Представлено квазістатичні характеристики роботи двопотокових гідрооб'ємно-механічних трансмісій (ГОМТ), що реалізуються в лабораторному стенді ГОМТ гусеничного трактору. Конструкція стенду дозволяє проводити дослідження схем ГОМТ типу з диференціалом «на виході», з диференціалом «на вході» як колісних, так і гусеничних машин. Схемне рішення стенду дає змогу моделювати роботу гідрооб'ємного механізму повороту та встановити ефективність системи рекуперації паразитної потужності, що циркулює в замкнутому контурі ГОМТ

Ключові слова: трактор, трансмісія, лабораторний стенд, планетарний механізм, циркуляція потужності, гідрооб'ємний механізму повороту

Представлены квазистатические характеристики работы двухпоточных гидрообъемно-механических трансмиссий (ГОМТ), реализуемые в лабораторном стенде ГОМТ гусеничного трактора. Конструкция стенда позволяет проводить исследования схем ГОМТ типа с дифференциалом «на выходе», с дифференциалом «на входе» как колесных, так и гусеничных машин. Схемное решение стенда позволяет моделировать работу гидрообъемного механизма поворота и установить эффективность системы рекуперации паразитной мощности, циркулирующей в замкнутом контуре ГОМТ

Ключевые слова: трактор, трансмиссия, лабораторный стенд, планетарный механизм, циркуляция мощности, гидрообъемный механизм поворота

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DESIGN OF THE LABORATORY BENCH FOR A HYDROVOLUMETRIC-MECHANICAL TRANSMISSION OF THE TRACKED TRACTOR

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1. Introduction

It is impossible to fully utilize the potential of modern energy-intensive tractor without a highly efficient transmission, which would make it possible to perform tillage operations at the required agrotechnical speed and would at the same time allow the tractor to move along general-purpose motor roads. All these properties are demonstrated by double-flow hydromechanical transmissions (HT). Stepless control of speed and traction effort in the tractor, equipped with HT in the machine-tractor assembly (MTA), makes it possible to reach maximum performance efficiency while the ease of control reduces psycho-emotional load on the driver-operator.

Transmissions of HT type with a differential «at the output» [1] have maximum performance efficiency coefficient within 78–83 %, which is typically lower than that in transmissions that switch gears under load, so called Power-Shift, however, the benefits of stepless regulation level off this drawback. The main advantage of transmissions of HT type with a differential «at the input» [2] is the ability to implement the so-called fast-reverse – switching from forward to backward course with no participation of additional levers and pedals, but only by controlling a parameter *e* of the hydromechanical transmission (HT). Their performance efficiency coefficient is approximately 3–5 % lower than that in the schemes of the first type.

Given the high demand for HT for wheeled tractors, and the lack of known industrial samples of HT for tracked tractor, it is an important task to design and implement new technical schemes of transmissions. A key requirement to them is a maximally possible performance efficiency coefficient, which can be achieved by aligning the ratios of toothed gears and planetary mechanisms that make up its structure, as well as the correct choice of standard size for volumetric hydraulic machines.

2. Literature review and problem statement

Introduction of HT to agricultural wheeled and tracked machines is the most promising fuel-saving technology. Given their considerable potential, lower emission of harmful substances, larger productivity and ease of control, tractors with HT are present in the model lines of most of the world's leading tractor manufacturers (Fendt, Case, Class, John Deere, HTZ, etc.).

Paper [3] described quantification of the degree of fatigue of drivers-operators of wheeled tractors with different types of transmissions, specifically HT. The author found, based on experimental research that involved tractors Fendt 936 Vario and HTZ 150K-09 with a manual step gear box during field and transportation operations, that a HTZ tractor driver experienced fatigue by 25–34 % earlier than that who drove the Fendt tractor with a stepless HT.

The structure of HT design enables wide opportunities for modernization. Specifically, a promising direction is the introduction of additional systems for power recuperation. At braking of a wheel tractor with HT, kinetic energy can be accumulated either using principle [4] – in a pneumatic hydro accumulator in the form of working fluid pressure or based on the principle described in ref. [5] – by electric machine in the battery, which is widely used in modern hybrid cars and electric vehicles.

Features of the calculation of HT, which implies determining the power and kinematic parameters at transmission chain, are tackled in many papers. A procedure for the calculation of transmission with a differential «at the output» was reported in [6]; with a differential «at the input» – in [7]. All of them are very close to known matrix approach [8].

In Ukraine, a small-scale production of stepless HT [9, 10] for general-purpose wheeled tractors with a hinge-jointed frame of traction class 4–5 was initiated at JSC «HTZ» and LLC «Speckran» (Kharkiv). The transmission employed highly technological differentials with coupled satellites with outer coupling; its drawback, however, is the presence of a hydraulic clamping clutch on the shaft of the hydraulic motor.

Success in the production attracted increased interest in circuit solutions that are given in the patents of Ukraine [11–13]. Each of them has its own advantages, it can be assembled in the dimensions of frames of known wheeled and tracked tractors, and the right choice of transfer ratios may help attain high technical and economic indicators.

The advantages of hydrovolumetric-mechanical transmission in a transportation vehicle [11] include the possibility to enable, under technological mode, the distribution of capacity flows to hydraulic part, 40 %, and mechanical part, 60 %. The assembly improved mass and dimensions, and the hydraulic system is maximally simplified due to utilizing only one hydraulic motor. The shortcomings of the design include two planetary mechanisms with internal coupling and 2 hydraulic clamping clutches, which significantly raises the cost of production.

The transmission of a vehicle, described in patent [12], by the introduction of a reverse reducer with cylindrical gears, blocking clutch and a brake with new links between elements of the transmission, enables an increase in the loading capability and torque, as well as reduces capacity losses while improving a performance efficiency coefficient. The main disadvantage is the complicated mechanical part.

A stepless HT [13] is characterized by a good layout (a single casing combines a planetary mechanism, a speed reducer, a distribution box). Transmission can be used for universal tilling and garden tractors, it has no hydraulic clamping clutches. The main disadvantage is the limited number of technological ranges.

Yet the industrial production of any transmission is preceded by a technical feasibility study, calculations and experimental tests.

The main problem of experimental research into working processes of double-flow HT is the presence of a large number of possible circuit solutions. They all need to be systemized by type; in practice, however, it is required to design an original bench-prototype of transmission for each scheme. In turn, transfer ratios of reducers, internal gear ratio of planetary mechanisms, and standard sizes of hydraulic machines, significantly impact a performance efficiency coefficient and technical-economic indicators of the transmission.

In 2015, at the Department of Automobile and Tractor Engineering at NTU «KhPI», researchers created a laboratory bench for HT of wheeled tractor (Fig. 1, *a*). Altering the arrangement of constituent elements made it work employing the circuit with a differential «at the output» (Fig. 1, *b*) or utilizing the circuit with a differential «at the input» (Fig. 1, *c*) [14].

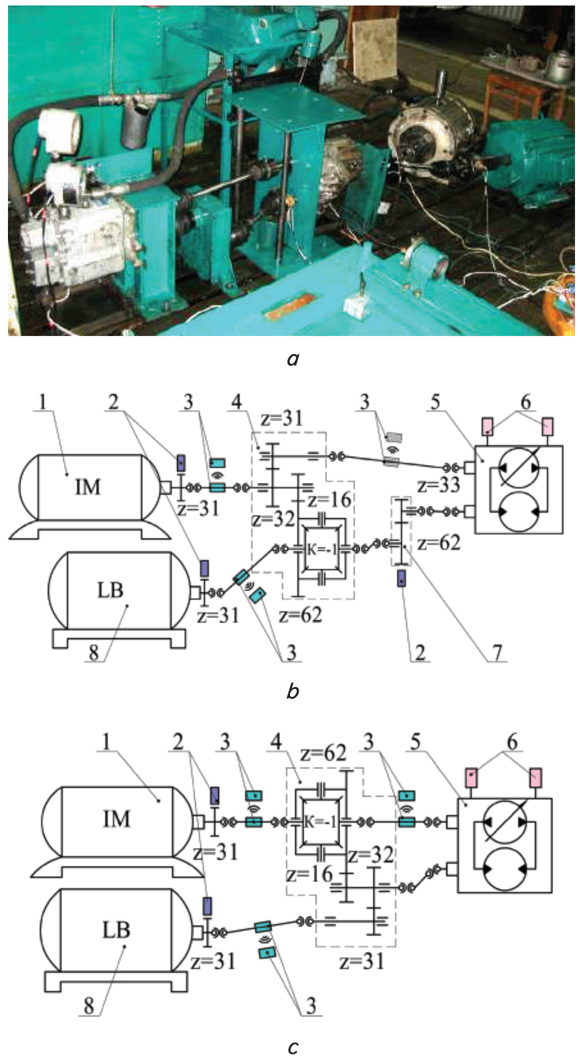

Fig. 1. Diagram of the laboratory bench for HT of wheeled tractor at the Department of Automobile and Tractor Engineering at NTU «KhPI»:
1 — induction motor (IM); 2 — inductive rotation frequency sensors; 3 — torque sensors; 4 — reducer with a planetary mechanism; 5 — hydrovolumetric transmission; 6 — excess pressure sensors; 7 — toothed aligning cylindrical reducer; 8 — powder loading brake (PLB); *a* — physical appearance; *b* — mode with a differential «at the output»; *c* — mode with a differential «at the input»

The main shortcoming of a given design (Fig. 1) is the difficulty of readjustment. In addition, because of the mismatch between transfer ratios of reducers for the HT control parameter $e = -1$, which must correspond to a zero speed of the transmission output shaft rotation frequency, we observed its turning.

Taking into consideration the experience gained when designing HT, the new laboratory bench must meet the following requirements:

1) modeling of operation of wheeled and tracked tractors (based on a board scheme) with a differential «at the input» and «output», as well as combined;

2) switching the mode of transmission work using control levers rather than altering the arrangement of constituent elements;

3) modeling of operation of hydrovolumetric turning mechanisms (HVTM) of tracked tractor of various types;

4) capability to change the number and the transfer ratio of reducers for the input, output, mechanical, and hydraulic branches of HT and HVTM;

5) variation of traction effort and speed of the output shaft of HT, which would make it possible to simulate the modes, typical for heavy tracked vehicles, tractors, construction and road machinery, and automobiles;

6) capability to create a resistance force, equal and different in magnitude, at the output shaft of the right and left board of HT, which would make it possible to model the regime «turning on a slope»;

7) the presence of outputs of the drive of electric generators to establish the possibilities of recuperation of parasite circulating power in the closed circuit of HT, as well as the use of electric generators to control the turning of a tracked machine;

8) availability of HT control system with feedback and a measuring complex. This would allow determining the angular speed of links, pressure in HT, the torque at shafts, regulation parameter e of hydraulic machines, temperature of the working fluid in a hydraulic system, and to simulate the acceleration, braking, turning, and steady motion of the tractor. In order to visualize a motion trajectory when applying control tools (a steering wheel or a joystick), it is planned to develop software;

9) the presence of friction coupling elements in the drive of hydraulic pumps and hydraulic motors that can be employed when studying the braking properties of transmission;

10) a potential for further modernization and improvement of the bench.

The absence of laboratory setups with such properties at universities and industrial organizations significantly slows down the development of tractor engineering and the rate of implementation of transmission circuit solutions developed by scientists and engineers. This is what makes our study promising from a scientific and practical point of view.

3. The aim and objectives of the study

The aim of present study is to propose a circuit for the laboratory bench that would make it possible to conduct experimental study into working processes of double-flow HT for wheeled and tracked tractors and to simulate operation of the hydrovolumetric turning mechanism. The circuit of the bench must combine all the most promising assembling schemes that were found in the course of literature search, as well as enable changing the gear ratios and the number of reducers at each link of HT or HVTM, in order to systemize, search for patterns, and establish interrelations.

To accomplish the aim, the following tasks have been set:

– to develop a kinematic circuit of the laboratory bench for HT of a tracked tractor, considering the above requirements;

– to construct a mathematical model for determining power and kinematic parameters of transmission under the mode of maximum engine load;

– to establish advantages and disadvantages of each HT circuit, implemented at a laboratory bench, and to identify objects to which a given transmission could be proposed.

4. Design and specifications of the laboratory bench for HT of a tracked tractor

4. 1. Design of the laboratory bench for HT

Fig. 3 shows a kinematic circuit of the bench, which demonstrates which gears are constantly coupled and which can move along the shaft, thereby creating new circuits for HT and HVTM.

The main feature and advantage of the bench under consideration is the use of spur gears with a single module. This allowed significant compaction of the assembly; switching the gears and modes of operation is performed at stopping the drive electric motor using a «tooth to tooth» technique.

The design presented in Fig. 3 has all the necessary properties to conduct field tests of HT circuits with a differential «at the input» and «output» of wheeled and tracked vehicles. Two adjustable hydraulic pumps and electric generators enable the simulation of the process of turning a tracked machine using HVTM, as well as combined – employing HT and a generator load. The second function of electric generators is the kinetic energy recuperation at braking from different branches of the transmission, which would make it possible to estimate the feasibility of its implementation in the manufacture of machines.

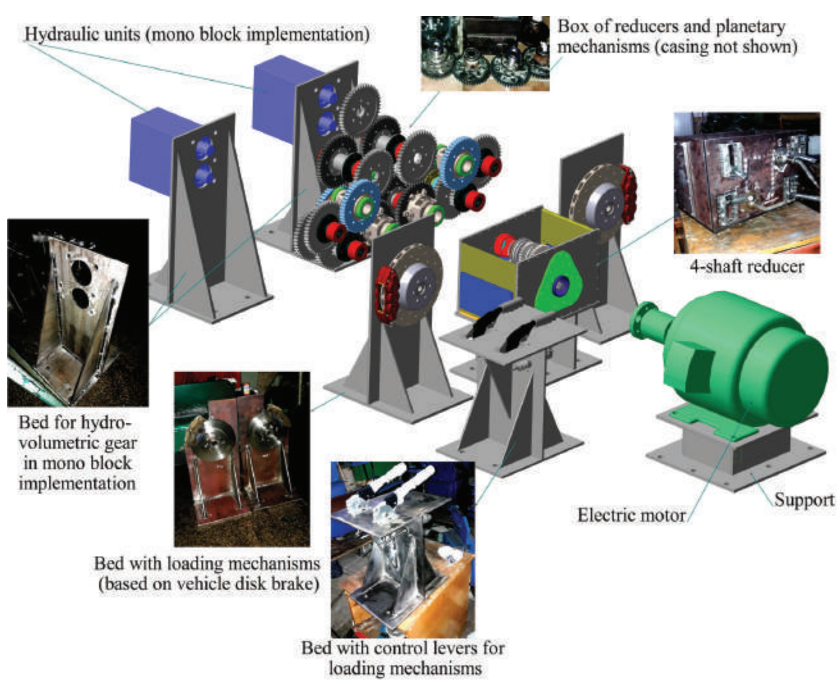

Fig. 2. Three-dimensional model of arrangement of basic elements of laboratory bench for HT of tracked tractor

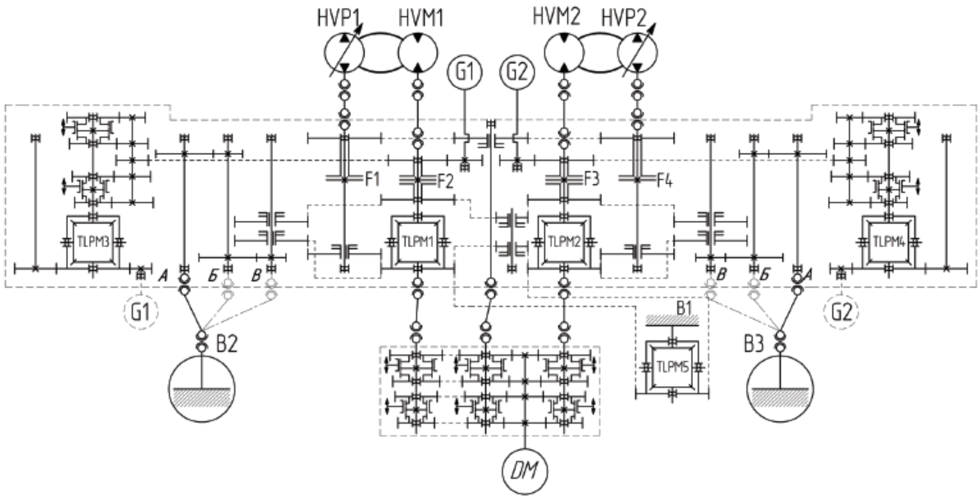

Fig. 3. Kinematic circuit of the laboratory bench for HT of tracked tractor:
DM — drive motor; TLPM — three-link planetary mechanism; VHP — volumetric hydraulic pump; VHM — volumetric hydraulic pump; VHP — volumetric hydraulic motor; B — brake; F — friction (hydraulic clamping clutch); G — generator

4. 2. Mathematical model for determining the power and kinematic parameters of transmission

The drawback of existing mathematical model for calculating operational parameters of HT, described in papers [1, 2, 8, 15], is the technique for determining the boundaries of circulation-free regime and the mode with power circulation in the closed circuit of HT. The identifier of the circulation mode was considered to be a power magnitude *N* that comes through the hydraulic branch of HT. At the degeneration of one of its elements to 0 (torque *M* at a link or angular speed *w*), it is impossible to reliably determine the boundaries of specific regions of work of a hydraulic gear and to separately explore the mechanical and hydraulic performance efficiency coefficient of hydraulic machines.

Paper [16] proposed an improved mathematical model that is devoid of this drawback, since the identification of operational mode of hydraulic machines was determined by a change in the sign at *M* and *w*. This approach will be used for the estimation of all the circuit modes in the operation of the laboratory bench.

Using a simple double-flow single-range HT as an example (Fig. 4), implemented at the laboratory bench (Fig. 3), we shall write a system of equations for determining the mechanical and volumetric performance efficiency coefficient VHM1 and VHM2 in the composition of HT in a direct and reverse flow of power considering a special region of work for a change in the sign at *M* and *w*.

In the considered structural diagram (Fig. 4), VHM1 operates in the reverse flow of power, when condition $M_{T1}+\Delta M_1<0$ is satisfied; VHM2 at $w_2\geq 0$. Mechanical performance efficiency coefficient of hydraulic machines is determined from equations:

$$\eta_{m\_HVM1} = \begin{cases} \dfrac{M_{T1}+\Delta M_1}{M_{T1}}, \text{ if } M_{T1}+\Delta M_1<0, \\ \dfrac{M_{T1}}{M_{T1}+\Delta M_1}; \end{cases} \quad (1)$$

$$\eta_{m\_HVM2} = \begin{cases} \dfrac{M_{T2}}{M_{T2}+\Delta M_2}, \text{ if } \omega_2 \geq 0, \\ \dfrac{M_{T2}-\Delta M_2}{M_{T2}}, \end{cases} \quad (2)$$

where $M_{T1,2}$ is the theoretical momentum at the shaft of hydraulic machines, Nm; $\Delta M_{1,2}$ is the loss of torque, Nm.

Overall mechanical performance efficiency coefficient of HT is derived from equation:

$$\eta_{m\_HVG} = \eta_{m\_HVM1} \cdot \eta_{m\_HVM2}. \quad (3)$$

Mechanical $\eta_м$ and volumetric η_V performance efficiency coefficient of HT can be derived from formulae for the direct and reverse flow of power:

$$\eta_V^{bk} = \frac{\omega_{HVM2}}{\omega_{HVM1}\cdot e}, \quad \eta_V^{st} = \frac{\omega_{HVM1}\cdot e}{\omega_{HVM2}}, \quad (4)$$

$$\eta_m^{bk} = \frac{_{HVM1}}{_{HVM2}\cdot e}, \quad \eta_m^{st} = \frac{_{HVM2}\cdot e}{_{HVM1}}. \quad (5)$$

where $\omega_{HVM1,2}$ is the angular speed at the shafts of hydraulic machines, rad/s.

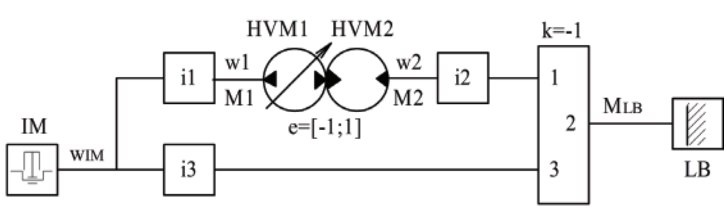

Fig. 4. Simplified structural diagram of a single-range HT:
IM — induction motor; *i* — gear ratio of reducers; VHM — volumetric hydraulic machine; LB — loading brake

We shall equate the right side of equation (3) to (4), substituting (1), (2) for direct flow of power in which we found discrepancy:

$$\frac{M_2 \cdot e}{M_1} = \frac{M_{T1}}{M_{T1} + \Delta M_1} \cdot \frac{M_{T2} - \Delta M_2}{M_{T2}}. \tag{6}$$

It is impossible to solve (6) relative to M_2 because M_{LB} will uniquely define it under all modes according to the Willis equations for a planetary series [9, 10].Upon transformations, we shall obtain an expression for determining M_1 in the direct power flow:

$$M_1 = \frac{M_2 (M_{T2} - \Delta M_2)}{(M_{T1} + \Delta M_1)}. \tag{7}$$

Therefore, the torque at the shafts of hydraulic machines in the reverse and direct flows, according to (4), (5) and (7), will be equal to:

$$M_2 = \begin{cases} M_{T2} + \Delta M_2, \text{if } \omega_2 \geq 0, \\ M_{T2} - \Delta M_2; \end{cases} \tag{8}$$

$$M_1 = \begin{cases} M_{T1} + \Delta M_1, \text{if } M_{T1} + \Delta M_1 < 0, \\ \dfrac{M_2 (M_{T2} - \Delta M_2)}{(M_{T1} + \Delta M_1)}. \end{cases} \tag{9}$$

To determine the volumetric performance efficiency coefficient, we shall consider a theoretical consumption Q_{T1} of the working fluid by hydraulic machines:

$$Q_{T1} = q_1 \cdot |e| \cdot |\omega_1|, \tag{10}$$

$$Q_{T2} = q_1 \cdot |e| \cdot |\omega_1| - \Delta Q_1, \tag{11}$$

$$\eta_{V\_HVM1} = \begin{cases} \dfrac{Q_{T1}}{q_2 \cdot |\omega_2| - \Delta Q_2}, \text{if } \left(-\dfrac{\omega_1 \cdot e}{\omega_2} \right) < 1, \\ \dfrac{Q_{T1} - \Delta Q_1}{Q_{T1}}; \end{cases} \tag{12}$$

$$\eta_{V\_HVM2} = \begin{cases} \dfrac{q_2 \cdot |\omega_2| - \Delta Q_2}{q_2 \cdot |\omega_2|}, \text{if } \omega_2 > 0, \\ \dfrac{q_2 \cdot |\omega_2|}{Q_{T1} - \Delta Q_1}. \end{cases} \tag{13}$$

The overall performance efficiency coefficient of HT is defined as the product of components:

$$\eta_{V\_HVG} = \eta_{V\_HVM1} \cdot \eta_{V\_HVM2}. \tag{14}$$

Such an approach is devoid of known shortcomings and makes it possible to explore, with high reliability, modes with a direct flow, a reverse flow, and specific regions of work of HT along the entire length of a regulation characteristic.

4. 3. Circuit No. 1. Double-flow hydrovolumetric-mechanical transmission of a tracked (wheeled) tractor with a differential «at the output»

A common advantage of all of the following circuits is the principle of construction based on a board scheme. Disconnecting a drive from one of the boards and shutting off the respective valve in a hydraulic system makes it possible to obtain a model of the transmission of a wheeled machine (tractor, forklift, road-building machinery, etc.). Reducers that are designated as $i1_V$, $i2_V$ can accept transfer ratios of 3.454; 2.056; 1.333; 0.969, allowing the estimation of influence of the gear ratio of each branch of HT on the technical-economic indicators in general.

A prototype for the circuit that is presented in Fig. 5 is the transmission HT-1C, which has proven highly effective and could be successfully modernized for standard tracked machines manufactured at JSC «KhTZ».

Fig. 6 shows results of the mathematical modeling of basic parameters of transmission operation (circuit No. 1) under the mode of a tracked machine; Fig. 7 – under the mode of a wheeled machine. In both cases, the overall performance efficiency coefficient is not less than 80 %.

The maximum estimated capacity of the bench is limited to 33 kW, according to the specifications for the drive induction motor AIR180M4. The rated engine revolutions are 1,500 rpm.

Pressure drop Δp, both for the tracked and wheeled modes, for circuit No. 1, does not exceed 40 MPa, which confirms its performance efficiency.

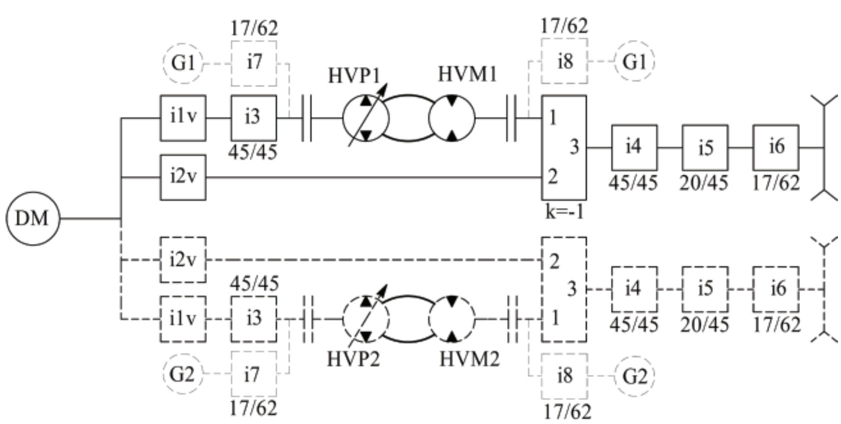

Fig. 5. Circuit No. 1. Double-flow hydrovolumetric-mechanical transmission of the tracked (wheeled) tractor with a differential «at the output»

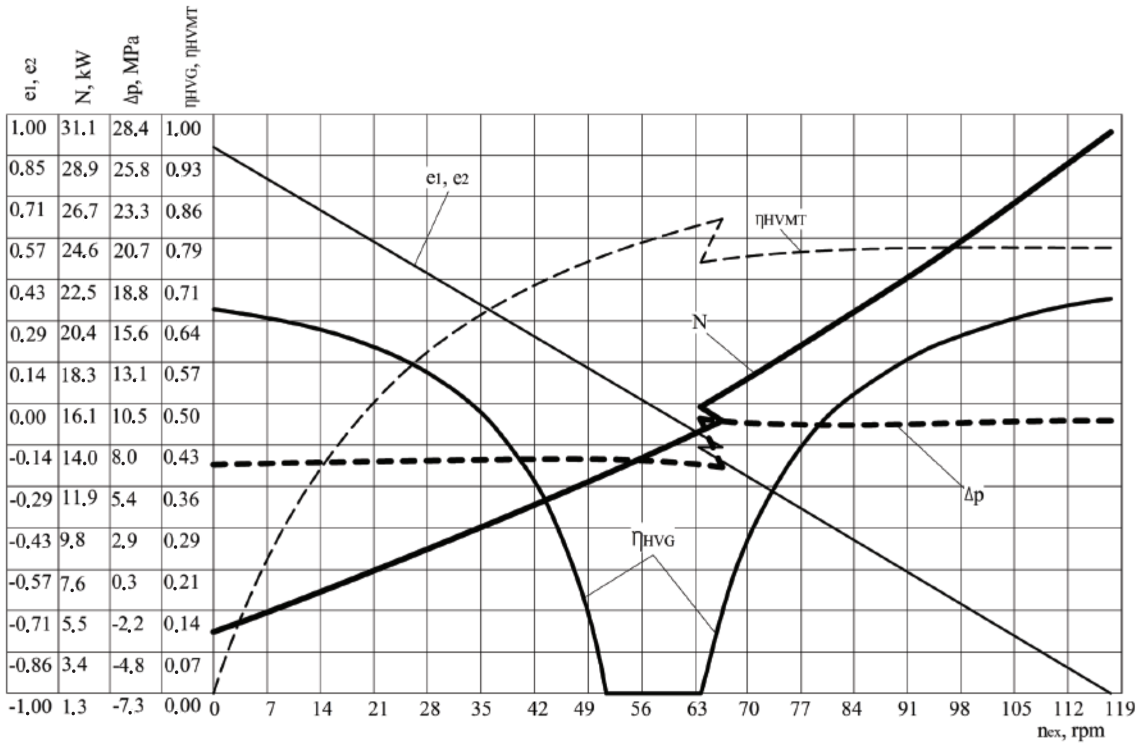

Fig. 6. Quasi-static characteristic of HT for circuit No. 1 (tracked mode)

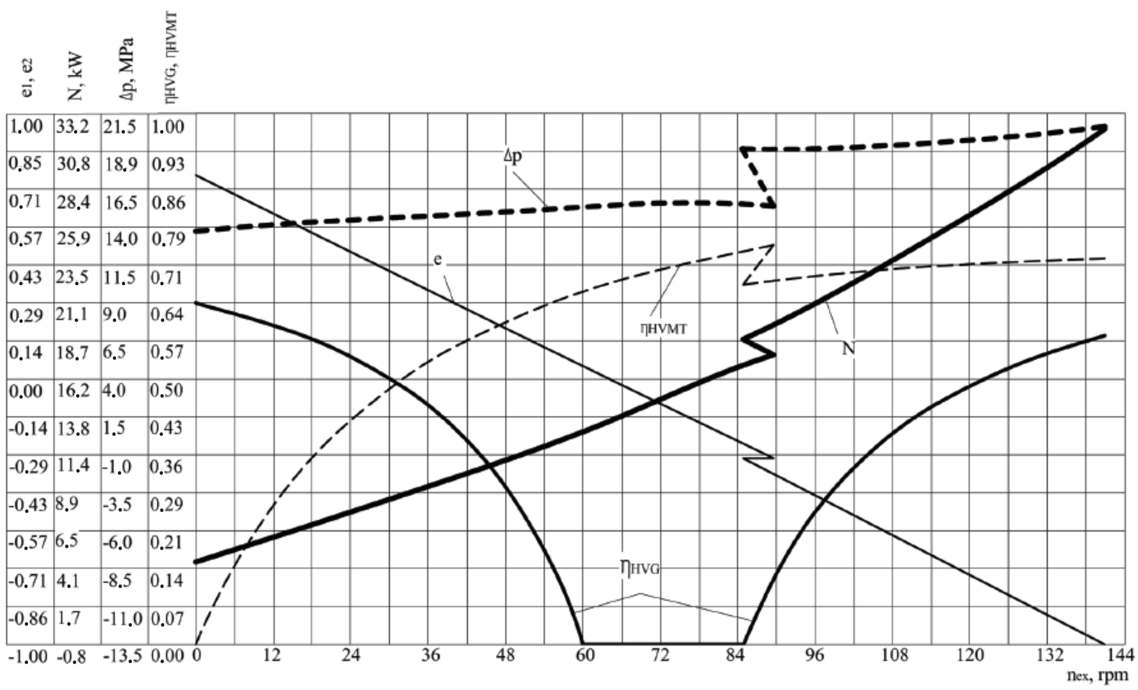

Fig. 7. Quasi-static characteristics of HT for circuit No. 1 (wheeled mode)

4. 4. Circuit No. 2. Double-flow hydrovolumetric-mechanical transmission of the tracked (wheeled) tractor with a differential «at the output»

The main advantage of the circuit presented in Fig. 8 is the so-called «rapid reverse» – switching from forward to reverse motion by a change in the regulation parameter *e* of HT. However, the transmission has low traction properties (Fig. 9) and, therefore, could be applied in a more sophisticated HT as the mode that would be used, for example, when making a *U*-turn or when equipped with tools.

Under a given mode, the maximum capacity cannot be achieved because of the exceeding of permissible Δp. In addition, low efficiency of HT (at the level of 40 %) negatively affects fuel consumption. Therefore, it is not appropriate to recommend it for production as an independent transmission.

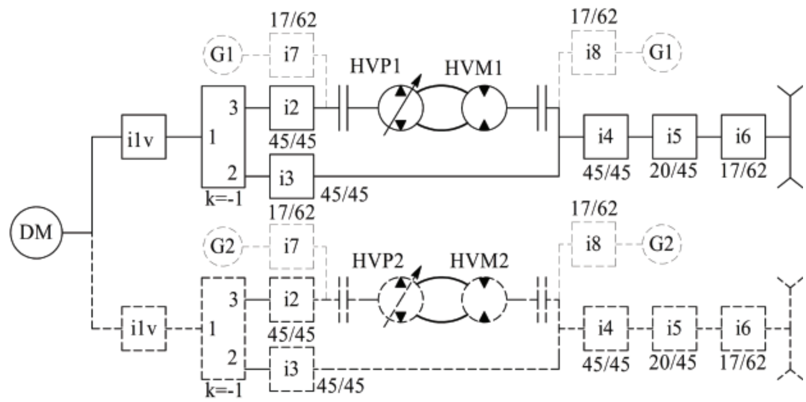

Fig. 8. Circuit No. 2. Double-flow hydrovolumetric-mechanical transmission of the tracked (wheeled) tractor with a differential «at the input»

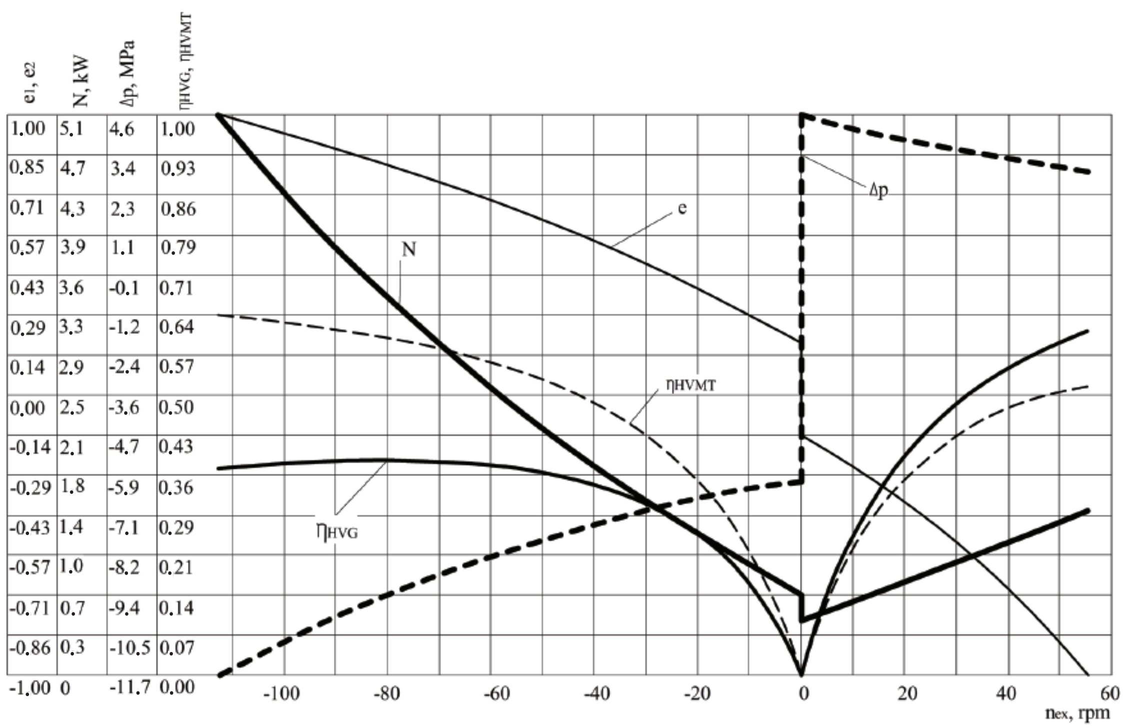

Fig. 9. Quasi-static characteristic of HT for circuit No. 2 (wheeled mode)

4. 5. Circuit No. 3. Double-flow hydrovolumetric-mechanical transmission of the tracked (wheeled) tractor with a differential «at the output»

The introduction of an additional differential and a brake mechanism to the mechanical branch of HT (Fig. 10) opens additional opportunities in terms of control over a turning of the tracked machine or braking of the wheeled machine. This makes it possible to stop using hydraulic clamping clutches in a hydraulic pump and hydraulic motor). Results of the simulation are shown in Fig. 11.

The advantage of a given circuit over circuit No. 1 (Fig. 6, 7) is the enhanced output speed. This is achieved by coupling a transmission output shaft with an epicycle of the planetary mechanism rather than a carrier. This circuit can be recommended for high-speed tracked vehicles.

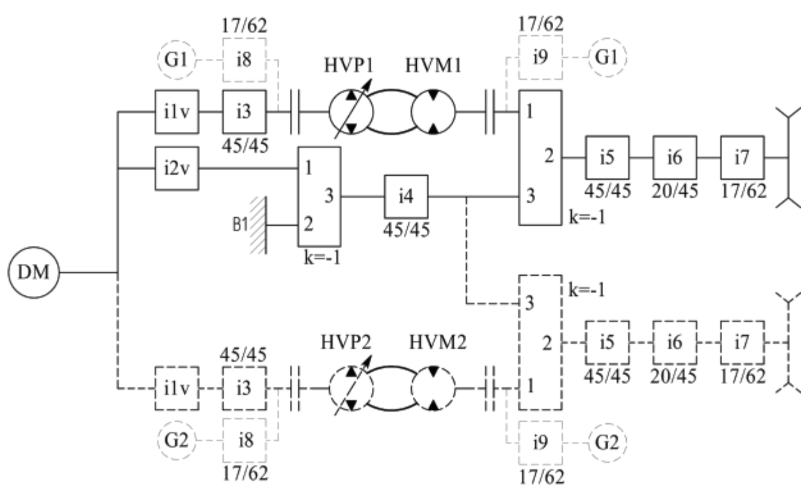

Fig. 10. Circuit No. 3. Double-flow hydrovolumetric-mechanical transmission of the tracked (wheeled) tractor with a differential «at the input»

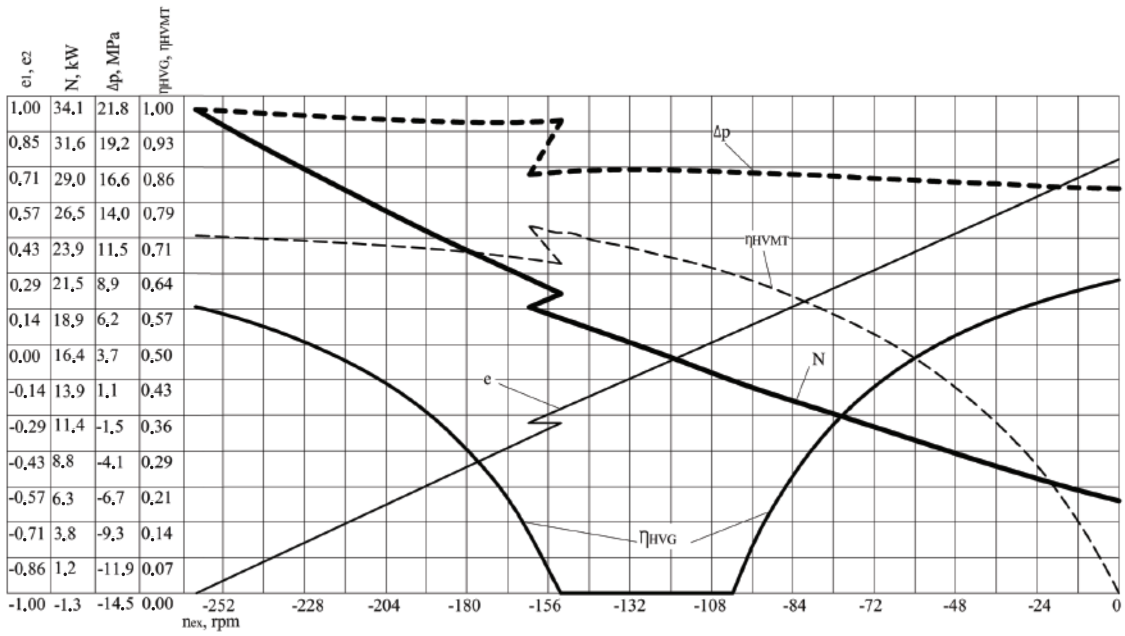

Fig. 11. Quasi-static characteristic of HT for circuit No. 3 (wheeled mode)

4. 6. Circuit No. 4. Double-flow hydrovolumetric-mechanical transmission of the tracked (wheeled) tractor with a differential «at the input»

The circuit that is presented in Fig. 12 is not appropriate to use as an independent HT. Similar to circuit No. 2 (Fig. 8), it is successfully combined with circuit No. 1 (Fig. 5) and can be used to move backward. In addition, when working under a given mode, one observes a rather high efficiency coefficient and thrust. Results of the calculation are shown in Fig. 13. At maximum load, HT control range is limited is equal to [+1; –0.05]. Thus, the system of control over such a transmission must include safety elements to avoid the overload of hydraulic machines.

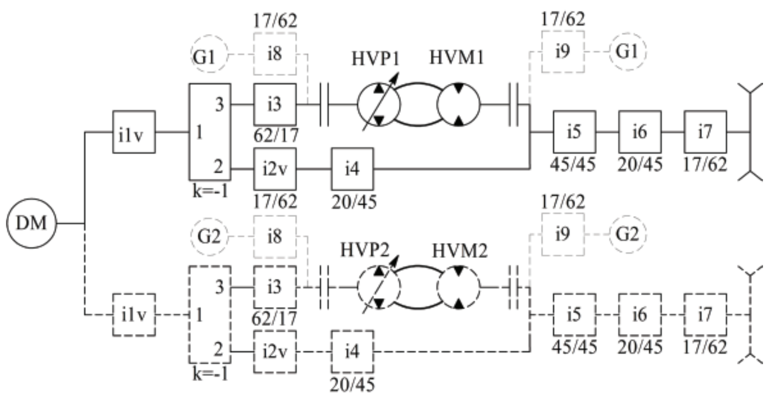

Fig. 12. Circuit No. 4. Double-flow hydrovolumetric-mechanical transmission of the tracked (wheeled) tractor with a differential «at the input»

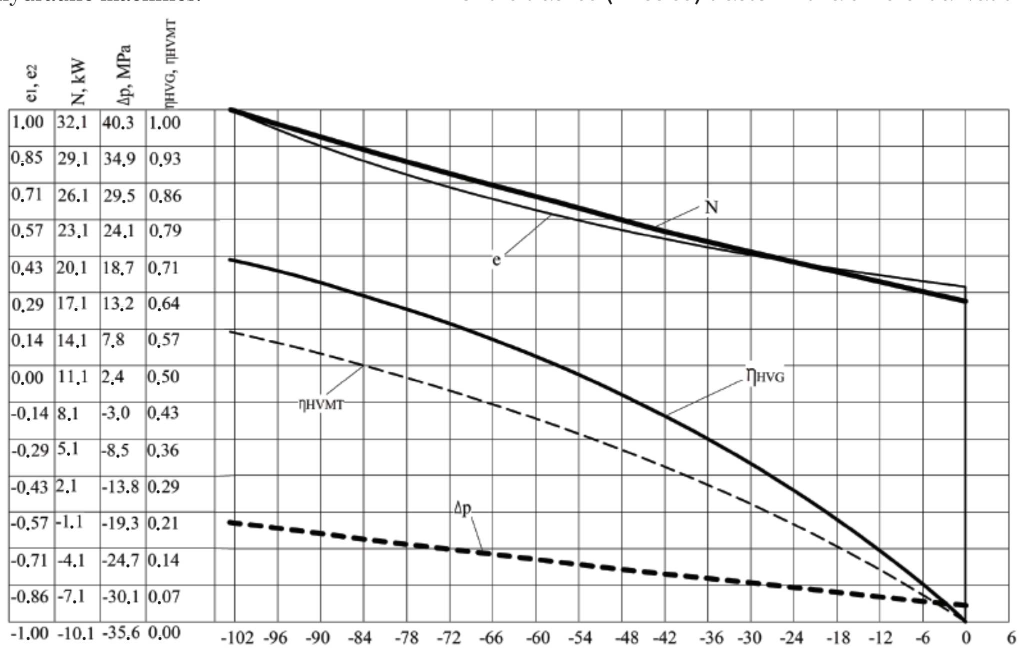

Fig. 13. Quasi-static characteristic of HT for circuit No. 3 (wheeled mode)

5. Discussion of results of theoretical study into working parameters of HT circuits at the laboratory bench

The task on structural and parametric synthesis of the double-flow stepless transmissions was partly solved owing to the availability of specialized software, specifically Trans [15], and mathematical models [1, 2, 8, 9], which can be implemented in the software Mathcad, MATLAB, Simulink, and others. However, the main problem is the experimental verification of operational efficiency and high performance indicators of the developed circuits for HT, which is impossible given the absence of specialized laboratory benches.

The main advantages of the bench under consideration is the ease of operation and a wide range of implemented circuits for HT and HVTM. Employing the unification of constituent elements, any toothed reducer can be replaced with another, with a new gear ratio while maintaining the interaxial distance.

The possibility of kinetic energy recuperation at braking, and parasite capacity of the closed HT circuit, as well as control system with feedback, open new research fields that are still unexplored because of the lack of a material base. The interface of software to control the motion and turning of a tracked (wheeled) machine is shown in Fig. 14.

The main disadvantage of the considered design is the use of conical differentials with an internal gear ratio of $k=-1$ as planetary mechanisms. It is known that it is necessary for such differentials, due to the fact that the axis of satellites has no rolling bearings, to restrict their angular speed. However, it is due to the conical differentials that we managed to achieve such a dense arrangement.

A given development is continuation of the research described in papers [14, 16]; however, it allows solving a much broader range of scientific and practical tasks. In the future, we plan to replace an induction drive motor with a diesel engine.

Such a modernization would make it possible to bring operational conditions of transmission created at a bench closer to actual conditions.

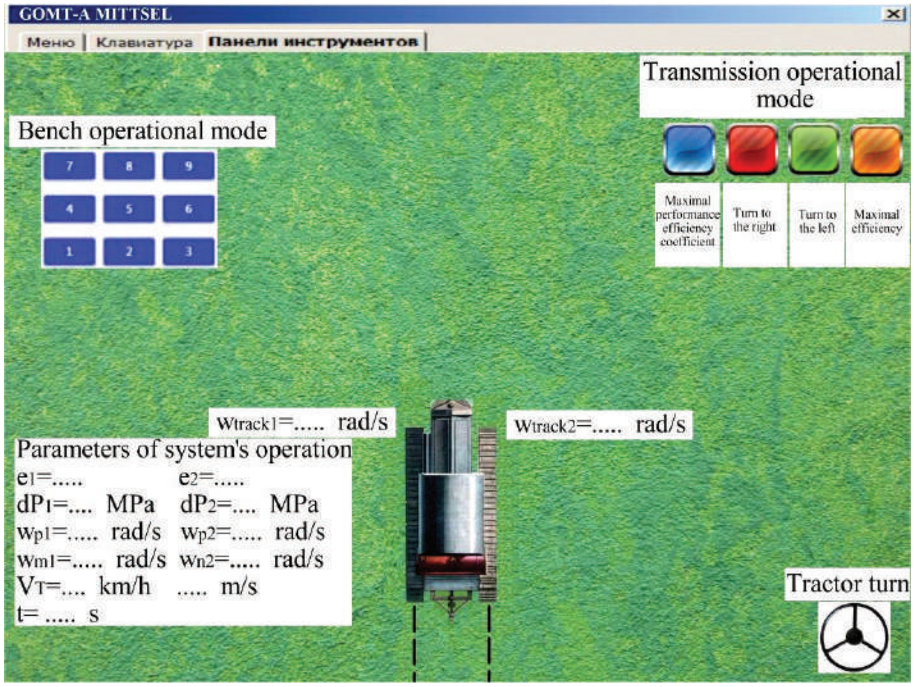

Fig. 14. Interface of software to control a laboratory bench of HT

6. Conclusions

1. We have developed a kinematic circuit for the laboratory bench of HT of the tracked tractor that combined all of the most promising transmissions, and the design itself acquired considerable potential for modernization.

2. We have constructed a mathematical model for determining the power and kinematic parameters of HT under the mode of maximum engine load, in which the identification of modes with the circulation of power is conducted based on the sign of torque M and angular speed w at the shafts of hydraulic machines.

3. We have built quasi-static characteristics at maximum engine load for each HT circuit, implemented at the laboratory bench. The objects were identified, for which a given transmission could be proposed, as well as the advantages and disadvantages in terms of design.

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