On the question of the rational distribution of power between the hydrostatic and hydrodynamic branches of the complex steering mechanism for the tracked vehicle

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Abstract: *The article presents a description of the mathematical model of the curvilinear movement of a tracked vehicle with a complex steering mechanism, which contains a control hydrostatic branch and an additional hydrodynamic branch for unloading the first. The adequacy of the presented mathematical model was verified by calculating the characteristics of the turn in place of the Leopard-2 tank. The resulting tool can become the basis for parametric optimization of similar steering mechanisms for tracked vehicles.*
KEYWORDS: *HYDROSTATIC STEERING MECHANISM OF A TRACKED VEHICLE, FILLABLE HYDRODYNAMIC COUPLINGS, MATHEMATICAL MODEL.*

1. Introduction

Modern military tracked vehicles (TV), as a rule, have two-line hydrostatic steering mechanism (TLHSM), which make it possible to obtain a stepless change in the turning radius and provide the ability to turn the vehicle on the spot. The turn of the track vehicle on the spot mode is the most difficult for a two-line steering mechanism. This is due to the fact that in this mode, all engine power goes through the steering mechanism, and the main gearbox is simply locked. For heavy military TV, the weight has long exceeded 50 tons, and the power of the main engine has long exceeded 1000 hp. These vehicles have a width limitation due to the need to transport them by rail, which means that their length increases with increasing weight. This significantly increases the resistance to both turning and turning on the spot these vehicles. This means that the hydrostatic unit, which provides turning of the vehicle, with the classical approach to design, should significantly increase in size and weight. Back in the second half of the last century, an effective technical solution was found, which makes it possible to significantly relieve the hydrostatic unit (HSU) of the steering mechanism due to the fillable hydrodynamic couplings (HDU) [1, 2]. However, the design process for such steering mechanisms has always involved a large amount of complex engineering calculations. The problem of the optimal solution is associated with finding the ratio of the HSU and HDU parameters, which simultaneously provides the minimum volume and weight of the control system and good controllability of the vehicle.

2. Materials research.

2.1 Formulation of the Problem

In this paper, the authors set themselves the task of developing a mathematical model and software product that would allow to adequately simulate the process of turning a TV on horizontal terrain and at the same time be able to control all kinematic and power parameters in the transmission, including the pressure in HSU. The model should take into account slipping and skid of the tracked mover and be able to set different intensities and the law of rotation of the control wheel.

2.2 The Description of the Mathematical Model of the TV and the Calculation Assumptions

Assumptions and Simplifications that was Accepted.

– The movement of the TV is considered on horizontal terrain.

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

– The entrance to the turn and the movement in the turn are considered in the selected gear without switching in the gearbox.

– The engine is considered running on an external characteristic with full fuel supply.

– The ductility and resilience of shafts and other transmission elements is not taken into account due to the consideration of only low-frequency transients in the system without taking into account torsional vibrations.

– When the clutch in the base transmission are turned on, a linear increase in the friction torque is assumed from zero to the maximum value in 0.5 s.

– A turn is always viewed in one direction (to the left) and, accordingly, not the right and left, but the running and lagging sides are considered.

Initial data for modeling

TV parameters used for simulation:
– weight or mass of the TV;
– track width of the TV;
– length of the bearing surface of the TV;
– drive wheel radius;
– the height of the mass center;
– maximum cross-sectional area of the TV;
– the moment of inertia of the TV during rotation about a vertical axis passing through the center of mass;
– the external characteristic of the engine (the dependence of the value of free torque on the frequency of rotation of the crankshaft at full fuel supply);
– moment of inertia of the engine with a flywheel;
– kinematic transmission scheme indicating the number of teeth on all gears;
– the diameter and dependence of the torque on the turbine wheel on the frequency of its rotation during engine operation in maximum power mode;
– the diameter and dependence of the torque on the HDU on the frequency of rotation of its turbine wheel at various levels of filling with the working fluid;

Terrain parameters used for modeling:
– The coefficient of resistance to movement of the machine f;
– The coefficient of friction of track with the road φ;
– The coefficient of resistance to turning μ_{max}.

№	Type of terrain (road)	f	φ	μ_{max}
1.	Dry soddy loamy soil (humidity <8%)	0,08	0.9	0,9
2.	Dry loamy dirt road	0,065	0,8	0,8
3.	Loam plowing	0,1	0,7	0,7
4.	Wet road on loam	0,125	0,6	0,35
5.	Deep loose snow	0,25	0,3	0,3
6.	Asphalt	0,045	0,8	0,8

Structure of the Mathematical Model.

The mathematical model for studying the turn entry and the curved motion of the TV is based on the integration of differential equations describing the perturbed movement of the system and containing 6 generalized velocities (coordinates) [3, 4]:
– the longitudinal velocity of the TV, taking into account the skid (slipping) – v_X ;
– lateral sliding speed (drift) of the TV – v_Y ;
– the angular speed of rotation of the body of the TV relative to the vertical axis passing through the center of mass – ω_Z ;

– angular speed of rotation of the crankshaft of the engine and the associated pump of the hydrostatic steering control (HSU) and pump wheels of the HDU – ω_E ;
– angular speed of the turbine wheel of the torque converter, gearbox and epicycles of the summing planetary gear sets (steering epicyclic – SE) – ω_T ;
– angular speed of rotation of the HSU hydraulic motor and the associated turbine wheels of the HDU and solar gears of the SE – ω_{HSU} .

For each of the generalized velocities, a differential equation is compiled, which are grouped into a system:

$$\begin{cases} \dfrac{dv_X}{dt} = \dfrac{P_{D1} + P_{D2} - P_{F1} - P_{F2}}{G_M / g}; \\ \dfrac{dv_Y}{dt} = \dfrac{P_Y - P_{\varphi 1} - P_{\varphi 2}}{G_M / g}; \\ \dfrac{d\omega_Z}{dt} = \dfrac{0{,}5B(P_{D2} - P_{D1}) - M_{RR}}{I_Z}; \\ \dfrac{d\omega_E}{dt} = \dfrac{M_E - M_{ER}}{I_E + I_{ETR}}; \\ \dfrac{d\omega_T}{dt} = \dfrac{M_T - M_{TR}}{I_{TTR}}; \\ \dfrac{d\omega_{HSU}}{dt} = \dfrac{M_{HSU} + M_{HDU} - M_{HSUR}}{I_{HSUR}}. \end{cases}$$

Here the variables are:

P_{D1} and P_{D2} – traction (braking) forces on the lagging and running sides taking into account the sign, calculated by the slip or skid coefficients;

P_{F1} and P_{F2} – the forces of resistance to movement on the lagging and running sides taking into account the redistribution of normal reactions between the sides due to centrifugal force P_Y ;

P_Y – centrifugal force arising from the curvilinear movement of the TV;

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

M_{RR} – the moment of resistance to turning, depending on the geometric characteristics of the machine, its weight, turning radius and the characteristics of the soil (road);

M_E – engine torque;

M_{ER} – carried over to the crankshaft of the engine moment of resistance of the pump wheels of the torque converter and HDU, as well as the pump HSU;

M_T – torque on the turbine wheel of the torque converter;

M_{TR} – carried over to the turbine wheel moment of resistance to movement of the TV;

M_{HSU} – torque on the shaft of the HSU hydraulic motor;

M_{HDU} – torque on the HDU turbine wheel;

M_{HSUR} – the drag moment, transmitted from the sun gears of the SE, carried over to the shaft of the HSU hydraulic motor.

Constants:

G_M – weight (gravity) of the TV;

B – track width of the TV;

I_Z – the moment of inertia of the TV during rotation (turning) about a vertical axis passing through the center of mass;

I_E – moment of inertia of the engine with a flywheel;

I_{ETR} – carried over to the engine crankshaft moment of inertia of the pump wheels of the torque converter and HDU, as well as the HSU pump;

I_{TTR} – carried over to the turbine wheel of the torque converter the moment of inertia of the rotating parts of the transmission from the SE epicycles to the turbine wheel of the torque converter and part of the moment of inertia of the rotating parts of the transmission from the drive wheels to the SE carriers (levers);

I_{HSUR} – carried over to the shaft of the HSU hydraulic motor, the moment of inertia of the rotating transmission parts from the SE sun gears and from the turbine wheels of the HDU to the HSU hydraulic motor shaft.

Model of curved motion of a tracked vehicle on the terrain.

Input parameters of the “The turning” module.

“The turning” module as input parameters at each integration step uses the current values of the following generalized speeds:
– the longitudinal velocity of the TV, taking into account the skid (slipping) – v_X ;
– the angular velocity of rotation of the body of the TV relative to the vertical axis passing through the center of mass – ω_Z ;
– angular speed of the turbine wheel of the torque converter, gearbox and epicycles of the SE – ω_T ;
– angular speed of rotation of the HSU hydraulic motor and the associated turbine wheels of the HDU and solar gears of the SE – ω_{HSU} ,

as well as the gear number and all the constants characterizing the TV and the condition of the road (ground) on which the traffic is carried out.

The algorithm of work “The turning" module.

Given the linear v_X and angular ω_Z speeds of movement of the machine body, the actual longitudinal speeds of movement of the chassis of the lagging and running sides relative to the supporting surface are calculated:

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

Further, the turning radii are determined for $\omega_Z \neq 0$ – central and along the sides:

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

Knowing the radius of turning of the machine, we can determine the actual coefficient of resistance to turning μ on a given road (terrain) and the centrifugal force P_Y that occurs when turning:

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

where $\mu_{\max}$ – maximum turning resistance coefficient corresponding to turning around a stopped side.

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

Knowing the centrifugal force P_Y and the actual coefficient of resistance to turning μ, we determine the absolute χ and relative χ_0 displacement of the pole of turning:

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

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

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

Having finally decided on the coefficient of resistance to rotation μ, we find the moment of resistance to rotation:

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

Taking into account: the rotation speeds of the turbine wheel of the torque converter ω_T and the shaft of the HSU hydraulic motor

ω_{HSU}, current gear number and all gear ratios, we find the driving speeds of the driving wheels on the lagging and running sides: ω_{DW1} and ω_{DW2}.

Accordingly, the linear speeds of rewinding of tracks on the sides will be:

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

where R_{DW} – drive wheel radius.

Knowing the speed of rewinding the tracks, we find the efficiency of the caterpillar mover separately on the sides (η_{cat1} and η_{cat2}) and the slip coefficients (skid) σ_1 and σ_2:

$$\eta_{cat1} = 0{,}95 - 0{,}018 \cdot V_{T1}; \quad \eta_{cat2} = 0{,}95 - 0{,}018 \cdot V_{T2};$$

$$\sigma_1 = \frac{V_{T1} - V_1}{V_{T1}} \quad \text{and} \quad \sigma_2 = \frac{V_{T2} - V_2}{V_{T2}}.$$

Knowing the centrifugal force P_Y, we determine the longitudinal component of the centrifugal force P_X and the actual normal reactions under the sides N_1 and N_2, taking into account the redistribution of weight due to the force P_Y:

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

Here h_C – the height of the mass center. In the process of calculating the values N_1 and N_2, it is necessary to control that the conditions $N_1 \geq 0$ and $N_2 \leq G_M$, that are responsible for the absence of the tipping of the machine are fulfilled.

Knowing the actual normal reactions under the sides N_1 and N_2, we determine the maximum adhesion forces $P_{\varphi 1}$, $P_{\varphi 2}$ and the total resistance to movement under the sides P_{F1}, P_{F2}:

$$P_{\varphi 1} = N_1 \varphi; \quad P_{\varphi 2} = N_2 \varphi;$$

$$P_{F1} = N_1 f + \frac{0{,}65 F_{\max} v_X^2 + P_X}{2} \quad \text{and}$$

$$P_{F2} = N_2 f + \frac{0{,}65 F_{\max} v_X^2 + P_X}{2}.$$

Here φ and f – accordingly, the coefficients of adhesion of the mover to the road (terrain) and resistance to movement; $F_{\max}$ – maximum cross-sectional area of the TV.

Knowing the slipping (skid) coefficients σ_1 and σ_2, we can determine the traction (braking) forces on the lagging P_{D1} and running P_{D2} sides taking into account the sign:

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

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

Knowing the traction (braking) forces on the lagging P_{D1} and running P_{D2} sides, the direction of the power flows and all the gear ratios, we determine the load moments on the turbine wheel of the torque converter M_{TR} and the HSU hydraulic motor M_{HSUR}.

Output parameters of "The turning" module.

P_{D1} and P_{D2} – traction (braking) forces on the lagging and running sides taking into account the sign;

P_{F1} and P_{F2} – forces of resistance to movement on lagging and running sides;

P_Y – centrifugal force arising from the curvilinear movement of the machine;

$P_{\varphi 1}$ and $P_{\varphi 2}$ – adhesion forces of the caterpillar mover with the soil in the transverse direction;

M_{RR} – turning resistance torque;

M_{TR} – carried over to the turbine wheel torque resistance to movement of the TV;

M_{HSUR} – the drag moment carried over to the shaft of the HSU hydraulic motor transmitted from the sun gears of the SE.

The remaining modules of the model.

The «Engine» module.

The «Engine» module uses the speed of rotation of the crankshaft ω_E as an input parameter. The output parameter of the module is the engine torque M_E, calculated for the full fuel supply mode.

The «Torque Converter» module.

The «Torque Converter» module uses the crankshaft ω_E and turbine wheel ω_T speed of rotation as input parameters. The output parameters of the module are the torques on the turbine wheel of the torque converter M_T and on the pump wheel of the torque converter (part of the torque M_{ER}).

The «HSU» module.

The «HSU» module uses the speed of rotation of the shafts of the HSU hydraulic motor ω_{HSU} and the engine crankshaft ω_E, the HSU control parameter in fractions of unity (relative steering angle) as input parameters. The output parameters of the module are the torques on the HSU hydraulic motor M_{HSU} and on the HSU pump (part of the torque M_{ER}), as well as the pressure difference Δp.

The «HDU» module.

The "HDU" module uses the speed of rotation of the HSU hydraulic motor ω_{HSU} and the crankshaft of the engine ω_E as input parameters, as well as the pressure difference Δp in the HSU. The output parameters of the module are the torques on the HDU turbine wheel M_{HDU} and on the HDU pump wheel (the part of the torque M_{ER}).

Central integration unit.

The central integration unit is a classic solver module for integrating a system of differential equations using the Runge-Kutta method with constant or variable pitch. In this module, the right-hand sides of differential equations are formed that describe the behavior of a dynamic system taking into account changes in its structure during operation. As the initial data in the described module are entered:
- counting time;
- integration step in fractions of a second and the desired accuracy;
- a list of print options in the file of calculation results;
- gearbox operating mode (gear number);
- type of terrain (road) along which the VC is moving;
- initial conditions for each of the differential equations;
- cyclogram rotation of the helm.

From the central block in the required sequence, all previously described blocks are called up for the correct formation of the right-hand sides of the differential equations. Then, taking into account the accepted initial conditions, the differential equations are integrated and the counting results are written to disk. Processing of calculation results and construction of the necessary graphs is performed in the MS Exel program.

2.3. Calculations results

Parameters used for simulation:
- vehicle weight G_M = 55 t;
- vehicle track width B = 2,785 m;
- length of the supporting surface of the tracks L = 4,73 m;
- radius of the driving wheel R_{DW} = 0,32125 m;
- the height of the center of mass h_C = 1,15 m;
- maximum cross-sectional area of the vehicle e $F_{\max}$ = 5 m^2;
- moment of inertia of the vehicle when rotating about the vertical axis passing through the center of mass I_Z = 196200 kgm^2;
- moment of inertia of the engine with a flywheel I_E = 78,48 kgm^2;

- a kinematic diagram of the transmission with an indication of the number of teeth on all elements (Fig. 1) [2];
- the external characteristic of the engine, the dependence of the torque on the turbine wheel of the torque converter on the frequency of its rotation when the engine is operating at maximum power, the dependence of the torque on the turbine of HDU on the its rotation frequency at different levels of filling with the working fluid, characteristics of HSU are taken from [1, 6].

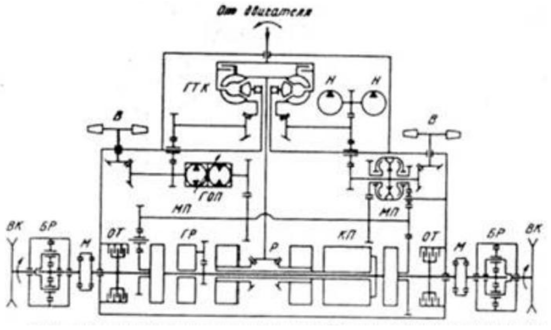

Figure 1 *– Kinematic diagram of the transmission*

When turning the vehicle on a horizontal asphalt surface, in the case of turning the steering wheel to the maximum position in 1 s, a 90° degree turn is completed in 3,6 s, 180° - in 6 s and 360° - in 10,7 s. At the same time, all transient processes in the engine and transmission completely calmed down in the first 2.5 seconds. Below (fig. 2-5) are the graphs of the main transient processes characterizing the distribution of power flows in the transmission when the vehicle turns in place.

In fig. 3-5, number 1 designates the parameter at the input to the device, and 2 - at the output from the device.

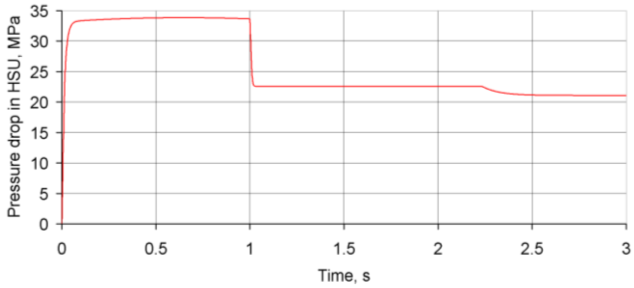

Figure 2 *– Pressure drop in HSU*

3. CONCLUSIONS

In accordance with the goals and objectives set for the results of the work done, the following conclusions can be drawn:

1) The calculated turn in place time of the Leopard-2 tank on a horizontal asphalt site using rubber asphalt pads when turning the steering wheel from the neutral position to the maximum in 1 second with an accuracy of 3% corresponds to those stated in the tactical and technical characteristics of this vehicle [1, 6]. This testifies to the adequacy of the developed mathematical model for this mode of motion.

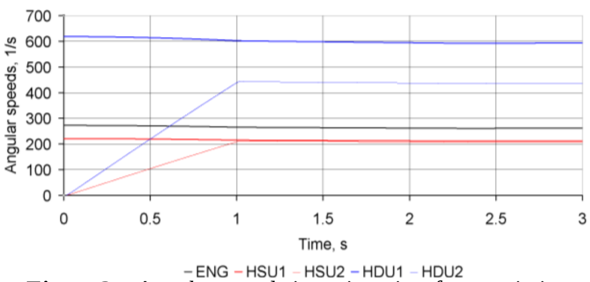

Figure 3 *– Angular speeds in main units of transmission*

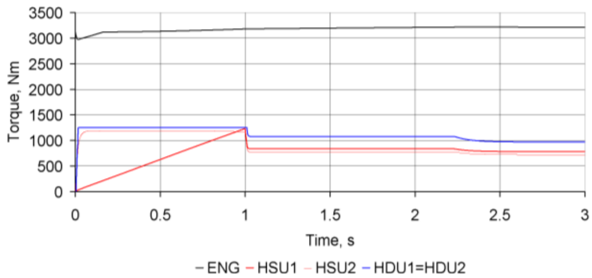

Figure 4 *– Torque in main units of transmission*

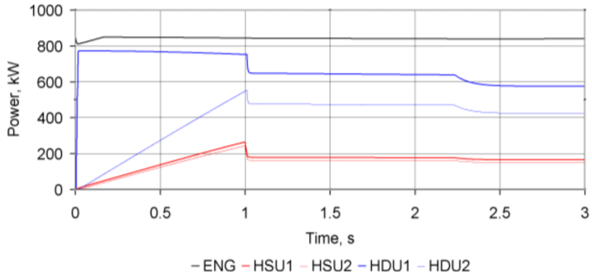

Figure 5 *– Power in main units of transmission*

2) The developed mathematical model and the software product that implements it make it possible to efficiently numerically study any planetary transmissions of tracked vehicles with TLHSM and hydrodynamic unloading in order to optimize their parameters at the design stage.

3) When designing TLHSM with hydrodynamic unloading for tracked vehicles, the power ratio of HDU and HSU should be approximately 3 to 1. This allows, on the one hand, to significantly relieve the HSU and, on the other hand, without disturbing the stability of the control system, to use the HSU as a link directly controlling the turning process.

4. LITERATURE

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