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ANALYSIS OF HYDROSTATIC MECHANICAL TRANSMISSION EFFICIENCY IN THE PROCESS OF WHEELED VEHICLE BRAKING

Summary. To analyze the braking process of a mine diesel locomotive with hydrostatic-mechanical transmission (HSMT), a mathematical model has been developed. In contrast to available models, this one takes into consideration rules of changes in controlling the parameters of hydrostatic drive (HSD) and implementation methods for the braking process. Moreover, the model makes it possible to study changes in kinematic parameters, power parameters, and energy parameters of HSMT under various conditions of a mine diesel locomotive operation. It has been determined that the use of HSMT operating according to the "output differential" scheme is possible, providing kinematic disconnection of the engine from the driving wheels (for emergency braking with complete stop of the diesel locomotive) or at the expense of HSD while preserving the kinematic connection between the engine and the wheels (for deceleration). Use of the braking system together with HSD or the braking system only while preserving the kinematic connection between the engine and the wheels will result in HSMT failure.

1. INTRODUCTION

Improved efficiency of rail transport functioning in mines results from the use of diesel locomotives equipped with hydrostatic mechanical transmissions (HSMTs). The most active share of their use (up to 80% in the modern world market) is in agricultural tractors [1, 2, 3]. The continuously variable hydromechanical transmission is an interesting solution for high-power vehicles subject to frequent changes of speed, in which comfort is a significant requirement [4].

It is considered [5, 6] that first of all implementation of HSMTs depends on the following advantages: a stepless power transmission from an internal-combustion engine to wheels provides a smooth start; decreases in dynamic loads within transmission in the context of varying operation modes; improved ergonomic features; and more possibilities to reduce the velocity down to a complete stop during brakeage stage owing to hydrostatic transmission, with unloading of the standard brakeage system. However, an increase in the traveling velocities of mine trains with diesel locomotives aggravates the problem of safety preservation in the context of the brakeage mode. Despite the popularity of HSMTs in the modern machine-building industry, transmissions of the type should be subjected to further detailed analysis to determine the required design parameters leading to an improvement in both the braking and the operational characteristics of diesel locomotives.

Thus, the necessity to modify available transmissions for diesel locomotives and to design advanced ones arises on the basis of the development of calculation and theoretical methods of analysis of "engine-transmission-diesel locomotive" systems in the context of braking mode.

2. STATE-OF-THE-ART

Since the time of development of wheeled and tracked vehicles to date, problems with the complete use of engine power as well as the design of transmission (gear box) enable the generation of continuous transformation of a torque within a wide range of velocities, analysis of machine dynamics depends on the transmission type, improvement of efficiency of a transmission as well as the technical and economic effectiveness of equipment owing to the expediency of transmission design, reliability, etc. have been still not solved.

As an example of a hydraulic pump drive, paper [7] proposes the use of an air turbine. The innovation made it possible to improve the efficiency of hydrostatic transmission by 17%. However, the technical solution has not been tested enough.

Paper [8] studies optimum control of efficiency balance between an internal-combustion engine and hydrostatic transmission. It is noted separately that to achieve optimum control, the control characteristics of both the hydraulic pump and the hydraulic motor should be determined.

Thesis [9] proposes an automated methodology for the design of complex multiple-mode hydro-mechanical transmissions. Design automation and optimization with detailed simulation models can support the industrial engineer in the design task and increase the available knowledge early in the design process. The proposed methodology uses simulation-based optimization to design the transmission for a specific vehicle application.

Owing to the studies of the selection of the most appropriate angle of hydraulic motor washer inclination [10], it has been determined that efficient operation of HSMT in a hydraulic branch involves an uncontrolled hydraulic motor. It is pointed out that in terms of low velocities and torques, a hydraulic system may operate more efficiently at a lower angle of hydraulic motor washer inclination.

Using mathematical modeling and experiments, the authors of [11] define the efficiency of HSD and machine control on the whole while applying one and two hydraulic motors. In addition, they emphasize that the current stage of the research should involve studying the dependence of characteristics of losses resulting from a torque and parameters of hydraulic machine control.

In the design of a hydro-mechanical transmission, the achievement of high efficiency is one of the most important objectives. Furthermore, the available space for the transmission is always limited, constraining the design process. In paper [12], therefore, the design of a hydro-mechanical transmission has been considered as a multi-objective optimization problem, where the goal is not only the best efficiency but also the smallest size of the transmission. A calculation procedure for a “Pareto optimization” of Input Coupled and Output Coupled transmissions has been carried out on MATLAB and AMESim platforms, in which the research of the optimum solution is carried out using the algorithm of the Multi-Objective Particle Swarm Optimizer.

The results of tests [13] on a full power-shift transmission, suitable for use on a 140 kW, agricultural tractor are reported. The aims of the research were to evaluate the working conditions where there are higher power losses and to identify the causes of energy dissipation.

The authors [14-16] identified the trends and prospects for application of stepless HSMT in automotive and tractor industry, and carried out a comparative analysis of stepless two-engine hydro-mechanical transmissions.

Paper [17] reports that designs of HSMTs are being developed tow decrease the number of frictional multidisc clutches depending on the decrease in the number of subranges and complex mechanical components. Moreover, it has not been determined finally which HSMT schemes may be more expedient to use: “input-differential” or “output-differential”.

The majority of the considered papers propose either too complicated designs of transmissions or transmissions whose efficiency is not more than 0.8. As a result, despite the wide popularity of HSMTs in the tractor-building industry and the great variety of schematic solutions, current designs of HSMTs have major disadvantages, in particular, complicated designs factoring into high costs. Switching from subrange to subrange within a contour with hydraulic transmission may change spasmodically the pressure of the working liquid, resulting in impact modes within HSD and its resource decrease. Braking from 15 km/h velocity and higher, incorrect braking mode and intensity of changes in controlling parameters of HSD involve not only spasmodic changes in working liquid pressure in

HSD but also a marked increase in the value of angular velocities of HSMT links. This is followed by overload of both HSD and planetary gear set and clutch.

Despite the availability of numerous papers considering the development and analysis of HSMTs for self-propelled machines, insufficient attention is paid to the problems of structural analysis and braking control of mine diesel locomotives with stepless HSMTs. Studies [18, 19] involve general recommendations of the rational law of the tractor braking process characterized by an effective braking process, stability, and controllability. Unfortunately, the papers present a graphical representation only for rational changes in controlling parameters of hydraulic machines for a tractor with HSMT operating in the mode of an input differential.

For a mine diesel locomotive, paper [20] proposes advanced HSMT #2 operating according to the "output differential" scheme, with the recorded maximum efficiency within 0.85-0.9 (depending on the movement range and the rolling resistance force).

3. MATERIALS OF THE STUDIES

3.1. Development of a Simulation Model

HSMT #2 (Fig. 1, 2) as a typical example of HSMT providing two two-stream range of movements (traction and transport) has been selected as the basic model considered in the paper as the object of mathematical modeling. The transmission concept presented in Fig. 1 is commonly defined as an "Input Coupled (IC) power-split transmission".

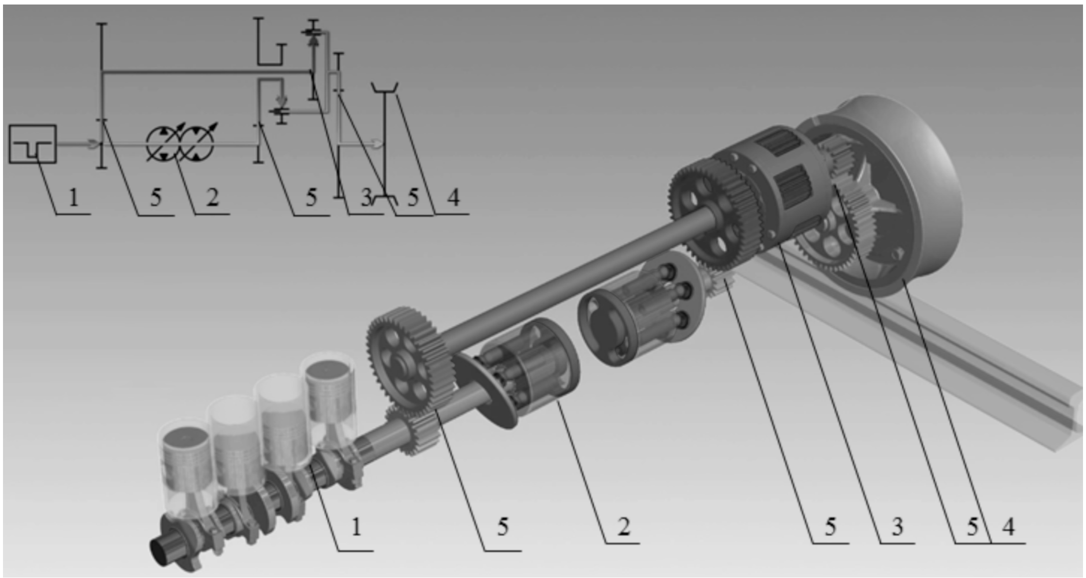

Fig. 1. 3-D model of HSMT #2: 1 – internal-combustion engine; 2 – hydrostatic drive; 3 – planetary reducing gear; 4 – wheel; 5 – reducing gears

The paper uses a method of transmission analysis. The method relies on kinematic scheme breakdown into structural components as well as the development of a matrix system on the basis of generalized kinematic and power basic matrices of each component of the transmission.

Kinematic link indexing is performed randomly in such a way that each of the links has a proper value of angular velocity. However, the moments are set with dual indexing; index one of a moment coincides with the number of kinematic link and index two is a Latin letter in line with the alphabet depending on the link complexity. Thus, each link involves no fewer than two moments: input moment and output one, for instance, M_{2a} and M_{2b} for kinematic link #2 associated with angular velocity ω_2. The method is specified in [21].

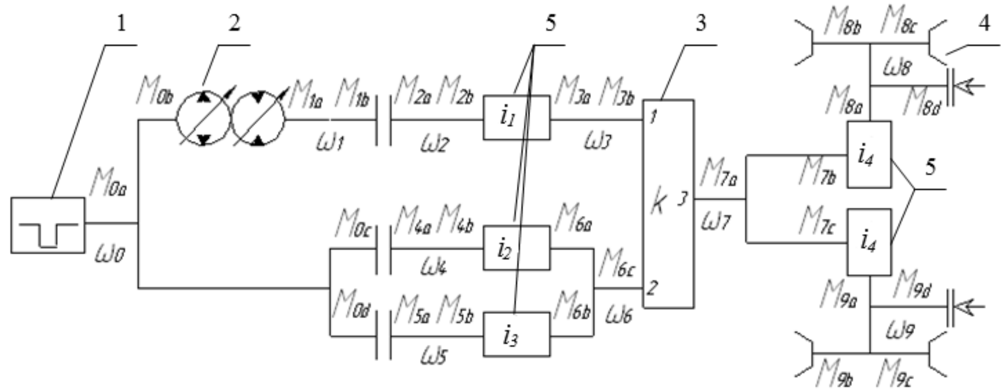

Fig. 2. Physical model of HSMT #2: 1 – 5 correspond to Fig.1; ω_i – angular velocity of transmission link; i_j – transmission ratio; k – internal transmission ratio of planetary gear set; M_{nm} – moments on HSMT links; *n* – indices corresponding to the number of link angular velocity; *m* – indexing letters correspond ing to the moments of the ends of links

The mathematical model of a transmission operation developed by the authors is a system of equations describing changes in power parameters, kinematic parameters, and energy parameters of HSMT #2 in the diesel locomotive braking process:
Thus, equation one explains the transmission-engine relation as follows:

$$\dot{\omega}_0 = \dot{\omega}_d; \tag{1}$$

Loss moment within hydraulic machines is determined using the following expression:

$$\begin{gathered}
e_1(t)\cdot q_1\cdot\dot{\omega}_0 + q_1\cdot\omega_0\cdot\dot{e}_1(t) - e_2(t)\cdot q_2\cdot\dot{\omega}_1 - q_2\cdot\omega_1\cdot\dot{e}_2(t) = \\
\left(\frac{V_{0*}}{2\cdot\pi\cdot E(g^*)}\cdot\left(|\omega_0|+|\omega_1|\right)\right)\cdot\Delta\dot{P} + \frac{V_{0*}}{2\cdot\pi\cdot E(g^*)}\cdot\left(\frac{d}{dt}|\omega_0| + \frac{d}{dt}|\omega_1|\right)\cdot\Delta P + \\
+\gamma\cdot\left(\frac{K_{1y}}{\mu}\cdot\left(1+C_{1y}\cdot|\omega_0|\right) + \frac{K_{2y}}{\mu}\cdot\left(1+C_{2y}\cdot|\omega_1|\right)\right)\cdot\Delta\dot{P} + \\
+\gamma\cdot\left(\frac{K_{1y}}{\mu}\cdot C_{1y}\cdot\frac{d}{dt}|\omega_0| + \frac{K_{2y}}{\mu}\cdot C_{2y}\cdot\frac{d}{dt}|\omega_1|\right)\cdot\Delta P;
\end{gathered} \tag{2}$$

Total loss of working liquid within hydraulic pumps and hydraulic motor is as follows:

$$\begin{gathered}
M_{0b} - e_1\cdot q_1\cdot\Delta P = -\Delta M_1\cdot sign(\omega_0); \\
\Delta M_1 = q_1\cdot\left[\bar{K}_1\cdot|\omega_0|\cdot(1+\bar{K}_2\cdot\bar{e}_1^2) + \frac{\bar{K}_5\cdot(1+\bar{K}_4\cdot|\bar{e}_1|)}{(1+\bar{K}_3\cdot|\omega_0|\cdot D_{q1})}\cdot\Delta P + \frac{\bar{K}_8\cdot(1+\bar{K}_7\cdot|\bar{e}_1|)}{(1+\bar{K}_6\cdot|\omega_0|\cdot D_{q1})}\right]; \\
D_{q1} = \sqrt[3]{2\cdot\pi\cdot q_1};\ M_{1a} + e_2\cdot q_2\cdot\Delta P = -\Delta M_2\cdot sign(\omega_1); \\
\Delta M_2 = q_2\cdot\left[\bar{K}_1\cdot|\omega_1|\cdot(1+\bar{K}_2\cdot\bar{e}_2^2) + \frac{\bar{K}_5\cdot(1+\bar{K}_4\cdot|\bar{e}_2|)}{(1+\bar{K}_3\cdot|\omega_1|\cdot D_{q2})}\cdot\Delta P + \frac{\bar{K}_8\cdot(1+\bar{K}_7\cdot|\bar{e}_2|)}{(1+\bar{K}_6\cdot|\omega_1|\cdot D_{q2})}\right]; \\
D_{q2} = \sqrt[3]{2\cdot\pi\cdot q_2};\ M_{1b} + M_{2a} = 0,\ \Psi = 1;\ M_{1b} = M_{2a} = 0,\ \Psi = 0;
\end{gathered} \tag{3}$$

HSMT kinematics is explained with the help of a system of the following equations:

$$\begin{aligned}
&\dot{\omega}_1 - \dot{\omega}_2 = 0,\ \Psi = 1; \quad \dot{\omega}_2 \cdot i_1 - \dot{\omega}_3 = 0;\\
&\dot{\omega}_0 - \dot{\omega}_4 = 0,\ Y = 1; \quad \dot{\omega}_4 \cdot i_2 - \dot{\omega}_6 = 0;\\
&\dot{\omega}_0 - \dot{\omega}_5 = 0,\ Y = 0; \quad \dot{\omega}_5 \cdot i_3 - \dot{\omega}_6 = 0;\\
&\dot{\omega}_3 - k \cdot \dot{\omega}_6 + (k-1) \cdot \dot{\omega}_7 = 0;\ \dot{\omega}_7 \cdot i_4 - \dot{\omega}_8 = 0;\ \dot{\omega}_7 \cdot i_4 - \dot{\omega}_9 = 0;
\end{aligned} \tag{4}$$

Power transmission parameters can be described using the following equations:

$$\begin{aligned}
&M_{2b} \cdot \eta_1^{\Theta \cdot sign(N_{2b})} + i_1 \cdot M_{3a} = 0;\\
&M_{0c} + M_{4a} = 0, \quad Y = 1;\ M_{0d} = M_{5a} = 0, \quad Y = 1;\ M_{4b} \cdot \eta_2^{\Theta \cdot sign(N_{4b})} + i_2 \cdot M_{6a} = 0;\\
&M_{0d} + M_{5a} = 0, \quad Y = 0;\ M_{0c} = M_{4a} = 0, \quad Y = 0;\ M_{5b} \cdot \eta_3^{\Theta \cdot sign(N_{5b})} + i_3 \cdot M_{6b} = 0;\\
&M_{3b} \cdot \eta_{13}^{\Theta \cdot sign(N_{3b})} + M_{6c} \cdot \eta_{23}^{\Theta \cdot sign(N_{6c})} + M_{7a} = 0;\\
&M_{3b} \cdot k \cdot \eta_{13}^{\Theta \cdot sign(N_{3b})} + M_{6c} \cdot \eta_{23}^{\Theta \cdot sign(N_{6c})} = 0;\\
&M_{7b} \cdot \eta_4^{\Theta \cdot sign(N_{7b})} + i_4 \cdot M_{8a} = 0;\ M_{7c} \cdot \eta_4^{\Theta \cdot sign(N_{7c})} + i_4 \cdot M_{9a} = 0;\\
&M_{8b} = M_{8c} = M_{V1};\ M_{8b} = M_{9c} = M_{V2}; \quad M_{0a} + M_{0b} + M_{0\tilde{n}} + M_{0d} = 0;\\
&M_{1a} + M_{1b} = 0; \quad M_{2a} + M_{2b} = 0;\\
&M_{3a} + M_{3b} = 0; \quad M_{4a} + M_{4b} = 0;\\
&M_{5a} + M_{5b} = 0; \quad M_{6a} + M_{6b} + M_{6c} = 0;\\
&M_{7a} + M_{7b} + M_{7\tilde{n}} = 0; \quad M_{8a} + M_{8b} + M_{8\tilde{n}} + M_{8d} \cdot \Omega = 0;\\
&M_{9a} + M_{9b} + M_{9\tilde{n}} + M_{9d} \cdot \Omega = 0,
\end{aligned} \tag{5}$$

where $\dot{\omega}_i$ is the angular deceleration of a link; i_j is the transmission ratio; k is the internal transmission ratio of the planetary gear set; Ψ is the coefficient characterizing the type of engine-wheels connection in the process of mine diesel locomotive braking ($\Psi = 1$ is without kinematic disconnection; $\Psi = 0$ is with kinematic disconnection); $e_1(t), e_2(t)$ are laws of changes in the parameters of HSD hydraulic machines control; V_{0*} is the volume of the compressible liquid; $E(g^*)$ is the modulus of elasticity of the operating liquid depending on g^* gas content per cent; ΔP is the operating pressure differential within HSD; K_{iy}, C_{iy} are loss coefficients for a hydraulic pump ($i = 1$) and a hydraulic motor ($i = 2$); μ is the coefficient of dynamic viscosity; Y is the coefficient characterizing movement ($Y = 1$ is traction movement, $Y = 0$ is transport movement); η_j is the reducing gear efficiency; η_{13}, η_{23} are efficiencies within gear-tooth systems of sun-satellite gear epicycle-satellite gear in terms of stopped carrier determining moment losses; Θ is the coefficient of loss recording within gear-tooth systems ($\Theta = 0$ is without loss consideration, $\Theta = -1$ is loss consideration within gear-tooth systems); N_{nm} is the power transmitted by HMST links; e_1, e_2 are parameters of HSD hydraulic machines control; q_1, q_2 are the maximum capacities of hydraulic machines; $\Delta M_1, \Delta M_2$ are the losses of moments within hydraulic machines; $\bar{K}_1, \bar{K}_2, ...\bar{K}_8$ are coefficients of hydromechanical losses; D_{qi} is the typical dimension of a hydraulic machine; M_{nm} is moments within the links of the HMST system; *n* is indices corresponding to the angular velocity of link rotation; *m* is indexing letters corresponding to the moments within the link ends; M_{Vaxis} are moments within the wheel axles whose components are braking moments; $axis$ are indices characterizing the number of axis ($axis = 1$ is front axis, $axis = 2$ is back axis); and Ω is the coefficient characterizing the state of braking mechanisms

($\Omega = 1$ means that braking mechanisms are switched on; $\Omega = 0$ means that braking mechanisms are switched off).

The distribution of kinematic parameters, power parameters, and energy parameters of HMST in the braking process depends strongly on the following:

– Transmission type;
– Initial velocity (the diesel locomotive braking process starts from the velocities of $V_{\max}$ and $0,5 \cdot V_{\max}$; in terms of HMST #2, $V_{\max}$ within the transport range is 17.96 km/h and within traction range it is 10.55 km/h);
– Drawbar force (as a rule, in terms of movement within the transport range, mine cars are empty, their maximum number is *n* for the selected operating conditions, which is equivalent to 2 loaded mine cars (it is assumed with ample); in terms of traction range, two loading variants are considered: 8 or 2 loaded mine cars);
– Ascending and descending angle (being equivalent to 50 ‰)
– Method of braking process.

The complex study of the braking process of a mine diesel locomotive with HMSTs involves an additional mathematical model [22] that yields quite an accurate qualitative description of the dynamics of mine diesel locomotive braking in terms of comparatively minor lateral forces. The model makes it possible to reveal the physical essence of the processes and to determine the effect of various factors on a braking dynamics.

Braking process modeling involves the following implementation methods:

1. In terms of a kinematic disconnection of engine from wheels (further, the implementation method will be marked as #1) and maximum possible braking moments within wheels and 2 different laws of changes in $e_1(t)$, $e_2(t)$. Law 1 ($e_1(t)$, $e_2(t)$) will be characterized by changes in controlling parameters of HSD hydraulic machines according to changes in the actual velocity of a diesel locomotive movement being marked as $e_{1\_1V\text{пот}}(t)$, $e_{2\_1V\text{пот}}(t)$. In case of law 2, during the entire braking process, controlling parameters of HSD remain unchangeable corresponding to the values that they had at the start of the braking process, i.e. $e_{1\_1V\max}(t)$, $e_{2\_1V\max}(t)$.

2. In terms of preserving the engine-wheels kinematic connection:

• decelerating at the expense of HSD and the braking system while preserving the engine-wheels kinematic connection (implementation method #2);
• decelerating at the expense of HSD while preserving the engine-wheels kinematic connection (implementation method #3); braking mechanisms form a moment equal to 0;
• decelerating at the expense of braking system while preserving the engine-wheels kinematic connection (implementation method #4). During the entire braking process, parameters of HSD control remain unchangeable corresponding to the values that they had at the start of the braking process, i.e. $e_{1\_4V\max}(t)$, $e_{2\_4V\max}(t)$.

As the braking process preserving the engine-wheels kinematic connection is usually the service one, consider $e_1(t)$, $e_2(t)$ laws to analyze the effect of initial braking velocity and drawbar force of a mine diesel locomotive on the distribution of kinematic parameters, power parameters, and energy parameters of HSMT. The laws stipulate time changes in e_1 and e_2 from values corresponding to the initial braking velocity to e_1 and e_2 values corresponding to zero velocity of a diesel locomotive at the level of 100.0 sec.

Qualitative evaluation of the effect of the initial velocity of braking and drawbar force of a mine diesel locomotive depends on the distribution of kinematic parameters, power parameters, and energy parameters of HSMT, detection, and systematization of basic regularities of power stream distributions within the closed contour of HSMT #2, whereas the braking process of a mine diesel locomotive takes place at the expense of a software support developed in MatLab/Simulink and combining mathematical models and [22].

The following parameters are studied in the process of braking modeling using four implementation methods:

– The maximum value of the working pressure differential in HSD $|dP|_{max}$ should not be more than the 40.0 MPa for hydraulic machines with a working volume of 90 cm^3;
– The maximum value of the angular velocity of a hydraulic pump shaft $|w_0|_{max}$ should not be more than 460.0 rad/s;
– The maximum value of the angular velocity of a hydraulic motor shaft $|w_1|_{max}$ should not be more than 460.0 rad/s;
– The maximum value of the angular velocity of a driven clutch shaft of HSMT #2 should be $|w_2|_{max}$;
– The maximum value of the angular velocity of the hydraulic branch of the closed contour of HSMT power output should be $|N_{gk}|_{max}$;
– The maximum value of the angular velocity of the mechanical branch of the closed contour of HSMT power output should be $|N_{mk}|_{max}$;
– Braking path is S;
– Braking period is t.

Angular velocities of transmission links and working pressure differential in HSD for HSMT #2 within transport range had the following initial values:

• V=17.96 km/h, w_0=210.0 rad/s, w_1=200.6 rad/s, ΔP=-6.26 MPa, e_1=1, e_2=1;
• V=8.98 km/h, w_0=210.0 rad/s, w_1=-127.7 rad/s, ΔP=-4.54 MPa, e_1=-0,58, e_2=1.

Despite the fact that the value of the working pressure differential within HSD ΔP is set in a mathematical model of the braking process as the initial condition, it is clarified according to the operating conditions as well as the method of braking process implementation. This is why ΔP set as the initial one will differ from a value obtained during the modeling process. As for the other parameters, values set as the initial ones will correspond to the first values obtained in the process of braking process modeling. Further, the paper will analyze only the maximum value of the parameters being analyzed as they characterize the loading level of the transmission.

3.2. Simulation results

As an example, the results of braking modeling of a diesel locomotive with HSMT #2 while moving within the transport range (V_{max} =17.96 km/h, n=2, descending) in the context of various methods of the braking process implementation are represented in the form of graphical curves in Fig. 3 (method #1) and Fig. 4 (method #2).

It should be noted that the dP operating pressure differential being clearly seen within 11.3 sec (Fig.4, c) can be explained by the specificity of the wheel–rail interaction. It is known [23] that the peculiarity of a locomotive wheel–rail interaction is in the fact that an insignificant increase in wheel slippage results in a spasmodic increase in the adhesion coefficient. In the process of braking, slippage changes within the range of 0.01 – 0.1. Changes in slippage took place from the value of 0.001 to the value of 0.003 within the range of 11.3 sec. One would believe that insignificant change in slippage has resulted in spasmodic changes in the adhesion coefficient ranging from 0.025 to a value of 0.075. As a result, we have load increase within the front wheelset and load relief within the rear one. As the rear wheelset is connected kinematically with the transmission, the decrease in reaction [22] within the rear wheelset results in pressure decrease in the transmission. Hence, this is the source of the slight operating pressure differential that can be seen in the graph.

Moreover, the modeling was also performed for methods #3 and #4. However, the paper does not report its graphic results.

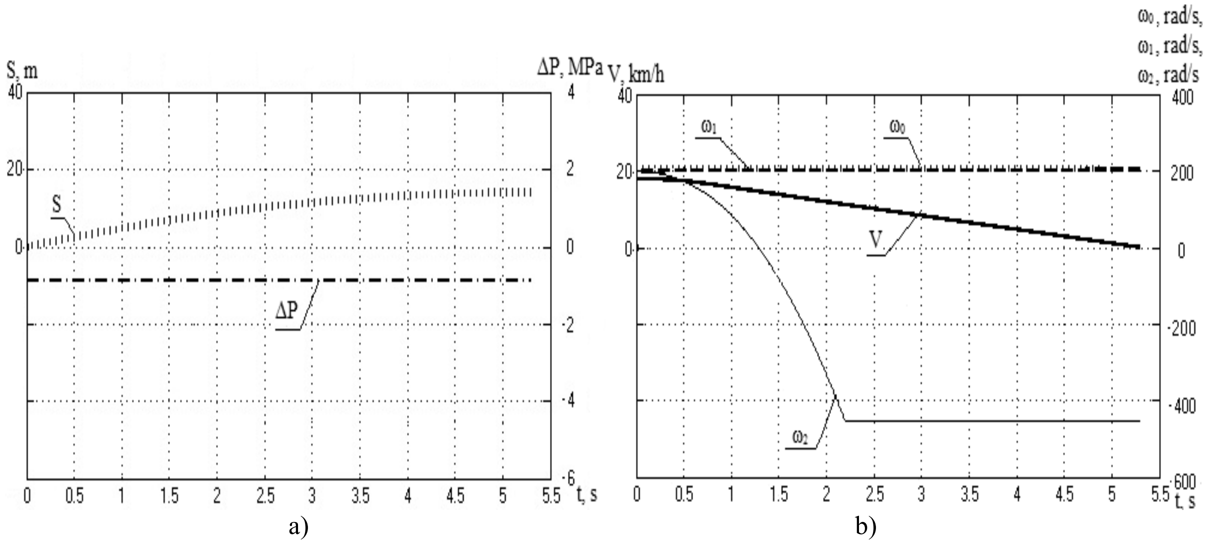

Fig. 3. Results of braking modeling of a diesel locomotive with HSMT #2 while moving within the transport range, method of braking process #1 implementation (laws $e_1(t)$, $e_2(t)$ – $e_{1\_1Vmax}(t)$, $e_{2\_1Vmax}(t)$): a) braking path dependence on the braking period as well as on working pressure difference within HSD; b) changes in the velocity of the diesel locomotive depending on its braking period as well as angular velocities of the transmission links. Angular velocities of transmission chains 0 and 1 are equal to ($\omega_0 = \omega_1$) as the HSD control parameter is equal to 1 (e=1)

The process of theoretical studies of the braking process of a diesel locomotive with HSMT #2 has yielded the following:

1. In terms of the kinematic disconnection of the engine from wheels (the implementation method #1), both mechanical and hydraulic branches of the closed contour of HSMT #2 are always unloaded as the moments within links of physical model (Fig. 2) "1", "2", "3", "6", and "7" are equal to 0.

Under one and the same operation conditions, the braking path of a diesel locomotive with a kinematic disconnection of engine from wheels on the ascent is less down to 87.5% (depending on the drawbar force) compared with the one on the descent.

With a decrease in the number of mine cars from 8 to 2, the braking path decreases on the descent down to 89.2% and down to 23.6% on the ascent.

As within closed contour of the HSMT kinematic disconnection of mechanical branch takes place owing to clutch, neither initial braking velocity nor ascending/descending angle, or drawbar power of a mine diesel locomotive has an effect on the distribution of kinematic parameters, power parameters, and energy parameters of HSMT.

2. In terms of deceleration at the expense of HSD and the braking system while preserving the kinematic engine–wheels connection (method of braking process #2 implementation), in the process of the mechanical branch, the closed contour of HSMT is more loaded in the braking process. The power stream transmitted through the mechanical branch of the closed contour is more (up to 21.5 times) compared with the hydraulic one.

The decrease in the drawbar power of a diesel locomotive from 8 mine cars to 2 ones in the process of braking on the ascent results in the decrease of power output from the hydraulic branch of the closed contour down to 36.6%; power output from the mechanical branch of the closed contour of HSMT down to 28.2%; and braking path shortening down to 22.1%. In turn, the angular velocity of the hydraulic pump shaft increases up to 38.7%. The braking process on the ascent results in an increase of the working pressure differential in HSD up to 41.2 times; the angular velocity of the hydraulic pump shaft up to 2.3 times; the angular velocity of the hydraulic motor shaft up to 3.22 times; power output from the hydraulic branch of the closed contour up to 12.3 times; power output from the mechanical branch of the closed contour of HSMT up to 8.5 times; and the braking path up to 3.9 times.

A clear increase in the values of the working pressure differential in HSD up to 22.0 times is observed. An increase in the following is also observed: the angular velocity of the hydraulic pump shaft up to 5.4 times; the hydraulic motor shaft up to 22.6 times; power output from the hydraulic branch of the closed contour of HSMT up to 46.9 times; power output from the mechanical branch of the closed contour of HSMT up to 15.9 times in terms of initial braking from velocity $V_{\max}$ instead of $0.5 \cdot V_{\max}$ velocity. It is not permitted to implement the method of braking process #2 as it is followed by excess allowable value of the working pressure differential within HSD up to 2.74 times.

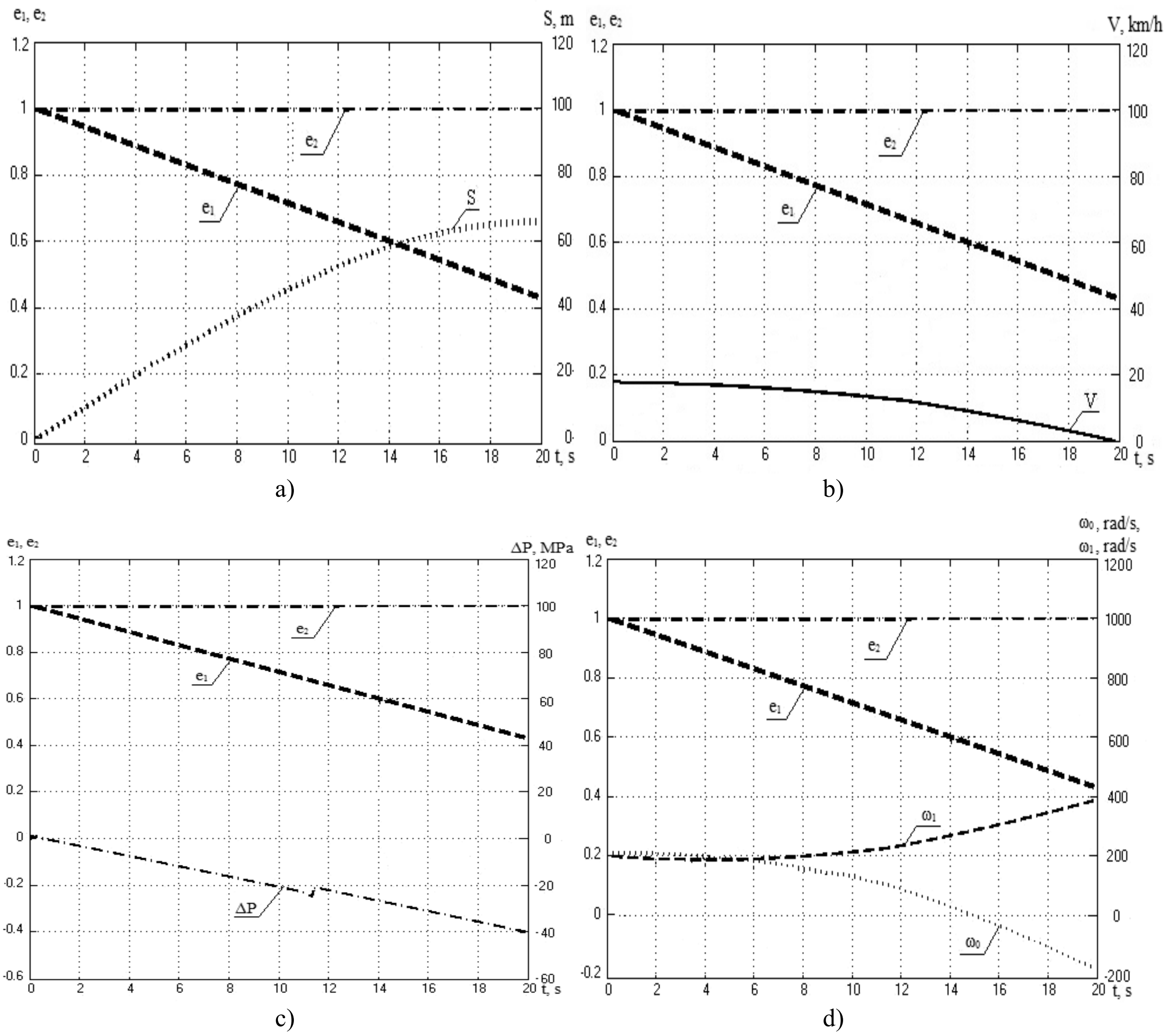

Fig. 4. Results of braking modeling of a diesel locomotive with HSMT #2 while moving within transport range, method of braking process #2 implementation: a) dependence of braking path on braking period and parameters of HSD control; b) changes in the diesel locomotive velocity on the basis of the braking period and parameters of HSD control; c) dependence of working pressure difference within HSD on the braking period and parameters of HSD control; d) dependence of angular velocities of transmission links on the braking period and parameters of HSD control

3. In terms of braking at the expense of HSD while preserving the kinematic engine-wheels connection (method of braking process #3 implementation), the mechanical branch of the closed contour of HSMT is more loaded in the braking process. Power up to 90.0 times more is transmitted through the mechanical branch of the closed contour to compare with the hydraulic one.

The decrease in the drawbar power of a diesel locomotive from 8 mine cars to 2 ones in the process of braking on the descent results in a decrease in the working pressure differential in HSD down to 81.5%; the angular velocity of the hydraulic pump shaft down to 43.0%; the angular velocity of the hydraulic motor shaft down to 42.9%; power output from the hydraulic branch of the closed contour down to 94.1%; power output from the mechanical branch of the closed contour of HSMT down to 93.6%; and braking path shortening down to 5.1%.

The braking process on the ascent results in a decrease in the working pressure differential in HSD down to 39.1 times; the angular velocity of the hydraulic pump shaft down to 75.8 times; and power output from the mechanical branch of the closed contour of HSMT down to 90.0 times. The braking path is increased up to 3.9 times. The angular velocity of the hydraulic pump shaft and power output from the hydraulic branch of the closed contour remains unchangeable.

Braking from $V_{\max}$ velocity instead of $0.5 \cdot V_{\max}$ velocity leads to a clear increase in values of the angular velocity of the hydraulic pump up to 97.0 %; the angular velocity of the hydraulic motor shaft increases up to 85.5 times; power output from the hydraulic branch of the closed contour of HSMT increases up to 84.0 times; and power output from the mechanical branch of the closed contour of HSMT increases up to 12.9 times. Use of the method of braking process #3 implementation makes it possible to decelerate diesel locomotive movement without an obligatory stop; however, each scheme of HSMT needs limitation of intensity of changes in e_1 and e_2 at a certain level to prevent the operator from applying the method for emergency braking.

4. In terms of braking at the expense of the braking system while preserving the kinematic engine–wheels connection (method of braking process #4 implementation), the mechanical branch of the closed contour of HSMT is more loaded in the braking process. Power up to 95.0 times more is transmitted through the mechanical branch of the closed contour compared with the hydraulic one.

The decrease in the drawbar power of a diesel locomotive from 8 mine cars to 2 ones in the process of braking on descent results in a decrease in the power output from the hydraulic branch of the closed contour down to 41.7 %; power output from the mechanical branch of the closed contour of HSMT down to 28.2%; and braking path shortening down to 24.8%. The angular velocity of the hydraulic pump shaft increases up to 38.7 times.

The braking process on the ascent results in an increase in the working pressure differential in HSD up to 4.05 times; the angular velocity of the hydraulic pump shaft up to 3.03 times; the angular velocity of the hydraulic motor shaft up to 4.4 times; power output from the hydraulic branch of the closed contour up to 15.04 time; power output from the mechanical branch of the closed HSMT contour up to 15.04 time; and the braking path is increased up to 3.9 times.

Braking from $V_{\max}$ velocity instead of $0.5 \cdot V_{\max}$ velocity shows a clear increase in the values of working pressure differential in HSD by up to 2.2 %; the angular velocity of the hydraulic pump shaft by up to 5.4 times and the hydraulic motor shaft by 465.5 times; power output from the hydraulic branch of the closed contour of HSMT by up to 1414.0 times; and power output from the mechanical branch of the closed HSMT contour by up to 15.9 times. Use of the method of braking process #4 implementation is not admissible as it is followed by excess allowable value of the working pressure differential in HSD by up to 2.74 times.

4. CONCLUSIONS

This paper considers the braking process of a mine diesel locomotive with hydrostatic mechanical transmission operating according to the “output differential” scheme. Showing up and systematization of basic regularities in the distribution of power flows within a closed transmission contour in the process of braking have been implemented using software support developed by means of MatLab/Simulink. The results of diesel locomotive braking simulation during its movement within various braking process implementations are represented in the form of graphical dependences. Analysis of the braking process indicated that the use of output-differential hydrostatic mechanical transmission in the context of diesel locomotives involves two braking methods – emergency braking with complete stop of a diesel locomotive in terms of kinematic disconnection of the engine from driving wheels and a decrease in speed at the expense of hydrostatic mechanical transmission; in this context, the kinematic connection between the engine and the wheels is preserved. Use of the braking system and hydrostatic mechanical transmission or the braking system alone while preserving the kinematic connection between the engine and the wheels will result in breakdown of hydrostatic mechanical transmission.

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