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OPTIMAL SYNTHESIS OF DRIVE SWING CONNECTIONS OF MOBILE CRANES HYDRAULIC MANIPULATING SYSTEMS

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The design, operation features and the optimal design method for the drive swing connections are considered. These are proposed as an alternative design variant of the articulation of adjacent links of hydraulic manipulation systems of mobile boom cranes are considered. The design of the device allows to combine two functions: to ensure a continuous, uninterrupted kinematic connection between adjacent links of the manipulation system by creating a cylindrical hinge for performing the reciprocating relative motion of adjacent links and to form an integrated rotary hydraulic gate type hydraulic motor. A mathematical model is developed, the problem of the optimal synthesis of the drive joint on the basis of minimization of the device mass is formulated and solved, while observing the nonlinear system of design, installation, power, strength and kinematic constraints. Analysis of the results of the performed optimization calculations showed that the mass of the optimal version of the device increases with the growth of the overcome moment from the moved cargo and decreases with the increase in the number of chambers. The operating pressure of the hydraulic system does not have a practical effect on the optimum mass, which allows the pump unit of lower power to be used to provide the movement of the links of the manipulation system. Optimum values of the main design dimensions of the drive articulation are determined both by the value of the operational load and by the installation conditions of the device taking into account the dimensions of the cross sections of adjacent links of the manipulation system. When designing manipulation systems, the swing joints allow to abandon remote power hydromotors, eliminate operational failures due to wear and fatigue failure of hinge elements, and also increase the energy efficiency of loader cranes by transferring the hydraulic system to lower operating pressure levels while maintaining the required load-altitude characteristics.

***Key words*:** manipulation system; mobile crane; drive swing connection; hydraulic drive; optimization; synthesis; weight

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Introduction. Hydraulic manipulation systems (boom cranes) installed on mobile transport-technological machines for various purposes, have become widespread in many sectors of the economy – industrial production, mining, gas and oil production, construction, etc. [1]. In Russia, they are a very in demand type of lifting equipment [7]. On the world market, the designs of such producing countries as South Korea, Japan, China, Germany, Italy, the USA, Austria, Russia, etc. are presented mainly [2].

The load bearing structure of the manipulator system of the boom crane consists of 3 to 12 series connected moving links which are in pairs by cylindrical or prismatic hinges [3]. Structurally cylindrical hinges are lugs fixed at the ends of adjacent links, having coaxial holes for mounting the hinge pin [7, 20]. Such a design provides only non-discontinuity of the kinematic chain of the manipulation system. To perform the links of the relative movement, it is necessary to install power hydraulic cylinders, which are fixed by means of pairwise located eyes to the surface of adjacent links [22].

Formulation of the problem. As the cylindrical swing connections are used, the gaps increase due to wear and crushing of their contacting surfaces between the hinge pins and the surfaces of the tower eye holes [7, 11, 12]. This leads to an increase in the stress level in the crane's metal structure to two or more times and a linear acceleration of the movement of the load up to 20 times [16, 17]. The use of intermediate anti-friction sleeves [3], polymer coatings [8, 14] or various hardening methods [15] can slow down, but not exclude the development of negative processes [11]. The presence of remote hydraulic motors of the movement reduces the carrying capacity [7] and the volume of the working area of the manipulator crane [3]. The elements of the fastening points of power hydromotors to the links are subject to failures due to the occurrence of fatigue cracks in them [7, 12].

An alternative with respect to the traditional design of articulated joints of the manipulation systems of mobile cranes can be the design of drive swing connections [10].

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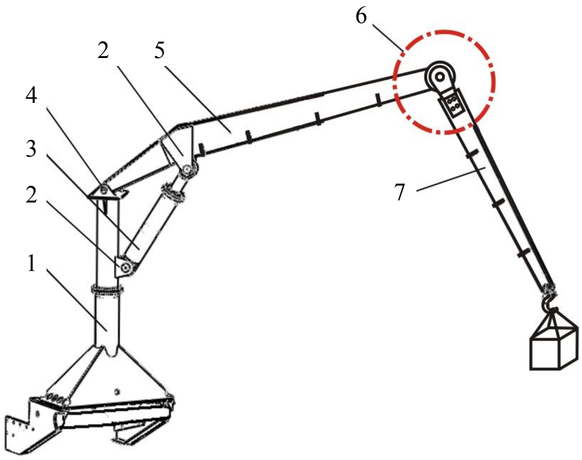

Fig.1. Three-link manipulation system of a mobile boom crane with a drive swing connection

The design and operation principle of the drive joints. Appearance of the manipulation system of the boom crane with the traditional and drive swing connections of adjacent links is shown in Fig.1. The articulation 4 of the column 1 and the boom 5 has a traditional design with the installation of a power cylinder 3 with the help of fastening elements 2 to adjacent links. The drive swing connection 6 of the boom 5 and the handle 7 provides a swivel motion of the handle relative to the boom, without requiring the installation of a remote hydraulic cylinder.

Figure 2 shows the design of the drive joint [10] and Fig.3 shows its design. Structurally, the drive articulation consists of a body 5, from both ends of the hermetically sealed removable covers 13. The body is fixedly fixed to the end of one of the adjacent links of the manipulation system. Inside the housing, a rotor 4 is located in the support bearings of the rolling 12. The involute splines for connecting the rotor with the clamps 10 are cut from both sides. The other ends of the clamps are secured with the help of the pins 9 on the side surfaces of the second adjacent link. Inside the body, three, four or five partitions 11 are fixedly fixed at an equal angular distance α. At the same distance α, an equal number of blades 2 is fixed on the surface of the sleeve 3. For example, as a result of the installation of three partitions, six chambers are formed: U_1, U_2, U_3 and V_1, V_2, V_3. Each of them is connected by an individual pipeline to the hydraulic system of the manipulator crane, the pipelines of the chambers U_1, U_2, U_3 are lifting 6, and the chambers V_1, V_2, V_3 – lowering pipelines 7. The sealing of the mating surfaces of the parts is achieved by a complex approach based on both modern sealing devices, and to improve the accuracy of their machining with the help of vibration-resistant tool systems [9, 18].

Thus, the function of the hinged connection of adjacent links is realized by the body, end caps, rotor and clamps. The contacting of these elements ensures a permanent inseparable kinematic between adjacent links and forms a cylindrical hinge. The function of the power cylinder is

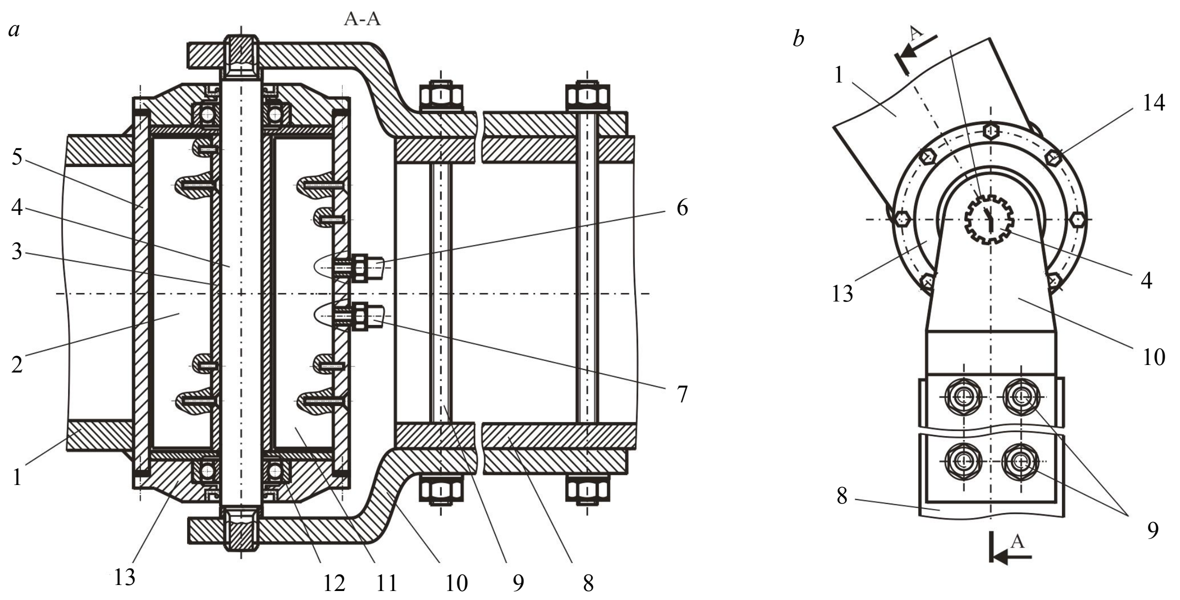

Fig.2. The design of the drive swing connection: *a* – longitudinal section; *b* – side view

realized with a casing with lifting and lowering pipelines fixed to it, end caps, partitions, blades, rotor and sealing elements. They form an integrated hydraulic motor, similar in principle to the action of the swivel rotary hydraulic motors [4, 13, 21].

The working fluid under high pressure p_1 enters the chambers U_1, U_2, U_3 when connecting the lifting pipelines to the pressure hydraulic line of the hydraulic system of the manipulator crane. Simultaneously the lowering pipelines of the chambers V_1, V_2, V_3 connect to a low-pressure drain line p_2. As a result of the pressure difference $\Delta p = p_1 - p_2$ working fluid in adjacent cells $U_1 - V_1$, $U_1 - V_1, U_2 - V_2, U_3 - V_3$, the forces R_1, R_2, R_3 appear. They create the total torque relative to the longitudinal axis of the rotor (point O)

$$T_t = R_1 h + R_2 h + R_3 h = n_h \Delta p A h,$$

where A – blade surface; h – shoulder action on the resultant force with respect to the longitudinal axis of the rotor; n_h – number of blades.

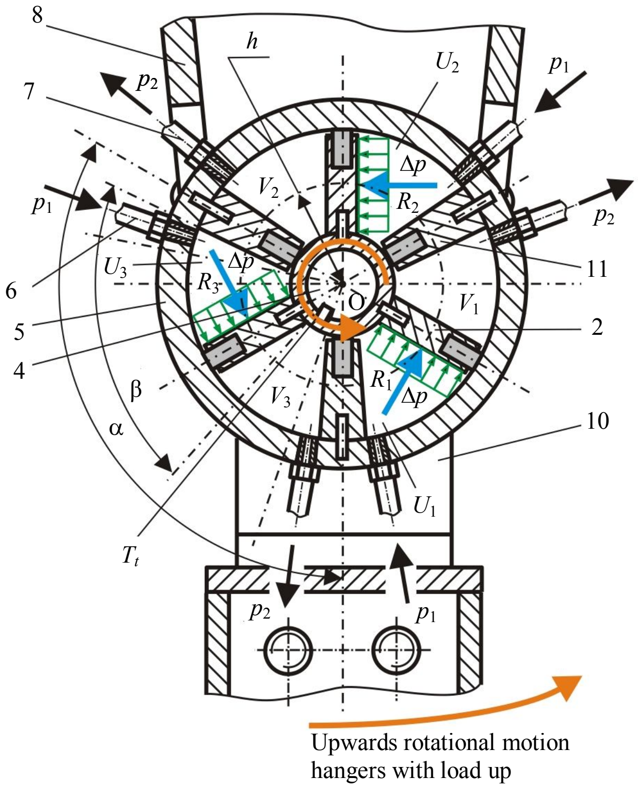

Fig.3. The design scheme of the six-dimensional drive swing connection

The maximum rotation angle of the handle is determined by the greatest possible angle of rotation of the blade between two adjacent partitions and approximately is $\beta \approx (0.70\text{-}85)\alpha$.

Optimal synthesis of drive joints statement of the problem. The mass of the elements of the drive swing connection increases the mass of mobile components of the manipulation system. Therefore, when designing a drive joint, it is important to ensure a minimum contribution of its elements to the total mass of the metal structure of the articulated manipulation system of the crane.

The mass of the drive swing connections consists of the body masses, end caps, partitions, blades, rotor, bushings, rolling bearings, connecting pins and filling chamber of the working fluid:

$$M \approx \pi\rho_b (D + s_w) s_w H + 0.5\pi\rho_l [(D + 2s_w)^2 - d_s^2] s_l + 0.5\rho_v [0.5\pi(d_v - d_s)^2 + n_h (D - d_v) h_v H] +$$
$$+ 0.25\pi\rho_s d_s^2 [H + 2(s_l + s_{bt})] + 2M_{br,1}(d_s) + 0.5 n_h \rho_d (D - d_v) h_d H + \rho_{bt} \frac{n_T T_t [H + 2(s_{bt} + d_{bt})]}{2[\tau]_s h_{bt}} +$$
$$+ 0.5\rho_{hf} (D - d_v)[0.5\pi(D + d_v) - n_h (h_d + h_v)] H, \quad (1)$$

where ρ_b, ρ_l, ρ_d, ρ_v, ρ_s, ρ_{bt}, ρ_{hf} – density of the body material, end caps, partitions, blades, bushing, rotor, pins and working fluid; D, H, s_w – inner diameter, length and thickness of the housing wall; s_l – thickness of the end cover; d_v, d_s, d_{bt} – diameters of bushing, rotor and studs; h_d, h_v – thickness of the baffle and blade; s_{bt} – clamps; h_{bt} – distance from the axial line of the cross-section of the final link to the center of the cross section of the hairpin**;** n_T –torque suppression margin ratio (n_T>1); $M_{br,1}(d_s)$ – bearing weight, determined by its size.

According to the relationship (1), the mass of the drive swivel joint under the operating load (torque T_t and working pressure p_1) depends on 11 design parameters: D, H, s_w, s_l, s_{bt}, d_v, d_s, d_{bt}, h_d, h_v, n_h. Taking into account the recommendations [6], two of these parameters should be considered as independent parameters and determining the dimensions of the entire device – internal diameter

D and body length H. As an independent parameter, it is also advisable to consider the diameter of the rotor d_s, Since it allows you to determine the structural dimensions of the connecting elements of the device with the end link. The remaining dimensions are either fixed, since they are determined by the design of the swing connection and the value of the operational load (s_{bt}, d_{bt}, n_h), or are uniquely determined depending on D, H и d_s (s_w, s_l, d_v, h_d, h_v). Number of blades of the gate n_h it is convenient to set it as the initial data for each optimization calculation, assigning successively its possible values from a complete set of values $n_h = 3, 4, 5$.

Thus, in formulating the problem of optimal synthesis of a drive swing connection as a vector of variables (variable) optimization parameters, one should use the internal diameter, the length of the hull and the diameter of the rotor, forming them a vector of unknown

$$\{x\}^T = \{x_1\ x_2\ x_3\} = \{D\ H\ d_s\}.$$

The remaining parameters are uncontrollable (fixed), from which a vector is formed

$$\{z\}^T = \{z_1\ z_2\ z_3\ z_4\ z_5\ z_6\ z_7\} = \{z\}^T = \{z_1\ z_2\ z_3\ z_4\ z_5\ z_6\ z_7\}\ .$$

Finally, the problem of optimizing the hydraulic drive joint of the adjacent links of the manipulation system reduces to minimizing the objective function of the form

$$F(\{x\},\{z\}) = \pi\rho_b(x_1+z_1)z_1x_2 + 0.5\pi\rho_l[(x_1+2z_1)^2 - x_3^2]z_2 + 0.5n_h\rho_d(x_1-z_4)z_5x_2 +$$

$$+\rho_{bt}\frac{n_T T_t[x_2+2(z_3+z_7)]}{2[\tau]_s h_{bt}} + 0.5\rho_v[0.5\pi(z_4-x_3)^2 + n_h(x_1-z_4)z_6x_2] +$$

$$+0.25\pi\rho_s x_3^2[x_2+2(z_2+z_3)] + 2M_{br,1}(x_3) + 0.5\rho_{hf}(x_1-z_4)[0.5\pi(x_1+z_4) - n_h(z_5+z_6)]x_2 \qquad (2)$$

under the following restrictions:

• by constructive conditions

$$x_1 > 0;\quad x_2 > 0;\quad x_3 > 0;$$

$$(x_1+2z_1) - 0.8H_A \ge 0;\quad 1.5H_A - (x_1+2z_1) \ge 0;$$

$$1.3B_A - x_2 \ge 0;\quad 1.5B_A - (x_2+2z_2) \ge 0;\quad x_1 - z_4 \ge 0;$$

$$(x_2+2z_2) - B_A \ge 0; \qquad (3)$$

$$z_3 - 0.8s_B \ge 0; \qquad (4)$$

$$2s_B - z_3 \ge 0; \qquad (5)$$

$$z_1 - s_A \ge 0; \qquad (6)$$

• by condition of avoiding the collapsing of the working surfaces of the keys of the coupling of the rotor with the hub

$$[\sigma]_{br,k} - 3.33\frac{n_T T_t}{x_3^2(z_4-x_3)} \ge 0;$$

• by condition of providing the maximum angle of relative rotation of the end link

$$\frac{2\pi}{n_h} - \frac{(z_5+z_6)n_h}{x_1} - [\varphi] \ge 0;$$

• by traction condition

$$0{,}125\Delta p n_h x_2 (x_1^2 - z_4^2) - n_T T_t \geq 0 ; \qquad (7)$$

• by condition for limiting the angular acceleration of the rotor during acceleration

$$0{,}125\Delta p n_h x_2 (x_1^2 - z_4^2) - T_t - J_и [\ddot{\varphi}] \geq 0 ;$$

• by condition for ensuring the torsional strength of the splined section of the rotor

$$x_3 - 2\sqrt[3]{\frac{2K_{\tau s} n_T T_t}{\pi [\tau]_t}} \geq 0 ;$$

• by condition for ensuring the crushing strength of the working surface of the involute splines of the rotor

$$x_3 - 3{,}16\sqrt{\frac{n_T T_t}{\psi\, z_3 \sigma_{us}}} - m_g \geq 0 ;$$

• by condition for ensuring the fatigue strength of the root section of the blade

$$\frac{\sigma_{-1}}{K} - \frac{3\Delta p (x_1 - z_4)^2}{4 z_6^2} \geq 0 ,$$

where H_A, B_A, s_A – overall height and width, wall thickness of the end section of the link to which the body is fixed; s_B – wall thickness of the end section of the end link; $[\sigma]_{br,k}$ – allowable crush stress of key material; $[\varphi]$, $[\ddot{\varphi}]$ – maximum angle of rotation and angular acceleration in the acceleration of the final link, regulated by the technical design for the design of the crane manipulator; $J_и$ – the moment of inertia of the rotating masses, reduced to the rotor of the joint; $K_{\tau s}$ – coefficient of tangential stress concentration at torsion of splined shaft [20]; $[\tau]_t$, σ_{us} – permissible shear stress and tensile strength of the rotor material; ψ – coefficient of accounting for the uneven distribution of the load between the splines and along the splines [5]; m_g – coefficient of accounting for the uneven distribution of the load between the splines and along the splines; K – coefficient of accounting for the uneven distribution of the load between the splines and along the splines.

Results and its discussion. The proposed mathematical model of the optimal synthesis of the drive hinged joint was implemented with reference to the design of the three-link articulated-articulated manipulation system of the mobile power machine AST-4-A for welding of main pipelines [7]. Its appearance corresponds to Fig.1. The main technical characteristics of the manipulation system: a nominal load capacity of 7.5 kN; torque T_t = 18 kNm; hydraulic operating pressure rating p_1 = 16 MPa; he length of the handle is 2.4 m; length of the boom is 3.6 m; maximum handle angle is 50°.

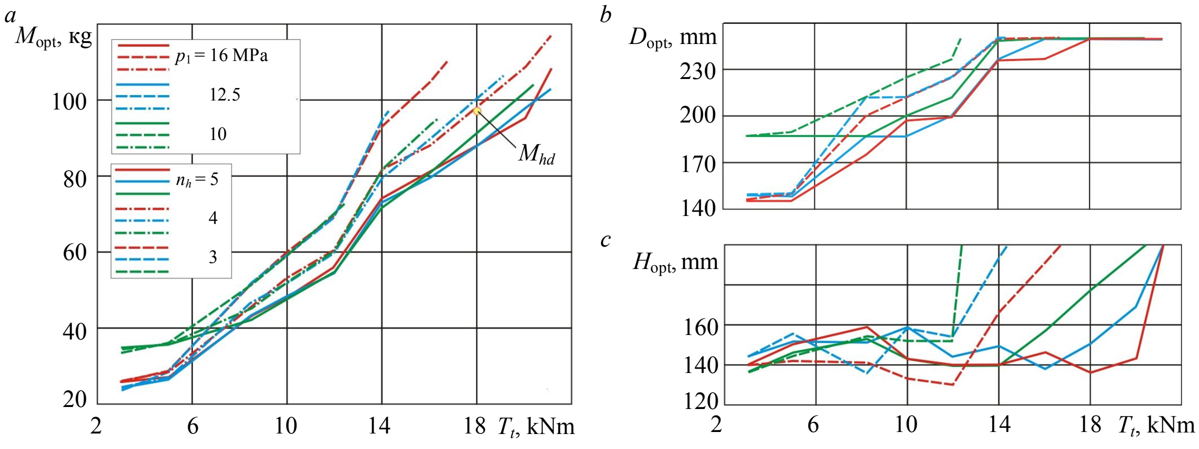

Fig.4. Optimal values of the main characteristics of the drive articulation: *a* – mass; *b* – inside diameter of the body; *c* – length of the case

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Figure 4 shows the results of the optimal synthesis of the swing connection of the hinge and boom of the manipulator crane in the torque interval T_t = 3-22 кNm at three standardized working pressure levels of the hydraulic system p_1 = 10; 12.5 и 16 MPa. Higher values p_1 were not considered, since according to the data of [4, 6, 22] at such pressures it is difficult to ensure reliable sealing of the internal cavities of the device.

Weight of the optimal version of the swivel joint M_{opt} approximately increases linearly with increasing torque T_t (Fig.4, *a*). With an equal value T_t an increase in the number of chambers (an increase in the number of blades n_h) allows it to be reduced. The following regularity is noticeable: the working pressure p_1 has a very small effect on M_{opt} (values M_{opt} for p_1 = 10-16 MPa lie in a narrow range of relative values ±3 %). This circumstance has a significant practical significance, as it makes it possible to use a lower working pressure in the hydraulic system of the loader crane and, accordingly, a pumping unit of lower power.

With reference to the example in question, a transition from a nominal working pressure of 16 MPa to a lowered working pressure of 12.5 or 10 MPa is possible. This significantly increases the energy efficiency of the hydraulic drive, since the power of the installed pumping unit can be reduced approximately 1.3 or 1.6 times, respectively, without deteriorating the basic technical characteristics of the manipulation system. Reducing the working pressure of the hydraulic system also has a positive effect on increasing the efficiency of the hydraulic drive of the manipulator crane and reducing the operating costs for its maintenance. In Fig.4, and also the mass of the power cylinder, the elements of its fastening and the articulated joint for the standard structure of the machine AST-4-A (point M_{hd}) is indicated, which is removed upon transition to the use of the drive articulation. It can be seen that the use of a ten-chamber connection (n_h = 5) makes it possible to obtain mass gain in the range of ~ 9 kg or ~ 9 %. The use of the eight-chamber connection (n_h = 4), on the contrary, leads to an increase in the mass of the manipulation system, although very small – in the range of 1-3 kg or 1-3 %.

Optimum values of the basic design dimensions of the drive joint D_{opt} and H_{opt} (Fig.4, *b*, *c*), depending on the change in the torque T_t, the working pressure p_1 and the number of the cameras of the device n_h vary quite differently. The general trend is that with increasing T_t these sizes vary from the minimum to the maximum values. In particular, the internal diameter of the body varies from 146 to 250 mm, the length of the body from 133 to 196 mm. The noted feature of the results of the optimal synthesis is due to the need to integrate the drive joint into the designed structure of the manipulation system, since its design dimensions are determined in priority by the dimensions of the cross sections of adjacent links, based on the conditions of their strength and rigidity.

The height of the cross section of the mating links limits the limiting (the smallest and largest) values of the body diameter, and the width of the cross section is the limiting values of the length of the hinged joint body. This is confirmed by the analysis of the location of the points of the global minimum of the objective function $F(\{x\},\{z\})$ in the admissible region of the space of controllable parameters $\{x\}$, which is established by the system of accepted constraints of the optimization problem.

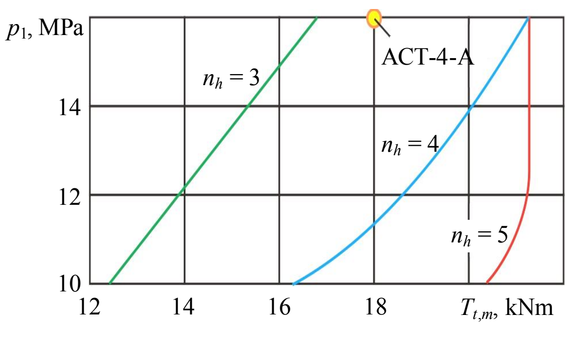

Fig.5. The limiting torque developed by the drive swing connection

At small values of overcome torque T_t < 6-9 kNm points of the global minimum lie mainly on the structural constraint (3), less often on the structural constraints (3) and (6) or (3) and (4). At large values T_t they lie on the constraint (7). The optimal version of the device is determined by the condition of providing traction of the drive articulation. At the limiting torque T_t, m, developed by the device, the global minimum points can also lie on the constructive constraint (5).

The values of the limiting torques $T_{t,m}$ depending on the torque T_t being overcome and the working pressure of the hydraulic system p_1 are shown in Fig.5.

Here, the point corresponding to the characteristics of the initial AST-4-A machine is indicated. It can be seen that when replacing the standard structure of a swivel joint with a swivel joint, a ten-chamber device (n_h = 5) at any values of the operating pressure of the hydraulic system p1 or an eight-chamber device (n_h = 4) at an operating pressure p_1 = 12.5 or 16 MPa. Using the Six-Channel Device (n_h = 3) it is impossible, since it does not provide the necessary torque T_t = 18 kNm (the limiting torque at 16 MPa is T_t, m = 16.75 kNm).

Conclusion. When designing mobile hydraulic cranes, the considered articulated joints of adjacent links of articulated-articulated manipulation systems can be considered as an alternative to traditional hinge joint designs with remote power hydromotors.

While providing equal cargo-altitude characteristics of the mobile crane manipulator, the drive swing connection allows to exclude a number of operational disadvantages of the conventional hinged joint: the development over time of the additional dynamic loading of the metal structure due to the increased gaps in the joint, the reduction of the working area of the crane due to the presence of remote power hydraulic motors, the appearance of failures due to fatigue failure of the elements of the fastening units of power hydromotors to the links It is possible to translate the hydraulic system to lower levels of working pressure, which leads to an increase in the energy efficiency of the manipulator crane and the efficiency of the hydraulic drive as a whole, as well as to a reduction in operating costs for maintenance.

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