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# Improving the controllability of the electro-hydraulic complex in emergency operating modes

***Abstract**. The possibility of using reversible operating modes of hydraulic and electric machines in the problems of improving the controllability of an electrohydraulic complex is shown. The schemes of energy balance in pumping and turbine operating modes of hydraulic machines are given. A mathematical description of the four-quadrant characteristics of the pumping unit is proposed. The complete characteristics of the pump operating on the pipeline network with back pressure, when the direction of rotation of the pump electric motor was changed and there was reverse of the liquid in the hydraulic system, are obtained. A structural diagram of a system for active regulation of pumping unit parameters based on the use of hydro turbine units or reversible hydraulic machines is proposed.*

***Streszczenie.** Pokazano możliwość użycia odwracalnego trybu pracy maszyn hydraulicznych i elektrycznych w polepszaniu sterowności zespołu elektrohydraulicznego. Podane zostały schematy równowagi energii w trybie pompowania i pracy turbinowej maszyny hydraulicznej. Dla jednostki pompowanie z czterokwadrantowym trybem pracy opracowano modele matematyczne. Otrzymano finalne charakterystyki działania pompy i sieci rurociągowej z ciśnieniem wstecznym w warunkach zmiany kierunku obrotów napędu elektrycznego pompy i cofania płynu w systemie hydraulicznym. Zaproponowany został diagram parametrów aktywnej regulacji jednostki pompowania, wykorzystujący jednostki turbin wodnych lub odwracalnych maszyn hydraulicznych. **(Poprawa sterowności zespołu elektrohydraulicznego w trybie pracy awaryjnej)***

***Keywords:*** controllability, four-quadrant characteristics, active regulation, reversible hydraulic machines.
***Słowa kluczowe:*** sterowałność, charakterystyka czterokwadrantowa, aktywna regulacja, odracalne maszyny hydrauliczne

## Introduction

In the operation of centrifugal pumps of electro-hydraulic complexes (EHC), emergency modes are possible, the reasons for which include: cessation of the supply of electricity to the electric drive (ED) of the pump, disconnection/activation of individual pipelines or their sections, a sharp change in the degree of opening of valves, etc. [1, 2]. Such modes are accompanied by the development of wave processes in the hydraulic system, leading to pressure inrush, surges, and premature wear of electro-hydraulic equipment.

Modern EHCs represent a system with low controllability of parameters due to the weak relationship between the operating modes of the pumping plant (PP) and the technological process, the lack of effective means of controlling the pumps ED and shut-off and adjustable valves in unsteady (emergency) modes of operation.

Currently, in world practice, there is a steady tendency to replace unregulated PP electric drives with regulated systems. This is due to a number of benefits: the ability to save electricity from 30 to 70 %, high quality control in static and dynamic modes, lower equipment loads due to a decrease in pressure in the hydraulic system, reduction of leaks and losses for transporting the working medium, etc. [3, 4]. The use of automatic systems controlled by variable-frequency ED based on scalar and vector control, taking into account continuous monitoring of electromechanical, hydraulic and energy parameters, makes it possible to implement energy and resource-saving operating modes, improve the reliability and energy efficiency of technological mechanisms ED [5, 6].

However, in most cases, the dynamic properties of the pump are considered for the pumping mode of a centrifugal machine, which is only a part of the possible modes of operation. The pumping equipment is characterized by modes similar to electric machines – their reversibility when the sign of the hydraulic power changes [2]. The prospect of using braking modes to control pumping unit (PU) in the event of an emergency power outage deserves the closest attention.

## Research method

Reversible centrifugal pumps operate in a wide range of power conversion modes [2]. With a negative value of flow, head and rotation frequency, the pump operates like a usual turbine, giving power to the motor shaft. With a positive value of the head and flow, but in the opposite direction of rotation, the pump operates as a reversible one, giving energy to the liquid flow. With negative flow and head, but a positive direction of rotation, the pump receives energy from the flow and is a reversible turbine. At positive head and rotation frequency, but negative flow, the pump operates as a hydraulic brake, dissipating the energy received from both the flow and the motor.

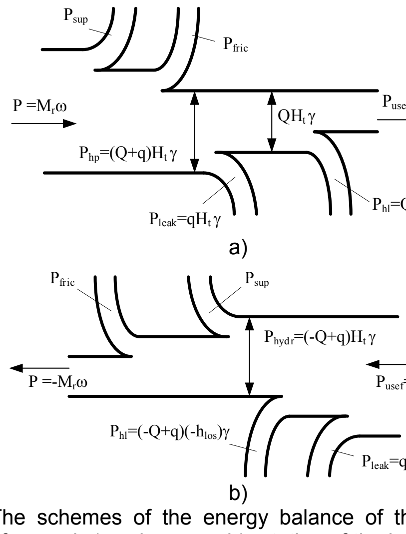


Fig. 1 The schemes of the energy balance of the pump in the modes of normal a) and reverse b) rotation of the impeller

Fig. 1 contains the pump energy balance schemes for the modes of normal ( $\omega > 0$ ) and reverse ( $\omega < 0$ ) rotation, where the following designations are adopted: $P$ – the power on the rotor shaft; $P_{sup}$ – the power of losses in the pump supports; $P_{hp}$ – hydraulic power; $P_{fric}$ – the power of disk friction losses; $P_{leak}$ – the power of leakage losses; $P_{hl}$ – the power of hydraulic losses in the pump; $P_{usef}$ – the useful (hydraulic power of the pump; $H_t$ – theoretic head; $h_{los} = \sum h_i$ – total hydraulic losses; $q$ – leakage through impeller groove seal; where $Q$ – discharge; $M_r$ – the pump moment of resistance; $\omega$ – PU angular rotation frequency; $\gamma = \rho g$ ; $\rho$ – liquid density; $g = 9.81$ m/s$^2$ – acceleration of gravity.

Thus, each operating mode of an electro-hydraulic unit is characterized by a well-determined energy distribution, the analysis of which in a wide range of flow rate variation is of undoubted interest.

In this regard, for the analysis of energy conversion processes in electromechanical complexes, it is advisable to use the instantaneous power method, which makes it possible to most fully characterize the change in power in the time domain. Currently, the instantaneous power method is successfully used in the problems of reactive power compensation [7], in the diagnosis of electromechanical devices [8, 9], in the assessment of energy controllability [10, 11], in systems for reducing dynamic loads in EHC [12, 13] and identification parameters of electrical systems [14, 15].

To research and analyze the unsteady operating modes of the EHC, complete (four-quadrant) characteristics of the pumps are required when the direction of liquid supply in the pipeline and the rotation frequency of the PU change.

For a mathematical description of the four-quadrant pump characteristics when operating on a hydraulic network with back pressure, it is advisable to introduce sign functions into the expressions for the characteristics of the PU:

head-discharge characteristic

(1) $$H(Q) = sign(\nu) H_0 \nu^2 + sign(Q) R_p Q^2 ;$$

power characteristic

(2) $$P(Q) = sign(\nu) A_3 \nu^2 Q + sign(Q) B_3 \nu Q^2 + D_3 \nu^3 ;$$

head-discharge characteristic of the pipeline network

(3) $$H(Q) = H_{st} + sign(Q) R_{net} Q^2$$

where $H_{st}$ – static back pressure; $R_p$ , $R_{net}$ – the hydraulic resistance of the pump and pipeline, respectively; $A_3$ , $B_3$ , $D_3$ – the coefficients of approximation of the rated power characteristics of the pump; $\nu = \omega_i / \omega$ – PU relative rotation frequency; $\omega_i$ – the current value of the PU rotation frequency; $sign(\nu)$, $sign(Q)$ – sign functions of the relative rotation frequency and pump flow, respectively.

From the joint solution of equations (1) and (3), the mathematical dependence of the pump flow on the rotation frequency was obtained when the PU operates on the hydraulic network with back pressure and the direction of rotation of the pump motor is changed:

(4) $$Q(\nu) = \begin{cases} \sqrt{\dfrac{H_0 \nu^2 - H_{st}}{R_b + R_{net}}}, \; at \; \nu_{kr} \le \nu \le 1; \\ -\sqrt{\dfrac{H_{st} - H_0 \nu^2}{R_b + R_{net}}}, \; at \; 0 \le \nu \le \nu_{kr} ; \\ -\sqrt{\dfrac{H_{st} + H_0 \nu^2}{R_b + R_{net}}}, \; at \; \nu < 0 \end{cases}$$

where $\nu_{kr}$ – the lower limit of the change in the pump rotation frequency, upon reaching which it becomes possible to transfer the hydraulic machine to the turbine mode of operation with the possibility of energy recovery when the motor is operating in the braking mode.

So, in the case of a single operation of a PU to a network with back pressure and a decrease in the supply to zero, the equality is true: $H_0 \nu_{kr}^2 = H_{st}$ . Then $\nu_{kr} = \sqrt{H_{st} / H_0}$ . In this case the lower boundary $\nu_{kr}$ of the change in the pump rotation frequency does not depend on the resistance of the pipeline, but is determined only by the value of the static head $H_{st}$ in the hydraulic system. In the case of group work of PU on the pipeline network, the definition of $\nu_{kr}$ becomes more complicated. In this case, it is necessary to take into account the number of simultaneously operating pumps, the PU connection scheme (parallel or series), the hydraulic resistance of the pumps and the pipeline.

To make four-quadrant pump characteristics when changing the direction of rotation of the pump motor, the following systems of equations were obtained:

the dependence of the head on the pump rotation frequency

(5) $$H(\nu) = \begin{cases} H_0 \nu^2 - R_b Q^2(\nu), \; at \; \nu_{kr} \le \nu \le 1; \\ H_0 \nu^2 + R_b Q^2(\nu), \; at \; 0 \le \nu \le \nu_{kr} ; \\ -H_0 \nu^2 + R_b Q^2(\nu), \; at \; \nu < 0; \end{cases}$$

the dependence of the hydraulic power of the pump on the pump rotation frequency

(6) $$P_h(\nu) = \rho g H(\nu) Q(\nu);$$

the dependence of the power supplied to the pump shaft

(7) $$P(\nu) = \begin{cases} A_3 \nu^2 Q(\nu) + B_3 \nu Q(\nu)^2 + D_3 \nu, \; at \; \nu_{kr} \le \nu \le 1; \\ A_3 \nu^2 Q(\nu) - B_3 \nu Q(\nu)^2 + D_3 \nu, \; at \; 0 \le \nu \le \nu_{kr} ; \\ -A_3 \nu^2 Q(\nu) - B_3 \nu Q(\nu)^2 + D_3 \nu, \; at \; \nu < 0; \end{cases}$$

the dependence of the moment of resistance created by the pump

(8) $$M_r(\nu) = \begin{cases} \dfrac{P(\nu) \cdot 10^3}{\nu \cdot 102{,}6}, \; at \; \nu_k \le \nu \le 1; \\ \dfrac{P(\nu) \cdot 10^3}{\nu \cdot 102{,}6}, \; at \; 0 \le \nu < \nu_k ; \\ -\dfrac{P(\nu) \cdot 10^3}{\nu \cdot 102{,}6}, \; at \; -1{,}0 \le \nu < 0. \end{cases}$$

The graphs of the full characteristics of a centrifugal pump with parameters $P_n = 760$ kW; $\eta_{max} = 0.75$ ; $Q_n = 0.556$ m$^3$/s; $H_n = 100$ m; $n_n = 980$ rev/min; $A_3 = 1181.818$ , $B_3 = -619.835, \; D_3 = 150$ when working on a pipeline network with back pressure are shown in Fig. 2.

Intersection points of pump characteristics $H(Q)_{\nu 1} .. H(Q)_{\nu 5}$ with pipeline characteristic $H(Q)_{net}$ (Fig. 2, a) in different quadrants of the coordinate axes correspond to different operating modes of the turbo mechanism. So, in the pumping mode of PF operation (p. $A_1$) with rated parameters at $\nu_{kr} < \nu \le 1.0$ the pressure developed by the pump is spent on raising the water to height $H_{st}$ and overcoming the hydraulic resistance of the pipeline. In this case the pump resistance moment $M_r$ is equal to the driving moment $M_e$ of the drive.

With a decrease in the rotation frequency, for example, in the event of an emergency shutdown of the ED from the power grid, the driving moment almost instantly drops to zero $M_e = 0$ , hydraulic moment $M_h$ remains on the impeller, which results in a sharp decrease in flow and head. Then, at p. A2 at $\nu = \nu_{kr}$ the pump flow is zero, the direction of liquid movement changes, the moment on the impeller is minimal, and the vacuum in the pressure pipeline is maximal.

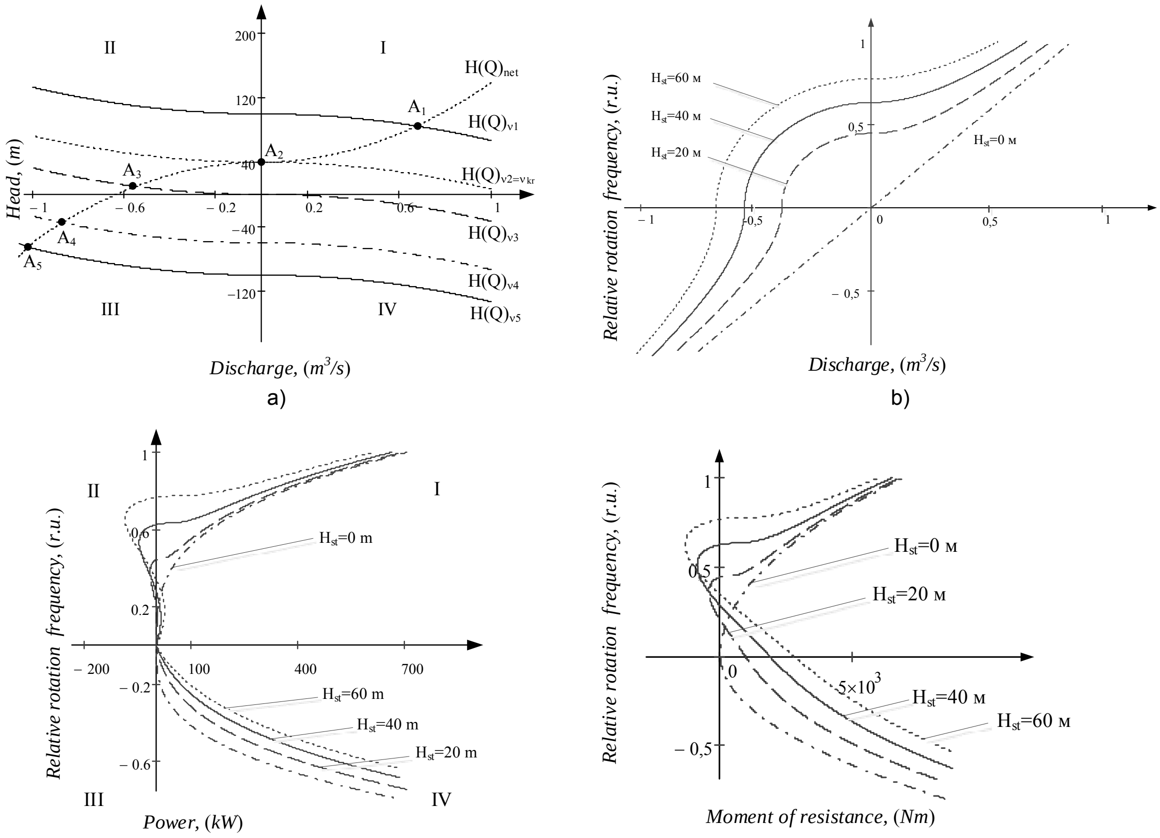


Fig. 2 Four-quadrant pump characteristics: pressure-discharge rate a), curves of change in flow b), shaft power c) and moment of resistance d) versus relative rotation frequency

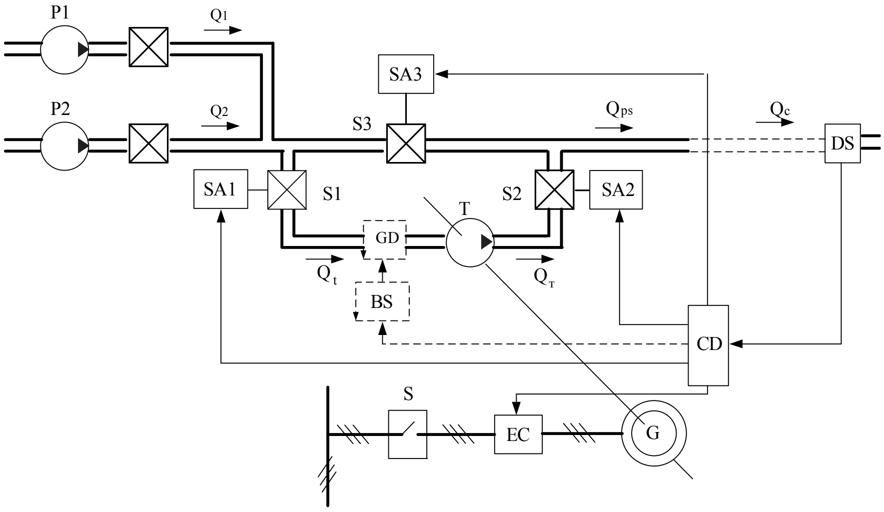


Fig. 3 The scheme of the PP with APCS installed in parallel with the control stopcock: CD – control device; BS – blade setter; GD – guide device; EC – energy converter; S – switch; P – pump; DS – discharge sensor; S – stopcock; SA – stopcock actuator; G – generator; T – turbine

In the counterflow mode, when $\nu < \nu_{kr}$, the impeller is decelerated by the flow of the reverse (turbine) direction, the pressure in the head pipeline increases, the rotor rotation frequency decreases to zero (p. A3). After a momentary stop, the rotor of the electric motor starts to rotate in the opposite direction ( $\nu < 0$ ) and the PU passes into the turbine mode (p. A4, p. A5). At the beginning of this mode, the hydraulic moment becomes maximum, and then decreases.

The analysis of the obtained curves (Fig. 2) revealed that in the absence of static pressure in the hydraulic network $H_{st}$ the four-quadrant pump characteristics pass through the point (0; 0) (Fig. 2, b-d). In the presence of $H_{st}$ the characteristics are distorted, which is especially noticeable in the second quadrant of the curves (Fig. 2, b-d), where, with a decrease in the rotation frequency, the liquid counterflow mode is observed. Moreover, the higher $H_{st}$ is the wider the area of the PP operating mode with reversal of the liquid supply in the pipeline is. When switching to turbine operation, the power curves on the pump shaft pass in the fourth quadrant (Fig. 2, c), which confirms the possibility of using hydraulic energy in the PP in emergency modes of operation on the basis of a four-quadrant power converter as part of a frequency-controlled ED.

Taking into account the foregoing, the active parameter control system (APCS) is an alternative option for increasing the EHC controllability. It consists of a turbine, an electric machine (generator) and an electromechanical energy converter (Fig. 3), where the hydro-turbine unit plays the role of an active control resistance, which makes it possible to regulate the supply within the required technological limits with the simultaneous return of part of the energy back to the power grid. At the same time, both hydraulic turbines (radial-axial, rotary-blade, bucket) and reversible hydraulic machines can act as active adjusting devices [16].

Active control devices based on hydro-turbine units allow changing the supply within the required limits (up to 50% of the rated one) with simultaneous power output (up to 30% of the power of the main pumping units) [16]. When using reversible hydraulic machines as part of pumping units, it is possible to implement various control schemes, depending on the technological requirements, especially in schemes with pumps group ED.

### Conclusions

It has been shown that the use of reversible operating modes of hydraulic and electric machines, which expand the control capabilities and protective functions of the pumping unit, is a promising way to increase the controllability of the electrohydraulic complex. This can be realized on the basis of centrifugal vane hydraulic machines and variable AC drives. The above is especially important in the systems of group electric drive of pumping facilities operating on a pipeline network with back pressure when the direction of movement of the liquid or the speed of rotation of one of the pumps connected in series or in parallel is changed.

A mathematical description of the four-quadrant characteristics of pumping facilities, when changing the direction of the rotation frequency of the drive motor and reversing the liquid supply, has been proposed.

This makes it possible to determine the permissible ranges of regulation of the pump performance and to substantiate the choice of an expedient variable electric drive system.

It has been shown that a promising option for increasing the controllability of the electrohydraulic complex consists in systems for active regulation of the parameters of pumping units as part of a high-power pumping station with a group load and a time-varying operating mode of the consumer.

At the same time, in addition to regulating the output technological parameters in the pipeline network, active control devices can act as means of hydro-protection of the electro-hydraulic complex against an unacceptable excess of pressure or a sharp decrease in performance to zero at the outlet of the hydraulic machine, caused by emergency operating modes of the equipment (sudden power outage of the pumps, abrupt closing of stopcocks, starting or stopping turbomachines, etc.).

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