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Testing of hydrodynamic drive of cranes mechanisms

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Abstract. The results of researches on the cranes (bridge, tower and portal) with hydrodynamic drives of travel and slewing mechanisms are given. Recommendations for a significant increase hydraulic drives elements limit number of fatigue cycles in relation to a drive with a phase rotor are given.

Key words: damages, load on cranes steel structure, hydrodynamic drives, fatigue of crane units, oscillograms of tests.

1. Introduction

The Department of LTM and E NTU "KhPI" has accumulated rich experience in scientific research and implementation of hydrodynamic drives. Among the researches of hydraulic drives can be mentioned the works of Losev P.G, Haidamaka V.F., Zhermunsky B.I., Pashkina S.A., Grigorov O.V., Petrenko N.O., Vyshnevetsky G.V., Dudnik V.A. [1-7].
During these studies, it was discovered by using of strain gauging and records on the oscilloscope that these drives provide a significant reduction in dynamic loads. The following hydrodynamic drives were investigated:
- slewing mechanism drive of the tower crane on a support with the 5 t capacity, Kharkiv, Ukraine;

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- hoisting mechanism drive;
- drive of the 30/5 t crane trolley travelling mechanism (design of NTU "KhPI");
- slewing mechanism drive of the tower crane KB 405.2 with the 9.3 t capacity (design of NTU "KhPI");
- separate drive of the travel mechanism of the bridge crane with the 20/5 t capacity (design of NTU "KhPI");
- tower crane KBM-401 P (design of NTU "KhPI") slewing mechanism drive (dual);
- travel mechanism drive of the tower crane KB 405.2 (4 drives, design of NTU "KhPI");
- tower crane «Ganz» slewing mechanism drive (design of NTU "KhPI");

In this work, methods of analytical mechanics, numerical modeling, planning and organization of the experiment are used.

2. Testing of hydrodynamic drive of cranes mechanisms and comparison results with the theoretical modelling

The objects of research are analyzed with taking into account the complex combination of mutual-influencing electro-hydro-mechanical processes, and an important feature is the pronounced multi-mass system.

We explain what is meant on the example of the tower cranes slewing mechanism (the corresponding equivalent schema is shown in Fig. 1).

In this case, a six-mass equivalent system is acceptable, which allows to take into account the torsional deflection of the tower, the bending of the jib, the deflection of the cargo. Masses of the engine, hydraulic coupling, platform, jib and the cargo are taken into account independently. The rigidity of individual elements of the steel structure and the resistance to movement from the forces of friction are also take into account.

When the actuator is switched on, the engine and the clutch pump wheel start to rotate. Therefore, the first mass J_P includes the moment of inertia of the engine and the pumping part of the hydraulic clutch (pump wheel and all rigidly coupled elements of the clutch). The rotation is transmitted to the turbine wheel, which is rigidly coupled to the gearbox because of the fluid filling the hydraulic clutch. The gearbox output shaft transmits significant torques. For the analysis of these phenomena the second concentrated mass J_1 includes the moments of inertia of the turbine part of the coupling and gearbox, reduced to the axis of rotation of the crane according to the general rules. Since the working fluid fills two clutch impellers simultaneously, its volume is divided into two components in the ratio÷ 0,4/0,6 (wheel/turbine wheel volumes). This ratio is caused by the difference in the angular velocity of the wheel/turbine wheels. When rotating the crane, the tower of long length is twisted.

The tower is seen as a fixed-end beam with masses centered at its attachment point and head.

The jib is modeled by two masses centered at the mounting point and at the head.

The third concentrated mass J_2 includes the moments of inertia of the slewing platform and part of the tower; the fourth J_3 includes the moments of inertia of the part of the tower and jib head; the fifth m_2 includes the mass of the jib. There are elastic bonds between them. The rigidity of the tower and the jib are determined by the method of splitting the structure into a series of simplest elements and summing the displacements from deformations.

To calculate the deflection of the load from the suspension plane, a sixth mass m_1 is introduced, corresponding to the mass of the lifted load and the gripping means.

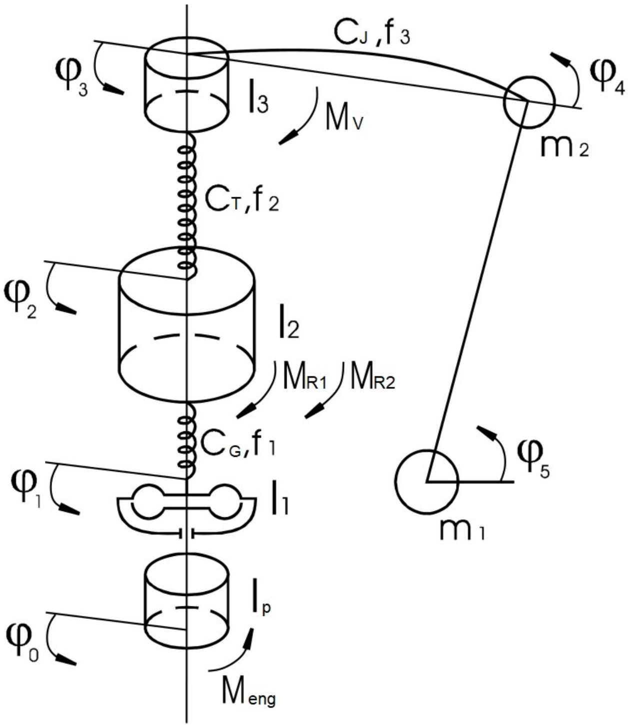


Fig. 1. Equivalent schema of the crane.

φ - reduced to the axis of rotation of the crane movement of the respective links; J_P, J_1, J_2, J_3 - moments of inertia of the rotor of the electric motor and pumping part of the hydraulic clutch, turbine part of the hydraulic clutch and gearbox, rotary platform and part of the tower, tower and jib, related to the tower head; m_1, m_2 - the weight of the load and the weight of the jib related to the jib head; C_G, C_T - reduced to the axis of rotation of the crane torsional rigidity respectively gear box and tower; C_J - reduced rigidity to the bend of the jib from the plane of suspension of cargo; f_1, f_2, f_3 - coefficients of inelastic losses in the links; M_{R1}, M_{R2} - moments of resistance from friction forces in the transmission and the supporting-rotary circle.

During the turn, there are bending strain of the tower and jib, the deflection of the load in the plane of suspension, etc., which due to the small importance of these processes in the conditions of this task are neglected.
Movement is described by a system of differential equations, based on the Lagrange equation:

$$J_R\ddot{\varphi}_0 = M_{eng} - M_{hc};$$

$$J_1\ddot{\varphi}_1 = M_{eng}l - M_{R1}sign\dot{\varphi}_1 - M_{br} - C_G(\varphi_1 - \varphi_2) - f_1(\dot{\varphi}_1 - \dot{\varphi}_2);$$

$$J_2\ddot{\varphi}_2 = C_G(\varphi_1 - \varphi_2) + f_1(\dot{\varphi}_1 - \dot{\varphi}_2) + C_{br}(\varphi_3 - \varphi_2) + f_2(\dot{\varphi}_3 - \dot{\varphi}_2) - M_{R2}sign\dot{\varphi}_2;$$

$$J_3\ddot{\varphi}_3 = C_T(\varphi_2 - \varphi_3) + f_2(\dot{\varphi}_2 - \dot{\varphi}_3) + C_JR^2(\varphi_4 - \varphi_3) + f_3(\dot{\varphi}_4 - \dot{\varphi}_3);$$

$$m_2\ddot{\varphi}_4 = C_JR^2(\varphi_3 - \varphi_4) + f_3(\dot{\varphi}_3 - \dot{\varphi}_4) + \frac{gm_1R^2}{l}(\varphi_5 - \varphi_4) + M_VB;$$

$$\ddot{\varphi} = \frac{g}{l}(\varphi_4 - \varphi_5).$$

Here $\varphi_1(i = 0........5)$ - displacement angles of the corresponding masses (see Fig. 1); M_{eng} - the engine moment described by the Kloss equation; M_{HC} - the moment transmitted by the hydraulic coupling; M_{R1}, M_{R2} - reduced to the axis of rotation of the crane the moments of resistance in the gearbox and the supporting and rotating wheel; $f_i(j = 1, 2, 3)$ - coefficient of inelastic loss of the corresponding link; M_{br} - the summed braking moment taking into account the electromagnetic inertia of the windings (begins to act upon reaching a certain angle by the crane) - is determined by the equation.

$$M_{br}sign\dot{\varphi}_1 = T_1\dot{M}_{br} + M_{br},$$

where: T_1 – electromagnetic time constant of the brake actuator, s;
C_{tr} - transmission rigidity; H·m/rad; R - jib length; M_V - moment of wind load; B - operator characterizing the direction of wind load;

$$B = (\cos(\varphi_1 + \varphi_2)(\cos(\varphi_4 + \varphi_3),$$

where φ_3 is the phase of wind load; T_T - time constant of the electromagnetic inertia of the windings of the brakes: ($T_T = (0,3 \div 0,5) \cdot 2\ \pi \cdot \phi$, where ϕ - frequency of the electrical network, $\phi = 50$ Hz).
The hydro coupling torque M_{HC}, with taking into account the hydrodynamic inertia of the flow formation, related to the $M_{HC\ st.}$ torque by static characteristic

$$\dot{M}_{HC.st} = \frac{1}{a}\dot{M}_{HC} + M_{HC}.$$

Note that a is the coefficient that takes into account the hydrodynamic inertia of the flow formation. It characterizes the intensity of the increasing of the torque transmitted by the hydraulic coupling and depends on the filling. For hydraulic couplings of the test size in case of nominal filling

$$a = e^{0,667\ln t}.$$

When stopping the crane, $M_{R1} = M_{R2} = 0$. On the other hand, the system must not accelerate until the driving torque exceeds the total resistance moment:

$$M_{eng}\, i < C_{tr}\left(\varphi_1 - \varphi_2\right),$$

where i is the transmission ratio;

$$C_{tr}\left(\varphi_1 - \varphi_2\right) < C_T\left(\varphi_2 - \varphi_3\right) .$$

Fig. 2 shows an example of solving the above system of equations for crane KB 405: changes in the speeds of the pump 1 and turbine 2 – parts of the hydraulic clutch, jib 3, load 4, the torque of the clutch 5, and also the path of the load 6 and the jib 7 as a function of time. From the analysis of fig. 2 a, it follows that the speed of the pumping part and the torque of the hydraulic clutch increase sharply during the launch. The speed of the turbine part changes smoothly and reaches its nominal value in 6...8 s. The pits in the curves of the initial launch period is due to the dynamic loading.

The movement of the jib and the load begins after some time, which is determined by the inertia of the system. The speed of the load, unlike jib speed, is described by a smooth line without dips. This is due to the flexible suspension of the load. The load acts on the boom with delay in relation to movement. Then after 5...6 s the curves of the jib and load paths merge. When braking there is a slight deviation of the load.

Fig. 2, b shows the changes of moments at the output of gearbox 1, at the root of tower 2, at the jib base 3. The gearbox output moment in the initial period corresponds to 120...140 kN·m, the moments of the tower and jib – 90...120 kN·m, which is 10...15 % less than in the electromechanical drive. After 3...4 s the dynamic loads drop and the oscillations quickly damp down.

We are aware of the complexity of the presented model, which is associated primarily with a significant number of its degrees of freedom. In this regard, it is appropriate to note the importance of the systematic experimental and design work carried out at the department on the subject under discussion.

In particular, at the request of the St. Petersburg’s Central design bureau (CDB) of tower cranes a hydrodynamic drive of slewing mechanism for the tower crane KB 405.2 (Fig. 3) was created and implemented. Fig. 4 shows the test equipment.

Fig. 5 represents typical oscillograms of the transient modes in hydrodynamic and electromechanical drives. The analysis shows that for the hydrodynamic drive of the slewing mechanism of the tower crane KB 405.2A, the dynamic loads in the gearbox, tower, and the crane jib are lower than in the electromechanical by 10...20 %, in addition, suppression of their oscillation are much faster; the deviation of the rope from the vertical position is in 1.4 times less. The average duration of the working cycle is in 18 % lower due to the lack of time spent on damping the load when it directed to the target. With a hydrodynamic drive due to better handling and reducing the swinging of cargo, placing a load to the target coordinates is carried out immediately, and in the electromechanical – time is required for stopping the swing.

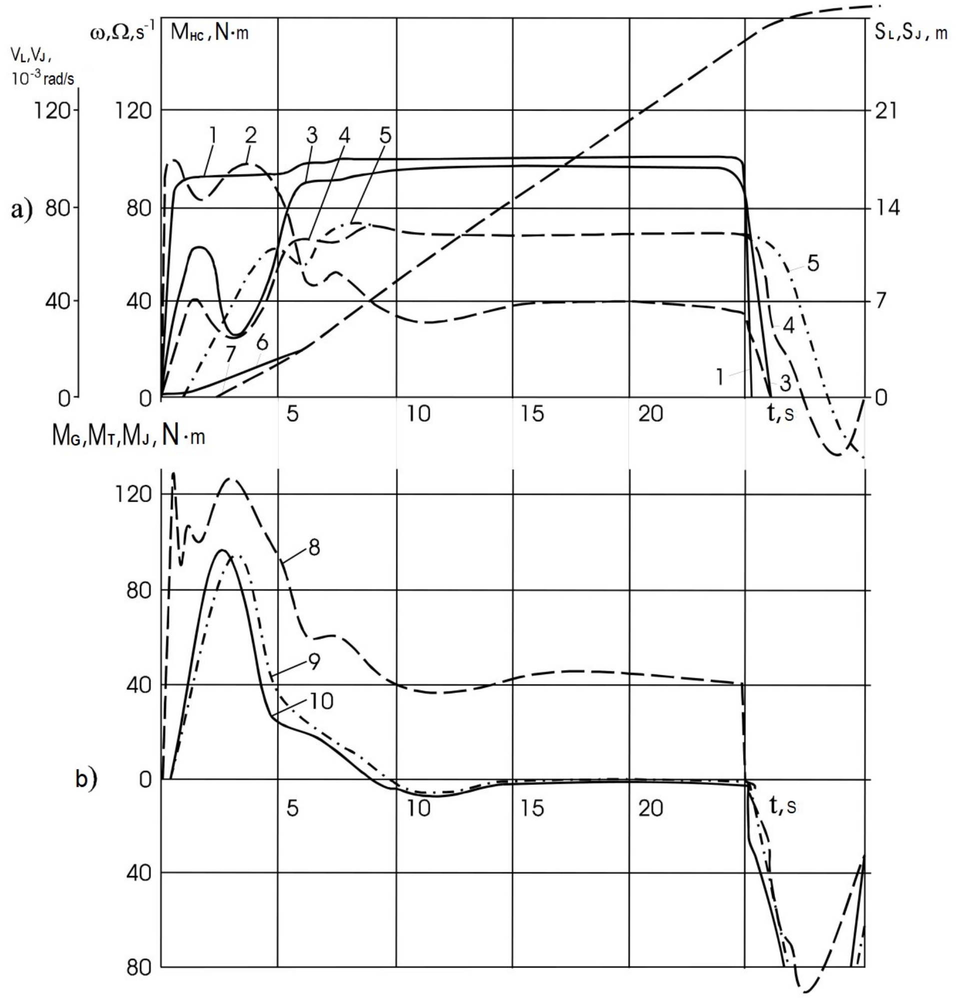


Fig. 2. Curves of transients in hydrodynamic drive
1-angular velocity of the pumping part of the hydraulic coupling; 2-moment hydraulic coupling; 3-angular velocity of turbine part of hydraulic coupling; 6,7- the path of the jib and cargo; 8, 9, 10-torque respectively at the output of the gearbox, at the root of the tower, at the root of the jib, reduced to the axis of rotation of the crane, at Q=6,9 т, R=25 м, l=20 м, M_V =0.

After the successful testing of the hydrodynamic drive of the crane KB 405.2, which was carried out at the Kharkiv DBK-1 in the presence of the director and chief designer of the St. Petersburg's CDB of tower cranes, it was recommended to expand the implementation of the drives at the Rzhevbashkran plant.

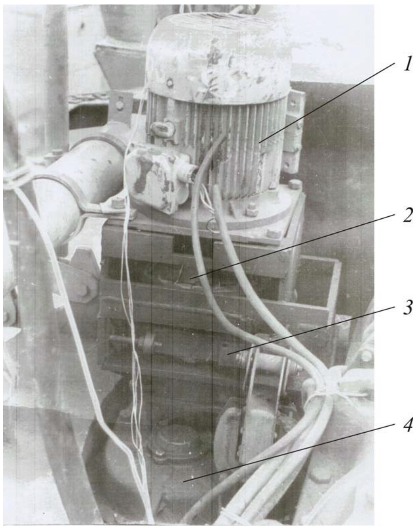


Fig. 3. Slewing mechanism hydrodynamic drive of the tower crane KB 405.2 with the 9.3 t capacity (design of NTU "KhPI"); Kharkiv DBK-1: 1 – electric motor; 2 – hydraulic coupling; 3 – brakes; 4 – gearbox

As a generalization, we note that the use of a hydraulic actuator, in our practice, gave a positive result for other crane mechanisms. In particular, at the request of the Rzhevbashkran plant and the Rzhevsky branch of VNIIBudmash, we created and tested a quadro-drive (on each lag of the crane there was one drive, i.e. 4 drives on one crane), of the tower crane KB-405.2 travel mechanism (see Fig. 6) and dual slewing mechanism of the crane KBM-401 P (see Fig. 7). Drives have greatly improved the operation of these two mechanisms, eliminating breakdowns of slewing and travel mechanisms by reducing the torque moments dynamics.

It should be noted that such complicated implementation, realization and testing of real drives on the real crane KB-410 in the shortest time became possible only due to the coincidence of various factors: the plant created a branch of the research institute of VND Budmash, created a research bureau that harmonized the drawing of drives NTU "KhPI" to the technological capabilities of the plant, created a research department, which conducts large-scale research.

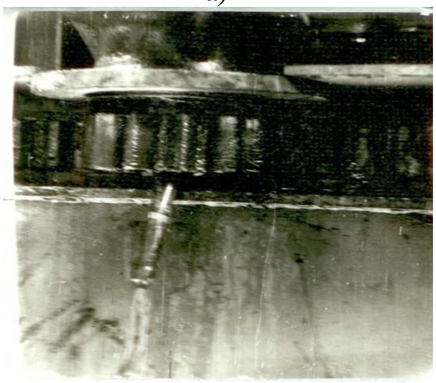
According to the request of the Klaipėda Marine Fishing Port, in NTU "KhPI" the hydrodynamic drive of the portal crane "Ganz" slewing mechanism with a capacity of 6.3/5 t was created, tested and implemented in practice.

Fig. 8 represents the construction of the hydrodynamic slewing mechanism of this crane with a centrifugal fan of cooling of the hydraulic coupling. This fan was more effective compared to the tested axial cooling fan.

The problem of cooling the hydraulic coupling and the replacement of a general-purpose electric motor with a crane type motor caused by the 6K tense mode of the crane Ganz operation. In this case, an upgraded coupling was used on the basis of the existing coupling GP 395 (plant "Svitlo Shakhtaria", Kharkiv).

a)

b)

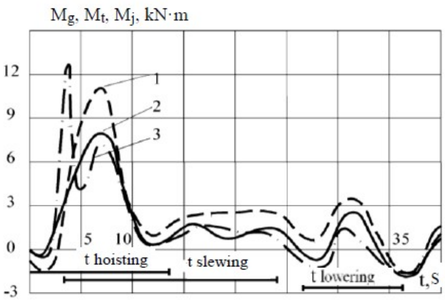


c)

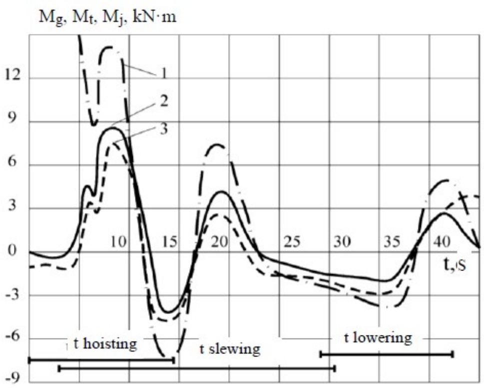
d)

e)

Fig. 4. Test equipment and torque measurement:
– the output shaft of the slewing mechanism gearbox;
– the elements of the jib and tower of the crane KB 405.2 with the 9.3 tons' capacity:
a) in the crane cab; b) installation of the artificial horizon; c) output gear of the slewing mechanism;
d) fragment of the crane tower with the installation of strain gauges;
e) the fragment of the tower with the installation of strain gauges.



a)



b)

Fig. 5. Chart of torque and bending moments of the crane KB 405.2 depending on the time: a with a – hydrodynamic drive; b – with an electromechanical drive:
1 – the moment in the gearbox; 2 – the moment in the tower; 3 – the moment at the root of the jib

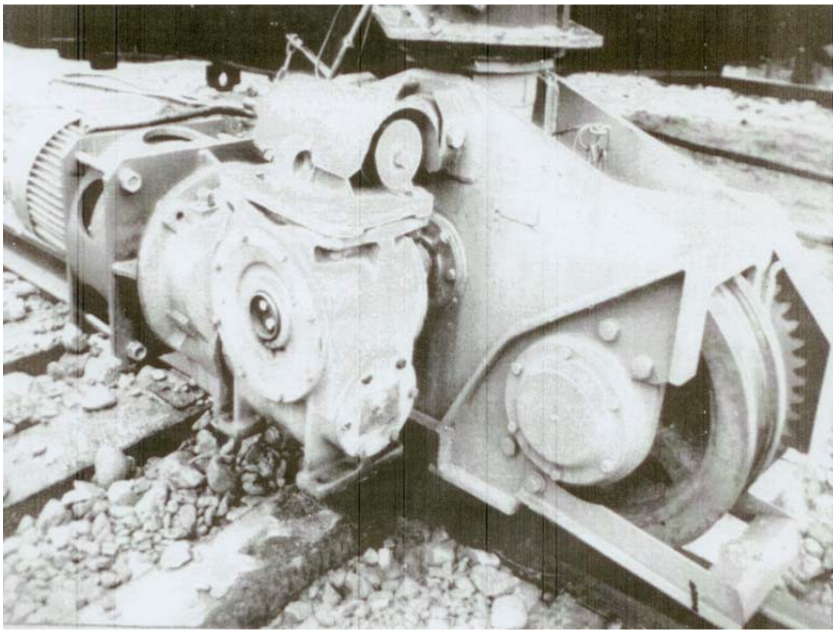


Fig. 6. "Quadro" drive of the tower crane KB-405.2 travel mechanism. Factory "Rzhevbashkran".

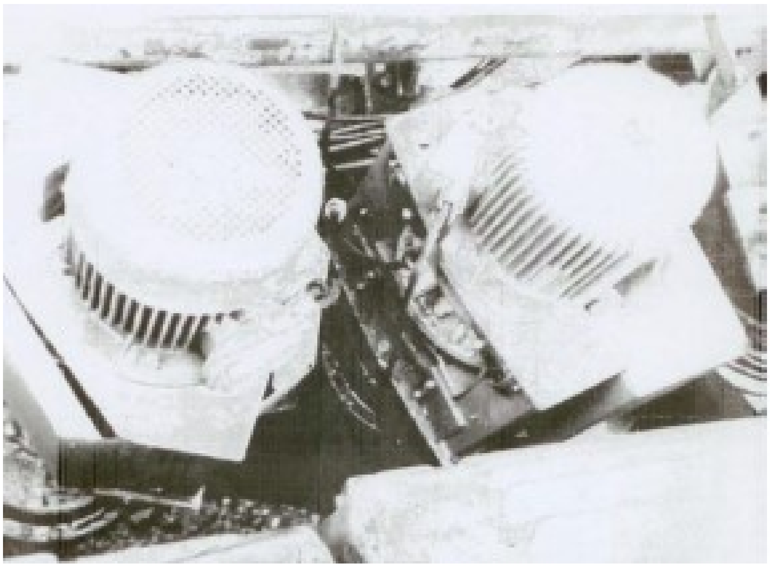


Fig. 7. Double drive of the tower crane KBM-401 P slewing mechanism. Factory "Rzhevbashkran".

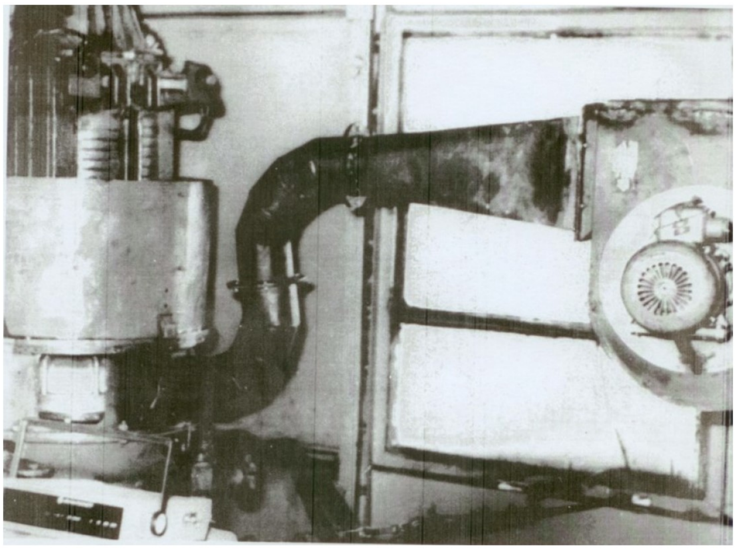


Fig. 8. Hydrodynamic drive of the portal crane "Ganz" slewing mechanism, Klaipeda port.

3. Determination of the limit number of fatigue cycles of Hydrostatic drive

The main result obtained in the cycle of the above works, recorded both theoretically and experimentally, was the effect of reducing the load on the metal structure using a hydraulic drive at least 10 % compared to the drive on the basis of an induction motor with a phase rotor and more than 10 % compared with an induction motor with a short-circuited rotor and direct connection to the network.
We consider this result important, since, as a result, a significant increase in the resource of mechanisms and metal structures is provided. We explain what was said.
According to the Weibull assumption, accumulated by N load cycles, fatigue damage d_N is: $d_N = \sum_{1}^{N} \Delta d = kN$, where Δd is the single fatigue damage in one cycle of loads; $k = \mathrm{tg}\alpha$ is the intensity of accumulation of fatigue damage.
The Wöhler curve reflects the dependence of the durability of the component of the design unit on the stresses level due to the stationary load. It is known that 80-95% of damages are created by cyclic tensions close to the maximum, which can occur at the moments of maximum loads, i.e. during start and braking.
This makes it possible to determine results of the limiting number of cycles for a hydrodynamic drive based on the Weibull theory of damage and on the Wöhler curves (fig. 9).

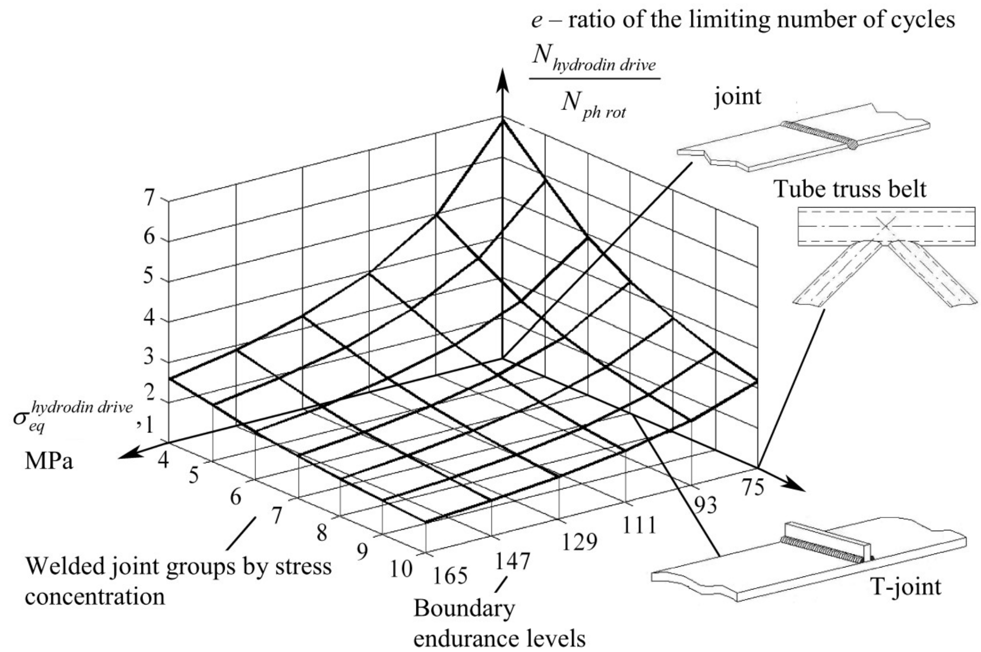


Fig. 9. The maximum number of fatigue cycles ratio for a hydrodynamic actuator and a motor with a phase rotor for groups of welded units at stress concentration with an equivalent stress reduction of 10 % when using a hydrodynamic drive.

4. Conclusions

It is clear from the calculations that the use of a hydrodynamic drive is an effective way to increase the permissible number of load cycles of welded joints of crane steel structures. Reducing the equivalent cycle strain even by 10 % can increase the number of permissible load cycles by at least 1.7 times, for example, for a stress concentration group of 10 for St3 material. The growth of the ratio (see parameter e1 in fig. 9), and accordingly the number of cycles to destruction, increases with a decrease in the equivalent tension of cycle and reaches the maximum value in the area of transition to an area of unlimited endurance. The bigger the item number of the group of the welded node by the concentration of stresses, so growth is more slowly, due to the gradual decrease of the limit of unlimited endurance.
Further scientific developments can be directed at consideration of specific constructions and cases of loads.

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