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## POWER FLOWS IN A HYDROSTATIC-MECHANICAL TRANSMISSION OF A MINING LOCOMOTIVE DURING THE BRAKING PROCESS

**Summary.** This paper considers the braking process of a mine diesel locomotive with hydrostatic mechanical transmission (HSMT) operating according to the “input differential” scheme. Braking process modeling involves four implementation methods. Identification and systematization of basic regularities in the distribution of power flows within a closed transmission contour in the process of braking have been performed with the help of software support developed by means of MatLab/Simulink. The simulation results of braking due to the hydrostatic transmission and the braking system during the movement of a diesel locomotive in the transport and traction ranges are presented in the form of graphical correlations. The process of theoretical studies of the braking process of a diesel locomotive with HSMT operating according to the “input differential” scheme has helped determine that, in terms of deceleration at the expense of a hydrostatic drive (HSD) and braking system while preserving kinematic engine-wheels connection, it is not permitted to implement this method of braking process as it is followed by excess of the allowable value of working pressure differential within HSD up to 2.8 times.

### 1. INTRODUCTION

The need to meet tight pollution standards and to reduce operational costs has resulted in the necessity to carry out research aimed at seeking new solutions that can improve the total effectiveness of a vehicle [1-5]. Efficiency of a transmission as well as its reliability have a sizable effect on the performance characteristics of a traction-transportation vehicle [6-7]. The current technologies make it possible to use the following variator types: hydrodynamic transmission (i.e. torque converter), hydrostatic transmission (i.e. hydraulic pump/motor), mechanical transmission, and electrical transmission. However, the hydrostatic drive (HSD) has relatively low efficiency due to the repeated energy conversion; hydrodynamic transmission usually achieves high efficiency in terms of high velocities of a vehicle [8]. Thus, complex hydrostatic mechanical transmissions (HSMTs) have become rather popular recently owing to their essential advantages (first of all, they have a higher coefficient of efficiency determined by the existence of the mechanical part), making them competitive for comparison with the converter-based transmissions as well as other types of full-flow transmissions [9-12].

In addition to proper efficiency of a transmission, HSMT provides a wide range of torque conversion/velocity conversion [13] as well as optimum performance of a diesel motor [14], helping the motor operate close to the maximum efficiency point. This paper also analyzes different architectures of the capacity distribution, demonstrating the advantages of such transmissions for heavy vehicles. The above-mentioned factors favor achievement of the objectives of fuel consumption and improvement of

the performance of a vehicle. It is especially relevant in the process of a mine diesel locomotive design [15] operating underground, where exclusive requirements need to be fulfilled to maintain ecological standards of the environment in a mine [16].

## 2. STATE OF THE ART

HSMT is a hybrid transmission combining stepless velocity characteristics of a traditional hydraulic transmission [17] and high performance of a mechanical power transmission [18]. A significant distinction of two-flow stepless HSMTs from the stepped mechanical transmissions and other transmission types lies in the fact that losses in hydrostatic drive (HSD) as well as its efficiency depend heavily on the operation mode of the transmission affecting the amount of power flows. Power circulation within the two-flow transmissions helps distribute the power flows along the closed HSMT circuits during the process of mine locomotive braking since it places restrictions on the structure of such a transmission and on its kinematics.

Early research on modeling of hydrostatic transmissions with continuous variable power represents nonlinear simulators describing a pump with variable capacity and a fixed-volume motor taking into consideration leakage losses [19] as well as linear models for adaptive control concepts [20]. Subsequently [21], a nonlinear model of a transmission with loss models in the stationary state of the two machines (i.e. a pump with the variable capacity and a motor) has been developed to determine values of inlet transmission velocity and a vehicle velocity.

Paper [22] performed system modeling of hydromechanical transmissions and developed a functional dynamic model of a transmission with distribution of power flows for a tractor. The authors of [23] have designed and simulated a hydrostatic variator with power distribution for high-capacity vehicles. Identification of the operational problems in the process of transmission dynamics his one the most interesting findings of that study.

A hydraulic hybrid system of the Cumulo type for urban buses is an early example for road vehicles [24]. Especially for wheel loaders, the designers [25] represented a multimode hydrostatic mechanical transmission with power distribution for which fuel economy potential is no less than 15% higher than that of the available solution with a hydraulic converter. Coauthors [26] have obtained similar findings.

To analyze a hydrostatic drive for road vehicles, the scientists of [27] applied constant parameters for linear coefficients and quadratic coefficients in terms of hydraulic flow and pressure loss, respectively; the researchers of [28] developed an expression of the hydraulic unit flow as a function of input flow for a control servovalve and introduced a pressure-dependent coefficient to assess flow leakage. In the process of analysis of fuel consumption by urban buses, the behavior of hydraulic modules has been simulated while applying two loss coefficients for an ideal module.

The coefficients were expressed in terms of polynomial functions of specific velocity, pressure, load, and shift location [29].

Models of subsystems for each functional component of a hydrostatic transmission have been designed in [30] as a tool of a system modeling; moreover, they have been integrated into the unified system model using MATLAB/Simulink, which was confirmed experimentally while analyzing energy flow of a wheel loader *ibid*.

In addition, commercial software Amesim with the inbuilt transient analytical equations for hydraulic, pneumatic, thermal, electric, or mechanical systems is also popular to analyze transmissions as multiple-domain systems [29, 31].

Article [32] make a point that it is expedient to use stepless two-flow HSMTs for heavy wheel vehicles and track-type vehicles while emphasizing the advantages of the transmissions. The basic advantages include the following: wide-range stepless control of velocity as well as tractive effort; high compactness in terms of a small weight and overall dimensions, which is extremely important for mine diesel locomotives; complex braking by means of a hydrostatic drive at the expense of changes in the controlling parameters of hydraulic machines as well as the standard brake system, which improves substantially both the efficiency and the reliability of brakes of the diesel locomotive; quick and symmetrical reverse for certain HSMT designs; improved movement stability owing to continuous

power flow and smooth torque variation; and increased reliability of the motor operation as well as mechanics of the transmission through the damping properties of a working fluid of the hydrostatic drive. Moreover, HSMT enables better automation compared with the stepped mechanical transmissions, which translates into improved labor conditions of an operator [33, 34].

While analyzing stepless two-flow HSMTs, it should be noted that they have proper design characteristics and may be manufactured according to the following schemes: “input differential” (for instance, Vario); “output differential” (for instance, S-Matic); and with a variable structure (for instance, MALI WSG 500) [35].

Comparison of the characteristics of two-flow hydrostatic mechanical transmissions should specify their basic dissimilarity. “Input differential” HSMTs have no power circulation within a two-flow circuit when a tractor is moving forward; however, the circulation is available if backward motion takes place. In turn, in the context of “output differential” HSMTs, power circulation within a two-flow circuit takes place when the tractor is moving forward within the first half of a velocity range toward the reversal sign of the control parameter of HSMT. Under similar conditions, circulation takes place in terms of backward motion as well [15].

### 3. METHODS

Hydrostatic mechanical transmission operating according to the “input differential” scheme (Fig. 1, 2), being a typical example of HSMT, has been selected as the subject of the mathematical modeling braking process of a mine diesel locomotive.

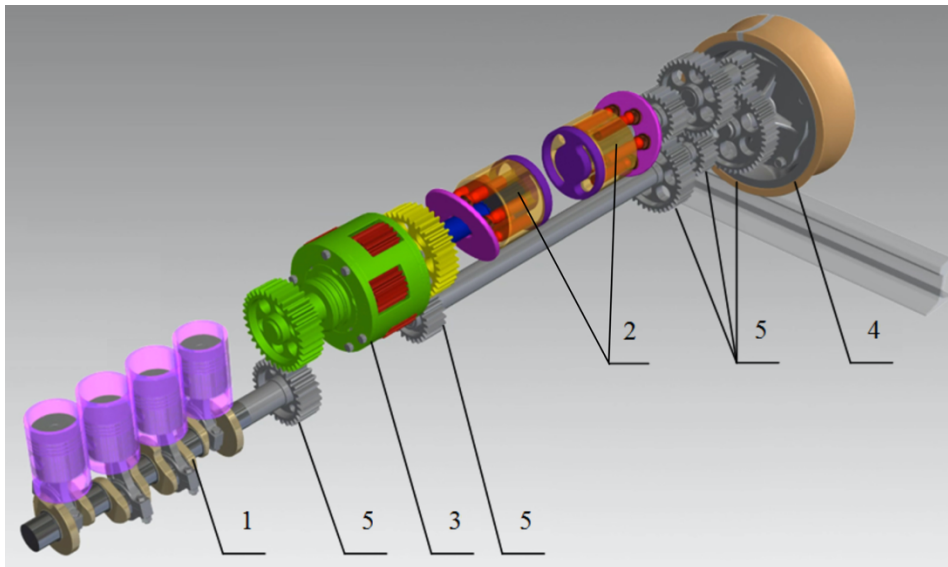


Fig. 1. 3-D model of HSMT #1: 1 – internal-combustion engine; 2 – hydrostatic drive; 3 – planetary reducing gear; 4 – wheel; 5 – reducing gears

This 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 [32].

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  $\omega_2$ .

The mathematical model describing changes in power parameters, kinematic parameters, and energy parameters of HSMT #1 is presented in the form of the following equations:

Thus, equation one explains the transmission–engine relation as follows:

$$\dot{\omega}_0 = \dot{\omega}_d; \quad (1)$$

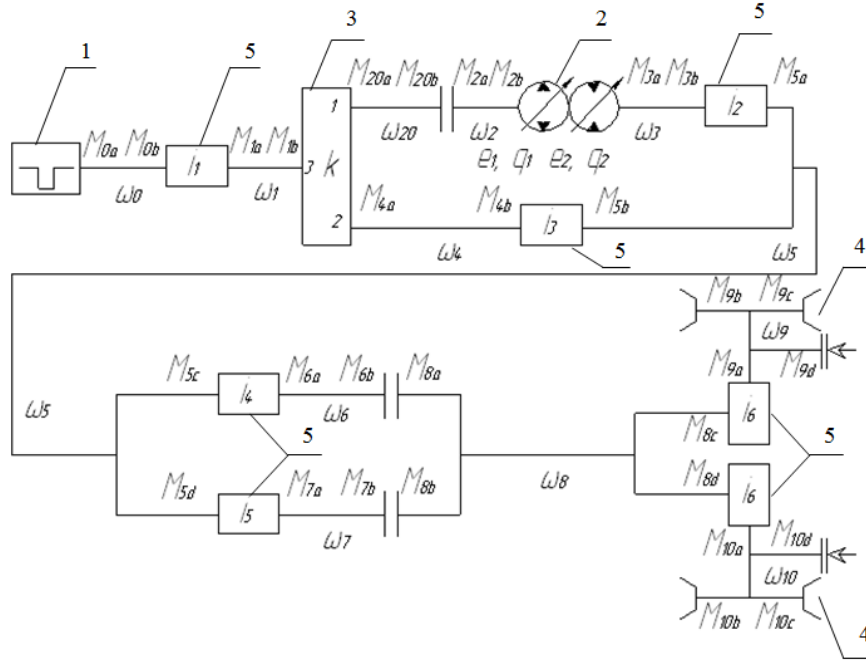


Fig. 2. Physical model of HSMT #1: 1 – 5 correspond to Fig. 1;  $\omega_i$  – angular velocity of transmission link;  $i_j$  – transmission ratio;  $k$  – internal transmission ratio of planetary gear set;  $M_{nm}$  – moments on HSMT links;  $n$  – indices corresponding to the number of link angular velocity;  $m$  – indexing letters corresponding to the moments of the ends of links;  $e_1, e_2$  – parameters of HSD hydraulic machines control; and  $q_1, q_2$  – maximum capacities of hydraulic machines

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

$$\begin{aligned} & e_1(t) \cdot q_1 \cdot \dot{\omega}_2 + q_1 \cdot \omega_2 \cdot \dot{e}_1(t) - e_2(t) \cdot q_2 \cdot \dot{\omega}_3 - q_2 \cdot \omega_3 \cdot \dot{e}_2(t) = \\ & \left( \frac{V_{0*}}{2 \cdot \pi \cdot E(g^*)} \cdot (|\omega_2| + |\omega_3|) \right) \cdot \Delta \dot{P} + \frac{V_{0*}}{2 \cdot \pi \cdot E(g^*)} \cdot \left( \frac{d}{dt} |\omega_2| + \frac{d}{dt} |\omega_3| \right) \cdot \Delta P + \\ & + \gamma \cdot \left( \frac{K_{1y}}{\mu} \cdot (1 + C_{1y} \cdot |\omega_2|) + \frac{K_{2y}}{\mu} \cdot (1 + C_{2y} \cdot |\omega_3|) \right) \cdot \Delta \dot{P} + \\ & + \gamma \cdot \left( \frac{K_{1y}}{\mu} \cdot C_{1y} \cdot \frac{d}{dt} |\omega_2| + 2 \cdot \frac{K_{2y}}{\mu} \cdot C_{2y} \cdot \frac{d}{dt} |\omega_3| \right) \cdot \Delta P; \end{aligned} \quad (2)$$

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

$$\begin{aligned}
M_{2b} - e_1 \cdot q_1 \cdot \Delta P &= -\Delta M_1 \cdot \text{sign}(\omega_2); \\
\Delta M_1 &= q_1 \cdot \left[ \bar{K}_1 \cdot |\omega_2| \cdot (1 + \bar{K}_2 \cdot \bar{e}_1^2) + \frac{\bar{K}_5 \cdot (1 + \bar{K}_4 \cdot |\bar{e}_1|)}{(1 + \bar{K}_3 \cdot |\omega_2| \cdot D_{q1})} \cdot \Delta P + \frac{\bar{K}_8 \cdot (1 + \bar{K}_7 \cdot |\bar{e}_1|)}{(1 + \bar{K}_6 \cdot |\omega_2| \cdot D_{q1})} \right]; \\
D_{q1} &= \sqrt[3]{2 \cdot \pi \cdot q_1}; \quad M_{3a} + e_2 \cdot q_2 \cdot \Delta P = -\Delta M_2 \cdot \text{sign}(\omega_3); \\
\Delta M_2 &= q_2 \cdot \left[ \bar{K}_1 \cdot |\omega_3| \cdot (1 + \bar{K}_2 \cdot \bar{e}_2^2) + \frac{\bar{K}_5 \cdot (1 + \bar{K}_4 \cdot |\bar{e}_2|)}{(1 + \bar{K}_3 \cdot |\omega_3| \cdot D_{q2})} \cdot \Delta P + \frac{\bar{K}_8 \cdot (1 + \bar{K}_7 \cdot |\bar{e}_2|)}{(1 + \bar{K}_6 \cdot |\omega_3| \cdot D_{q2})} \right]; \\
D_{q2} &= \sqrt[3]{2 \cdot \pi \cdot q_2};
\end{aligned} \tag{3}$$

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

$$\begin{aligned}
\dot{\omega}_0 \cdot i_1 - \dot{\omega}_1 &= 0; \quad \dot{\omega}_{20} - k \cdot \dot{\omega}_4 + (k-1) \cdot \dot{\omega}_1 = 0; \quad \dot{\omega}_{20} - \dot{\omega}_2 = 0, \quad \Psi = 1; \\
\dot{\omega}_3 \cdot i_2 - \dot{\omega}_5 &= 0; \quad \dot{\omega}_4 \cdot i_3 - \dot{\omega}_5 = 0; \quad \dot{\omega}_5 \cdot i_4 - \dot{\omega}_6 = 0; \quad \dot{\omega}_5 \cdot i_5 - \dot{\omega}_7 = 0; \\
\dot{\omega}_6 - \dot{\omega}_8 &= 0, \quad Y = 1; \quad \dot{\omega}_7 - \dot{\omega}_8 = 0, \quad Y = 0; \\
\dot{\omega}_8 \cdot i_6 - \omega_9 &= 0; \quad \dot{\omega}_8 \cdot i_6 - \omega_{10} = 0;
\end{aligned} \tag{4}$$

Power transmission parameters can be described using the following equations:

$$\begin{aligned}
M_{0b} \cdot \eta_1^{\ominus \cdot \text{sign}(N_{0b})} + i_1 \cdot M_{1a} &= 0; \\
M_{20a} \cdot \eta_{13}^{\ominus \cdot \text{sign}(N_{20a})} + M_{4a} \cdot \eta_{23}^{\ominus \cdot \text{sign}(N_{4a})} + M_{1b} &= 0; \\
M_{20a} \cdot k \cdot \eta_{13}^{\ominus \cdot \text{sign}(N_{20a})} + M_{4a} \cdot \eta_{23}^{\ominus \cdot \text{sign}(N_{4a})} &= 0; \quad M_{20b} + M_{2a} = 0, \quad \Psi = 1; \\
M_{20b} = M_{2a} = 0, \quad \Psi = 0; \quad M_{3b} \cdot \eta_2^{\ominus \cdot \text{sign}(N_{3b})} + i_2 \cdot M_{5a} &= 0; \\
M_{4b} \cdot \eta_3^{\ominus \cdot \text{sign}(N_{4b})} + i_3 \cdot M_{5b} &= 0; \\
M_{5c} \cdot \eta_4^{\ominus \cdot \text{sign}(N_{5c})} + i_4 \cdot M_{6a} &= 0; \\
M_{6b} + M_{8a} = 0, \quad Y = 1; \quad M_{7b} = M_{8b} = 0, \quad Y = 1; \\
M_{5d} \cdot \eta_5^{\ominus \cdot \text{sign}(N_{5d})} + i_5 \cdot M_{7a} &= 0; \\
M_{7b} + M_{8b} = 0, \quad Y = 0; \quad M_{6b} = M_{8a} = 0, \quad Y = 0; \\
M_{8c} \cdot \eta_6^{\ominus \cdot \text{sign}(N_{8c})} + i_6 \cdot M_{9a} = 0; \quad M_{8d} \cdot \eta_6^{\ominus \cdot \text{sign}(N_{8d})} + i_6 \cdot M_{10a} &= 0; \\
M_{9b} = M_{9c} = M_{V1}; \quad M_{10b} = M_{10c} = M_{V2}; \quad M_{0a} + M_{0b} = 0; \quad M_{1a} + M_{1b} = \\
M_{2a} + M_{2b} = 0; \quad M_{20a} + M_{20b} = 0; \\
M_{3a} + M_{3b} = 0; \quad M_{4a} + M_{4b} = 0; \\
M_{5a} + M_{5b} + M_{5c} + M_{5d} = 0; \quad M_{6a} + M_{6b} = 0; \quad M_{7a} + M_{7b} = 0; \\
M_{8a} + M_{8b} + M_{8c} + M_{8d} = 0; \quad M_{9a} + M_{9b} + M_{9c} + M_{9d} \cdot \Omega = 0; \\
M_{10a} + M_{10b} + M_{10c} + M_{10d} \cdot \Omega &= 0;
\end{aligned} \tag{5}$$

where  $\dot{\omega}_i$  is the angular deceleration of a link;  $\Psi$  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 in terms of 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 compressible liquid;  $E(g^*)$

is the modulus of elasticity of the operating liquid depending on  $g^*$  gas content per cent;  $\Delta P$  is the operating pressure differential within HSD;  $K_{iy}, C_{iy}$  are loss coefficients for a hydraulic pump ( $i = 1$ ) and a hydraulic engine ( $i = 2$ );  $\mu$  is the coefficient of dynamic viscosity;  $Y$  is the coefficient characterizing movement ( $Y = 1$  is traction movement,  $Y = 0$  is transport movement);  $\eta_j$  is the reducing gear efficiency;  $\eta_{13}, \eta_{23}$  are efficiencies within gear-tooth systems of sun-satellite gear and epicycle-satellite gear in terms of stopped carrier determining moment losses;  $\Theta$  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 HSMT links;  $\Delta M_1, \Delta M_2$  are losses of moments within hydraulic machines;  $\bar{K}_1, \bar{K}_2, \dots, \bar{K}_8$  are coefficients of hydromechanical losses;  $D_{qi}$  is the typical dimension of a hydraulic machine;  $M_{Vaxis}$  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);  $\Omega$  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).

#### 4. RESULTS AND DISCUSSION

Distribution of kinematic parameters, power parameters, and energy parameters of HSMT in the braking process depends heavily on the following:

- Transmission type;
- Initial velocity (diesel locomotive braking process starts from the velocities of  $V_{max}$  and  $0,5 \cdot V_{max}$ , in terms of HSMT #1,  $V_{max}$  within transport range is 20,24 km/h; within traction range, it is 10.02 km/h);
- Drawbar force (as a rule, in terms of movement within transport range, mine cars are empty and their maximum number is  $n$  for the selected operating conditions that is equivalent to 2 loaded mine cars (it is assumed to be ample); in terms of traction range, two loading variants are considered: 8 or 2 loaded mine cars);
- Ascending and descending angles (being equivalent to 50 ‰); and
- Method of braking process.

The braking process involves the following implementation methods:

1. In terms of kinematic disconnection of engine from wheels (further, the implementation method will be marked as #1).
2. In terms of preserving engine–wheels kinematic connection:
  - decelerating at the expense of HSD and the braking system while preserving engine–wheels kinematic connection (implementation method #2);
  - decelerating at the expense of HSD while preserving engine–wheels kinematic connection (implementation method #3); braking mechanisms form a moment equal to 0; and
  - decelerating at the expense of the braking system while preserving engine–wheels kinematic connection (implementation method #4).

Consideration of method #2 of implementing the braking process is the focus of this work.

As the braking process preserving engine–wheels kinematic connection is usually service one (in [6], it was proved that it should only be in the presence of HSMT), 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 of  $e_1$  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. Such change times as

$e_1(t)$  and  $e_2(t)$  have been selected since implementation of a breakage process 3 cannot provide deceleration similar to method 1. Thus, the braking time is specified with a margin that should be sufficient to stop the diesel locomotive with the help of any of the analyzed methods. Selection of the unified time of the breakage process will help determine easily one or another factor affecting the kinematic parameters, power parameters, and energy parameters of the HSMT.

In order to evaluate the effect of the initial velocity of the braking and drawbar force of a mine diesel locomotive upon the distribution of kinematic parameters of HSMT as well as detect and systematize basic regularities of power stream distribution within the closed contour of HSMT #1 during braking process of a mine diesel locomotive, software support developed in MatLab/Simulink was applied.

Angular velocities of transmission links and working pressure differential in HSD for HSMT #1 had the following initial values:

– within transport range:

- $V = 20,24$  km/h,  $\omega_1 = 210,0$  rad/s,  $\omega_2 = -61,2$  rad/s,  $\omega_3 = -102,9$  rad/s,  $\Delta P = 11,85$  MPa,  $e_1 = 1$ ,  $e_2 = 0,5$ ;
- $V = 10,12$  km/h,  $\omega_1 = 210,0$  rad/s,  $\omega_2 = -200,5$  rad/s,  $\omega_3 = -51,36$  rad/s,  $\Delta P = 19,6$  MPa,  $e_1 = 0,36$ ,  $e_2 = 1$ ;

– within traction range:

- $V = 10,02$  km/h,  $\omega_1 = 210,0$  rad/s,  $\omega_2 = -53,3$  rad/s,  $\omega_3 = -105,8$  rad/s,  $\Delta P = 13,55$  MPa,  $e_1 = 1$ ,  $e_2 = 0,4$ ;
- $V = 5,01$  km/h,  $\omega_1 = 210,0$  rad/s,  $\omega_2 = -195,9$  rad/s,  $\omega_3 = -53,05$  rad/s,  $\Delta P = 21,0$  MPa,  $e_1 = 0,384$ ,  $e_2 = 1$ .

The following parameters are studied in the process of braking modeling:

- Maximum value of the working pressure differential in HSD  $|\Delta P|_{\max}$ , which should not be more than the 40.0 MPa for hydraulic machines with a working volume of 90 cm<sup>3</sup>;
- Maximum value of angular velocity of a hydraulic pump shaft  $|\omega_2|_{\max}$  should not be more than 460.0 rad/s;
- Maximum value of angular velocity of a hydraulic engine shaft  $|\omega_3|_{\max}$  should not be more than 460.0 rad/s;
- Maximum value of output power by a hydraulic branch of the closed circuit of HSMT power output should be  $|N_{gk}|_{\max}$ ;
- Maximum value of output power by a mechanical branch of closed circuit of HSMT power output should be  $|N_{mk}|_{\max}$  (Preliminary identification of the power values is impossible since due to the power circulation, more potential energy may be transferred through a hydraulic branch or a mechanical one, being even higher than the motor power. Theoretically, the total of the two powers should not exceed the motor output (ignoring its coefficient of efficiency));
- Braking path is  $S$ ; and
- Braking period is  $t$ .

Despite the fact that the value of the working pressure differential within HSD  $\Delta P$  is set in the mathematical model of braking process as the initial condition, it is clarified according to the operating conditions as well as a method of braking process implementation. The mathematical model, used by this research, relies on the actual differential pressure search. The initial value at the beginning of the modeling stage is specified as being equal to 0. This is why  $\Delta P$  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.

The results of integrated research on the braking modeling of a diesel locomotive at the expense of a hydrostatic drive (HSD) and braking system in terms of preserving the engine–wheels kinematic connection is presented in tab. 1.

The results of braking modeling of a diesel locomotive with HSMT #1 while moving within the transport and traction ranges in the context of a method of braking process implementation at the expense of hydrostatic drive and braking system are represented in the form of graphical curves in Fig. 3 and Fig. 4. The graphs demonstrate variations of parameters of interest,  $e_1, e_2, V, S, \Delta P, \omega_2, \omega_3$ , in the breakage process up to the complete stopping of the diesel locomotive. The braking time is plotted on the X-axis; the change in parameters of  $e_1, e_2, V, S, \Delta P, \omega_2, \omega_3$  is plotted on the Y-axis.

## 5. CONCLUSIONS

This paper considers the braking process of a mine diesel locomotive with HSMT operating according to the “input differential” scheme. Braking process modeling involves four implementation methods. Identification and systematization of basic regularities in the distribution of power flows within the closed transmission contour in the process of braking have been implemented with the help of software support developed by means of MatLab/Simulink.

Table 1

Results of braking modeling of a diesel locomotive  
at the expense of a hydrostatic drive and braking system

$V$ max, km/h	$n$	Descending angle, +50 ‰ ( $\alpha$ =2,862)/ascending angle, -50 ‰ ( $\alpha$ =-2,862)	$ \Delta P $ max, MPa	$ \omega_2 $ max, rad/s	$ \omega_3 $ max, rad/s	$ N_{gk} $ max, кVt	$ N_{mk} $ max, кVt	Braking path $S$ , m	Braking period $t$ , s	Note
Transport range										
20,24	2	50 ‰	41,8	63,3	102,9	17,5	80,4	62,22	16,16	*
20,24	2	-50 ‰	41,8	62,5	102,9	17,2	81,0	57,57	15,34	*
10,12	2	50 ‰	110,4	200,9	51,36	32,8	33,5	29,75	15,51	*
10,12	2	-50 ‰	110,9	200,6	51,5	33,1	34,3	26,89	14,7	*
Traction range										
10,02	2	50 ‰	36,5	57	105,8	13,2	70,2	33,49	17,59	
10,02	2	-50 ‰	36,2	56,2	105,8	13,2	72,9	29,87	16,75	
10,02	8	50 ‰	85,8	224,1	250,8	29,6	80,0	59,96	35,81	*
10,02	8	-50 ‰	16,1	56,2	105,8	10,36	64,8	9,84	7,04	
5,01	2	50 ‰	81,7	196,3	53,1	24,6	26,5	18,18	19,15	*
5,01	2	-50 ‰	64,9	196,1	53,1	25,0	27,0	9,44	13,46	*
5,01	8	50 ‰	103,0	202,6	53,7	25,0	26,5	24,76	25,35	*
5,01	8	-50 ‰	16,1	196,1	53,1	10,8	13,4	2,46	3,51	

\* – version is not workable, as the above-mentioned factors depend on the fact that the differential pressure exceeds the allowable pressure of 40 MPa.

The process of theoretical studies of the braking process of a diesel locomotive with HSMT #1 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) has helped determine that the mechanical branch of the closed contour of HSMT is more loaded in the braking process. Power



stream transmitted through the mechanical branch of closed contour is more (up to 6.3 times) compared with the hydraulic one. This is an advantage since the HSD unloads during braking.

The decrease in the drawbar power of a diesel locomotive from 8 mine cars to 2 in the process of braking on the ascent results in the decrease of the working pressure differential in HSD down to 57.5%; angular velocity of the hydraulic pump shaft down to 74.5%; angular velocity of hydraulic engine shaft down to 57.8%; power output from the hydraulic branch of the closed contour down to 55.4%; power output from the mechanical branch of the closed contour of HSMT down to 12.3%; and braking path shortening down to 44.2%. The braking process on the ascent results in the increase of the working pressure differential in HSD up to 4.03 times; power output from the hydraulic branch of the closed contour up to 2.3 times; power output from the mechanical branch of the closed contour of HSMT up to 2.02 times; and braking path up to 3.84 times. The angular velocities of the hydraulic pump shaft and hydraulic engine shaft remain unchanged.

Braking from  $V_{\max}$  velocity instead of  $0,5 \cdot V_{\max}$  velocity demonstrates a clear decrease in the values of the working pressure differential in HSD down to 62.3% and an increase of the angular velocity of the hydraulic motor shaft up to 4.67 times; power output from the mechanical branch of the closed contour of HSMT increased up to 4.83 times. The results of complex research on the braking modeling of a diesel locomotive with HSMT operating according to the “input differential” scheme helped determine that, in terms of deceleration at the expense of hydrostatic drive and braking system while preserving the kinematic engine–wheels connection, it is not permitted to implement this method of braking process as it is followed by excess of the allowable value of working pressure differential within HSD up to 2.8 times. It is expedient to analyze the scheme separately during the motor braking, and separately while using brakes or to pay more attention to the “output differential” schemes. However, this is beyond the scope of this paper and will be a part of future research.

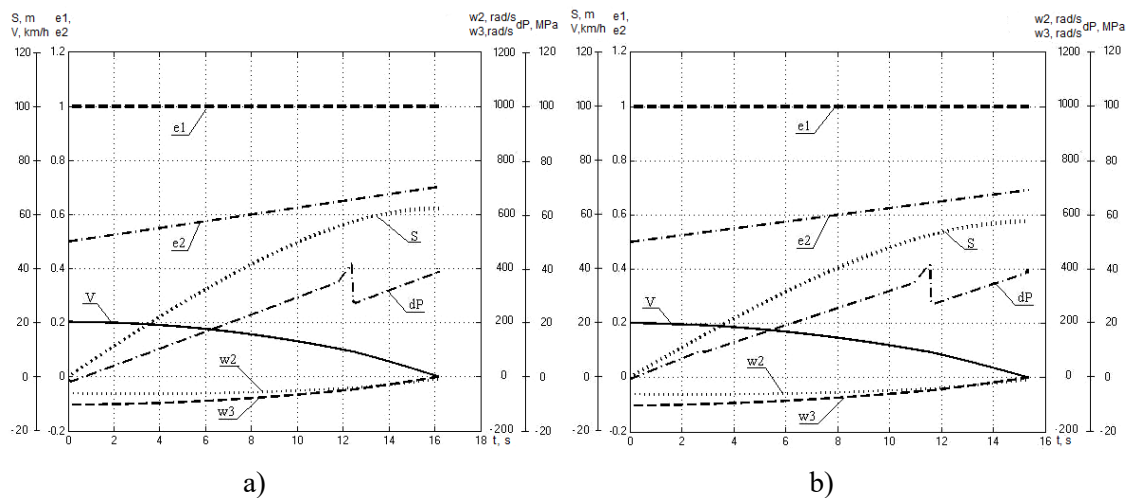


Fig. 3. Results of braking modeling of a diesel locomotive with HSMT #1 while moving within the transport range in the context of the method of braking process implementation at the expense of a hydrostatic drive and braking system: a) descending angle and b) ascending angle

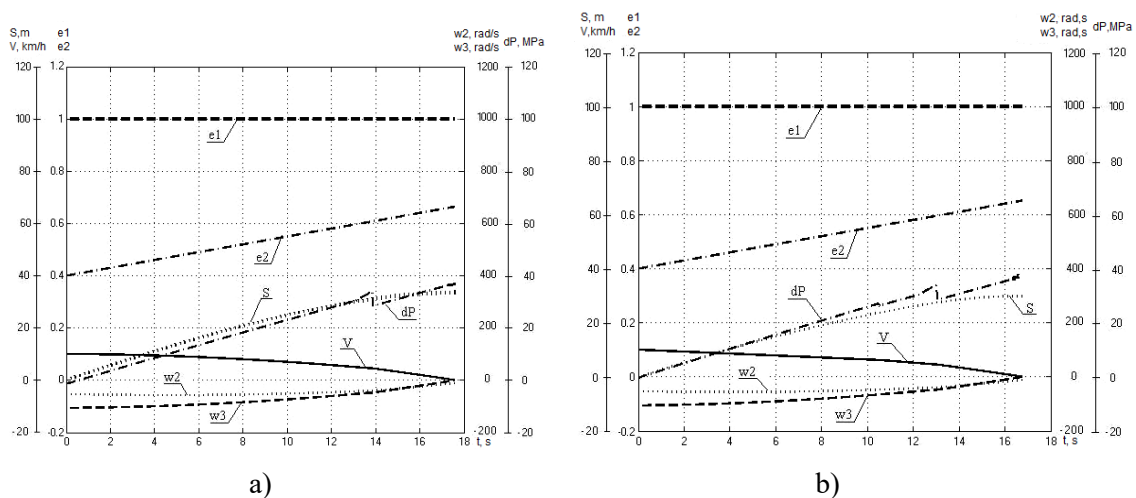


Fig. 4. Results of braking modeling of a diesel locomotive with HSMT #1 while moving within the traction range in the context of method of braking process implementation at the expense of a hydrostatic drive and braking system: a) descending angle and b) ascending angle

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