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RESEARCH ARTICLE

Diagnosis of Hydrostatic Transmission via Generation of Active Hydraulic Load

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ABSTRACT

Most actuation processes in agricultural machinery are hydraulically powered. Compactness, high specific power, and stepless speed control determine the application of hydraulic drives in the propulsion systems of grain harvesters and tractors. High operational intensity and complex supporting surface topography often lead to overload conditions and, consequently, to failure of hydrostatic transmission (HST) components. The complexity of installation, dismantling, and repair operations for HST components necessitates efficient and reliable diagnostics. Unambiguous rejection of tested units is only possible with accurate determination of parameters such as leakage volume and volumetric efficiency. Determination of these parameters requires the creation of test operating conditions for HST units, characterized by high flow rates and pressures in hydraulic lines. In addition, verification or adjustment of high-pressure valves in HST is carried out under maximum pressures and low flow rates. Comprehensive bench testing of refurbished units requires the application of an external load to the hydraulic motor shaft. This approach formed the basis for the development of a test bench for diagnosing HST components. An axial piston hydraulic motor with a displacement of 160 cm3 was used as the external loading device, with load applied through throttling. The pump was driven by an asynchronous motor with inverter control. For accurate measurement of input and output power and determination of losses in the supports, the loading hydraulic motor and the electric motor were mounted on strain gauges. Monitoring of real-time readings, shaft rotational speeds, pressures, and flow rates in hydraulic lines was ensured by an appropriate sensor system. Calculation of current output characteristics for the hydraulic motor and pump, based on established relationships, was performed by an embedded computer. This approach made it possible to vary loads over a wide range and not only record current HST characteristics but also calculate leakage, volumetric and hydromechanical efficiency, evaluate the power balance and the accuracy of its calculation, as well as rotational speeds. Control of pump shaft speed and monitoring of its current value, along with the creation of load on the hydraulic motor shaft, enabled precise determination of the flow coefficient of hydraulic machines with unknown displacement.

INTRODUCTION

Modern agricultural, tractor, and road machinery is characterized by a high degree of hydraulic actuation of working motions. The operating conditions of such equipment are distinguished by high intensity, movement over unprepared road surfaces or off-road terrain, and prolonged operation under heavily loaded regimes. These conditions adversely affect the service life of hydraulic machines used in actuators and propulsion systems [1–4]. One of the most heavily loaded hydraulic systems in agricultural machinery is the hydrostatic transmission (HST). The HST is based on axial piston hydraulic machines, whose failure rate during the pre-repair operational period accounts for approximately 20% of the total number of machine failures [4]. Downtime of grain harvesters due to HST failures can reach 22%. Moreover, it has been established that more than 75% of pumps and hydraulic devices are incorrectly rejected [6] and unjustifiably disassembled during repair attempts. A distinctive feature of HST operation is the closed-loop configuration, high stiffness of high-pressure hoses, elevated pressures and flow rates of the working fluid in the lines, and the positioning of hydraulic motors significantly below the hydraulic reservoir level, which complicates or renders impossible the effective use of existing portable diagnostic devices under field conditions. However, during field testing of HST, no difficulties arise in creating operational loads at its output elements.

Among the existing methods for diagnosing the condition of HST hydraulic machines, dynamic and static approaches can be distinguished [7]. The static method requires prior determination of coefficients describing the dependence of clearances and leakage in the hydraulic machine on pressure and rotational speed. The dynamic method, which is closest to the actual HST operating process, involves determining volumetric characteristics of hydraulic machines under their operating conditions. Within this approach, statoparametric and kinematic methods are of particular interest. The first method involves installing sensors to measure pressures and flow rates in the hydraulic drive lines, enabling real-time determination of flow and energy characteristics of the fluid in the lines. Its drawback is the need to integrate sensors into the system under study, which is not always feasible in field conditions due to the requirement for specialized fittings, hydraulic components, and connection points that are not available in standard machinery. Additionally, installation and removal of sensors may lead to contamination of the working fluid. The second method involves measuring input and output mechanical parameters using speed and force sensors, allowing analysis of both power and energy losses in the system. While monitoring rotational speeds of pump and hydraulic motor shafts is feasible under field conditions with existing instruments, real-time control of torque on the shafts of hydraulic units in production machines is difficult to implement and requires structural modification of machine components. Effective combination of the advantages of both methods while minimizing their limitations is achievable only in specialized test bench equipment.

Hydraulic test benches for HST employ both static testing methods [GOSNIITI] and dynamic ones. In the static method, the main characteristics of the hydraulic drive are recorded with the hydraulic machine shaft stationary; examples include KI test benches developed by GOSNITI [8] and later by the Federal Scientific Agroengineering Center VIM [9]. In practice, limiting characteristics of the HST are obtained under activation of high-pressure relief valves. The disadvantage of this method is that the operating condition of the hydraulic machines is emergency (fault) mode, and only a portion of the hydraulic motor pistons is under load, which restricts testing to short durations. Taking these features into account, the KI-28097-03M test bench provides for adjustable loading on the hydraulic motor shaft. More versatile is the testing of hydraulic machines and HST under dynamically generated load [10, 11]. This approach is implemented in domestic test benches such as SGN110 with tank module and BIM hydraulic motor testing attachment [12]; SI-110 test bench manufactured by PSM-Engineering LLC; and the universal test bench SIU 55-3 produced by LLC “Spets-Proekt” [13]. Among foreign manufacturers, dynamic testing methods are implemented in systems such as the Model 850 Hydraulic Test Center by AIDCO Test Systems and the Model HTB (hydraulic test bench) by Schroeder Industries. Analysis of the characteristics of the listed equipment shows that real-time monitoring of such an important parameter as torque on the hydraulic motor shaft is implemented either indirectly or as an optional feature, and only in foreign models, whereas hydraulic motor manufacturers specify both nominal and maximum torque values [14].

Torque on the shaft of a hydraulic machine can be determined indirectly if the current displacement and pressure differential are known; however, the actual value of hydromechanical efficiency cannot be taken into account in this case. Direct measurement of torque and rotational speeds of HST hydraulic machine shafts enables not only determination of this critical parameter but also an energy audit of test efficiency.

The aim of the study is to develop test bench equipment capable of monitoring the input and output parameters of HST hydraulic machines, as well as a methodology for their periodic and acceptance testing.

The objectives of the study are as follows

To propose an effective diagnostic methodology for HST hydraulic units;

To develop a test rig for diagnostics of HST units;

To confirm the validity of the applied methodology and the measurements performed.

MATERIALS AND METHODS

For acceptance and periodic testing of hydraulic machines used in positive-displacement hydraulic drives, the following parameters must be controlled and measured in real time: rated pump delivery and hydraulic motor flow rate; pump discharge pressure,pressure differential across the hydraulic motor; working fluid temperature; torque on the pump and hydraulic motor shafts; and rotational speeds of the pump and hydraulic motor shafts. For hydraulic machine diagnostics, the following indicators must be determined in real time: pump delivery coefficient, pump volumetric efficiency; effective torque; and hydromechanical efficiency of the pump and hydraulic motor. For monitoring test energy consumption, input and output power must also be determined in real time.

To address these objectives, a statoparametric method for measuring HST parameters was employed. Pressures and flow rates in the hydraulic lines, rotational speeds of the pump and hydraulic motor shafts, and torques on these shafts were measured continuously. Monitoring of system parameters was implemented using appropriate sensors, the data from which were transmitted to digital indicators and to the test bench controller. The indicators primarily served as setpoint devices for threshold values of pressure and temperature, which the operator could promptly adjust before or during testing. Display of current pressure and temperature values on the indicators was of secondary importance, since all current parameters were shown on the mnemonic diagram displayed on the test bench computer monitor.

The strain-gauge-based torque measurement system for the pump and hydraulic motor shafts made it possible not only to determine the power balance and hydromechanical efficiency, but also to provide real-time verification of the performance of the computational and measurement system, thereby preventing gross errors and drift in the determination of the operating parameters of HST components. Torque and power balance were determined from direct measurements of rotational speed and torque, but could always be indirectly compared with nominal theoretical values based on the measured pressures, flow rates, and displacement of the hydraulic machines. This method enabled calculation not only of the input and output power of the tested units, but also of hydromechanical and volumetric efficiency, upon reaching threshold values of which the unit was rejected.

Testing of repaired or tested hydraulic units should be classified primarily as acceptance testing rather than periodic testing. The purpose of such testing is to verify that the parameters of the hydraulic machine under test conform to reference (nameplate) values. The measuring equipment of test benches intended for acceptance testing must comply with accuracy class 3, whereas equipment for periodic testing must comply with accuracy class 2. According to GOST 17108-86, the permissible total measurement errors for accuracy class 3 in acceptance testing are as follows: pressure, flow rate, and rotational speed, ±2.5%; torque, ±2.0%. In acceptance and periodic testing of positive-displacement hydraulic drives, both foreign and domestic manufacturers consider monitoring of the following parameters essential. The permissible total measurement errors for accuracy class 2 in periodic testing are as follows: pressure and flow rate, ±1.5%; rotational speed, ±1.0%; torque, ±1.5%.

To implement the tests, the bench was equipped with load-generating and torque-measuring devices, flowmeters for high-pressure lines, and pressure and rotational speed sensors.

In designing the bench, account was taken of the fact that, in accordance with GOST 14658-86, acceptance and periodic testing of pumps must be performed at rated operating pressure, rated shaft speed, and rated displacement. However, for pump power up to 200 kW, the following parameters may be reduced below their nominal values: shaft rotational speed by 40%, and pressure or pressure differential by 20%. Likewise, in accordance with GOST 20719-83, all types of testing of hydraulic motors and reversible machines must be carried out at maximum displacement, rated operating pressure, and rated shaft speed. However, for power up to 200 kW, a reduction of up to 20% from the nominal values of operating pressure and shaft rotational speed of the hydraulic motor is permitted. Based on these requirements, the test bench electric motor power was substantiated at 75 kW, which is consistent with earlier studies by Russian researchers [15]. With a 20% reduction in nominal values, this power rating makes it possible to test the operability of hydraulic machines with a displacement of up to 250 cm3. To control the pump shaft speed and provide soft starting, the bench electric motor was equipped with an inverter.

Depending on the type of hydraulic machine under test, whether a pump or a hydraulic motor, the working load may be generated by hydraulic or mechanical resistance. The load on the pump is generated and regulated either by throttling devices or by loading a reference hydraulic motor with an external braking device. Hydraulic motor tests are performed in both motor and pump modes; that is, the load may be generated in different ways, either by a hydraulic throttle or by an external braking device, while either a reference pump or the diagnostic pump from the tested HST may be used.

To implement the selected methodology, a test bench was designed and constructed, the schematic hydraulic circuit of which is shown in Figure 1. In the presented scheme, the diagnostic HST pair, the pump (HP) and the hydraulic motor (HM1), are interconnected by high-pressure lines and hoses through flange connections FA1.1, FA1.2, FA2.1, and FA2.2. The shaft of pump HP is connected to the shaft of electric motor EM through flexible coupling EC1. The shaft of hydraulic motor HM1 is connected to the shaft of the pump-motor HM2 of hydraulic brake HB by means of flexible coupling EC2. Rotational speeds of the hydraulic machine shafts are measured using speed sensors: sensor RS1 for the pump, and sensor RS2 for hydraulic motor HM1 and pump-motor HM2.

The bench hydraulic system is protected against excessive pressure in the high-pressure lines by relief valve SV. Hydraulic brake HB includes hydraulic motor HM2. The load on the shaft of the tested hydraulic motor HM1 is generated by throttle valve D in the hydraulic system of hydraulic brake HB. The pressure of the generated load is monitored by pressure sensor PS3. Protection against excessive pressure in the hydraulic brake circuit is provided by a pilot-operated relief valve. The hydraulic system of hydraulic brake HB is supplied from a dedicated tank T2 equipped with a filter, heat exchanger, and systems for monitoring the level and temperature of the working fluid.

The hydraulic units under diagnosis, HP and HM1, are supplied from tank T1, which is also equipped with a filter, heat exchanger, and level and temperature monitoring systems.

To prevent dry start-up, the casings of pump HP and hydraulic motor HM1 can be prefilled. This solution makes it possible not only to fill the hydraulic units, but also to remove air from the main hydraulic system.

Pre-start flushing of the hydraulic systems of pump HP and hydraulic motor HM1 is performed by prefill pumps FP1 and FP2, respectively. The hydraulic system of each of pumps FP1 and FP2 includes an electric motor EM, rigid coupling, check valve, and relief valve.

For preliminary filling of the diagnostic pump HP, the lower point of its casing is connected by a low-pressure hydraulic line, through quick-release coupling QRC2, to filling pump FP1. The upper point of the casing of pump HP is connected by a low-pressure hydraulic line, through quick-release coupling QRC3, to rotameter RFMP. The working fluid is then drained into tank T1 through filter F.

For preliminary filling of diagnostic hydraulic motor HM1, the lower point of its casing is connected by a low-pressure hydraulic line, through quick-release coupling QRC5, to filling pump FP2. The upper point of the casing is connected by a low-pressure hydraulic line, through quick-release coupling QRC4, to rotameter RFMM. The working fluid is then drained into tank T1 through filter F.

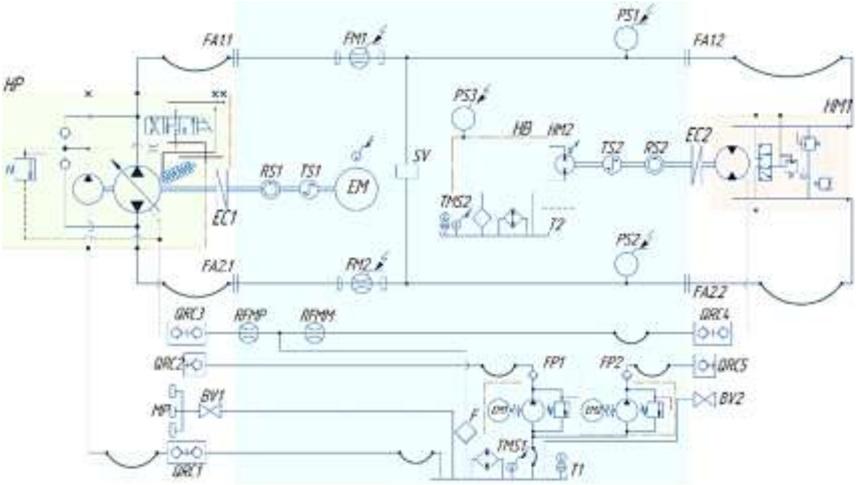


Figure 1: Schematic hydraulic diagram of the HST test bench

For filling the diagnostic hydraulic motor HM1, the filling pump FP2 is connected via a low-pressure line, through QRC5, to the lower point of the hydraulic motor casing. The upper point of the HM1 casing is connected by a low-pressure line, through quick-release coupling QRC4, to the rotameter flowmeter RFMM, after passing through which the working fluid is discharged into tank T1. The housings of rotameters RFMP and RFMM are transparent (Fig. 3), which allows visual monitoring of the presence of air and contaminants in the hydraulic systems and casings of the tested hydraulic units. Due to the possibility of contamination in the tested units, discharge from the filling system into tank T1 is carried out through filter F.

For improved filling of the cavities of hydraulic units and to prevent starvation of pump HP, tanks T1 and T2 are positioned on the upper part of the test bench. The suction line is equipped with a ball valve BV1 connected to the supply multiport MP associated with tank T1 (Fig. 3). The multiport is designed to facilitate simultaneous supply to multiple consumers from a single point. Ball valve BV2 is installed on the side of the tested hydraulic motor for technological operations or for functional expansion during testing of hydraulic devices and components such as hydraulic testers, control valves, and metering pumps.

Actual flow rates in high-pressure lines F1.1–F1.2 and F2.1–F2.2 are determined using flowmeters FM1 and FM2, respectively. Pressures in the high-pressure lines are measured by sensors PS1 and PS2. Monitoring of shaft rotational speeds, torques on the shafts of HP and HM1,2, and working fluid temperatures is performed using speed sensors RS1,2, torque sensors TRS1,2, and temperature sensors TMS1,2, respectively.

Sensor data are processed by the embedded computer of the test bench and displayed in real time on the monitor (Fig. 3).

Figure 2: External view of the assembled 3D model and the physical prototype

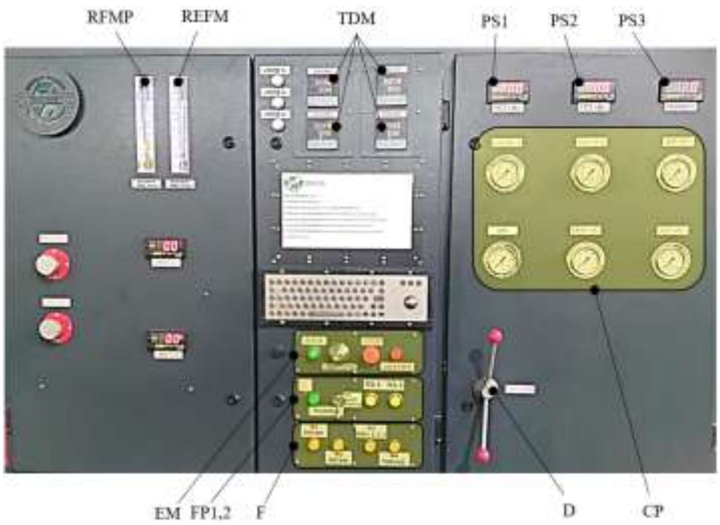


Figure 3: Front view of the test bench, control panel

On the central section of the control panel (Fig. 3), the following components are located: TDM indicators of temperature sensors for the electric motor EM, hydraulic motor HM2, and tanks T1 and T2; a monitor with keyboard; a control unit for electric motor EM; a control unit for prefill pumps FP1 and FP2 used for pre-filling hydraulic units; and a filter condition indication unit for the hydraulic test bench. On the right side of the control panel, the following are located: a group of mechanical pressure gauges CP and pressure indicators PS1, PS2, and PS3; and throttle valve D (handle removed) for controlling the load unit HB. The CP pressure gauges are connected to the monitoring points of the HST pump and hydraulic motor via microhoses and breakaway microunions located on the right and left side panels of the test bench. On the left side of the control panel, rotameters RFMP and RFFP are installed.

On the left side panel of the test bench (Fig. 4), the pump service points are located: flange adapters for high-pressure lines FA1.1 and FA1.2; supply multiport MP; supply valve BV1; quick-release couplings for the charge pump supply ORC1, inlet ORC2, and outlet ORC3 of the casing of the tested pump, respectively. Above the quick-release couplings QRS1, QRS2, and QRS3, five microunions of pressure monitoring points are located: pressure in the control channels of the swashplate tilt mechanism of the pump; discharge pressure of the charge pump (monitoring of the relief valve setting of the charge pump); manovacuum gauge upstream of the charge pump filter; and an additional pressure monitoring point.

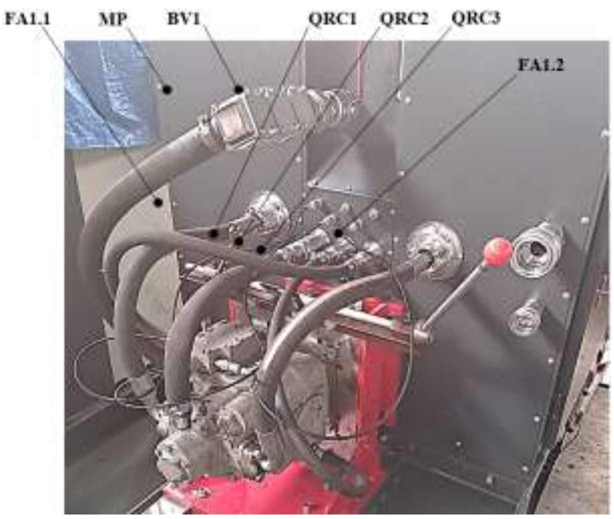


Figure 4: Elements of the left side panel of the test bench: FA1.1, FA2.1 – flange adapters of high-pressure lines; MP – supply multiport; BV1 – supply valve; ORC1, ORC2, ORC3 – quick-release couplings for the charge pump supply, and casing inlet and outlet, respectively.

Figure 5. Elements of the right side panel of the test bench: FA2.1, FA2.2 – flange adapters of high-pressure lines; BV2 – auxiliary supply valve; ORC4, ORC5 – quick-release couplings of the hydraulic motor drain circuit.

The right-side panel of the test bench (Fig. 5) contains the hydraulic motor connection points: flange connections FA2.1, FA2.2; quick-release couplings ORC4, ORC5; auxiliary supply valve BV2; and an additional pressure monitoring point. The pressure monitoring points are used to evaluate service pressure readings inside the pump and hydraulic motor.

One of the most common diagnostic methods for hydraulic machines is the determination of their volumetric efficiency under operating conditions or conditions close to them. A distinctive feature of this test is the continuous measurement of the hydraulic machine shaft rotation speed and flow rates in the high-pressure lines F1.1–F1.2 and F2.1–F2.2 at a specified operating pressure. Calculating the volumetric efficiency of a hydraulic machine requires continuous monitoring of pump delivery HP, shaft rotation speed, and precise knowledge of the current displacement of the hydraulic machine.

The volumetric efficiency is calculated using established formulas [16–18].

The volumetric efficiency of the pump is defined as:

$$\eta_{Vp} = \frac{Q_p \cdot 1000}{q_{Vp} \cdot n_p}, \tag{1}$$

where η_{vp} is the volumetric efficiency of the pump; q_{vp} is the pump displacement, cm^3; Q_p is the actual pump delivery, L/min; n_p is the pump shaft rotation speed, rpm.

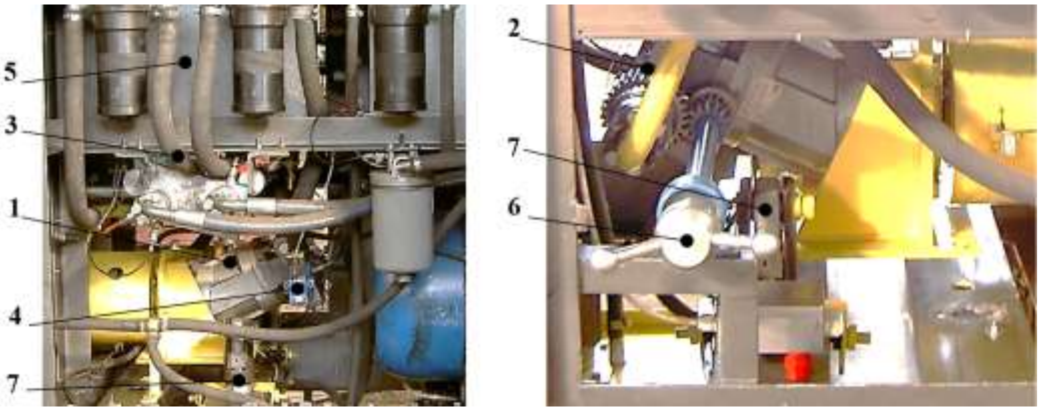
On the test bench described, the volumetric efficiency of the pump is calculated in real time using formula (1). The actual delivery under flow reversal is measured by flow meters FM1 and FM2 (Fig. 6), while the shaft rotation speeds of the hydraulic machines are measured by rotation sensors RS1 and RS2 (Fig. 7(a, b)

Figure 6: Flow meters FM1 and FM2 of the high-pressure lines of the test bench

a) b)

Figure 7: Hydraulic machine shaft rotation sensor assemblies: a – pump; b – hydraulic motor

The working pressure is generated by a load unit comprising a reversible axial-piston hydraulic machine (Fig. 8, item 1), a mechanically actuated throttle (Fig. 8, item 2) with a handwheel (Fig. 8, item 6), hydraulic control equipment (Fig. 8, item 3), overpressure protection (Fig. 8, item 4), and a reservoir (Fig. 8, item 5). Torque is measured by two strain gauge supports (Fig. 8, item 7). The load unit is connected to the shaft of the hydraulic motor under test via a coupling.



a) b)

Figure 8: Active load unit of the test bench: a – front view (door open); b – rear view (door open)

The calculated volumetric efficiency values are compared against the nameplate specifications. The criterion for reaching the pump's limit condition is a reduction in volumetric efficiency of more than 20% below the nameplate value [19]. Pump manufacturers' operating manuals often include volumetric efficiency in formulas while presenting the delivery coefficient in tabular form. A significant drawback of diagnosing a pump by determining its actual volumetric efficiency is the requirement to know the hydraulic machine displacement — which is impractical when it is necessary to reduce the power consumed during testing.

The second common diagnostic method for hydraulic machines is the determination of their delivery coefficient [16–18].

The advantage of this approach is that knowledge of the actual displacement value is not required. A distinctive feature of the delivery coefficient determination is that pump or motor flow measurements must be carried out in two stages. The first stage involves measuring the delivery or flow rate at the rated shaft rotation speed and the lowest achievable pressure, i.e., minimum external load. The second stage involves measuring the delivery or flow rate at the rated shaft rotation speed and rated external load.

The pump delivery coefficient is calculated as the ratio of delivery at minimum pressure to delivery at rated pressure

$$K_Q = \frac{Q_p \cdot n_{p0}}{Q_{p0} \cdot n_p}, \qquad (2)$$

where K_Q is the pump delivery coefficient; Q_p is the pump delivery at rated load, L/min; Q_{p0} is the pump delivery at minimum load, L/min; n_{p0} is the pump shaft rotation speed at minimum load, rpm; n_p is the pump shaft rotation speed at rated load, rpm.

The tests are characterized by the fact that any change in the loading condition invariably causes a change in the electric motor shaft speed — a phenomenon known as speed droop. This affects the instantaneous pump delivery. To compensate for speed droop, a correction factor in the form of the ratio of the rotation speed at minimum load to the rotation speed at rated load is introduced into formula (2).

It should also be noted that the HST pump incorporates a charge pump with a valve system, while the hydraulic motor includes a valve block with a flushing circuit. Consequently, flow rates in the HST lines may vary. For this reason, determining the hydromechanical efficiency of the hydraulic motor is particularly relevant, and the corresponding algorithm is implemented in the test bench described.

The hydromechanical efficiency of the pump is determined by the following formula [16–18]

$$\eta_{hp} = \frac{T_{tp}}{T_{pp}} = \frac{q_{Vp} \cdot \Delta p_p}{2\pi} \cdot \frac{1}{T_{pp}}, \quad (3)$$

where η_{hp} is the hydromechanical efficiency of the pump; T_{pp} is the actual (measured) pump torque, N·m; T_{tp} is the theoretical (ideal) pump torque, N·m; q_{vm} is the pump displacement, cm^3; Δp_m is the pressure differential across the pump, MPa.

The hydromechanical efficiency of the hydraulic motor is determined by the following formula [16–18]

$$\eta_{hm} = \frac{T_{pm}}{T_{tm}} = T_{pm} \frac{2\pi}{q_{Vm} \cdot \Delta p_m}, \quad (4)$$

where η_{hm} is the hydromechanical efficiency of the hydraulic motor; T_{pm} is the actual (measured) motor torque, N·m; T_{tm} is the theoretical (ideal) motor torque, N·m; q_{vm} is the hydraulic motor displacement, cm^3; Δp_m is the pressure differential across the hydraulic motor, MPa.

Torque measurement on both the pump and the hydraulic motor is performed using strain gauge supports connected in a full-bridge configuration to the bracket of each hydraulic machine. The strain gauges serve as the support for the hydraulic motor bracket of the active load unit (Fig. 8, item 7). The strain gauge supports are calibrated for a torque of 1,000 N·m, determined by the limit capabilities of the pump-motor of the loading device: displacement — 160 cm^3; maximum pressure — 42 MPa; hydromechanical efficiency — 0.96.

The overall efficiency of the pump is determined by the following formula:

$$\eta_p = \frac{N_{tp}}{N_{pp}} = \frac{Q_p \cdot \Delta p_p}{60} \cdot \frac{9549,3}{T_p \cdot n_p}, \quad (5)$$

where η_p is the overall pump efficiency; N_{pp} is the power consumed by the pump, kW; N_{tp} is the effective (ideal) pump power, kW; Q_p is the measured pump delivery, L/min; Δp_p is the pressure differential across the pump, MPa; T_p is the torque at the pump shaft, N·m; n_p is the measured pump shaft rotation speed, rpm.

Based on the dependencies presented above, the power and efficiency of the hydraulic machines are determined as follows.

Effective pump power:

$$N_{pp} = \frac{Q_p \cdot \Delta p_p}{60}, \quad (6)$$

Power consumed by the pump

$$N_{tp} = \frac{T_p \cdot n_p}{9549{,}3}, \tag{7}$$

The overall efficiency of the hydraulic motor is determined by the following formula:

$$\eta_m = \frac{N_{tm}}{N_{pm}} = \frac{T_m \cdot n_m \cdot 60}{Q_m \cdot \Delta p_m \cdot 9549{,}3}, \tag{8}$$

where η_{pm} is the overall hydraulic motor efficiency; N_{pm} is the power consumed by the hydraulic motor, $N_{pm} = N_{pp}$, kW; N_{tm} is the effective (output) power of the hydraulic motor, kW; Q_m is the hydraulic motor flow rate, L/min; Δp_m is the pressure differential across the hydraulic motor, MPa; T_m is the measured torque at the hydraulic motor shaft, N·m; n_m is the measured hydraulic motor shaft rotation speed, rpm.

Effective power of the hydraulic motor

$$N_{tm} = \frac{T_m \cdot n_m}{9549{,}3}, \tag{9}$$

Overall HST efficiency

$$\eta = \frac{N_{tm}}{N_{pp}}, \tag{10}$$

Figures 9–12 present a fragment of the acceptance test results for the HST 112 unit manufactured by JSC Hydrosila. The HST was configured with high-pressure relief valves set to 26 MPa and comprised a serviceable axial-piston variable-displacement pump of type PVH112/MH and an overhauled axial-piston fixed-displacement hydraulic motor MFH112.

Figures 9 and 10 show a fragment of tests conducted at the maximum pump displacement (111–112 cm^3) and a shaft rotation speed of approximately 1,000 rpm. The parameters of the overhauled hydraulic motor with a displacement of 111–112 cm^3 were recorded and calculated. All measured parameters are displayed on the bench monitor as a mimic diagram of the "HST Paired Check" test mode (Fig. 9). The test type is selected in advance from the main menu. The dynamic variation of test parameters is displayed over a 20-second time window in graphical form (Fig. 10). The plots show the test dynamics over a 30-second period. The graphical dependencies shown in Fig. 11 are examined in greater detail below. The first window shows the variation in pressures and flow rates in the HST high-pressure lines. The second window displays the dynamic variation in pump shaft rotation speed and torque. The third window shows the variation in hydraulic motor shaft rotation speed and torque. The fourth window presents the dynamic variation in the overall efficiencies of the pump, the hydraulic motor, and the entire hydraulic system. By default, the data table displays the hydraulic system parameters at the moment the test is stopped. Moving the cursor along the time axis of the plots allows the parameters to be reviewed for the entire test period.

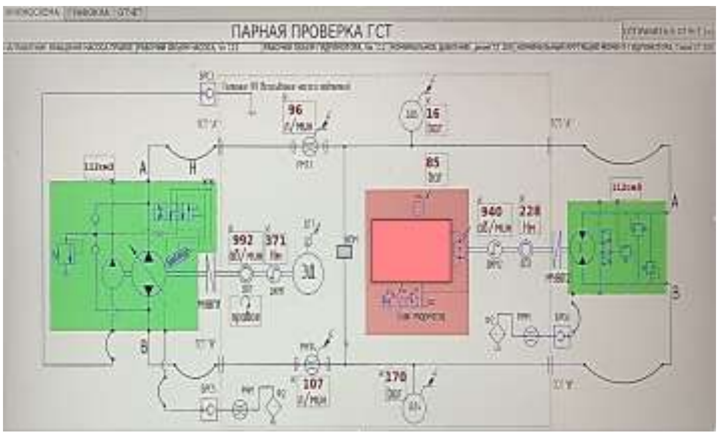


Figure 9: Mimic diagram of the "HST Paired Check" test mode at maximum pump displacement

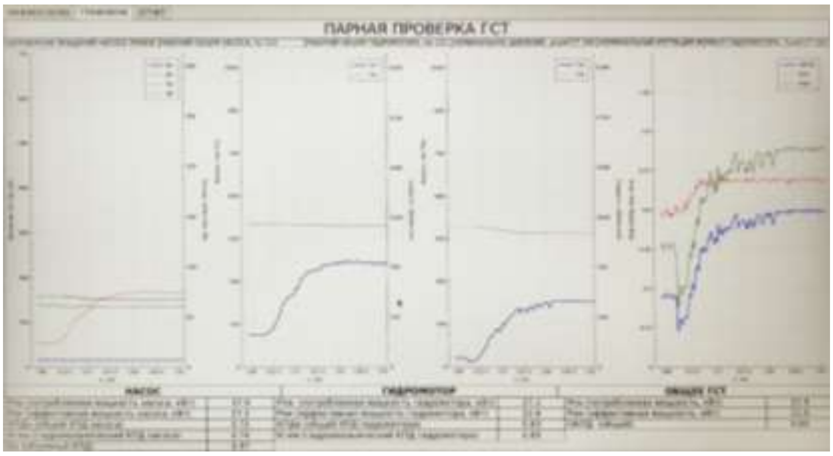


Figure 10: Dynamic parameter variation display for the "HST Paired Check" test mode at maximum pump displacement

The data displayed in the lower table of Fig. 10 are calculated using the formulas presented in the table below.

Table 1: Calculation formulas for parameters of the "HST Paired Check" test mode

Parameters	Formula
Pump	
C_{PP} (consumed pump power, kW)	(7)
E_{PP} (effective pump power, kW)	(6)
OE_{PN} (overall pump efficiency)	(5)
HM_{PN} (hydromechanical pump efficiency)	(3)
K_V (volumetric efficiency)	(1)
Hydraulic motor	
C_{PM} (consumed motor power, kW)	(6)
E_{PM} (effective motor power, kW)	(9)
OE_{PM} (overall motor efficiency)	(8)
HM_{PM} (hydromechanical motor efficiency)	(4)
HST overall	
C_{PP} (consumed pump power, kW)	(7)
E_{PM} (effective motor power, kW)	(9)
OE (overall HST efficiency)	(10)

Figures 11 and 12 present the mimic diagram and parameter plots for the hydraulic motor and pump operating at an unknown displacement. The pump swashplate has been set to an intermediate position that does not correspond to its maximum displacement of 112 cm^3. Since the pump is known to be serviceable, the mimic diagram data (Fig. 11) indicate that at a pump shaft rotation speed of 1,492 rpm, the delivery is 61 L/min; furthermore, compared to the tests shown in Fig. 9, the flow direction in the HST high-pressure lines has been reversed. Based on the volumetric efficiency of the serviceable pump, it can be concluded that the current pump displacement is approximately 42 cm^3.

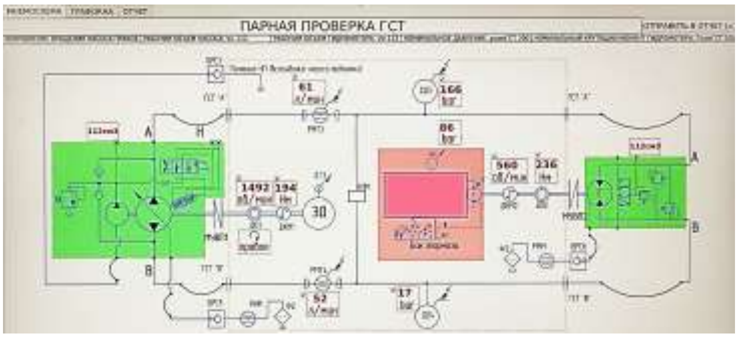


Figure 11: Dynamic parameter variation display for the "HST Paired Check" test mode at an unknown pump displacement value

The purpose of testing at maximum and intermediate displacements is to evaluate the measurement error of the torque at the shaft of the hydraulic motor under test. During the verification carried out, the only factor influencing the hydraulic motor shaft torque was the pressure differential across it. In the tests at maximum pump displacement, the pressure differential across the hydraulic motor was 15.4 MPa (Fig. 9), while in the tests at the unknown pump displacement it was 14.9 MPa (Fig. 11). In the first case the torque was 228 N·m (Fig. 9), and in the second — 236 N·m (Fig. 11).

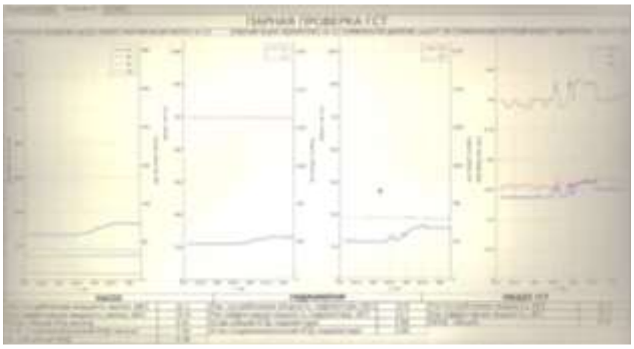


Figure 12: Dynamic parameter variation display for the "HST Paired Check" test mode at an unknown pump displacement value

When the obtained data are recalculated — taking into account the hydromechanical efficiencies determined from the tests (Fig. 10, 12) — into the hydraulic motor displacement value corresponding to each test, the following results are obtained: at maximum pump displacement, the hydraulic motor displacement is 109.44 cm^3; at the unknown pump displacement, the hydraulic motor displacement is 113.09 cm^3. The relative error of the determined displacement values, against the nameplate hydraulic motor displacement of 110.8 cm^3, is 3.3%. Given that the torque at the hydraulic motor shaft is directly proportional to the pressure differential and the displacement, it can be concluded that the torque measurement accuracy is 3.3%. This is consistent with the torque measurement accuracy requirements for acceptance testing of ±2.0%.

DISCUSSION AND CONCLUSIONS

The designed test bench enables acceptance testing of HST hydraulic machines. A diagnostic methodology for HST hydraulic units using the proposed bench has been developed. The validity of the proposed methodology has been confirmed by preliminary testing of the HST 112. The accuracy of the obtained parameters corresponds to the declared Class 3 accuracy, which meets the requirements for acceptance testing of positive-displacement hydraulic machines. To conduct periodic hydraulic machine tests more effectively, it is necessary to improve measurement accuracy and refine the methodology for determining hydromechanical efficiency.

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