarizing the results obtained and making them applicable for a wide range of manipulators of the mobile transport and production machines, including loader cranes.

6. Acknowledgements

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References

- [1] Shevchenko V P 1986 *The recovery of articulated joints of the timber cutting machines by arc welding metallization* (Khimki: VLSA Publ.)
- [2] Serebryanskiy A I, Gudkov A Yu and Sviridov V G 2013 The transformation of a reversible friction in a unilateral friction in the joints of manipulators *Actual directions of scientific researches* **2**, **2** 326-330
- [3] Lagerev I A 2016 *Working processes simulation for manipulation systems of the mobile multi-purpose transport and technological machines and complexes* (Bryansk: BSU Publ.)
- [4] Antsev V Yu, Tolokonnikov A S, Vorobev A V and Sakalo V I 2017 Methods of determining the margin of cyclic crack resistance of metal structures for hoisting machinery *IOP Conference Series: Materials Science and Engineering* **177** 559-564
- [5] Krasnyi V A and Maksarov V V 2017 Improving wear resistance of friction assemblies of oil-well pumps having seals from directionally reinforced polymer composites *Chemical and Petroleum Engineering* **53(1-2)** 121-125
- [6] Krasnyy V A, Maksarov V V and Olt J 2016 Increase of wear and fretting resistance of mining machinery parts with regular roughness patterns *Annals of DAAAM and Proceedings of the International DAAAM Symposium* **27(1)** 151-156
- [7] Jönsson A, Bathelt J and Broman G 2005 *Proc. IMechE* **219 K 3** 299-305
- [8] Featherstone R 2008 *Rigid body dynamics algorithms* (New York: Springer)
- [9] Lagerev A V, Lagerev I A and Milto A A 2014 Tool for Preliminary Dynamics and Stress Analysis of Articulating Cranes *International Review on Modelling and Simulation* **6**, **4** 644-652
- [10] Lagerev A V, Lagerev I A and Milto A A 2015 Preliminary Dynamics and Stress Analysis of Articulating Non-Telescoping Boom Cranes using Finite Element Method *International Review on Modelling and Simulations* **8**, **2** 223-226
- [11] Luh J Y S, Walker M W and Paul R P C 1980 On-Line Computational Scheme for Mechanical Manipulators *Journal of Dynamic Systems, Measurement, and Control* **102**, **2** 69-76
- [12] Kuznetsov N K, Makhno D E and I A Iov 2017 Damping elastic oscillations of digging mechanism *IOP Conf. Series: Earth and Environmental Science* **87** 022011
- [13] Korkealaakso P, Mikkola A, Eantalainen T and Rouvinen A 2008 Description of Joint Constraints in the Floating Frame of Reference Formulation *Proc. IMechE* **223**, **K** 133-144
- [14] Skibin G M, Shutova M N, Evtushenko S I and Chutchenko I A 2017 Reliability increase of running gears elements of mining traction locomotives using finite-element analysis package *IOP Conf. Series: Earth and Environmental Science* **87** 022021