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Mathematical model of the volumetric hydrodrive's dynamics, sensitive to the loading variation

Alexandra Ivanovskaya[1,*], and *Vladimir* Popov[1]

[1] Kerch State Maritime Technological University, Kerch, Russian Federation

Abstract. The paper deals with the problems of mathematical modeling of the dynamic processes proceeding in a complex hydromechanical system of the ship's deck load-lifting devices. The characteristic feature of this class of equipment is the operation in special conditions, i.e, under the influence of external hydrometeorological factors, taking into account variations in the parameters of lifted or lowered cargo. With a view to gradual regulation of the change of torque on the working drum to reduce dynamic loads, it is offered to use a hydraulic drive that is sensitive to the variation of loading. The operating principle of the considered volumetric hydraulic drive is described. The received mathematical model allows to establish influence of the hydrodrive parameters on transients in a drive system.

A charactristic feature of most deck load-lifting devices is the ability to operate under special conditions. As special conditions it is necessary to consider influence of a wind, rolling or pitching peculiarities, variation of parameters of «cable – cargo» system (its forms, weights, hydrodynamical and inertial factors, length of a cable), the transition from one milieu to another [1]. Nonstationary loadings that occur during the operation of such class of equipment take place not only over the period of transients, but also during the steady-state operating mode, and it has a negative effect on their performance and reliability. Therefore, the researches which have been aimed at improving the system of a drive deck load-lifting devices, are relevant and caused by requests.

The ship's deck load-lifting devices can be driven directly from the internal combustion engine, the drive itself can be either electric or hydraulic.

Among the requirements to drive such equipment are the optimal operating parameters, such as the conversion of speeds over a wide range, the possibility of reversing, the realization of modulating control as for speed and torque and also ability to withstand loadings. The hydromechanical drive complies with these requirements in a greater degree. It is proposed to use a hydraulic drive sensitive to changes in load parameters for smoother regulation of operating options, reduction of dynamic loadings, increasing of durability and reliability of deck's load-lifting equipment [2].

* Corresponding author: invkerch@yandex.ru

The aim of this work is the compilation of the mathematical model of dynamic processes taking place in complex hydromechanical system of ship's deck load-lifting devices operating under special circumstances.

At modeling such types of systems there are some complexities connected with the non-evolutionary nature of the process. They can arise during the hooking of the cargo due to the roughness of the bottom or other underwater objects, at breakage of a cable, and because of the changing the parameters of the loading and the length of the cable.

The calculation scheme of the hydraulic drive, sensitive to load changes, is shown in figure 1.

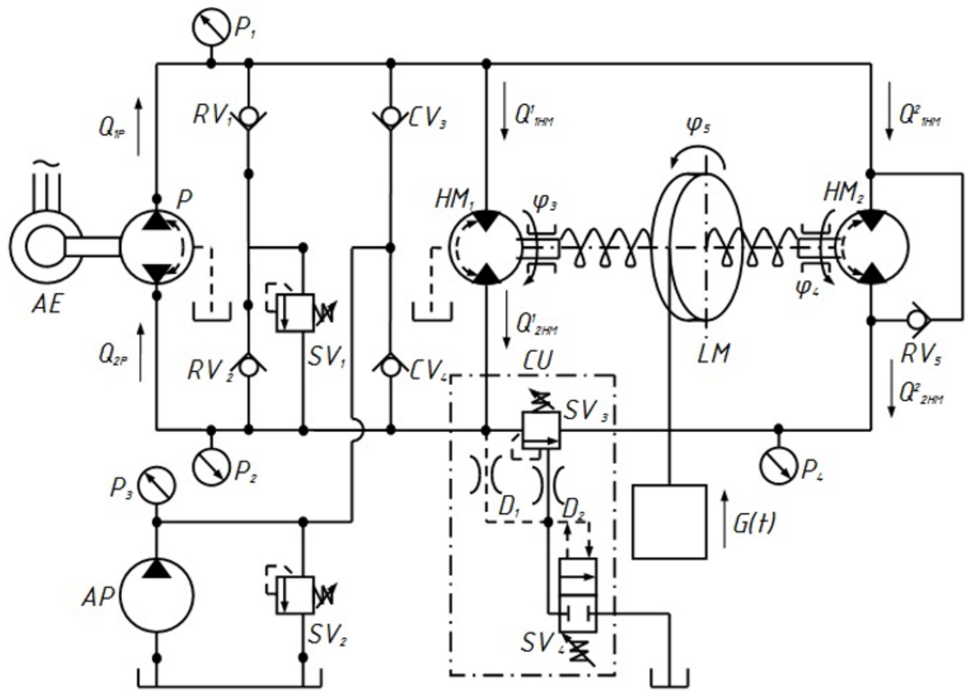

Fig. 1. The calculation scheme of the hydraulic drive, sensitive to load changes

The structure of a drive includes two hydromotors HM_1 and HM_2 for which the standpipe and drain line are connected. They are connected through a gear with a lifting mechanism *LM*. The maximum pressure generated by the pump *P*, driven by the asynchronous engine *AE*, is determined by setting of the safety valve SV_1, which is connected during the reverse operation of the pump with a discharge line through non-return valves RV_1 and RV_2. The total reversal of hydraulic drive is carried out by the pump P. The operation of check valves CV_3 or CV_4 is performed depending on the direction of rotation of the hydraulic motor HM_1. The auxiliary pump *AP* is provided for replenishment of leaks, from there, at the preset constant pressure, which is determined by the overflow pump OP_2, the working liquid is supplied to the annular line through the return valve RV_3.

The control unit *CU* is installed in the hydraulic main, which connects the semi-axis pressure cavity with the hydraulic motor HM_2. The input of the hydraulic motor HM_2 at the switched off control unit is connected to the drain through the return valve RV_5. At rated loading, only the hydraulic motor HM_1 operates. At sudden increase in loading, the pressure at the inlet of the hydraulic system reaches the value P_4, it exceeds the nominal P_1, in this case the control unit *CU* is triggered, owing to that the working liquid enters the working chamber of the hydraulic motor HM_2 and connects it in parallel to the hydraulic motor $_{HM1}$. The ability to control the torque on the shaft of the drum and the lifting speed is achieved through the regulation of the flow rate and pressure of the working liquid.

The equations of the mechanical part motion and the equation of fluid flow balance are the basis of the mathematical model of the output link of the hydro-mechanical drive.

During the modeling of the mechanical system, some assumptions were made:

– discrete masses of the load-lifting device are absolutely rigid bodies;
– the connections of discrete weights - are elastic weightless connecting links with constant factor of rigidity;
– the chosen cable is presented by a weighty elastic-viscous thread of variable length;
– the absence of transverse vibrations of the cable;
– the slipping a cable on a drum is absent ;
– the moment of inertia of the body coiling the winch drum, during the period of the system's acceleration .

At modelling of the hydraulic system the following assumptions have been accepted:

– the pump output capacity is constant (Q_P = const);
– the total amount of working liquid in each line and in the connecting channels at current pressure is constant;
– pressure connecting lines are short, therefore, it is possible to neglect the hydraulic resistance and the wave processes in them ($\Delta P_f = 0$);
– the temperature and viscosity of the working fluid flow are taken as constant ($t0$ = const, μ = const);
– the forces of dry friction are insignificant, therefore they can be neglected;
– there is no cavitation and break of the flow in the system;
– for this range of pressure variation, the ductility of the cavity, and the compressibility of the working fluid, are taken into account as average values;
– the pressure of the liquid in the pressure and drain lines is approximately constant over the entire length and does not exceed those to open the safety valves;
– the inertial pressure in the throttle line is insignificant in comparison with the total loss of fluid pressure;
– the deflection angle of the flow is approximately constant and does not vary at minor vibrations of closing the valve near to the settled position;
– the losses at the injection line are proportional to the pressure;
– the overflowing in hydraulic machines are considered as the linear functions of the pressure difference in the lines;
– the leakages in hydraulic machines are represented by a linear pressure function.

The equations that describe the dynamics of the hydraulic drive operation we will write in the form of separate equations of individual elements of the drive, taking as a basis the equation of flow rate and moments of the hydro-mechanical system.

The equation of the working fluid flow rate that passes through the pump:

- into the pressure main

$$q_P \frac{d\varphi_2}{dt} - K_{lP} P_2 - Q_{1P} - K_{oP}\left(P_1 - P_2\right) = 0 \ ;$$

- into the drain line

$$q_P \frac{d\varphi_2}{dt} + K_{lP} P_2 - Q_{2P} - K_{oP}\left(P_1 - P_2\right) = 0 \ ,$$

where $q_P = k_P \gamma$;

- on the leaks in the pump

$$K_{lP}\left(P_1 - P_2\right) - Q_{lP} = 0\,.$$

At the nominal loading in the equation that describes the operation of the hydromechanical drive, at first we take into account the operation of only one hydraulic motor HM_1.

The equations of working fluid flow rate:

- in the pressure line of the hydraulic motor HM_1

$$q^1_{HM}\frac{d\varphi_3}{dt} - Q^1_{1HM} + C^1_{oHM}\left(P_1 - P_2\right) + C^1_{lHM}P_1 = 0\,;$$

- in the drain line of the hydraulic motor HM_1

$$q^1_{HM}\frac{d\varphi_3}{dt} - Q^1_{2HM} + C^1_{oHM}\left(P_1 - P_2\right) - C^1_{lHM}P_2 = 0\,;$$

- for the leakage in the hydraulic motor

$$Q_{lHM} = C_{lHM}\left(P_1 + P_2\right).$$

In that case the general equations of the working fluid flow rate that determine the linkage between the fluid volumes flowing through the pump and the hydraulic motor, taking into account the strain of liquid and pipelines, they will have the form:

- in the pressure line

$$KV_L\left(\frac{dP_1}{dt}\right) = Q_{1P} - Q^1_{1HM}\,; \tag{1}$$

- in the drain line

$$KV_L\left(\frac{dP_2}{dt}\right) = Q^1_{2HM} - Q_{2P}\,,1 \tag{2}$$

where $K = \dfrac{d_0}{E_w\delta} + \dfrac{1}{E_l}$.

It is necessary to consider, that $P_{CU} \geq P_1$

When the loading value exceeds the nominal during the operation of the ship's deck load-lifting equipment, the pressure P_4 is set in the hydraulic system, it exceeds the nominal pressure P_1, in this case the control device is activated, as a result the working fluid enters the process chamber of the hydraulic motor HM_2 and switches it on parallel to the hydraulic motor HM_1.

The equations of working fluid flow rate presented here is:

- in the pressure line

$$KV_L\left(\frac{dP_1}{dt}\right) = Q_{1P} - Q^1_{1HM} - Q^2_{1HM}\,; \tag{3}$$

- in the drain line

$$KV_L\left(\frac{dP_2}{dt}\right) = Q^1_{2HM} + Q^2_{2HM} - Q_{2P}\,. \tag{4}$$

The following identification marks are used in the settlement scheme and the resulting equations : Q_{1P} , Q_{2P} - this is the flow rate of the pump's working fluid through the pressure and drain lines, respectively; Q^1_{1HM}, Q^2_{1HM} , Q^1_{2HM}, Q^2_{2HM} - it's the working fluid's consumption of the hydraulic motors GM_1 and GM_2 through the pressure and drain lines, correspondingly; q_P - it is specific supply of the pump; k_P - it is the factor of the pump's specific supply; K_{yP}, K_{nP} - these are the quotients of the overflow and leaks in the pump; C^1_{oHM}, C^1_{lHM} - these are the factors of the overflow and leaks in the hydraulic motor *HM1*; P_1, P_2 , P_4 , P_{CU} - these are the pressure indicators in the pressure and discharge lines, the settings of the supply valve *SV2* in the main line of the hydraulic motor *HM2* after the control device's *CU* activation in the hydraulic line control unit; φ_2 φ_3 - the rotation angle of the pump shaft and hydraulic motor HM_1,respectively; V_L - the volume of fluid in each hydraulic line; d_0 δ - the inner diameter and wall thickness of the pipeline; E_w, E_l - the modules of elasticity of the pipeline's wall material and of the liquid.

At design of the hydraulic drive, sensitive to variation in loading, it is necessary to consider all possible cases of dynamic effects on lifting mechanism relative to the mechanical system « a cable – a cargo ».

As the cable length of the investigated class of ship's deck lifting devices is quite large, we can present it as a system with distributed parameters in the form of the variable-length thread with moving boundaries.

The mathematical model of the mechanical part is formulated on the basis of Lagrange's equations II [3] and has the form:

$$l\left(Q(t)+\frac{ql}{3}\right)\frac{d^2u}{dt^2}+\left(\frac{q}{3}-\frac{ql}{6}+\frac{dQ}{dt}\right)\frac{dl}{dt}\frac{du}{dt}+\left[2g\left(C_c l+EF\right)\frac{dQ}{dt}\left(\frac{dl}{dt}\right)^2\right]u=$$

$$=\left(Q(t)+\frac{ql}{2}\right)\left(g-\frac{d^2l}{dt^2}\right)+\left(q-\frac{dQ}{dt}\right)\left(\frac{dl}{dt}\right)^2. \qquad (5)$$

Under boundary conditions:

- as $z=l_0$ at the lower end of the cable: $\left(\frac{Q(t)}{g}\frac{\partial^2 u}{\partial t^2}+EF\frac{\partial u}{\partial z}\right)_{z=l_0}=Q(t)\left(1+\frac{\dot{v}_c}{g}\right);$ (6)

- as $z=l$ at the point C of the cable winding: $\frac{du}{dt}=\left(\frac{\partial u}{dz}\frac{dl}{dt}+\frac{\partial u}{dt}\right)_{z=l}.$ (7)

where *Q(t)* – the lifted cargo of variable weight; $u(z,t)$ - the absolute elongation of the part of a cable at length z.

Conclusions

The received equation (1), (2), (5) to (7) and (3) to (7) describe the motion of the hydromechanical systems of ship's deck lifting devices, respectively, at nominal operating mode and after exceeding the loading. The solution of differential equations, which are included in the mathematical model makes it possible to estimate the parameters of the hydraulic drive under different types of loads.

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