При проектуванні гідравлічного приводу в якості робочої схеми розподілу енергії від одного його джерела до споживачів прийнята індивідуальна схема. Теоретичні дослідження виконано на основі використання системного аналізу для гідравлічного приводу з індивідуальною схемою розподілу енергії і послідовним підключенням двох її підсистем. Цими дослідженнями встановлені залежності для розрахунку та вибору параметрів для насоса і гідродвигунів

Ключові слова: привод гідравлічний, проектування, система, потужність, коефіцієнт корисної дії, насос, гідроциліндр, гідромотор

При проектировании гидравлического привода в качестве рабочей схемы распределения энергии от одного его источника до потребителей принята индивидуальная схема. Теоретические исследования выполнены на основе использования системного анализа для гидравлического привода с индивидуальной схемой распределения энергии и последовательным подключением двух ее подсистем. Этими исследованиями установлены зависимости для расчета и выбора параметров для насоса и гидродвигателей

Ключевые слова: привод гидравлический, проектирование, система, мощность, коэффициент полезного действия, насос, гидроцилиндр, гидромотор

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METHODOLOGY FOR THE HYDRAULIC DRIVE DESIGN BASED ON THE APPLICATION OF THE SYSTEMS ANALYSIS

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1. Introduction

For fulfilling different types of operations, execution equipment of most modern machinery is set into motion by hydraulic drive (HD). It is known from the scientific sources that machines with HD were created in 1938 at the Mykolaiv Plant of Road Machines (Ukraine). The machines in question were a bulldozer and a scraper. Growth [1] in the production of machines with HD has continued for the last 70...85 years. Since 1950, manufacturing of machines with HD has become commonplace [2].

The structure of modern machines, in addition to HD, includes a primary engine, operating equipment, working environment and the human for sure. Ties between the components of the machine are shown in Fig. 1.

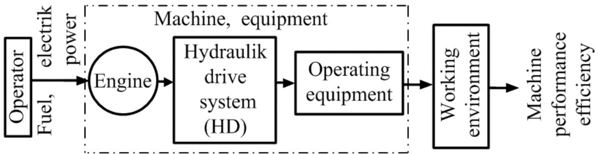

Fig. 1. Position of HD in the structure of complex system

The schematic diagram (Fig. 1) clearly shows position of the main elements of the machine, including the location of HD as a separate component of the machine. In the course of functioning of the machine, a process of energy transfer from the primary engine to the operating equipment for such complex system [3–5] is determined by a generalizing parameter. This parameter is characterized by multiplying the two indicators – overall performance efficiency coefficient (PE) of machine's HD η_{hp} and indicator k_N, which reflects utilization level of the capacity of the machine's primary engine. Generalizing parameter of machine functioning is determined by formula

$$\eta_{hp} \times k_N = \int_{t_{1hp}}^{t_{2hp}} (N_{hp} \times dt) / (N_{nom} \times (t_{2m} - t_{1m})), \quad (1)$$

where N_{hp} is the useful power consumed by HD for the execution of work; N_{nom} is the nominal capacity of primary engine; t_{2hp}, t_{1hp} is the duration of HD work; t_{2m}, t_{1m} is the duration of work of the machine's primary engine.

The first indicator η_{hp} of dependence (1) is determined by formula

$$\eta_{hp} = \int_{t_{1hp}}^{t_{2hp}} (N_{hp} \times dt) / \int_{t_{1hp}}^{t_{2hp}} (N_{dvz} \times dt) = N_{hp} / N_{dvz}, \quad (2)$$

where N_{dvz} is the capacity of primary engine of the machine for using HD.

The second indicator k_N of dependence (1) is calculated by formula

$$k_N = \int_{t_{1hp}}^{t_{2hp}} (N_{dvz} \times dt) \,/\, (N_{nom} \times (t_{2m} - t_{1m})). \qquad (3)$$

An analysis of the indicated dependences reveals that for the hydroficated machines of cyclic action, the duration of HD operating time t_{2hp} and t_{1hp} and the primary engine t_{2m} and t_{1m} are different magnitudes; while for the machines of continuous action they are the same by the duration of operating time, which is why results of the calculation by formula (1) will be different.

Indicator η_{hp} does not depend on the functional purpose of machine, that is, on the peculiarities of machine's operation process, which is characterized by cyclic or continuous action. The given indicator η_{hp} reflects quality (level) of the conversion of supplied power (energy) into the power at the output of the machine (useful power). It should be noted that the magnitude of the given indicator η_{hp} for HD is provided mainly at the design stage and the level of this indicator is maintained under operation mode. At the design stage of HD, the magnitude of η_{hp} is at the level of 0.6...0.7 and this indicator affects largely the functional capacities of the machine as a whole relative to its engine. Due to this, considerable attention is still paid to the issues of HD design.

A substantiation to this circumstance is the fact that η_{hp} characterizes the relationship between fuel consumption and machine's productivity as the components of parameters of the input power and output power of the machine. The task of improving productivity and reducing energy losses by engine (fuel or electric power) is relevant today for the hydroficated machines and equipment. Considering the aforementioned, there is a need to conduct additional research into the HD design stage; hence, studies in this direction are also an important issue.

As the object of research, to address the relevant problem, let us consider the stage of HD design.

2. Literature review and problem statement

To create a HD with the specified indicators of operational reliability, it is necessary at the stage of its design to analyze all known technical and scientific developments, which are presented in the scientific literature. The application of the most important of them can considerably increase the efficiency of HD in the composition of machine over all stages of the life cycle of these machines.

The element of HD, which provides for the distribution of fluid between the consumers, is the hydroallocator. Depending on the design of the main controlling element of hydroallocator, they are divided into the slide valve, valve and crane. The most common are the slide valve hydroallocators with a mono-block and sectional design of case [6]. Hydraulic connection between a pump and hydro engines through hydroallocators is ensured by applying different circuits of their connection.

As an example, let us consider the known [7] circuit decision of HD, shown in Fig. 2, which, when using one pump, enables motion of the rod of single-acting hydraulic cylinder and movement of shaft of hydraulic engine by applying various elements of the drive. Their compatible utilization allows the creation of different circuits of their connection simultaneously.

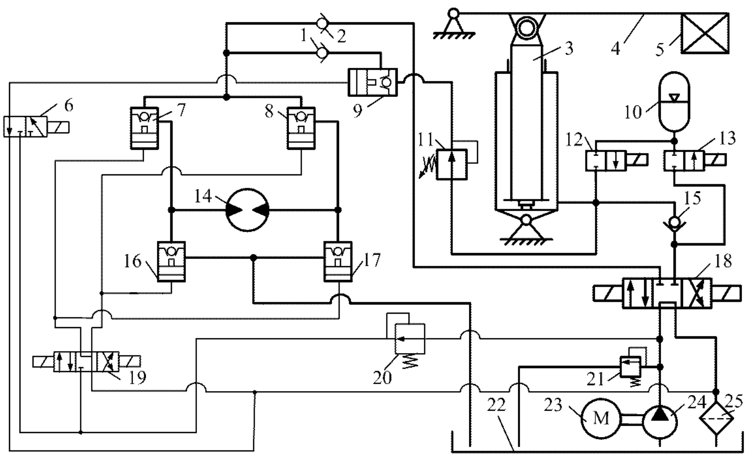

Fig. 2. Schematic of HD with hydraulic engine of translational and rotational motion

In Fig. 2 the following designations are used:

– 1, 2, 15 – return valves;
– 3 – single-acting hydraulic cylinder;
– 4, 5 – boom with the load;
– 6, 12, 13, 18, 19 – solenoid operated liquid distributors;
– 7, 8, 9, 16, 17 – single-acting controlled valve;
– 10 – hydroaccumulator;
– 11, 20 – reducing valve;
– 14 – hydraulic engine;
– 21 – safety valve;
– 22 – oil tank;
– 23 – drive motor;
– 24 – pump;
– 25 – filter.

The hydraulic circuit, which is shown in Fig. 2, enables the following functions:

– lifting and lowering boom 4 with load 5;
– rotational motion by feeding fluid to hydraulic engine 14 from the pump or from cylinder 3 from the piston cavity;
– energy storage by hydroaccumulator 10 at connecting it to pump 24 or to the piston cavity of cylinder 3 and its return to the system;
– creation of different variants of circuit distribution and combining the flow of fluids from power source – pump 24, from power source – hydraulic cylinder 3 and from power source – hydroaccumulator 10.

It follows from the presented schematic (Fig. 2) that HD is also quite a complex system with branching of energy flows in the form of sequential, parallel and individual circuits.

At the design stage of HD, according to [8–11], it is necessary to provide such basic indicators as:

– reliability of performance over a specified period of operation;
– manufacturability with the possibility of standardization and unification of units, nodes and parts in their fabrication;
– ergonomics in the composition of the sysytem "man – machine – environment";

– patent-legal norms that define the degree of protection when implementing new technical solutions, aimed at applying principles of energy saving by reducing friction forces in movable joints of parts and energy accumulation;

– aesthetics to assess outer properties;

– cost efficiency for setting the level of expenses for the design, production and operation (especially, creating conditions to provide for the process of diagnosing the condition of hydraulic drive).

Results of the improvement and uniformity of HD in the course of its design are presented in the scientific papers [6, 9, 12].

At the design stage of new machines with HD in line with the national and international standards, it is recommended to enable development and implementation of means for diagnosing its condition. To ensure this, it is necessary to design special sections in the pipeline for mounting the sensors of diagnostic devices [12].

Diagnosing the condition of HD under operating conditions by using modern tools is addressed in the research articles [12–14].

In particular, the diagnosing instrument [14] provides for determining the HD workability, registers information at the display of computer and highlights three levels of the condition; one of them, in particular, indicates the unacceptable level of using HD.

Application of the principle of energy recuperation necessitates designing reliable devices for its accumulation and return to the hydraulic system of machine. Positive results on the implementation of the principle of energy recuperation are represented in the studies, conducted for stationary and mobile machinery, in particular in papers [7, 15] and others.

The process of wearing out resilient seals and friction surfaces in the elements of HD requires consideration of their causes [16] and the development of measures to reduce their impact.

Articles [17, 18] describe structural solutions that are able to ensure conditions for the occurrence of centering hydrostatic and hydrodynamic forces to reduce friction forces in the movable joints of hydraulic drive's elements.

The information presented is intended to improve the process of HD functioning under operating conditions; however, particular attention should be paid to the process of HD design as it is, in particular, to the calculation of parameters for the pump and the hydraulic engines. Such studies are presented in [6, 19, 20]. At the HD design stage, calculations are performed and passport or standardized parameters of HD are selected.

In paper [19], the magnitude of diameter of the piston of hydraulic cylinder Dp (inner diameter of cylinder sleeve), provided it performs direct or reverse motion of the rod, respectively, is determined according to formulas:

$$\mathrm{Dp}=\sqrt{(4000\times \mathrm{Rz})/(\pi\times \mathrm{Pn}\times z)}, \tag{4}$$

$$\mathrm{Dp}=\sqrt{(4000\times \mathrm{Rz})/(\pi\times(1-\varepsilon^2)\times \mathrm{Pn}\times z)}. \tag{5}$$

Formulas (4) and (5) contain the following designations:
– Dp – estimated cylinder piston diameter, mm;
– Rz – effort transmitted by the hydraulic cylinder rod, kN;
– Pn – nominal pressure received by HD of machine, MPa;
– ε – coefficient, which is determined by the ratio of diameter of the rod to the diameter of piston and its magnitude is 0.3...0.9;
– z – number of hydraulic cylinders that work in parallel.

For a pump, the magnitude of volume of its working chamber, according to [19], is determined by formula

$$\mathrm{Vkp} = (\mathrm{Rz}\times \mathrm{Vw})/(n_n\times \mathrm{Pn}), \tag{6}$$

where n_n is the nominal rotation frequency of the pump shaft, rev^{-1}; V_w is the motion velocity (displacement) of the rod of hydraulic cylinder, m/s.

According to [20], the main element of HD is the pump whose magnitude of volume of the working chamber Vkp is calculated by formula

$$\mathrm{Vkp} = (\mathrm{Rz}\times \mathrm{Vw})/(n_n\times \mathrm{Pn}\times\eta_{hs}), \tag{7}$$

where η_{hs} is the total PE of hydraulic drive of the machine.

Based on Vkp, by the technical specifications, the pump is selected that is serially produced with the magnitude Vk. According to the results of choosing technical specification volume of the working chamber of pump Vk, hydraulic cylinder's piston diameter is determined by the following calculation dependences, with regard to direct or reverse direction of motion of the rod, respectively:

$$D_p=4{,}6\times\sqrt{(\mathrm{Vk}\times \mathrm{nn}\times\eta_{n\text{-}o})/\mathrm{Vw}}, \tag{8}$$

$$D_p=4{,}6\times\sqrt{(\mathrm{Vk}\times n_n\times\eta_{n\text{-}o})/(\mathrm{Vw}\times(1-\varepsilon^2))}, \tag{9}$$

where $\eta_{n\text{-}o}$ is the volumetric PE of the pump.

Based on the analysis of calculation dependences for the HD design stage, it was established that the difference between formulas (6) and (7) is characterized by the magnitude, which is defined as $(1/\eta_{hs})$. A discrepancy between dependences (4) and (5), and (8) and (9), is the magnitude $(\eta_{h\text{-}o}/\eta_{hs})^{1/2}$. It should be noted that such a discrepancy between the indicated dependences when fabricating HD can cause to its overload under operating conditions at the action of loads that correspond to the level of initial data, accepted at the stage of design. Based on the aforementioned, one should conclude that to eliminate this shortage and to identify causes of the occurrence of these differences, it is necessary to conduct additional study based on the systems approach.

3. The aim and tasks of the study

The aim of present work is to substantiate a design scheme of HD with the allocation of energy flow from the hydraulic pump to the hydraulic engines in the form of parallel, serial and individual circuits with hydraulic engines of translational or rotational motion of their controlling elements of rod or shaft. By the results of the accepted scheme of energy distribution from the pump to the hydraulic engines, to provide for the development of calculation technique and for the choice of technical specifications or standardized parameters of the pump and hydraulic engines when designing HD by employing the systems analysis.

To achieve the set aim of scientific study, the following tasks are to be solved:

– to determine appropriate circuit connection "pump – hydroallocator – hydraulic engines" out of the known energy distribution schemes – sequential, parallel and individual – which one is the most suitable for use in hydroficated machines;

– to compose simplified schemes of HD and the calculation model "pump – hydroallocator – engine" with translational and rotational motion of their controlling elements;

– to obtain dependences for the calculation and selection of parameters of hydraulic pump and hydraulic engines at the HD design stage based on employing the systems analysis.

4. Materials and methods of examining HD at the design stage

As shown in Fig. 1, the structure of modern machines is quite a complex system, which is considered in the form of an organized cybernetic system. Each component of the system and the system as a whole can be tackled by the systems analysis [21]. In particular, when considering HD as the object of study, it is represented in the form of a complex system with many elements, each of which is characterized by its inner state that depends on the purpose, design and functional features and its input and output parameters.

Based on [21], HD is represented in the form of a "black box" with three components: input and output, as well as structural, parameter the latter of which reflects the inner state of the system. A model of such an object of research can be represented as shown in Fig. 3.

The examined HD model is actually under the influence of multidimensional vectors. In particular: $X=\{x_1, x_2..., x_n\}$ and $Y=\{y_1, y_2..., y_n\}$ characterize the input and output multi-dimensional vectors; $S=\{s_1, s_2..., s_n\}$ characterize vector of (inner) structural parameters.

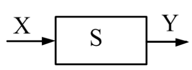

Fig. 3. Cybernetic model (model of the examined system): X — input; Y — output; S — inner state of the object of study

If input X and output Y are known, then [21] the inner state of such a system is determined by dependence

$$S=Y/X, \tag{10}$$

where S is the inner state of a complex system that expresses the proportional conversion of system's input into output.

The inner state of a complex system – of hydraulic drive and its separate elements [22], in the essence of the physics of process, characterizes an indicator that is represented in the form of an overall PE of the system and its components. This indicator is accepted as the main one at the HD design stage.

5. Results of examining HD at the design stage

It is known, based on article [6], that the distribution of energy when connecting the pump to the hydraulic engine through the slide valve hydroallocators can be achieved according to the parallel, sequential and individual circuits, which are shown in Fig. 4, *a–c*.

The application of one or another scheme depends on the peculiarities of functioning of machines various by purpose. Characteristic differences in the operation of such schemes are as follows.

Thus, Fig. 4, *a* shows a parallel scheme of energy distribution when connecting one pump, for example, to two hydraulic engines that set into motion various operating equipment of machine when the fluid under pressure P is fed through pressure channel H into working cavities of both hydraulic engines, for example, A and A1 or to cavities B and B1 through the channels of two included hydroallocators. From the non-working cavities of both hydraulic engines, through discharge channels C1 or C2 from both hydroallocators, the fluid is displaced into the oil tank through drain pipeline T.

The magnitude of volume of the fluid that is fed to two hydraulic engines through the channels of both hydroallocators through such a connection scheme is divided between them in inverse proportion to their external loads. In this case, fluid pressure depends on the magnitude of external loads, which are received by these hydraulic engines.

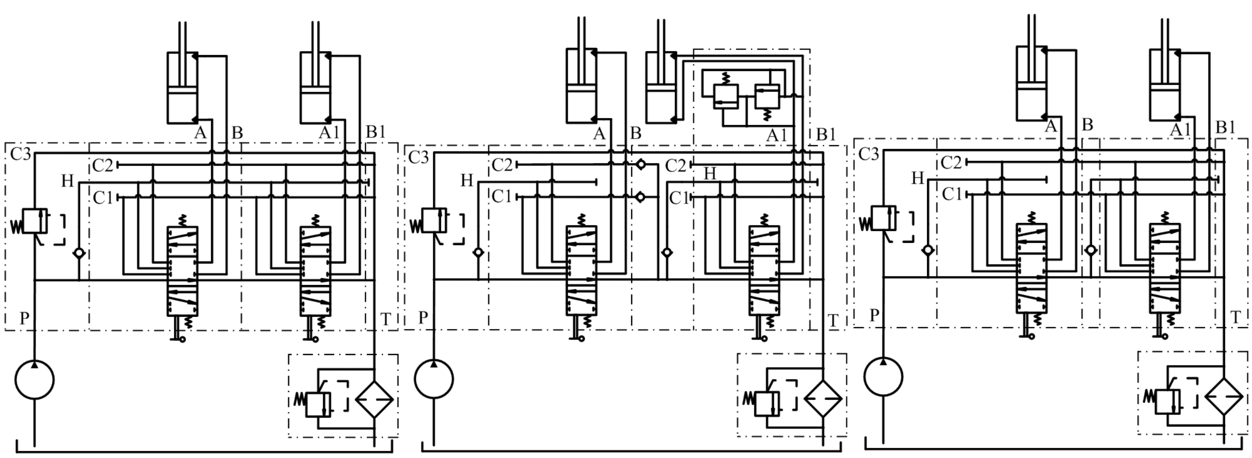

Fig. 4. Schemes of energy distribution from the pump to the hydraulic engines: *a* — parallel; *b* — sequential; *c* — individual. Designations in formulas (4) and (5) characterize the following: A, B, A1, B1 — cavities of hydraulic engines, the first and the second, respectively; C1, C2 — drain pipelines that are connected with the allocator; C3 — drain pipeline that is connected to the safety valve; H — pressure channel; P — total pressure provided by the pump; T — pressure in the drain pipeline

In the sequential scheme of energy distribution that is shown in Fig. 4, *b*, connection of one pump to two hydraulic engines is enabled by supplying the fluid under pressure P through pressure channel H to one of the cavities of the nearest hydraulic engine, for example, A. From another cavity of the same (first) hydraulic engine, liquid B passes through the channels of the first and second hydroallocator into the working cavity, for example, A1, of the second hydraulic engine. From the non-working cavity B1 of the second hydraulic engine, the fluid is displaced through the channel of the second hydroallocator and through pipeline T to discharge into the oil tank. According to this, the volume of fluid consumed by each of hydraulic engines is the same.

Based on this, the same velocity of the plunger rod motion of two hydraulic cylinders or shafts of two hydraulic engines is ensured while the working pressure in each of the next hydraulic engine is equal to the pressure at the output of the preceding one. Therefore, pressure at the output of pump is determined by the sum of pressure resistances of the turned hydraulic engines that provide for the execution of useful work.

Individual scheme of energy distribution to two hydraulic engines that is shown in Fig. 4 is enabled by feeding the fluid under pressure P along pressure channel H from the pump, which is sent to the working cavity of hydraulic engine at its turning of one hydroallocator, and another one when it is not turned. Discharge of the fluid into the oil tank from the non-working cavity of hydraulic engine is provided through the second channel in the same hydroallocator. Supply of the fluid to the next cavities of the second hydraulic engine is shut by the slide valve of hydroallocator, which was switched on first. In order to perform useful work by another hydraulic engine, it is necessary to turn off the slide valve of the first hydroallocator and turn on the slide valve of the second hydroallocator. The individual scheme is designed to enable alternate motion of different controlling elements of machine in the hydraulic systems with several hydraulic engines.

An analysis of all three schemes revealed that the individual circuit of connecting one pump to hydraulic engines is predominant. Such a scheme of energy distribution from the pump to the hydraulic engines provides for the optimal HD loading of machines different by purpose.

At the HD design stage, it is implied that the design specialist detected in advance the need for applying one of the examined circuits for connecting multiple hydraulic engines to the one source of hydraulic energy – the pump.

Given the aforementioned, we shall consider the process of designing HD for the individual scheme of connection between the pump and the hydraulic engine. Based on the systems approach, HD as an object of research is represented (simplified) in the form of two subsystems functionally interconnected as shown in Fig. 5, *a*, *b*. In Fig. 5, *a*, *b*, the first subsystem I consists of a pump and an oil tank and the second subsystem II consists of a hydroallocator, pipelines and a hydraulic engine.

The difference of the second subsystem is characterized by the following. In Fig. 5, *a*, hydraulic engine is presented with the translational motion of the rod of the hydraulic cylinder. In Fig. 5, *b*, hydraulic engine is presented with the rotational motion of the shaft of hydraulic engine.

Designations in Fig. 4 are as follows:
– 1 – safety valve;
– 2 – return valve;
– 3 – pump;
– 4 – pressure pipeline;
– 5 – filter;
– 6 – oil tank;
– 7 – fluid distributor;
– 8 – electromagnetic control of allocator;
– 9 – single-acting hydraulic cylinder;
– n_n – pump shaft rotation frequency;
– Pn – fluid pressure at the pump output;
– Vw – motion velocity of hydraulic cylinder rod;
– Rz –external load received by the hydraulic cylinder rod;
– M_k – torque received by the hydraulic engine shaft;
– ω_m – hydraulic engine shaft rotation frequency;
– Vk, Vm – technical specification value of volume of the working chamber of the pump and the engine, which is determined based on the calculated data;
– D, d – standardized values of diameters of piston and rod of the hydraulic cylinder that are determined based on the calculated data;
– I – subsystem pump;
– II – subsystem of allocator, pipeline, hydraulic engine;
– A, B1, B2 – hydroallocator's positions, neutral and two working, respectively.

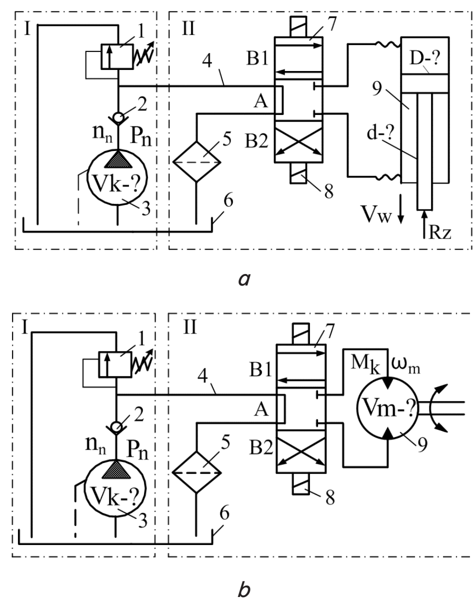

Fig. 5. Schematic of HD with two subsystems when using: *a* — hydraulic cylinder with the translational motion of rod; *b* — hydraulic engine with the rotational motion of shaft

The inner state of the system and of each element of this system is determined by the magnitude of PE and depends on the input and output of the system.

The system and each of the constituent elements of the system are characterized by their input, output and inner state. For the system or its element, input is the magnitude of power consumption. The output, for the system or its element, characterizes the magnitude of useful power.

Thus, these two components (input and output) are different in the magnitude of power. Input and output, both for the system and each element, differ from each other by the magnitude of overall performance efficiency.

According to the HD circuits shown in Fig. 5, we developed estimated models for the objects of research, divided, accordingly, also into two subsystems I and II with their inputs, outputs and inner state. Simplified models of HD are shown in Fig. 6.

Designations in Fig. 6 are as follows:

– Nvx, Nvux – power at the input and output of the object of research;

– η_n, η_{rtz}, η_{rtm} – PE of subsystem of the pump and of subsystem of allocator, pipelines and hydraulic engine, respectively, hydraulic cylinder and hydraulic motor;

– η_{hsz}, η_{hsm} – total PE of HD with hydraulic cylinder and hydraulic motor, respectively;

– Nn – pump power, defined as Pn·Qn;

– Pn – fluid pressure at the output of the pump;

– Qn – pump feed;

– Rz – load received by the hydraulic cylinder rod;

– Vw – motion velocity of the hydraulic cylinder rod;

– M_k – torque received by the hydraulic engine shaft;

– ω_m – rotation frequency of the hydraulic motor shaft.

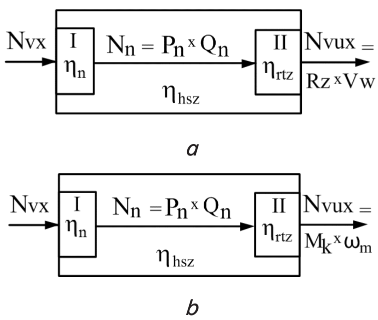

Fig. 6. Estimated models of HD with hydraulic engine: *a* — translational motion of the cylinder rod; *b* — rotational motion of the hydraulic motor shaft

Fig. 6, *a*, *b* shows HD in the form of mathematical models based on the application of systems approach.

Based on examining the subsystem I, according to Fig. 6, *a*, the inner state is determined as

$$\eta_n = Nn / Nvx, \tag{11}$$

where Nn is the power at the output of the pump.

By solving equation (11) we shall determine power at the input of HD model with regard to components Pn and Qn for the subsystem I and obtain the following equation

$$Nvx = Nn / \eta_n = (Pn \times Qn)/\eta_n . \tag{12}$$

Pump feed at its output is determined by formula

$$Qn = Vkp \times n_n, \tag{13}$$

where Vkp is the estimated magnitude of volume of the working chamber of the pump; n_n is the rotation frequency of the shaft of the pump.

The optimal shaft rotation frequency in the majority of modern pumps n_n is the magnitude at the level of 1500 min^{-1}. It is known that according to this shaft rotation frequency, its optimal PE is ensured. If this condition is satisfied, the maximum PE of HD as a whole is also provided.

For this purpose, it is necessary to employ special mechanisms that align the rotation frequency of the primary motor shaft with the rotation frequency of the pump shaft. To align various rotation frequencies of the shaft of the pump and the engine (electric motor or internal combustion engine), it is necessary to install a reducer or a multiplier, the gear ratio of which is determined relative to the values of their rotation frequency.

Dependence (12) with regard to (13) will take the form

$$Nvx = Nn / \eta_n = (Pn \times Vkp \times n_n)/\eta_n . \tag{14}$$

Based on examining the object of study as system as a whole, the inner state is defined by formula

$$\eta_{hsz} = Nvux / Nvx. \tag{15}$$

Through parameters Rz, Vw that characterize the output of the 2nd subsystem, we shall determine power input for the HD model HD as a whole, employing (15), we obtain the following equation

$$Nvx = (Rz \times Vw)/\eta_{hsz} . \tag{16}$$

Since equations (14) and (16) characterize the input of the same system, their right components can be equated, as a result we shall obtain

$$(Pn \times Vkp \times n_n)/\eta_n = (Rz \times Vw)/\eta_{hsz} . \tag{17}$$

From equation (17) we shall determine the estimated magnitude of volume of the working chamber of pump Vkp by formula

$$Vkp = (Rz \times Vw \times \eta_n)/(Pn \times n_n \times \eta_{hsz}). \tag{18}$$

The inner state η_{hsz} of the object of study, shown in Fig. 6, *a*, and, taking into account the connection of subsystem I to subsystem II, based on the systems approach, is determined by formula

$$\eta_{hsz} = (Nvux/ Nn) \times (Nn/ Nvx) = \eta_n \times \eta_{rtz}. \tag{19}$$

As a result of substituting equation (19) into (18), we receive

$$Vkp = (Rz \times Vw)/(Pn \times n_n \times \eta_{rtz}). \tag{20}$$

Upon determining by the calculation the magnitude of volume of the working chamber of pump Vkp, based on reference data, we determine technical specification value for this pump parameter, which we denote as Vk.

The inner state η_{rtz} for the subsystem I is determined by equation

$$\eta_{rtz} = (Rz \times Vw)/(Pn \times Qn). \tag{21}$$

It is a common knowledge that PE η_{rtz} for the subsystem II is determined through two components, in particular

$$\eta_{rtz} = \eta_{rtz\text{-}o} \times \eta_{rtz\text{-}hm}, \tag{22}$$

where $\eta_{rtz\text{-}o}$, $\eta_{rtz\text{-}hm}$ are the volumetric and hydromechanical PE of the subsystem II, respectively.

Pump feed at the input pump of the subsystem II is determined by formula

$$Qn = (F_n \times Vw)/\eta_{rtz\text{-}o}, \tag{23}$$

where F_n is the working surface of hydraulic cylinder, which receives the appropriate fluid pressure, which provides for the effort at the level Rz.

Based on (23), dependence (21) will take the form

$$\eta_{rtz} = (Rz \times \eta_{rtz\text{-}o})/(Pn \times F_n). \quad (24)$$

Using (24), we shall define components Rz and Pn for this dependence:

$$Rz = (Pn \times F_n \times \eta_{rtz}) / \eta_{rtz-o}, \quad (25)$$

$$Pn = (Rz \times \eta_{rtz-o}) / (F_n \times \eta_{rtz}). \quad (26)$$

The inner state for the entire model, which is characterized by dependence (15), when replacing the input and output parameters with their constituents, will be determined by formula

$$\eta_{hsz} = (Rz \times Vw \times \eta_n)/(Pn \times Qn). \quad (27)$$

Substituting equations (13), (25) and (26) into formula (27), we shall receive dependence in the form

$$\eta_{hsz} = (Pn \times F_n^2 \times \eta_{rtz}^2 \times Vw \times \eta_n) / (Rz \times \eta_{rtz-o} \times Vk \times n_n). \quad (28)$$

By solving equation (28) relative to F_n, we shall obtain

$$F_n^2 = (Rz \times \eta_{rtz-o} \times Vk \times n_n \times \eta_{hsz}) / (Pn \times \eta_{rtz}^2 \times Vw \times \eta_n). \quad (29)$$

Working surface F_n of the piston and rod cavities of hydraulic cylinder is determined, respectively, by formulas:

$$F_n = (\pi \times D_p^2)/4, \quad (30)$$

$$F_n = (\pi \times D_p^2 \times (1-\varepsilon^2))/4, \quad (31)$$

where D_p is the estimated diameter of the hydraulic cylinder piston.

By solving equation (29), with regard to (30), (31) and (19) and the motion direction of the hydraulic cylinder rod for direct and reverse motion, respectively, we shall receive:

$$D_p = \sqrt[4]{\frac{16 \times Rz \times Vk \times n_n}{\pi^2 \times Pn \times Vw \times \eta_{rtz\text{-}hm}}}, \quad (32)$$

$$D_p = \sqrt[4]{\frac{16 \times Rz \times Vk \times n_n}{\pi^2 \times (1-\varepsilon^2)^2 \times Pn \times Vw \times \eta_{rtz\text{-}hm}}}. \quad (33)$$

Based on the estimated magnitude of diameter of the hydraulic cylinder piston, we accept its standardized value D, by which the cylinder will be made.

For the object of study shown in Fig. 6, *b*, the inner state of the system as a whole, which is denoted as η_{hsm}, taking into account that $Qn = Vkp \times n_n$, then the state of the whole system is determined by formula

$$\eta_{hsm} = Nvux/Nvx = \\ = (M_k \times \omega_m \times \eta_n)/(Pn \times Vkp \times n_n). \quad (34)$$

The solution of equation (34) relative to Vkp will take the form

$$Vkp = (M_k \times \omega_m \times \eta_n) / (Pn \times n_n \times \eta_{hsm}). \quad (35)$$

Based on the relationship between subsystem I and subsystem II, the inner state η_{hsm} of the HD model as a whole, according to Fig. 6, *b*, is determined by formula

$$\eta_{hsm} = Nvux / Nvx = \\ = (Nvux/ Nn) \times (Nn/ Nvx) = \eta_n \times \eta_{trm}. \quad (36)$$

By replacing parameter η_{hsm} in (36) with dependence (34), we shall receive

$$Vkp = (M_k \times \omega_m) / (Pn \times n_n \times \eta_{rtm}). \quad (37)$$

By the results of calculating Vkp, as the magnitude of volume of the working chamber of the pump for the model shown in Fig. 6, *b*, we shall select, based on reference data, technical specification value of the pump parameter. The accepted technical specification value of the pump parameter that is serially produced, will be denoted as Vk.

Parameter η_{rtm} that characterizes the overall PE of the subsystem II is determined by formula

$$\eta_{rtm} = \eta_{rtm\text{-}o} \times \eta_{rtm\text{-}hm}, \quad (38)$$

where $\eta_{rtm\text{-}o}$, $\eta_{rtm\text{-}hm}$ are the volumetric and hydromechanical PE of the subsystem II, respectively.

The examined system operates reliably under condition that all of the pump feed is consumed by hydraulic engine, that is,

$$Qn = Qm. \quad (39)$$

By replacing Qm in equation (39) with its components, we shall obtain

$$Qn = (Vkpm \times n_m)/\eta_{rtm\text{-}o}, \quad (40)$$

where Vkpm is the estimated volume of the working chamber of hydraulic engine.

The inner state of the subsystem II η_{rtm}, shown in Fig. 6, *b*, is determined by formula

$$\eta_{rtm} = (M_k \times \omega_m)/(Pn \times Qn). \quad (41)$$

By replacing in equation (41) parameter Qn with formula (40) and, upon determining Vkpm in the received dependence, we shall obtain

$$Vkpm = (M_k \times \omega_m \times \eta_{rtm\text{-}o}) / (Pn \times n_n \times \eta_{rtm}). \quad (42)$$

By calculating, according to formula (42), the magnitude of volume of the working chamber of hydraulic engine Vkpm for the model shown in Fig. 6, *b*, we shall select, based on reference data, its technical specification value. By the results of choosing technical specification value close to the estimated parameter, we shall denote it as Vm.

When comparing the results obtained in the calculation of diameters of hydraulic cylinder piston D by dependences (32) and (33) relative to (4), the difference is $(1/\eta_{rtz\text{-}hm})^{1/4}$. The same comparison of results for dependences (8) and (9) relative to (4) yields difference $(\eta_{n\text{-}o}/\eta_{hs})^{1/2}$. When comparing results for dependence (4), which was accepted as the basis, the difference is 1. Since $\eta_{hs} < \eta_{n\text{-}o}$, and $\eta_{rtz\text{-}hm} \approx \eta_{n\text{-}o}$, then, based on this, it is possible to make up a ratio, which is represented in the form

$$\sqrt[4]{1/\eta_{rtz\text{-}hm}} < \sqrt{\eta_{n\text{-}o}/\eta_{hs}} > 1. \quad (43)$$

Comparison of the results obtained for the calculation of volume of the working chamber of pump Vкр by dependence (20) relative to (6) yields the $1/\eta_{rtz}$ difference. Similar comparison for dependence (7) relative to (6) yields the $1/\eta_{hs}$ difference. The same comparison for dependence (6), which was accepted as the basis, yields the difference of 1. Since $\eta_{hs}<\eta_{rtz}$, then, according to the results obtained, it is possible to make up a ratio, which is represented in the form

$$1/\eta_{rtz} < 1/\eta_{hs} > 1. \quad (44)$$

Dependences (43) and (44) demonstrate the difference between those received and those known results of the study. The level of difference between the obtained results is determined through the overall HD PE and the PE components.

6. Discussion of research results obtained for the HD design stage

Theoretical dependences, obtained for the calculation of parameters of the pump, hydraulic cylinder and hydraulic engine at the HD design stage based on the application of systems analysis, allow us to state the following. The calculation of parameters of the pump and hydraulic cylinder to connect the HD elements according to the circuit "pump – hydroallocator – hydraulic cylinder" when using research data, formulas (20), (32) and (33), compared with the well-known studies [19], formulas (4), (5) and (6), reveals the understatement of their estimated magnitudes. When comparing results of the present research relative to the well-known studies [20], formulas (7), (8) and (9), the results of calculating parameters of the pump and hydraulic cylinder are overstated.

The calculation of parameters of the pump and hydraulic engine to connect the HD elements according to the scheme "pump – hydroallocator – hydraulic engine" is performed based on dependences (37) and (42).

In comparison with the well-known studies, the dependences obtained are presented for the first time and they have a substantial advantage. In particular, they provide for a selection of technical specification values for the pump and the hydraulic engine and the standardized parameters of hydraulic cylinder piston and rod at maximum accuracy. Based on this, we manage to maintain with high accuracy the input and output parameters of HD, accepted at the stage of its design. Consequently, when manufacturing such an HD, under operation conditions, a decrease in the load is assured at its functioning. When designing other elements of HD, the calculation and selection of technical specification values and standardized parameters coincide with the known technique [19].

We plan to conduct experimental studies when creating HD according to the circuit "pump – allocator – hydraulic motor".

7. Conclusions

Research results allowed us to establish the following.

1. Of the known schemes of energy distribution from the pump to the hydraulic engines, which enable sequential, parallel and individual circuit, we accepted in the present study the individual scheme of supplying energy to the hydraulic motor. The advantage of this decision is the circumstance, which we also defined, that due to it, the maximum level of pump load is attained according to the maximum load of the HD hydraulic engine. Due to this, the advantages of individual scheme of energy distribution over other schemes are used to the fullest.
2. Based on the application of individual scheme of energy distribution to the hydraulic engines, we compiled simplified circuits of the drive and calculation models in the form of "pump – hydroallocator – cylinder" and "pump – hydroallocator – engine". Their characteristic feature is that they are presented in the form of two subsystems interconnected sequentially.
3. For the HD design stage, by the results of employing the systems analysis, we obtained new dependences for the calculation and, based on them, the selection of hydraulic pump and hydraulic engine technical specification values and the standardized parameters of hydraulic cylinder. They represent a dependence for the calculation, at first, energy source – the pump, which takes into account its functioning under conditions of complex system in contrast to the known dependences. The subsequent dependences are the formulas for calculating the parameters of hydraulic engines with the translational and rotational motion of the rod and shaft. These dependences are obtained with regard to the work of hydraulic engines under conditions of functioning of a complex system and the action of all influential factors received by the system at input and output, as well as the inner state.

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