Сформовані та обґрунтовані раціональні зміни параметрів регулювання гідромашини в гідрооб'ємній передачі на етапах розгону та гальмування колісного трактора ХТЗ-21021 з безступінчастою двопотоковою гідрооб'ємно-механічною трансмісією за рахунок завантаженості гідравлічної гілки. Визначено вплив використання раціональної та лінійної зміни параметрів регулювання гідромашини на показники гідрооб'ємно-механічної трансмісії та в цілому колісного трактора. Встановлено звуження зони найвищого значення коефіцієнта корисної дії гідрооб'ємно-механічної трансмісії

Ключові слова: гідрооб'ємно-механічна трансмісія, раціональна зміна параметрів, гідравлічна гілка трансмісії, коефіцієнт корисної дії, критерії оцінки

Сформированы и обоснованны рациональные изменения параметров регулирования гидромашины в гидрообъемной передаче на этапах разгона и торможения колесного трактора ХТЗ-21021 с бесступенчатой двухпоточной гидрообъемно-механической трансмиссией. Определено влияние использования линейного и рационального нелинейного изменения параметров регулирования гидромашины на показатели гидрообъемно-механической трансмиссии и в целом колесного трактора. Установлено сужение зоны наивысшего коэффициента полезного действия гидрообъемно-механической трансмиссии

Ключевые слова: гидрообъемно-механическая трансмиссия, рациональное изменение параметров, гидравлическая ветвь трансмиссии, коэффициент полезного действия, критерии оценивания

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FORMATION OF A RATIONAL CHANGE IN CONTROLLING CONTINUOUSLY VARIABLE TRANSMISSION AT THE STAGES OF A TRACTOR'S ACCELERATION AND BRAKING

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1. Introduction

In the world tractor engineering, every year more and more attention is paid to creating the tools for the transfer of power from the power unit to the engines, which are alternative to mechanical, these are primarily transmissions with a variator, hydrovolumetric-mechanical and electromechanical transmissions, etc. [1].

Particularly widespread for wheeled tractors is certainly a continuously variable two-flow hydrovolumetric-mechanical transmission (HVMT). In the West European markets, manufacturers have been encouraged to equip tractors with HVMT for 30 years now. The leaders in this direction are such world famous companies as CNH, John Deere, AGCO with tractor brands Case IH, John Deere, Fendt, Massey Ferguson [2–4]. As for the Ukrainian market, a model manufactured by PAT "Kharkiv tractor plant", HTZ-21021 with HVMT-1S, was for the first time introduced to the market [5, 6].

The tendency of implementation of HVMT is most of all connected to a chain of advantages, in particular, this is a continuously variable transfer of power from internal combustion engine to the wheels, which predetermines provision of smooth motion from the start (stage of acceleration) [7]; reduces dynamic loads in the transmission at variable modes of operation; enhances ergonomic properties during technological and transporting activities; adds up a capacity to reduce speed to a full stop by hydrovolumetric transmission (stage of braking), thus unloading a standard braking system, etc.

An important factor that affects the work of a wheeled tractor with HVMT when performing transporting and technological operations is the control of the tractor (i. e., a change in the parameter of adjustment of hydraulic machine). It is known that in the process of acceleration and at the braking stage, excitatory forces are created that affect technical-economic indicators and the safety of motion of a tractor in general, so the study of the impact of the change in the parameter of adjustment of hydraulic machine at acceleration and braking is a relevant task.

2. Analysis of scientific literature and the problem statement

The unsolved problem of loading efficiency of hydraulic branch of HVMT is important because, with the increase of quenching the excitatory forces by a hydraulic branch, it will significantly reduce the load on the mechanical one. Based on this, one can stress the use of the lower internal combustion engine power both at the stages of acceleration and braking of a wheel tractor.

In the paper [9], the authors propose to increase the efficiency of hydrovolumetric transfer by adding air turbine as a hydraulic pump, which will increase the efficiency by 17 %. However, this innovative approach has not been tested for HVMT in a wheeled tractor.

The materials [10] examined optimal balance control between the efficiency of internal combustion engine and hydrovolumetric transmission. However, the authors stress that, to achieve optimum control, it is necessary to separately determine adjustment characteristics of both a hydraulic pump and a hydraulic motor.

The paper [11] determined that for the efficient work of HVMT in a hydraulic branch, it is expedient to use unregulated hydraulic motor. Such statement was proposed by the author owing to the research into selection of the most appropriate angle of inclination of a hydraulic motor's washer. It was found in the process of selection that in some cases, especially at low speeds and torques, hydraulic system can produce greater efficiency at a smaller angle of inclination of hydraulic motor's washer.

The study of efficiency of HVT as part of an earthmoving machine is the subject of the work [12]. In this work, the author, using mathematical modeling and experiment, determines the efficiency of HVT and the control of machine in general when using one and two hydraulic engines. By carrying out the research, the author stresses that at this stage of the study, it is necessary to pay attention to the dependency of characteristics of the losses on the torque load and parameters of adjustment of hydraulic machines.

The issues that determine the rational change in the HVMT control by changing parameters of regulation of hydraulic machines in hydrovolumetric transmission both at the stages of acceleration and braking are covered in many papers.

In the article [13], the author proposes a mathematical model using a generalized equation of energy balance, finding a parameter of adjustment of HVMT, close to optimum convex-concave law, on the time, for mobile tracked vehicles at a constant coefficient of the motion resistance.

The materials to the paper [14] determined rational changes in regulation parameters of hydraulic machines for tractors with HVMT that work with a "differential at the input" and a "differential at the output", in terms of enhancing technical and economic indicators of a machine-tractor unit at performing a technological operation of "plowing" in the process of entering a technological mode.

In the papers [15, 16], the author gives general recommendations on the rational law of braking, which is characterized by the efficiency of braking, stability and manageability. Unfortunately, the author in these works comes up graphically only with the rational change in parameters of adjusting hydraulic machines for a tractor with HVMT that works with the differential at the output. The article [16] determines the change in the parameters of regulation of hydraulic machines during braking of a wheeled tractor from 60 km/h using an antiblocking system. However, this approach cannot be used because it is known, by forming a rational change in the parameter of adjustment of hydraulic machine for a wheeled tractor HTZ-21021 with HVMT-1S, that the maximum speed amounts only to 41.5 km/h. Furthermore, the tractor HTZ-21021 is equipped with pneumatic braking drive, whose principle of work differs significantly from the antiblocking system.

Based on the analysis of the above-mentioned papers, there is a need to form a rational change in the parameter of adjustment of hydraulic machine for the wheeled tractor HTZ-21021 with HVMT-1S by the criteria of loading the efficiency of hydraulic branch of HVMT. It is this criterion at the stages of acceleration and braking, at which a rational operation of the hydrovolumetric transmission (HVT) would be observed, which will make it possible to reduce the load on the mechanical part of the transmission, internal combustion engine and will increase safety of the motion.

3. The purpose and objectives of the study

The aim of this work is the formation of rational change in the parameter of adjustment of the hydraulic machine of the wheeled tractor HTZ-21021 with continuously variable HVMT-1S by optimizing the shape of characteristics of the parameter of adjustment of the hydraulic machine at the stages of acceleration and braking. It is necessary for the formed change to confirm the idea of the convex-concave law of change in the regulation of HVMT [13] as the most efficient one at acceleration or braking of mobile vehicles with HVMT.

To achieve the set goal, it is necessary to solve the following tasks:

– to formalize the criteria of optimization, which take into account main technical parameters of HVMT, as well as indicators of the wheeled tractor HTZ-21021 with continuously variable HVMT-1S during acceleration and braking;

– to assess the impact of the formed rational changes in the parameters of adjustment of hydraulic machines on the kinematical, power and energy indicators of continuously variable HVMT-1S at the stages of acceleration and braking of the wheeled tractor HTZ-21021.

4. Methods used for the formation of a rational control system of continuously variable transmission in the process of acceleration and braking of a wheeled tractor

In this work, in order to achieve the set objectives, it is necessary to use a number of methods, namely: method of mathematical modeling at differential and integral calculations of dynamic processes in the motion of the wheeled tractor HTZ-21021 with HVMT-1S, the theory of optimization, as well as statistical analysis to compare the data when using linear and rational change of parameters of hydraulic machine adjustment during acceleration and braking.

The process of acceleration of the wheeled tractor with HVMT is modeled based on the papers [17], in which they built a generalized mathematic model of transient processes, taking into account properties of the internal combustion engine performance, HVMT and interaction of the wheels with the bearing surface.

The braking of the wheeled tractor with HVMT is modeled with the help of generalization of scientific papers. In particular, the materials from the article [18] are used to determine the dynamics of the performance of internal combustion engine. On the basis of the work [17], we compile a mathematical model of the work of HVMT, which takes into account the change in the parameter of regulation of the hydraulic machine of HVT, the volume of hydraulic machines, the moment of losses in hydraulic machines and allows describing the process of braking of the elements of transmission of various engineering designs. The paper [19] describes determining parameters of work of the braking mechanism of the wheeled tractor HTZ. As for the study of the braking system of a tractor, it is necessary to describe the processes that occur in a pneumatic braking drive, then we will use a mathematical model, which is formed in the article [20].

For the formation of the rational change in the parameter of adjustment of hydraulic machine both at the stage of acceleration and braking, we will introduce the generalized criterion (K_Σ), (which is characterized by partial criteria and must have the maximum value).

It should be noted that the processes of acceleration and braking are not equivalent, but common partial criteria will be those, which characterize the load of efficiency of the hydraulic transmission branch, as well as acceleration and braking time. One also needs to note that, for the process of braking, there is a very important indicator, which determines the braking distance of the tractor and, as a result, requires a separate partial criterion.

To assess the efficiency of the braking of a wheeled tractor with HVMT, the criteria are used that determine the braking distance and braking time. Thus, it is expedient to represent the criteria in the following expression

$$K_1(e_1) = 1 - \frac{S(e_1)}{S^*_{max}}; \tag{1}$$

$$K_2(e_1) = 1 - \frac{t\,(e_1(t))}{t^*_{max}}, \tag{2}$$

where $S(e_1)$ is the current value of braking distance; S^*_{max} is the maximum value of braking distance; $t(e_1)$ is the current value of time; t^*_{max} is the maximum value of time.

The maximum value of braking distance is accepted as 100 m. The maximum value of time, both of acceleration and braking, will equal the time of change in the parameter of adjustment of hydraulic machine, i. e., $t^*_{max}=t(e_1)$.

The best efficiency of the wheeled tractor at braking is observed at the maximum value $K_1(e_1)$ (1). As for the time of acceleration, the best efficiency is observed at the maximum value $K_2(e_1)$ (2).

Let us use the criteria that characterize efficiency of HVT and efficiency of HVMT. It should be noted that determining the maximum indicator of efficiency of HVT η^*_{HVTmax} will be carried out both in direct (DIR) and reverse (REV) power flows in the closed circuit of HVMT. Since the papers [14] already contain attempts to describe the criterion of efficiency of a machine-tractor unit, then let us take this approach

$$K_3(e_1) = \frac{\eta_{HVTDIR}(e_1)}{\eta^*_{HVTmax}}; \tag{3}$$

$$K_4(e_1) = \frac{\eta_{HVTREV}(e_1)}{\eta^*_{HVTmax}}; \tag{4}$$

$$K_5(e_1) = \frac{\eta_{HVMT}(e_1)}{\eta^*_{HVMTmax}}, \tag{5}$$

where $\eta_{HVT\,DIR}(e_1)$ is the current value of efficiency of HVT in direct flow of power; $\eta_{HVT\,REV}(e_1)$ is the current value of efficiency of HVT in reverse flow of power; $\eta^*_{HVT\,max}$ is the maximum value of efficiency of HVT; $\eta^*_{HVMT}(e1)$ is the current value of efficiency of HVMT; $\eta^*_{HVMT\,max}$ is the maximum value of efficiency of HVMT.

In the calculation, the maximum values of indicators of efficiency of HVT $\eta^*_{HVT\,max}=\eta^*_{O\,max}\cdot\eta^*_{M\,max}$ tend to equal 0.99, then $\eta^*_{HVT\,max}=0{,}98$. It is known that the maximum value of efficiency of the transmission directly depends on the number of connecting elements from internal combustion engine and wheels, but from the paper [21] it is known that the maximum efficiency of HVMT equals $\eta^*_{HVT\,max}=0{,}82$.

It should be noted that the partial criteria are within the range of $0<K_1(e_1)$, $K_2(e_1)$, $K_3(e_1)$, $K_4(e_1)$, $K_5(e_1)\leq 1$. The best load of hydraulic branch of HVMT of the wheeled tractor at braking is observed at the maximum values of $K_3(e_1)$, $K_4(e_1)$ та $K_5(e_1)$ (3)–(5).

Let us combine received criteria (1)–(5) in the generalized one with assigning penalty functions [10–12]

$$K_\Sigma = \sum_{i=1}^{n} Z_i \cdot K_i + \sum_{j=1}^{m} Z_j \cdot P_j, \tag{6}$$

where Z_i, Z_j are the weight coefficients [10]; P_j is the penalty function, which reduces the value of generalized criterion at the exit of variable parameter beyond permissible values [10].

The author of the paper [14, 15], in the study of rational changes in the parameter of adjustment of a hydraulic machine, introduced penalty functions for such indicators as the difference in working pressure in HVT, the angular speed at a satellite and on the links of hydraulic pump and hydraulic engine. As noted, the introduction of penalty functions is directly related to the workability of HVMT.

The maximum value of working pressure drop in HVT does not exceed the mark of 45 MPa ($45\cdot10^6$ Pa), which is provided by the work of safety valves. The maximum values of the angular velocities at the satellite in the planetary mechanism and on the links of hydraulic pump and hydraulic engine should not exceed 600 rad/s. Such peculiarities are primarily related to the design features of HVMT.

$$P_{\omega_S}(|\omega_S|) = \begin{cases} 1 - \dfrac{|\omega_S^*|}{600}, & \text{if } |\omega_S^*| > 600 \text{ rad/s;} \\ 0, & \text{if } |\omega_S^*| \leq 600 \text{ rad/s;} \end{cases} \tag{7}$$

$$P_{\omega_4}(|\omega_4|) = \begin{cases} 1 - \dfrac{|\omega_4^*|}{600}, & \text{if } |\omega_4^*| > 600 \text{ rad/s;} \\ 0, & \text{if } |\omega_4^*| \leq 600 \text{ rad/s;} \end{cases} \tag{8}$$

$$P_{\omega_5}(|\omega_5|) = \begin{cases} 1 - \dfrac{|\omega_5^*|}{600}, & \text{if } |\omega_5^*| > 600 \text{ rad/s;} \\ 0, & \text{if } |\omega_5^*| \leq 600 \text{ rad/s,} \end{cases} \tag{9}$$

where ω_S^* is the current value of the angular velocity of the satellite; ω_4^*, ω_5^* are the current values of the angular velocities on the links of hydraulic pump and hydraulic engine.

The values of penalty functions are within $-\infty<P_{\omega S}(|\omega_S|)$, $P_{\omega 4}(|\omega_4|)$, $P_{\omega 5}(|\omega_5|)\leq 0$.

Thus, we will form a general view of the generalized criterion for the process of acceleration

$$K_{A\Sigma}(e_1)=Z_{A1}\cdot\left(1-\frac{t\ (e_1(t))}{t^*_{\max}}\right)+$$
$$+Z_{A2}\cdot\left(\frac{\eta_{HVTDIR}(e_1)}{\eta^*_{HVT\max}}\right)+Z_{A3}\cdot\left(\frac{\eta_{HVTREV}(e_1)}{\eta^*_{HVT\max}}\right)+$$
$$+Z_{A4}\cdot\left(\frac{\eta_{HVMT}(e_1)}{\eta^*_{HVMT\max}}\right)+Z_{\omega_S}\cdot P_{\omega_S}(|\omega_S|)+Z_{\omega_4}\times$$
$$\times P_{\omega_4}(|\omega_4|)+Z_{\omega_5}\cdot P_{\omega_5}(|\omega_5|). \quad (10)$$

Similar for the acceleration, let us compile generalized criterion for the process of braking

$$K_{B\Sigma}(e_1)=Z_{B1}\cdot\left(1-\frac{S(e_1)}{S^*_{\max}}\right)+Z_{B2}\cdot\left(1-\frac{t\ (e_1(t))}{t^*_{\max}}\right)+$$
$$+Z_{B3}\cdot\left(\frac{\eta_{HVTDIR}(e_1)}{\eta^*_{HVT\max}}\right)+Z_{B4}\cdot\left(\frac{\eta_{HVTREV}(e_1)}{\eta^*_{HVT\max}}\right)+$$
$$+Z_{B5}\cdot\left(\frac{\eta_{HVMT}(e_1)}{\eta^*_{HVMT\max}}\right)+Z_{\omega_S}\cdot P_{\omega_S}(|\omega_S|)+Z_{\omega_4}\cdot P_{\omega_4}(|\omega_4|)+$$
$$+Z_{\omega_5}\cdot P_{\omega_5}(|\omega_5|). \quad (11)$$

Assessing the obtained equations, one may notice that the generalized criterion is affected by the magnitude of weight coefficients. Since the sum of weighting coefficients for the criteria should equal 1, and their magnitude is accepted as the equilibrium, then for the acceleration we will take Z_{A1}=0,25, Z_{A2}=0,25, Z_{A3}=0,25, Z_{A4}=0,25, and for the braking – Z_{B1}=0,2, Z_{B2}=0,2, Z_{B3}=0,2, Z_{B4}=0,2, Z_{B5}=0,2. When determining weight coefficients for the penalty functions, it is necessary for their magnitude to help achieve the fastest determining the crossing of permissible values, since realization of the search for a rational change will be performed accurate to hundredths, then $Z_{\omega s}=10^5$, $Z_{\omega 4}=10^5$, $Z_{\omega 5}=10^5$.

The solution to the set problem, determining a rational change in the parameter of adjustment of a hydraulic machine in the process of braking and acceleration, is carried out by one of the methods of optimization. In this paper we applied a method of direct search, namely, the Hook-Jeeves method.

5. Results of the study of indicators when applying rational control system of continuously variable transmission in the process of acceleration and braking of a wheel tractor

In the process of optimization, we formed rational changes in the parameter of adjustment of hydraulic machines at acceleration and braking for 5 s, 10 s, 15 s (Fig. 1, 2). When applying the obtained dependencies, it is expedient to explore the change in the indicators of the wheeled tractor HTZ-21021 with HVMT-1S. Table 1 summarizes the results when applying linear and rational change in the parameter of adjustment of the hydraulic machine. Fig. 3, 4 present graphical change in the indicator of efficiency of HVT η_{HVT} in the process of acceleration and braking of the wheeled tractor.

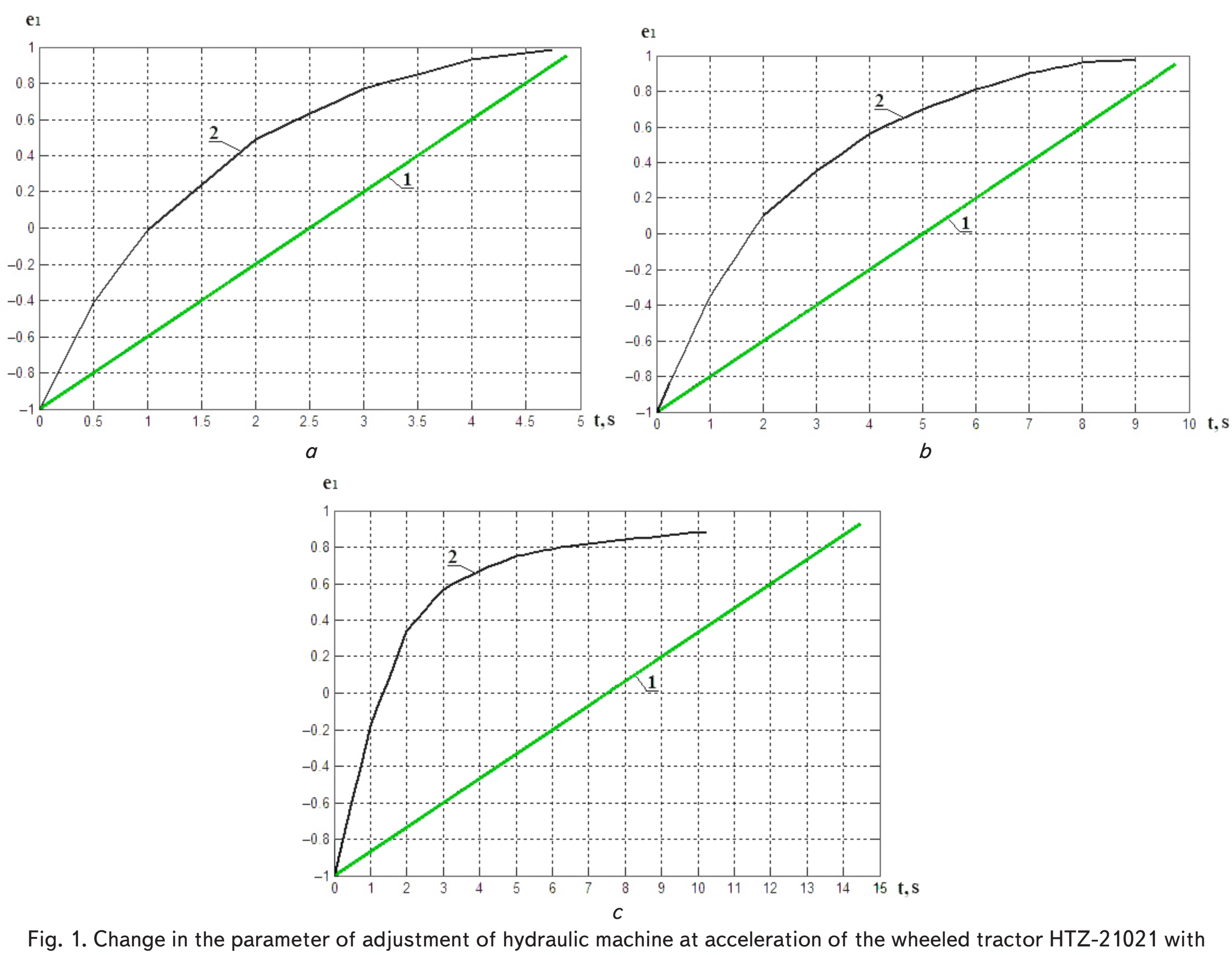


Fig. 1. Change in the parameter of adjustment of hydraulic machine at acceleration of the wheeled tractor HTZ-21021 with HVMT-1S: *a* — for 5 s; *b* — for 10 s; *c* — for 15 s; 1 — by linear dependency; 2 — by rational law

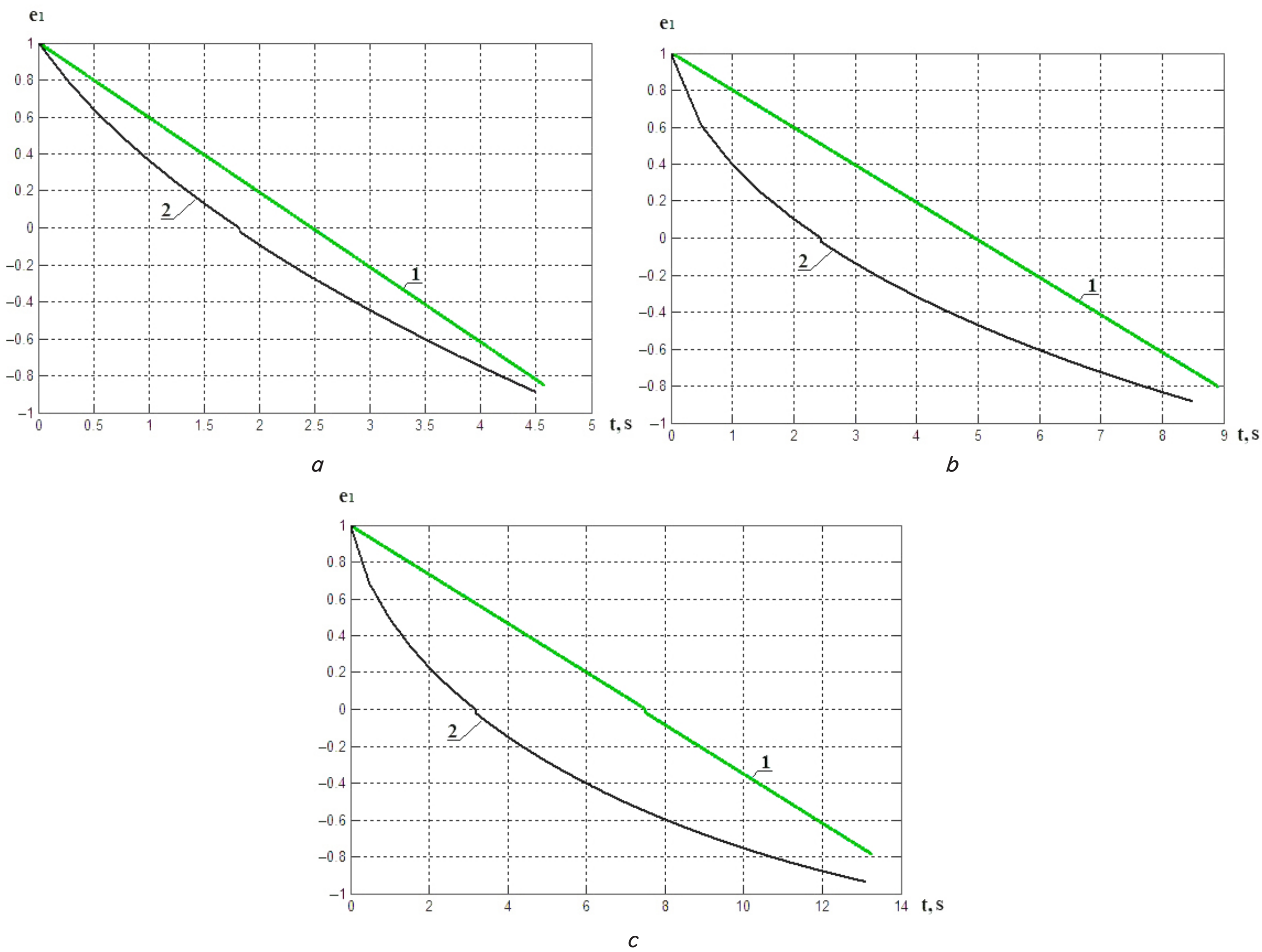
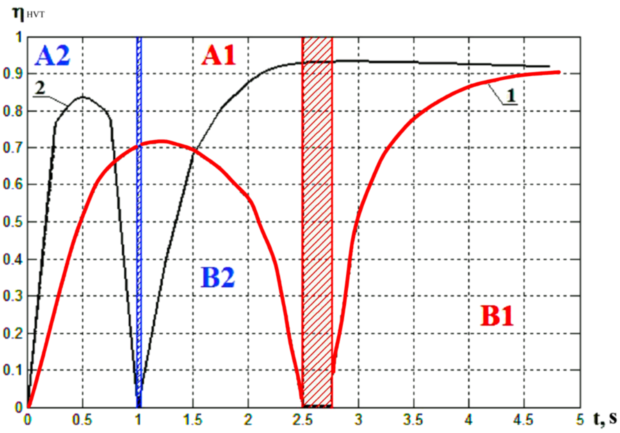


Fig. 2. Change in the parameter of adjustment of hydraulic machine at braking of the wheeled tractor HTZ-21021 with HVMT-1S: *a* — for 5 s; *b* — for 10 s; *c* — for 15 s; 1 — by linear dependency; 2 — by rational law

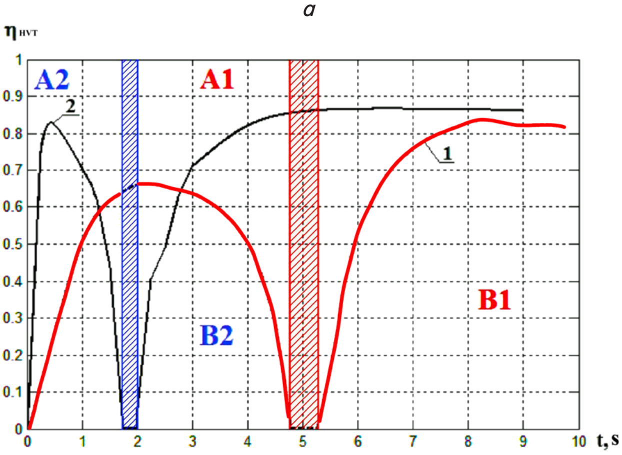
Table 1

Results when applying linear and rational change in the parameter of adjustment of the hydraulic machine at the stage of acceleration and braking of the wheeled tractor HTZ-21021 with HVMT-1S

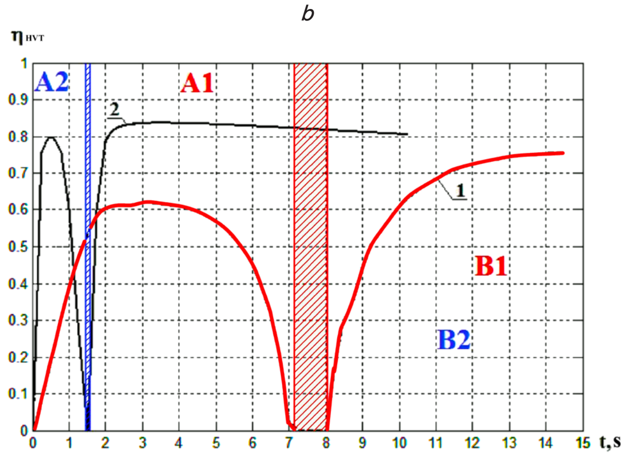
$t(e_1)$, s	t^*_{max}, s	S, m	$\eta_{HVMTmax}$	η_{HVTmax}		η_{Omax}		η_{Mmax}		$\lvert\Delta P\rvert_{max}$, MPa		$K_\Sigma(e_1)$
				DIR	REV	DIR	REV	DIR	REV	DIR	REV	
Acceleration												
Linear change in the parameter of adjustment of hydraulic machine												
5	4,81	–	0,79	0,9	0,72	0,95	0,83	0,95	0,93	15,7	22,8	0,66
10	9,75	–	0,78	0,84	0,66	0,94	0,82	0,9	0,83	12,4	15,4	0,63
15	14,5	–	0,75	0,75	0,62	0,93	0,82	0,8	0,76	11,0	7,5	0,59
Rational change in the parameter of adjustment of hydraulic machine												
5	4,75	–	0,8	0,93	0,87	0,97	0,87	0,96	0,94	27,2	33,4	0,72
10	9,0	–	0,79	0,85	0,82	0,96	0,85	0,95	0,95	18,3	29,6	0,69
15	10,2	–	0,78	0,84	0,8	0,95	0,85	0,95	0,98	11,0	25,1	0,74
Braking												
Linear change in the parameter of adjustment of hydraulic machine												
5	4,66	28,03	0,66	0,71	0,58	0,81	0,87	0,9	0,75	11,0	39,3	0,58
10	8,93	52,53	0,66	0,71	0,29	0,81	0,8	0,9	0,43	11,0	33,7	0,48
15	13,2	77,11	0,66	0,71	0,11	0,81	0,78	0,9	0,15	11,0	16,5	0,4
Rational change in the parameter of adjustment of hydraulic machine												
5	4,5	25,51	0,73	0,88	0,71	0,98	0,97	0,93	0,81	38,1	39,5	0,67
10	8,5	38,79	0,72	0,86	0,61	0,97	0,97	0,93	0,64	39,7	36,4	0,63
15	13,1	50,45	0,7	0,8	0,45	0,88	0,94	0,92	0,54	34,8	31,9	0,55



a

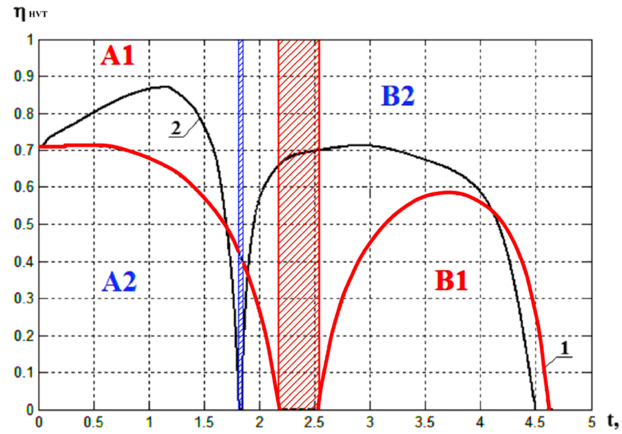


b

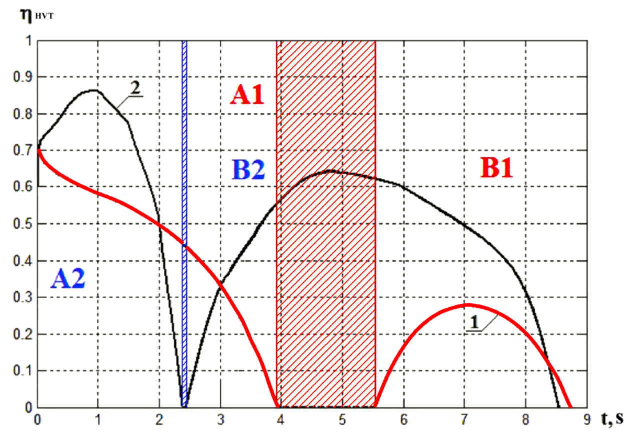


c

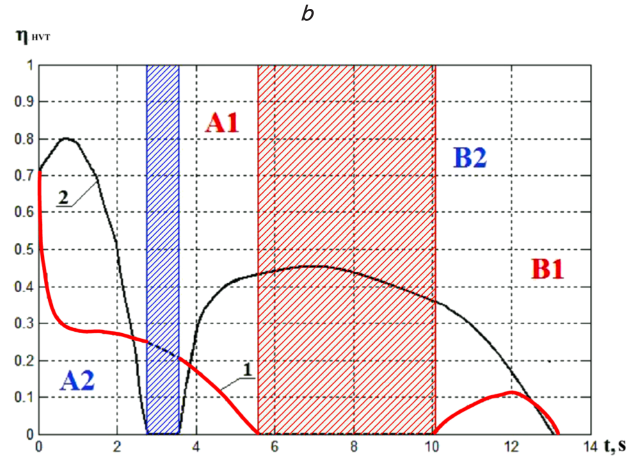
Fig. 3. Change of efficiency of HVT η_{HVT} in the process of acceleration: *a* – 5 s; *b* – 10 s; *c* –15 s; A1, A2 is the zone of reverse flow; B1, B2 is the the zone of direct flow; 1 – e_1 changed by linear change; 2 – e_1 changes by rational change; ▨ is the area of the highest efficiency of HVMT at linear dependency e_1; ▨ is the area of the highest efficiency of HVMT at rational dependency e_1



a



b



c

Fig. 4. Change of efficiency of HVT η_{HVT} in the process of braking: *a* – for 5 s; *b* – 10 s; *c* – 15 s; A1, A2 is the zone of reverse flow; B1, B2 is the zone of direct flow; 1 – e_1 changes by linear change; 2 – e_1 changes by rational change; ▨ is the area of the highest efficiency of HVMT at linear dependency e_1; ▨ is the area of the highest efficiency of HVMT at rational dependency e_1

Analyzing the dependency of efficiency of HVT on the time of acceleration and braking, it was found that the use of the rational change in the parameter of adjustment of a hydraulic machine narrows the zone of the highest efficiency of HVMT (η_{HVT}=0) – this phenomenon points to the larger load of hydraulic branch of HVMT-1S.

6. Discussion of the results of study of the influence of the use of control systems for a wheeled tractor with continuously variable transmission

In the course of theoretical research, by applying rational, instead of linear, change in the parameter of adjustment of the hydraulic machine, the following is observed:

– in the process of acceleration for 5 s:
- decrease: time of acceleration of tractor t^* by 1.25%;
- increase:

– efficiency of HVMT $\eta_{HVMT\,max}$ by 1,27 %;
– efficiency of HVT in direct flow $\eta_{HVT\,DIR\,max}$ by 3,33 %;
– efficiency of HVT in reverse flow $\eta_{HVT\,REV\,max}$ by 20,83 %;
– efficiency volumetric in direct flow $\eta_{O\,DIR\,max}$ by 2,11 %;
– efficiency volumetric in reverse flow $\eta_{O\,REV\,max}$ by 4,82 %;
– efficiency mechanical in direct flow $\eta_{M\,DIR\,max}$ by 1,05 %;
– efficiency mechanical in reverse flow $\eta_{M\,REV\,max}$ by 1,08 %;
– the difference of working pressure in HVT in direct flow $|\Delta P|_{DIR\,max}$ by 73,25 %;
– the difference of working pressure in HVT in reverse flow $|\Delta P|_{REV\,max}$ by 46,5 %;
– in the process of braking for 5 s:
- decrease:

– of the time till full stop of tractor t^* by 3,43 %;
– of the braking distance S by 8,99 %;
- increase:

– efficiency of HVMT $\eta_{HVMT\,max}$ by 9,85 %;
– efficiency of HVT in direct flow $\eta_{HVT\,DIR\,max}$ by 23,94 %;
– efficiency of HVT in reverse flow $\eta_{HVT\,REV\,max}$ by 22,41 %;
– efficiency volumetric in direct flow $\eta_{O\,DIR\,max}$ by 20,99 %;
– efficiency volumetric in reverse flow $\eta_{O\,REV\,max}$ by 11,5 %;
– efficiency mechanical in direct flow $\eta_{M\,DIR\,max}$ by 3,3 %;
– efficiency mechanical in reverse flow $\eta_{M\,REV\,max}$ by 8,0 %;
– the difference of working pressure in HVT in direct flow $|\Delta P|_{DIR\,max}$ by 2,46 times;
– the difference of working pressure in HVT in reverse flow $|\Delta P|_{REV\,max}$ by 0,51 %;
– in the process of acceleration for 10 s:
- decrease: in the time of acceleration of tractor t^* by 7,69 %;
- increase:

– efficiency of HVMT $\eta_{HVMT\,max}$ by 1,41 %;
– efficiency of HVT in direct flow $\eta_{HVT\,DIR\,max}$ by 1,8 %;
– efficiency of HVT in reverse flow $\eta_{HVT\,REV\,max}$ by 24,24 %;
– efficiency volumetric in direct flow $\eta_{O\,DIR\,max}$ by 2,13 %;
– efficiency volumetric in reverse flow $\eta_{O\,REV\,max}$ by 3,66 %;
– efficiency mechanical in direct flow $\eta_{M\,DIR\,max}$ by 5,56 %;
– efficiency mechanical in reverse flow $\eta_{M\,REV\,max}$ by 14,46 %;
– the difference of working pressure in HVT in direct flow $|\Delta P|_{DIR\,max}$ by 47,58 %;
– the difference of working pressure in HVT in reverse flow $|\Delta P|_{REV\,max}$ by 92,2 %;
– in the process of braking for 10 s:
- decrease:

– of the time till full stop of tractor t^* by 4,82 %;
– of the braking distance S by 26,16%;
- increase:

– efficiency of HVMT $\eta_{HVMT\,max}$ by 9,09 %;
– efficiency of HVT in direct flow $\eta_{HVT\,DIR\,max}$ by 21,13 %;
– efficiency of HVT in reverse flow $\eta_{HVT\,REV\,max}$ by 1,1 times;
– efficiency volumetric in direct flow $\eta_{O\,DIR\,max}$ by 19,75 %;
– efficiency volumetric in reverse flow $\eta_{O\,REV\,max}$ by 21,25 %;
– efficiency mechanical in direct flow $\eta_{M\,DIR\,max}$ by 3,3 %;
– efficiency mechanical in reverse flow $\eta_{M\,REV\,max}$ by 49,53 %;
– the difference of working pressure in HVT in direct flow $|\Delta P|_{DIR\,max}$ by 2,6 times;
– the difference of working pressure in HVT in reverse flow $|\Delta P|_{REV\,max}$ by 8,01 %;
– in the process of acceleration for 15 s:
- decrease in the time of acceleration of tractor t^* by 29,66 %;
- increase:

– efficiency of HVMT $\eta_{HVMT\,max}$ by 4,0 %;
– efficiency of HVT in direct flow $\eta_{HVT\,DIR\,max}$ by 12,0 %;
– efficiency of HVT in reverse flow $\eta_{HVT\,REV\,max}$ by 29,03 %;
– efficiency volumetric in direct flow $\eta_{O\,DIR\,max}$ by 2,15 %;
– efficiency volumetric in reverse flow $\eta_{O\,REV\,max}$ by 3,66 %;
– efficiency mechanical in direct flow $\eta_{M\,DIR\,max}$ by 18,75 %;
– efficiency mechanical in reverse flow $\eta_{M\,REV\,max}$ by 28,95 %;
– the difference of working pressure in HVT in reverse flow $|\Delta P|_{REV\,max}$ by 2,3 times;
– in the process of braking for 15 s:
- decrease:

– of the time till full stop of tractor t^* by 0,98 %;
– of the braking distance S by 34,57 %;
- increase:

– efficiency of HVMT $\eta_{HVMT\,max}$ by 6,06 %;
– efficiency of HVT in direct flow $\eta_{HVT\,DIR\,max}$ by 12,68 %;
– efficiency of HVT in reverse flow $\eta_{HVT\,REV\,max}$ by 3,09 times;
– efficiency volumetric in direct flow $\eta_{O\,DIR\,max}$ by 8,64 %;
– efficiency volumetric in reverse flow $\eta_{O\,REV\,max}$ by 20,43 %;
– efficiency mechanical in direct flow $\eta_{M\,DIR\,max}$ by 2,78 %;
– efficiency mechanical in reverse flow $\eta_{M\,REV\,max}$ by 2,6 times;
– the difference of working pressure in HVT in direct flow $|\Delta P|_{DIR\,max}$ by 2,16 times;
– the difference of working pressure in HVT in reverse flow $|\Delta P|_{REV\,max}$ by 93,3 %.

7. Conclusions

1. On the basis of the theory of optimization it was formalized that the most rational change in the parameter of adjustment of the hydraulic machine of HVMT, which operates by the scheme of the differential at the output, there is a change in line with a convex-concave law: convex (relative to linear) in the process of acceleration, concave in the process of braking. Such patterns are related to the fact that when using the linear dependency of change in the parameter of adjustment of hydraulic machine, excitatory forces that occur during acceleration and braking are absorbed evenly by hydraulic and mechanical branches of HVMT. The application of the obtained rational dependencies provides the larger absorption of the excitatory forces by hydraulic branch of the transmission.

2. When applying the rational changes in the parameters of adjustment of hydraulic machine specifically for the wheeled tractor HTZ-21021 with HVMT-1S, the following is observed: decrease in the time of acceleration by 1.25–29,66 %, in the time of braking by 0.98–4,82 %; decrease in the braking distance by 8.99–34,57 %; increase in efficiency of HVT at acceleration by 1.27–4.0 %; the maximum increase in efficiency of HVT at acceleration by 29.03 % and at braking by 3.09 times. Such indicators are predetermined by the greater load of hydraulic branch of continuously variable HVMT-1S compared to the use of linear change in the parameters of adjustment of hydraulic machines.

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