Modeling Curvilinear Motion of Tracked Vehicle with the Dual-Flux Electromechanical Turning Mechanism

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ABSTRACT. Tens of thousands of old tracked vehicles with no ergonomic and ineffective the turning mechanisms are exploited in the world. Light multipurpose tracked transporter-tractor MTLB – is one of these machines, which is equipped with mechanical dual-flux turning mechanisms. These turning mechanisms are controlled by levers instead of a steering wheel and in the turning the part of the kinetic energy is converted to heat by friction mechanisms. This article analyzes the problems of the installation of electric drive in the mechanical dual-flux turning mechanisms of tracked vehicles. Previously, by the authors of this article were published materials, which by the analytical methods investigated the possibility of using the electric drive in the dual-flux turning mechanisms on the example of the light multipurpose tracked transporter-tractor MTLB. To solve this problem, the authors proposed an algorithm and a program that allow by numerical modeling techniques make the parametric optimization of the turning mechanisms and use them to assess their effectiveness. In contrast to the analytical determination of the trajectory of movement of the machine numerical simulation make possible to calculate trajectory considering real-slipping and skidding of caterpillar tracks. In the next publication is planned to bring the results of the comparative modeling of the entry process in the turning of light multipurpose tracked transporter-tractor MTLB with the standard mechanical and two selected scheme of the electromechanical dual-flux turning mechanisms.

Introduction. Previously, by the authors of this article were published materials, which investigated the possibility of using the electric drive in a dual-flux turning mechanisms on the example of the light multipurpose tracked transporter-tractor MTLB.

On the basis of the research it was concluded that, in carrying out the shallow modernization of the tractor transmission the installation and use of electric motors in the traction mode in mechanisms of turning at this stage of Ukrainian electric drive development is impractical. Calculations have shown that to maintain the turning speed indicators on heavy soils without mechanical branch in the turning mechanisms must be installed electric motors with capacity of at least 40 kW for each side. Installation of these motors requires installation of a much more powerful generator and more capacious batteries. As a result, the desired increase in mobility and controllability machines without a substantial increase in engine power, fuel consumption, weighting machine and its rise in price will not be possible.

Thus, it was decided to leave for further consideration two options of the use of electric drive in the dual-flux turning mechanisms:

– with one motor and the sun gears of the summation planetary rows different sides, rotating in opposite directions (Fig. 1);

– with two motors while maintaining the mechanical branch in the turning mechanisms (Fig. 2).

Mathematical models and algorithms to simulate the behavior of the tracked vehicles have been designed with all three of the above options turning mechanisms for a detailed study of controllability of caterpillar multipurpose transporter-tractor with the proposed turning mechanisms and its comparison with the controllability of basic machine (Fig. 3).

The initial data for modeling. For modeling were used geometric, kinematic, power, weight and inertia characteristics caterpillar multipurpose transporter-tractor MT-LB [5]. Below in Tables 1 and 2 shows the numerical values of the machine parameters and of soils, that were used for all three embodiments schemes of the dual-flux turning mechanisms.

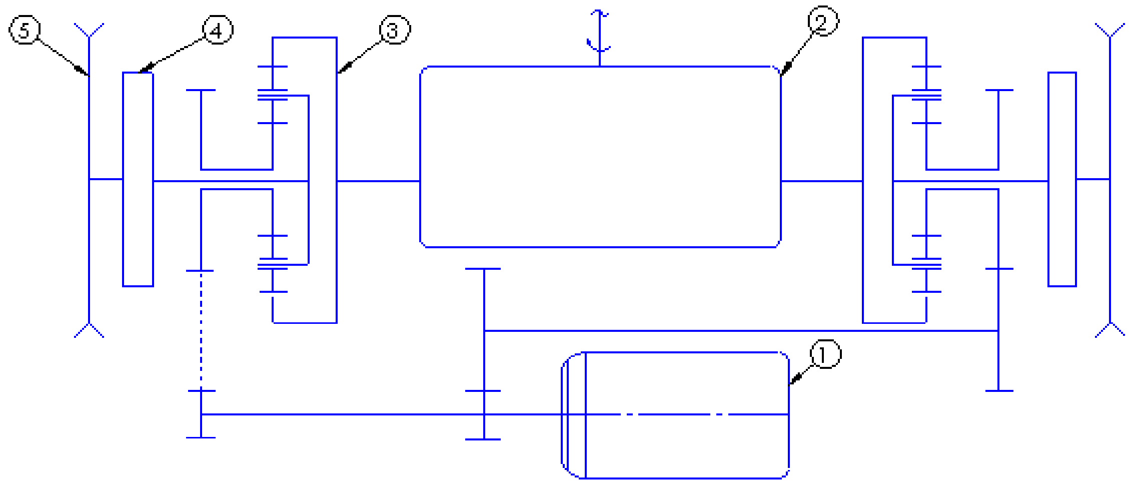

Fig. 1. Kinematic scheme of transmission with one electric motor and the sun gears of the summation planetary rows of different sides, rotating in opposite directions: 1 – Electric drive; 2 – Gearbox; 3 – Summation planetary rows; 4 – Final drives; 5 – Driving wheels

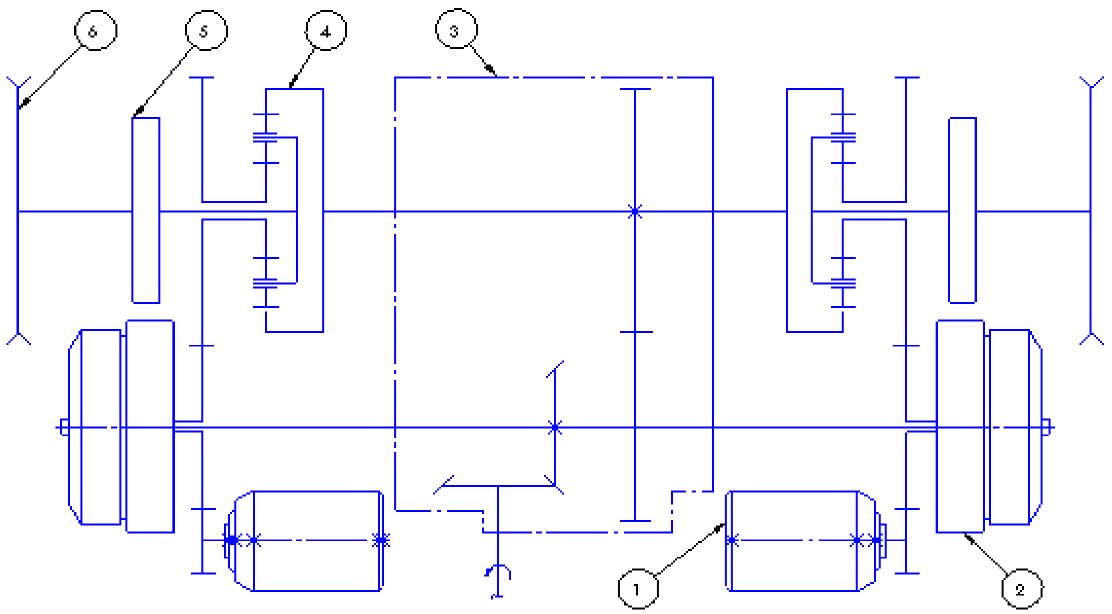

Fig. 2. Kinematic scheme of transmission with two electric motors while maintaining the mechanical branch in the turning mechanisms: 1 – Electric drives; 2 – Native turning mechanisms; 3 – Gearbox; 4 – Summation planetary rows; 5 – Final drives; 6 – Driving wheels

The overall structure of the mathematical model. Mathematical model for the study enter in the turning and curved movement of the light multipurpose tracked transporter-tractor MTLB is based on the integration of the differential equations describing the motion of the perturbed system containing 6 generalized velocities:

– the longitudinal velocity of the machine taking into account skidding (slipping) – v_X ;

– the transverse velocity of the side slip of machine – v_Y ;

– the angular velocity of rotation of the machine frame about a vertical axis passing through the center of mass – ω_Z ;

– the angular velocity of the engine crankshaft – ω_E ;

– the angular velocity of rotation of the input link of the lagging mechanism of turning – ω_1 ;

– the angular velocity of rotation of the input link of the more fast mechanism of turning – ω_2 .

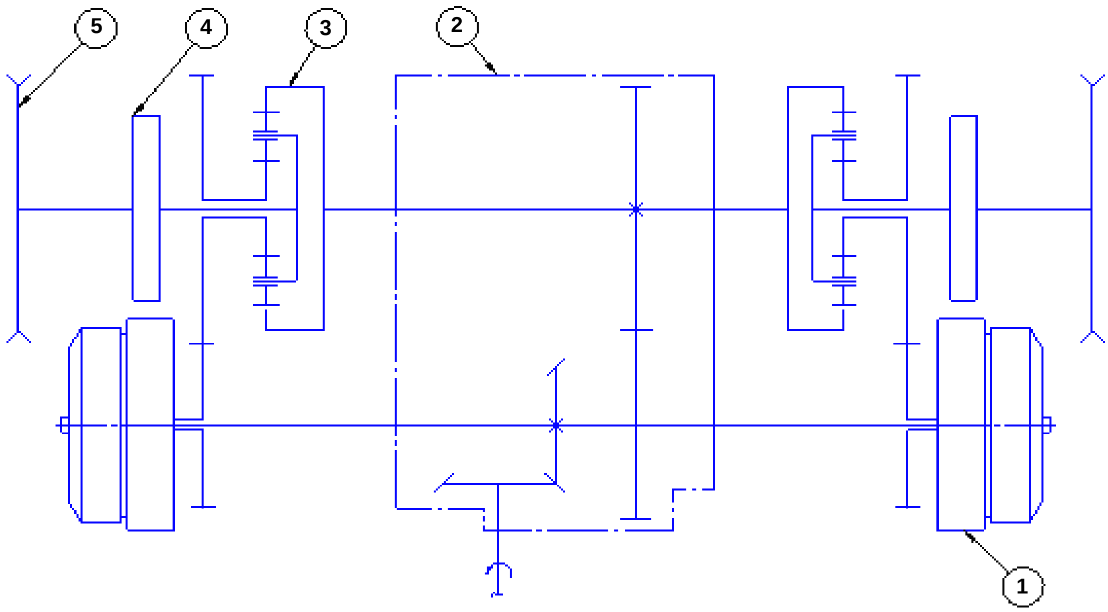

Fig. 3. Kinematic scheme of native transmission of the light multipurpose tracked transporter-tractor MTLB: 1 – Turning mechanisms; 2 – Gearbox; 3 – Summation planetary rows; 4 – Final drives; 5 – Driving wheels

Table 1. Machine parameters

№	Parameter	Value
1.	Machine weight, t	12,5
2.	Engine's type	YaMZ-238V
3.	Engine power, kW (hp)	176 (240)
4.	Engine speed at maximum power, rev/min	2100
5.	Maximum torque, Nm (kgm)	883 (90)
6.	Engine speed at maximum torque mode, rev/min	1250 – 1450
7.	The gear ratio of the input bevel gear	0,905

8.	Gearbox ratio: I II III IV V VI	∞ 3,125 1,500 0,833 0,585 0,435
9.	The internal gear ratio of the summation planetary rows	– 2,41
10.	Gear ratio of final drive	6
11.	The radius of the drive wheel, m	0,265
12.	The width of machines on the track centers, m	2,5
13.	The support caterpillar length, m	3,7
14.	The height of the center of mass, m	1,1
15.	Moment of inertia of the engine with flywheel, kgm^2	4,61
16.	The moment of inertia of the machine in rotation about a vertical axis passing through the center of mass, kgm^2	50000

Table 2. Soils parameters

№	Soil type (road)	f	φ	μ_{max}
1.	Turfy dry clay soil (moisture content <8%)	0,08	0.9	0,9
2.	Dry dirt road on the loam	0,07	0,8	0,8
3.	Plowing on the loam	0,1	0,7	0,7
4.	Wet road on the loam	0,125	0,6	0,35
5.	Snow loose	0,25	0,3	0,3

For each of the generalized velocity compiled its differential equations, which are grouped in:

$$\begin{cases} \dfrac{dv_X}{dt} = \dfrac{P_{D1} + P_{D2} - P_{F1} - P_{F2}}{G_M}; \\ \dfrac{dv_Y}{dt} = \dfrac{P_Y - P_{\varphi 1} - P_{\varphi 2}}{G_M}; \\ \dfrac{d\omega_Z}{dt} = \dfrac{0{,}5B\left(P_{D2} - P_{D1}\right) - M_{RR}}{I_Z}; \\ \dfrac{d\omega_E}{dt} = \dfrac{M_E - M_R}{I_E + I_{TR}}; \\ \dfrac{d\omega_1}{dt} = \dfrac{M_{MR1} - M_{R1}}{I_{MR}}; \\ \dfrac{d\omega_2}{dt} = \dfrac{M_{MR2} - M_{R2}}{I_{MR}}. \end{cases} \quad (1)$$

Here variables are:

P_{D1} and P_{D2} – traction force (braking) on the lagging and more fast sides taking into account the sign, calculated from the coefficients of slipping or skidding;

P_{F1} and P_{F2} – forces of resistance to movement on the lagging and more fast sides based on the redistribution of the normal reactions between the sides in the turning due to centrifugal force P_Y;

P_Y – the centrifugal force generated by the curvilinear motion of the machine;

$P_{\varphi 1}$ and $P_{\varphi 2}$ – cohesion forces caterpillar mover with the ground in the transverse direction (value $P_Y - P_{\varphi 1} - P_{\varphi 2}$ can be positive or zero – the friction force can not be the driving);

M_{RR} – the moment of resistance turning, which depends on the geometrical characteristics of the machine, its weight, the radius of turning and soil characteristics (road);

M_E – engine torque;

M_R – moment of movement resistance reduced to the engine shaft (which depends on P_{F1} and P_{F2});

M_{MR1} and M_{MR2} – torques on the input links of the turning mechanisms of lagging and more fast sides. They depend on the torque in the friction clutches or in the friction brakes for the base transmission and they depend on the torque on the motors of the turning mechanisms if they have electric drive.

M_{R1} and M_{R2} – torques of movement resistance reduced to the input links of the turning mechanisms of lagging and more fast sides, which depends on the P_{F1} и P_{F2}.

Constant values (constants):

G_M – weight (gravity) of the machine;

B – width of machines on the track centers;

I_Z – moment of inertia of the machine in rotation about a vertical axis passing through the center of mass;

I_E – moment of inertia of the engine with flywheel;

I_{TR} – moment of inertia of the rotating parts of the transmission and chassis, reduced to the crankshaft of the engine;

I_{MR} – moment of inertia of the turning mechanism, reduced to its input link.

Assumptions and simplifications. The movement of the machine is considered on level ground.

The coordinate system associated with the machine has the beginning in the center of mass of the machine and the axis: ***OX*** – forward, ***OY*** – to the left, ***OZ*** – up.

Entrance to the turning and movement in turning is considered in a selected gear ratio without switching in the gearbox.

The engine is considered working on the external characteristic with the full fuel feed.

Transmission work is only considered at a high speed range of the presence of two power flows through the transmission and the turning mechanism.

Malleable shafts and other drivetrain components are not taken into account.

When the command to transfer frictional controls to the ON state in the base transmission is assumed that the friction torque increases linearly from zero to a maximum value of 0.5 s.

The turning is always seen in the same direction (to the left) and, accordingly, is not considered right and left side, and more fast and a lagging side.

The model of diesel engine YaMZ-238V. Since the operation of the engine is considered only at full fuel feed, the engine simulation is made possible by the Leiderman's formula:

$$N_e = N_{e\max}\left[a\frac{\omega}{\omega_N} + b\left(\frac{\omega}{\omega_N}\right)^2 - c\left(\frac{\omega}{\omega_N}\right)^3\right], \qquad (2)$$

where $N_{e\max}$ =176 kWt – the maximum effective power of the engine when the crankshaft rotational speed $\omega_N = \frac{\pi \cdot 2100}{30} = 219{,}9\,\text{s}^{-1}$;

$a = 0{,}49;\ b = 1{,}7;\ c = 1{,}19$ – coefficients for the Leiderman's formula YAMZ-238V;

ω – crankshaft rotational speed. In the area of regulatory power characteristics decreases linearly from $N_{e\max}$ to 0 for $\omega = 240\,\text{s}^{-1}$.

Accordingly, it is possible to write the formula for the transition to the free torque:

$$M_E = \frac{N_e - a_\Delta\left(\frac{\omega}{\omega_N}\right)^3 N_{e\max}}{\omega},$$

where $a_\Delta = 0{,}14$ – power loss factor for fan cooling system.

To simulate the behavior of the engine at varying load **module "engine"** has been developed, input parameter for which each step of the integration is the current value of the rotation speed of the engine crankshaft ω_E.

The output parameter is the actual value of the free torque M_E, which is supplied from the engine into the transmission.

Model of curvilinear motion of a tracked vehicle on the terrain. To model the curvilinear motion of a tracked vehicle on the terrain were used depending listed in [6].

For this purpose, the module "Rotate" has been developed, input parameters for which on each step of the integration are the current values of the following generalized velocities:

– the longitudinal velocity of the machine taking into account skidding (slipping) – v_X;

– the transverse velocity of the side slip of machine – v_Y;

– the angular velocity of rotation of the machine frame about a vertical axis passing through the center of mass – ω_Z;

– the angular velocity of the engine crankshaft – ω_E;

– the angular velocity of rotation of the input link of the lagging mechanism of turning – ω_1;

– the angular velocity of rotation of the input link of the more fast mechanism of turning – ω_2,

as well as the gear ratio in the gearbox and characteristics road (soil) in which make machine movement.

Given the linear v_X and angular ω_Z velocities of the body of the machine calculates the actual speed of longitudinal movement of the chassis for lagging and for more fast sides relative to the support surface:

$$V_1 = v_X - \omega_Z \frac{B}{2} \text{ and } V_2 = v_X + \omega_Z \frac{B}{2}.$$

Further, for $\omega_Z \neq 0$ determined a central turning radius and turning radius on the sides:

$$R_C = \frac{v_X}{\omega_Z},\ R_1 = \frac{v_X}{\omega_Z} - \frac{B}{2} \text{ and } R_2 = \frac{v_X}{\omega_Z} + \frac{B}{2}.$$

Knowing the turning radius of the machine, we can determine the actual drag coefficient of the turning μ on a given road (terrain) and the centrifugal force P_Y generated by turning:

$$\mu = \frac{\mu_{\max}}{0{,}85 + 0{,}15 \cdot R_2 / B},$$

where $\mu_{\max}$ – maximum drag coefficient of the turning corresponding to turn around stopped caterpillars.

$$P_Y = \frac{G_M v_X^2}{R_C}.$$

Knowing the centrifugal force P_Y and the actual drag coefficient of the turning μ we determine the absolute χ and relative χ_0 bias of the pole of turning:

$$\chi = \frac{P_Y}{\mu G_M} \text{ and } \chi_0 = \frac{2\chi}{L}.$$

If we get $\chi_0 > 1$, then we equate it to $\chi_0 = 1$ and recalculate the actual drag coefficient of the turning μ:

$$\mu = \frac{P_Y}{G_M}.$$

When finally determined the actual drag coefficient of the turning, we find the torque of resistance to turning:

$$M_{RR} = 0{,}25\mu G_M L\left(1 + \chi_0^2\right).$$

In view of the rotational speeds of the engine crankshaft ω_E and of the input links of the mechanism of turning ω_1 and ω_2, and the gear ratio of transmission we determine the drive wheels speed for lagging and for more fast sides:

$$\omega_{DW1} = \frac{\left(\omega_1 / i_{MR} + (\omega_E k_S)/(i_{GB} i_{IG})\right)/\left(k_S + 1\right)}{i_{FD}};$$

$$\omega_{DW2} = \frac{\left(\omega_2 / i_{MR} + (\omega_E k_S)/(i_{GB} i_{IG})\right)/\left(k_S + 1\right)}{i_{FD}}.$$

Here:

k_S – internal gear ratio of the summation planetary rows;

i_{GB} – the gear ratio of the gearbox at the selected level;

i_{IG} – the gear ratio of the input bevel gear;

i_{MR} – the gear ratio of the turning mechanism from the input link to the sun gear of the summing planetary row;

i_{FD} – gear ratio of final drive.

Accordingly, the speed of rewinding caterpillar tracks on the sides will be:

$$V_{T1} = \omega_{DW1} R_{DW} \text{ and } V_{T2} = \omega_{DW2} R_{DW},$$

where R_{DW} – drive wheel radius.

Knowing the speed of rewinding the caterpillar tracks, we find the efficiency of caterpillar tracks separately on the sides η_{cat1} and η_{cat2} as well as the coefficients of slipping (skidding) σ_1 and σ_2:

$$\eta_{cat1} = 0{,}95 - 0{,}018 \cdot V_{T1};\ \eta_{cat2} = 0{,}95 - 0{,}018 \cdot V_{T2};\ \sigma_1 = \frac{V_{T1} - V_1}{V_{T1}} \text{ and } \sigma_2 = \frac{V_{T2} - V_2}{V_{T2}}.$$

Knowing the centrifugal force P_Y, we define a longitudinal component of the centrifugal force P_X and the actual normal reaction under the caterpillar tracks N_1 and N_2, taking into account the redistribution of weight due to a lateral centrifugal force P_Y:

$$P_X = \frac{P_Y \chi}{R_C};\ N_1 = \frac{G_M}{2} - \frac{P_Y h_C}{B} \text{ and } N_2 = \frac{G_M}{2} + \frac{P_Y h_C}{B}.$$

Here h_C – the height of the center masses.

In the process of calculating the values N_1 and N_2 need to be monitored, to satisfy the conditions $N_1 \geq 0$ and $N_2 \leq G_M$ that are responsible for the lack of overturn machine.

Knowing the actual normal reaction under the caterpillar tracks N_1 and N_2, to determine the maximum forces of adhesion caterpillar mover with road (terrain) $P_{\varphi 1}$, $P_{\varphi 2}$ and total resistance to movement P_{F1}, P_{F2} under the boards:

$$P_{\varphi 1} = N_1 \varphi;\ P_{\varphi 2} = N_2 \varphi;\ P_{F1} = N_1 f + \frac{0{,}65 F_{\max} v_X^2 + P_X}{2} \text{ and } P_{F2} = N_2 f + \frac{0{,}65 F_{\max} v_X^2 + P_X}{2}.$$

Here, φ – the coefficient of adhesion caterpillar mover with road (terrain);

f – coefficient of resistance to movement;

$F_{\max}$ – the maximum cross-sectional area of the machine.

Knowing the factors slipping (skidding) σ_1, σ_2, we can determine the traction force (braking) on the lagging P_{D1} and on the more fast P_{D2} boards with sign:

$$P_{D1} = \frac{\sigma_1}{0{,}3} R_{\varphi 1}, \text{ if } \sigma_1 > 0{,}3, \text{ then } P_{D1} = R_{\varphi 1}; \text{ if } \sigma_1 < -0{,}3, \text{ then } P_{D1} = -R_{\varphi 1};$$

$$P_{D2} = \frac{\sigma_2}{0,3} R_{\varphi 2}\text{, if } \sigma_2 > 0,3\text{, then } P_{D2} = R_{\varphi 2}\text{; if } \sigma_2 < -0,3\text{, then } P_{D2} = -R_{\varphi 2}.$$

Knowing traction force (braking) on the lagging P_{D1} and on the more fast P_{D2} boards, define the torques of engine load M_R and the turning mechanisms M_{R1}, M_{R2}:

$$M_R = \frac{M_{GB} + M_{FMR1} + M_{FMR2}}{i_{IG}\eta_{IG}}. \tag{3}$$

Here, i_{IG} and η_{IG} – gear ratio and efficiency of the input bevel gear;

M_{FMR1} and M_{FMR2} – torques on the input links of the lagging and of the more fast turning mechanisms, depending on the friction torque on the friction clutches of the base transmission ($M_{FMR1} = M_{FMR2} = 0$ for the turning mechanism with one electric motor);

M_{GB} – load torque on the input shaft of the gearbox, is given by:

$$M_{GB} = \frac{2(P_{D1} + P_{D2})R_{DW}k_S}{i_{FD}i_{GB}(k_S+1)(\eta_{cat1} + \eta_{cat2})\eta_{FD}\eta_S\eta_{GB}}\text{, if } P_{D2} > P_{D1} > 0 \text{ and}$$

$$M_{GB} = \frac{R_{DW}k_S}{i_{FD}i_{GB}(k_S+1)}\left(\frac{P_{D2}}{\eta_{cat2}\eta_{FD}\eta_S\eta_{GB}} + P_{D1}\eta_{cat1}\eta_{FD}\eta_S\eta_{GB}\right)\text{, if } P_{D2} > 0 \text{ and } P_{D1} \le 0.$$

The load on the turning mechanisms M_{R1}, M_{R2} is calculated according to the formulas

$$M_{R2} = \frac{P_{D2}R_{DW}}{i_{FD}i_{GB}(k_S+1)i_{MR}\eta_{cat2}\eta_{FD}\eta_S\eta_{MR}};$$

$$M_{R1} = \frac{P_{D1}R_{DW}}{i_{FD}i_{GB}(k_S+1)i_{MR}\eta_{cat1}\eta_{FD}\eta_S\eta_{MR}}\text{, if } P_{D1} > 0 \text{ and}$$

$$M_{R1} = \frac{P_{D1}R_{DW}\eta_{cat1}\eta_{FD}\eta_S\eta_{MR}}{i_{FD}i_{GB}(k_S+1)i_{MR}}\text{, if } P_{D1} \le 0.$$

As a result, the output parameters of the module are:

P_{D1} and P_{D2} – the traction force (braking) on the lagging and on the more fast boards with sign;

P_{F1} and P_{F2} – the forces of total resistance to movement on the lagging and on the more fast boards;

P_Y – the centrifugal force, generated by the curvilinear motion of the machine;

$P_{\varphi 1}$ and $P_{\varphi 2}$ – the maximum forces of adhesion caterpillar mover with road (terrain);

M_{RR} – torque resistance to the turning;

M_R – torque resistance to the motion, is reduced to the crank shaft of the engine;

M_{R1} and M_{R2} – torques resistance to the motion, is reduced to the input links of the turning mechanisms of the lagging and of the more fast boards.

Model of the dual-flux turning mechanism.

Three modules have been developed to simulate the operation of a dual-flux turning mechanisms:

– “The turning mechanism with one electric motor and the sun gears of the summation planetary rows of different sides, rotating in opposite directions” (fig. 1);
– “The turning mechanism with two electric motors operating in braking mode while maintaining the mechanical branch in the turning mechanisms” (fig. 2);
– “The mechanical turning mechanism (base machine)” (fig. 3).

The input parameters for all modules at each step of the integration are the current values of the generalized velocities of rotation of the input links of the lagging ω_1 and more fast ω_2 the turning mechanisms, as well as the angular velocity of the engine crankshaft ω_E for all modules except the first (fig. 1).

For the mechanical turning mechanism and the turning mechanism with two electric motors on more fast board the friction clutch of the turning mechanism is always enabled. This allows us to eliminate the last equation of the system (1) and to express the angular velocity of rotation of the input link of the more fast turning mechanism ω_2 through the angular velocity of the engine crankshaft ω_E:

$$\omega_2 = \frac{\omega_E}{i_{IG}}.$$

When calculating the load on the engine (3) value M_{FMR2} of friction torque determined by the load torque of the friction clutch considering the sign $M_{FMR2} = M_{R2}$. If these schemes considered the linear motion from the system (1) is deleted and the penultimate equation and $\omega_1 = \omega_2$ and the respectively $M_{FMR1} = M_{R1}$. When turning with a free radius for the mechanical turning mechanism of the lagging board the friction clutch is switched off $M_{FMR1} = 0$ and the angular velocity ω_1 is determined by integration of the penultimate equation of the system (1).

For the mechanical turning mechanisms when developing fixed turning radius on the lagging board after the clutch is turned off the brake is turned on. Within 0.5 seconds the negative friction torque M_{MR1} increases linearly in absolute value to its maximum value and is kept so until $\omega_1 = 0$. After this, the penultimate equation of the system (1) is excluded from the integration process.

Asynchronous electric motor is switched as a generator for the turning mechanisms with two electric motors on the lagging board. In this case $M_{FMR1} = 0$ and $M_{MR1} = M_{EM1} < 0$. This mode continues until the $\omega_1 = 0$. After this the turning mechanism is placed on the friction brake and the penultimate equation of the system (1) is excluded from the integration process.

For the turning mechanism with one electric motor and the sun gears of the summation planetary rows of different sides, rotating in opposite directions (fig. 1) the last two equations are replaced by one:

$$\frac{d\omega_{EM}}{dt} = \frac{M_{EM} - M_{R2} + M_{R1}}{I^*_{MR}}.$$

Here I^*_{MR} – moment of inertia of the turning mechanism comprising two boards and one electric motor is reduced to the electric motor shaft.

The torque of asynchronous motor M_{EM} (fig. 4, [7]) with frequency adjustment (fig. 5, [7]) determined depending on the speed of its shaft by a generalized characterization (fig. 6).

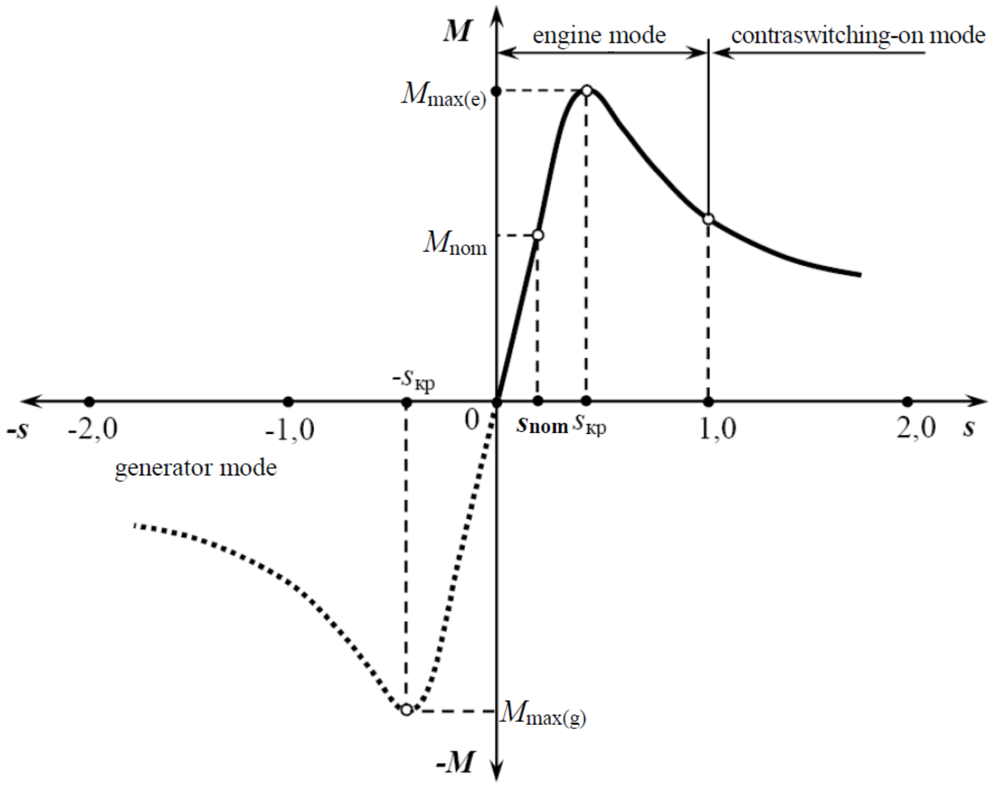

Fig. 4. Mechanical characteristics of asynchronous motor, depending on the slipping s

Asynchronous motor, developed to drive the motor-wheel armored personnel carrier (State enterprise “Kharkov Machine Building Design Bureau named after A.A. Morozov”) with a nominal power of 45 kW at a shaft speed of 3150 rev / min (330 s^{-1}) was adopted as a basis for calculations. Maximum torque in the traction mode of the engine was limited by the current force and overheating of the value of 230 Nm. An electric motor is switching to constant power mode when $\omega_X = 195{,}6\ s^{-1}$. The required reduction in the basic power an electric motor is achieved fold reduction in the axial dimension of the engine, in which only changed in proportion to the torque without changing the speed characteristics.

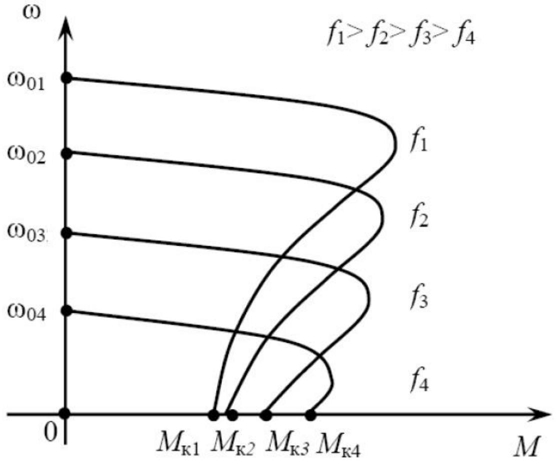

Fig. 5. Mechanical characteristics of the asynchronous motor with frequency speed control

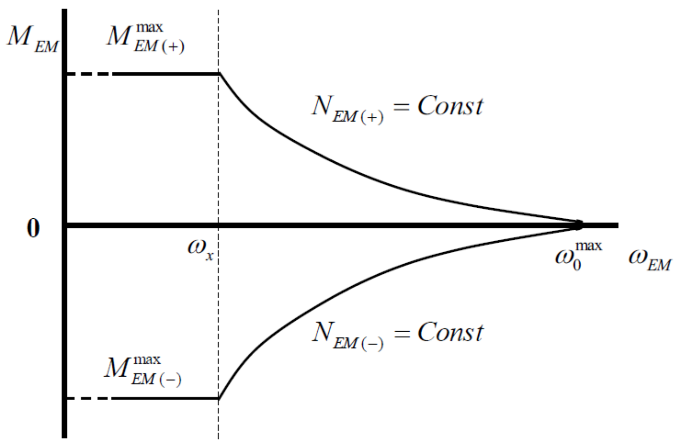

Fig. 6. The dependence of the traction (braking) of the electric motor of the turning mechanism from its speed of rotation of the armature with the management system

Central integration unit. The central integrating unit is a classic solver module for integration of differential equations by the Runge-Kutta method with constant or variable pitch. In this module, formed the right parts of the differential equations describing the behavior of dynamic system considering changes in its structure in the process. In the module following initial data is entered:

– counting time;

– the integration step in fractions of a second and the desired accuracy;

– a list of print parameters to file of the calculate results;

– the type of the dual-flux turning mechanism (one of the three according to fig. 1–3);

– mode of transmission (number of the gear);

– the type of road (terrain), on which machine is moving;

– the initial conditions for each of the differential equations;

– sequence diagram switching of the turning mechanisms.

From the central unit in the desired sequence caused all the previously described blocks for the correct formation of the right parts of differential equations. Then, considering accepted initial conditions or the results of the previous step the differential equations is integrating and the account of the results is written to disc.

Summary. As a result of this work was to create a tool with which on the example of light multipurpose tracked transporter-tractor MTLB became possible to simulate the curvilinear motion of the machine on a horizontal surface with different characteristics and different schemes of the dual-flux turning mechanisms.

In contrast to the analytical determination of the trajectory of movement of the machine numerical simulation make possible to calculate trajectory considering real-slipping and skidding of caterpillar tracks.

In the next publication is planned to bring the results of the comparative modeling of the entry process in the turning of light multipurpose tracked transporter-tractor MTLB with the standard mechanical and two selected scheme of the electromechanical dual-flux turning mechanisms.

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