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Original Study Article

The method of development of the electronic control system for curvilinear motion of a high-speed tracked vehicle with dual-flow transmission

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ABSTRACT

BACKGROUND: Handling and safety requirements to high-speed tracked vehicles (HSTV) rise in tandem with the growth of average motion velocities. The issue of ensuring continuously variable turn radius at curvilinear motion is relevant for HSTVs. Current layouts of steering mechanisms are able to meet this requirement, however they have certain disadvantages and are not compatible with electronic systems improving motion safety and lowering demands to mechanic-drivers' skills.

AIMS: The synthesis of control laws for dual-flow transmission with a hydrostatic steering mechanism (HSSM) controlled by an electromechanical actuator which exclude "hard" links between steering handwheel and working volume adjustment mechanism of the HSSM.

METHODS: The study methods are based on using numerical simulation and ensuring real-time operation of the developed models. In addition, the study methods include synthesis of control algorithms for vehicle's mechanical systems, used in on-board controllers, with adequacy assessment at virtual and laboratory experiments.

RESULTS: The method of development of control systems (CS) making possible to develop and to debug CSs without a HSTV prototype has been put into force. With using the described method, the total time of CS development and debugging reduces. Workability of the method is proved with the example of development of the CS for curvilinear motion of the HSTV with dual-flow transmission.

CONCLUSIONS: The study aim has been achieved, the accomplished work shows validity of the given methof of CS development.

Keywords: *high-speed tracked vehicle; curvilinear motion; control system; dual-flow transmission; hydrostatic steering mechanism.*

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Оригинальное исследование

Метод разработки электронной системы управления криволинейным движением быстроходной гусеничной машины с двухпоточной трансмиссией

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АННОТАЦИЯ

Обоснование. Совместно с ростом средних скоростей движения быстроходных гусеничных машин (БГМ) растут и требования к управляемости и безопасности движения. Актуальным для БГМ является вопрос обеспечения бесступенчатого изменения радиуса поворота при криволинейном движении. Существующие схемы механизмов поворота могут обеспечить данное требование, но при этом обладают определенными недостатками и не позволяют применять электронные системы, повышающие безопасность движения и снижающие требования к квалификации механиков-водителей.

Цель работы — синтез законов управления двухпоточной трансмиссией с гидрообъемным механизмом поворота (ГОМП), управляемым электромеханическим актуатором, исключающим «жесткую» связь между штурвалом и механизмом регулирования рабочего объема ГОМП.

Методы: исследования основываются на применении имитационного моделирования и обеспечении функционирования разрабатываемых моделей в режиме реального времени, а также включают в себя синтез алгоритмов управления механическими системами транспортного средства для бортовых контроллеров управления с проведением оценки их адекватности в виртуальном и полунатурном экспериментах.

Результаты: реализован метод разработки систем управления (СУ), позволяющий проводить разработку и отладку СУ при отсутствии опытного образца БГМ. При использовании описанного метода сокращается общее время разработки и отладки алгоритмов СУ. Применимость данного метода доказана на примере разработки СУ криволинейным движением БГМ с двухпоточной трансмиссией.

Заключение: поставленная цель достигнута, проведенная работа показывает состоятельность приведенного метода разработки СУ.

Ключевые слова: *быстроходная гусеничная машина; криволинейное движение; система управления; двухпоточная трансмиссия; гидрообъемный механизм поворота.*

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BACKGROUND

The development of highly mobile wheeled and tracked vehicles is implemented under increasing requirements related to improving their operational properties. In particular, increased speed must be accompanied by improved methods to ensure safe traffic control.

For high-speed tracked vehicles (HSTV), with an increase in the average and maximum speeds, the issue of stepless change in the radius (curvature) of rotation remains relevant. A solution to this problem can be achieved through the use of traditional dual-flow transmissions with hydrostatic steering mechanism (HSSM), as well as through the use of traction electric motors to drive the traction wheels to control the curvilinear movement of a tracked vehicle like a car. In addition, this type of transmission is largely suitable for the implementation of remote control and robotization [1, 2].

For a dual-flow transmission scheme with an HSSM in the presence of a “hard” connection between the steering wheel and a mechanism for regulating the working volume of the HSSM pump, the actual turning radius with a constant steering wheel position will depend on the selected transmission gear and driving conditions [3]. To eliminate this dependence, the steering wheel should not be mechanically connected to the pump. This communication must be implemented through a curvilinear motion control system (CS) using information to estimate the actual curvature of the turn.

In the absence of a mechanical connection between the steering wheel and the mechanism for regulating the displacement of the pump, the CS for the curvilinear movement of the HSTV is implemented based on an electronic control unit (ECU) with operating algorithms that are determined by laws that establish a connection between the influence on the controls for the curvilinear movement of the HSTV, the dynamics of the machine, and the operation of a dual-flow transmission with HSSM.

When developing laws and algorithms, the most important aspect is ensuring that the car responds predictably to the actions of the driver. The control of the curvilinear movement of the HSTV occurs because of the difference in the rotation of speeds of the tracks on opposite sides [4]. The predictable response of the machine to the control action is ensured by the correct choice of function that connects the control action with changes in the rotation of speeds of the tracks on the lagging and leading sides, considering the assessment of the actual curvature of the trajectory.

The use of mathematical simulation models of the HSTV motion [1, 2, 5–7] to test algorithms for a curvilinear motion CS at the early stages of design will significantly speed up the process of implementing a prototype CS at the physical level.

METHOD FOR DEVELOPING ONBOARD CS

The development of onboard CS by mechanisms and assemblies of ground vehicles, which is an operation determined not only by quantitative criteria but also by qualitative criteria (based largely on feedback from test drivers on the behavior of objects), involves the use of special design and research methods that will ensure compliance with the requirements in the absence of a prototype HSTV. One of these methods for designing onboard CS is the use, at the early stages of development work, of real-time mathematical simulation models in conjunction with the developed electronic CS [8], which can be used to describe the interaction of the driver with the CS and evaluate the reaction of the object to various control inputs.

Fig. 1 presents the process of transition from the use of real-time mathematical simulation models to working with a physical CS for curvilinear motion and an HSTV sample. The CS development process includes three stages.

In Stage 1, it is assumed to work only with mathematical simulation models of the developed CS and the biomass model. Data exchange uses virtual channels available in the software used for mathematical simulation.

At this stage, a synthesis of the basic control laws underlying the CS operation with curvilinear motion is performed. Because the CS is developed with curvilinear movement, it is necessary to provide an interface for the interaction of the mathematical simulation model with the driver. This interface must include the following:

- graphical interface for displaying the current position of the virtual control object;

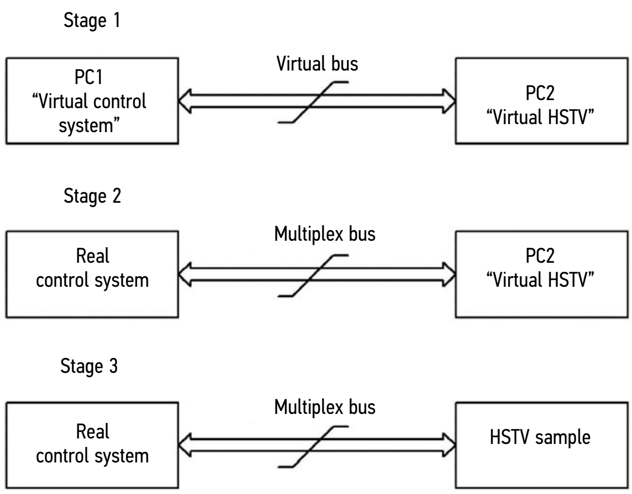

Fig. 1. Sequence of transition from numerical simulation models of the developed CS and the HSTV to real prototypes of the CS and the HSTV.
Рис. 1. Последовательность перехода от имитационных математических моделей проектируемой СУ и БГМ к действующим образцам СУ и БГМ.

- controls for receiving command inputs from the test driver.

As a result of a series of virtual races performed by a group of test drivers using real-time motion models with different control laws, the most suitable law is selected based on expert assessments and the requirements for CS with curvilinear movement.

In Stage 2, the operation of a prototype CS based on an ECU is implemented. At the same time, the prototype CS functions together with a mathematical simulation model of the HSTV in real time.

One of the main tasks solved at this stage is confirming the compliance of the performance and stability indicators of the CS with the specified requirements. At this stage, the CS is also debugged under conditions close to real work.

In Stage 3, direct research into the developed CS for curvilinear motion as part of a prototype HSTV is conducted, with confirmation of the capability to perform the required control tasks.

DESCRIPTION OF THE OPERATION OF THE TRANSMISSION AND HYDROSTATIC STEERING MECHANISM OF THE MACHINE DESIGNED

The developed electronic system for controlling the curvilinear movement of the HSTV is designed for joint operation with the HSSM elements of the dual-flow transmission [9, 10], as shown in Fig. 2.

The implementation and control of the curvilinear movement of the HSTV are performed by adjusting the working volume of the axial piston pump of the HSSM to change the speed and direction of the flow of the working fluid.

When moving in a straight line, the delivery flow of the HSSM pump is zero, the small central wheels (SCW) of the summing planetary gears are stopped, and the entire power of the internal combustion engine (ICE) is supplied to the carriers of the summing planetary gears through the gearbox (GB). Stopping the SCW ensures the stability of straight-line movement.

Stepless changes in turning radii are implemented by adjusting the HSSM pump working volume. The presence of an idler gear in driven engagement between the output shaft of the HSSM motor and one of the summing planetary gears ensures the opposite rotation of the corresponding SCW with equal angular velocities.

An increase in the HSSM pump flow rate leads to an increase in the speed of the SCW rotation in opposite directions, which will be accompanied by first an increase and then a decrease in the speed of rotation of the drive wheels of the opposite sides. In the absence of skidding and slipping of the supporting surfaces of the tracks, the theoretical turning radius of the HSTV is equal to

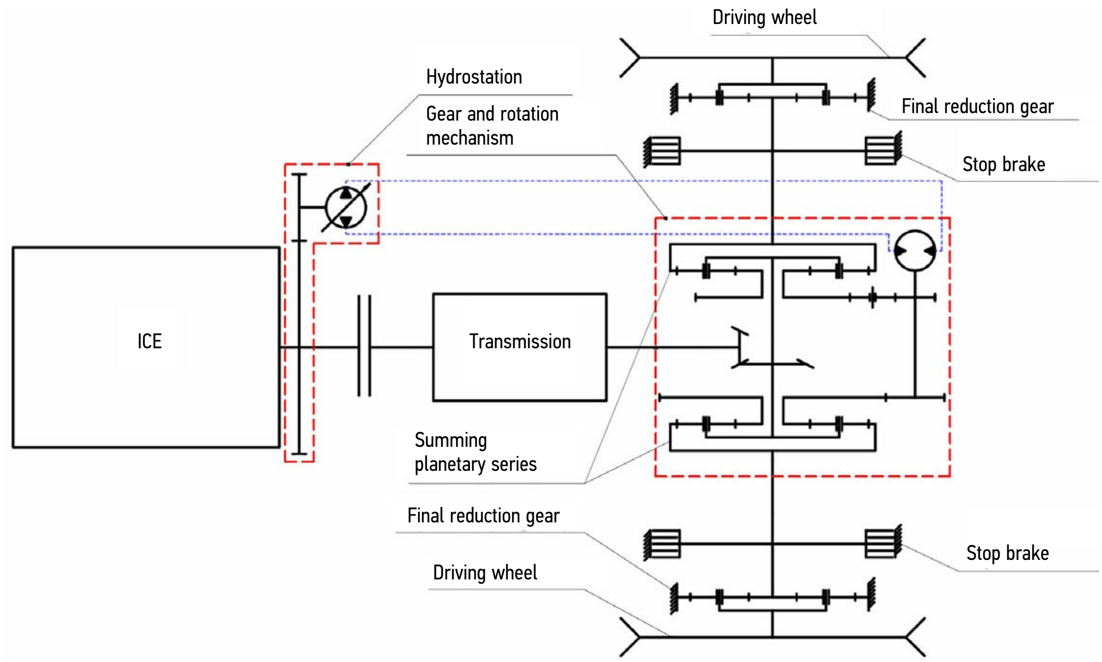

Fig. 2. Kinematic scheme of the dual-flow transmission of the HSTV.
Рис. 2. Кинематическая схема двухпоточной трансмиссии БГМ.

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$$R_{\mathrm{T}} = \frac{v_t}{\omega_{zt}} = \frac{B(\omega_{\mathrm{dw}2} + \omega_{\mathrm{dw}1}) r_{\mathrm{dw}}}{2(\omega_{\mathrm{dw}2} - \omega_{\mathrm{dw}1}) r_{\mathrm{dw}}} = \frac{\omega_{\mathrm{dw}2} + \omega_{\mathrm{dw}1}}{\omega_{\mathrm{dw}2} - \omega_{\mathrm{dw}1}} \cdot \frac{B}{2}, \quad (1)$$

where $\omega_{\mathrm{dw}1}$, $\omega_{\mathrm{dw}2}$ are the angular velocities of the driving wheels (DW) of the lagging and advancing sides, respectively; $\omega_{zt} = (\omega_{\mathrm{dw}2} - \omega_{\mathrm{dw}1}) \cdot r_{\mathrm{dw}} / B$ is the theoretical angular speed of rotation of the HSTV in the absence of skidding and slipping; B is the HSTV track; r_{dw} is the DW radius; and v_t is the theoretical speed of movement of the center of mass of the HSTV.

Regulation of the working volume of the HSSM pump is performed by turning the steering wheel. Based on Eq. (1) and the transmission and turning mechanism diagram presented in Fig. 2, the theoretical turning radius of the HSTV will depend on the GB transmission based on the presented dependence:

$$R_{\mathrm{T}} = \frac{(1 + k_{\mathrm{SPS}})\omega_{\mathrm{ICE}}}{i_{\mathrm{GB}} i_{\mathrm{FD}} \omega_{\mathrm{SCW}}} \cdot \frac{B}{2}, \quad (2)$$

where k_{SPS} is the parameter of the summing planetary series;; ω_{ICE} is the angular velocity of the ICE crankshaft; ω_{SCW} is the angular velocity of the SCW of the summing planetary series; i_{GB} is the GB ratio; and i_{FD} is the final drive (FD) gearing ratio.

The absolute values of the angular velocities of the SCW of the summing planetary gears at a constant angular velocity of the ICE are determined based on the current value of the working volume of the HSSM pump. If there is a “hard” connection in the CS for the curvilinear movement of the HSTV, then the steering wheel position uniquely determines the working volume of the HSSM pump.

To ensure the independence of the theoretical turning radius from the GB transmission, it is necessary to abandon the “hard” connection between the steering wheel and the mechanism for regulating the working volume of the HSSM pump through the implementation of an electronic CS for the curvilinear movement of the HSTV. The ECU will control the actuator for regulating the working volume of the HSTV pump based on the reference action (the current value of the steering wheel rotation angle) and the current values of the angular velocities of the DW (adjustable parameter).

The angular speed of the output shaft of the HSSM motor is determined based on the following equation:

$$\omega_{\mathrm{m}} = \frac{\omega_{\mathrm{ICE}} \cdot \eta_{\mathrm{vol\,p}} \eta_{\mathrm{vol\,m}} \left(\dfrac{q_{\max \mathrm{p}} e_{\mathrm{p}}}{q_{\mathrm{m}}} \right)}{i_{\mathrm{PTO}}}, \quad (3)$$

where ω_{m} is the angular velocity of the HSSM motor shaft; ω_{ICE} is the angular velocity of the ICE crankshaft; $\eta_{\mathrm{vol\,p}}$ is the volumetric efficiency of the HSSM pump; $\eta_{\mathrm{vol\,m}}$ is the volumetric efficiency of the HSSM motor; $q_{\max \mathrm{p}}$ is the maximum working volume of the HSSM pump; e_{p} is the parameter for regulating the working volume of the HSSM pump; q_{m} is working volume of the HSSM motor (constant value); and i_{PTO} is the gear ratio of the power takeoff (PTO) of the HSSM pump drive.

According to Eqs. (2) and (3), in an electronic curvilinear motion CS, in the absence of a “hard” connection between the steering wheel and the actuator for regulating the working volume of the HSSM pump, the dependence of the theoretical turning radius on the GB transmission can be reduced, as follows:

$$R_{\mathrm{T}} = \frac{(1 + k_{\mathrm{SPS}}) i_{\mathrm{PTO}} i_{\mathrm{add}}}{i_{\mathrm{GB}} i_{\mathrm{FD}} \eta_{\mathrm{vol\,p}} \eta_{\mathrm{vol\,m}} \left(\dfrac{q_{\mathrm{p}}}{q_{\mathrm{m}}} \right)} \cdot \frac{B}{2}, \quad (4)$$

where i_{add} is the gear ratio between the motor shaft of the HSSM and the SCW of the summing planetary gear ($\omega_{\mathrm{SCW}} = \omega_{\mathrm{M}} / i_{\mathrm{add}}$).

The working volume of the pump q_{p} in Eq. (4) is a variable value, which is regulated by the ECU. Thus, the implementation of the independence of the theoretical turning radius, which is set by the steering wheel position, from the GB transmission is possible by a corresponding change in the working volume of the HSSM pump.

The considered dual-flow transmission scheme with HSSM and an automated CS also enables to implementation of the following capabilities:

- maintaining the speed of straight-line motion while turning by the center of mass (Type 1 turning mechanism reported in Ref. [3]);
- the capability to rotate around the center of mass (when the drive wheels rotate only from the HSSM and neutral gear in the GB).

MATHEMATICAL SIMULATION MODEL OF CURVILINEAR MOTION

When describing the dynamics of the curvilinear motion of the HSTV, the following coordinate systems (i.e., CS) are used (Fig. 3):

- fixed CS connected to the supporting surface;
- movable CS connected to the HSTV center of mass;
- movable CS connected to the HSTV support rollers (K_i).

The mathematical simulation model was developed considering the following assumptions:

- the HSTV moves on a nondeformable support base of the “dense soil” type;

- the dynamics of the track contour are not considered, and the speed of rewinding the track is equal to the peripheral speed of the DW;
- transmission links are considered rigid;
- transient processes when shifting GB are not considered;
- the directions of tangential reactions from the side of the supporting surface under the active sections of the tracks are opposite to the direction of their sliding speeds;
- the properties of the hydraulic system working fluid do not depend on its temperature;
- modeling of hydrostatic transmission occurs without considering local resistance.

The dynamics of the curvilinear motion of the HSTV on a flat, nondeformable support base are described based on the system of equations expressed in Eq. (5) [11], which includes six equations. The first three equations describe the movement of the machine and represent a projection of the vector expression of the theorem on the change in momentum on the *X* and *Y* axes and a projection of the vector expression on the change in angular momentum on the *Z* axis (direction perpendicular to the plane shown in Fig. 3). The last three equations are necessary for the transition from the moving coordinate system *XOY* to the fixed coordinate system *X'OY'* and obtaining the trajectory of the HSTV motion.

$$\begin{cases} a_x = \dfrac{dv_x}{dt} - \omega_z v_y = \dfrac{1}{m}\left(\sum\limits_{i=1}^{n} R_{xi} - P_{wx}\right), \\ a_y = \dfrac{dv_y}{dt} + \omega_z v_x = \dfrac{1}{m}\sum\limits_{i=1}^{n} R_{yi}, \\ J_z \dfrac{d\omega_z}{dt} = \sum\limits_{i=1}^{n} M_{\mathrm{PC}i} + \sum\limits_{i=1}^{n} M\left(\overrightarrow{R_i}\right), \\ \dfrac{dx'}{dt} = v_x \cos\theta - v_y \sin\theta, \\ \dfrac{dy'}{dt} = v_x \sin\theta + v_y \cos\theta, \\ \omega_z = \dfrac{d\theta}{dt}, \end{cases} \qquad (5)$$

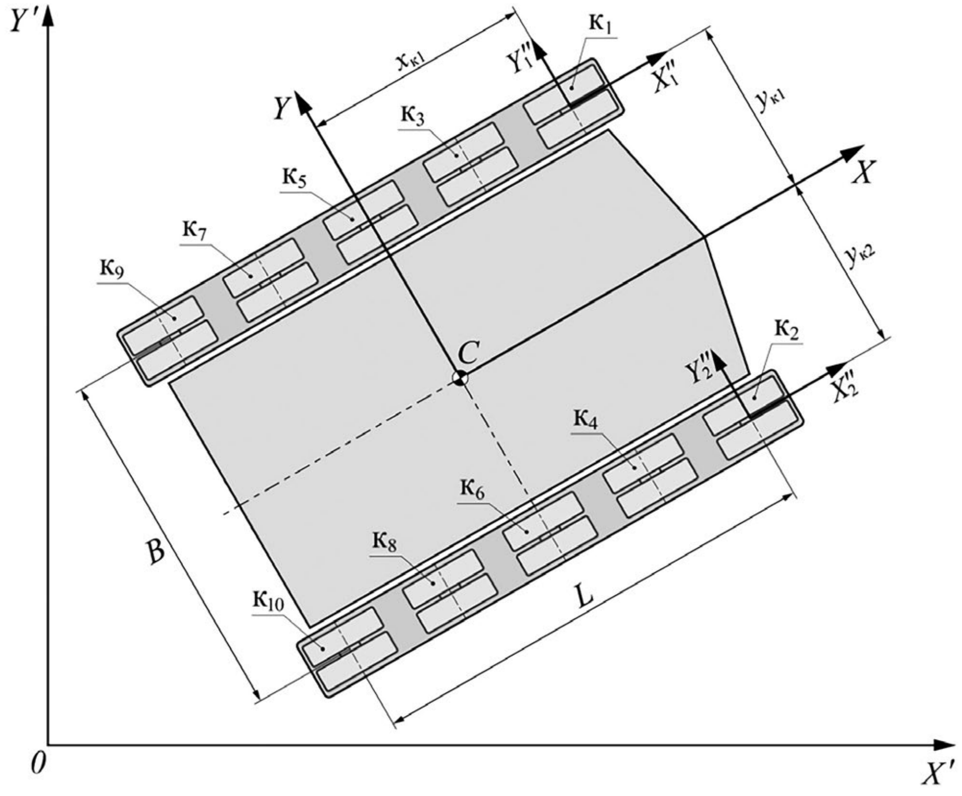

Fig. 3. Coordinate frames used for definition the dynamics of curvilinear motion of the HSTV: *C* – the HSTV's center of gravity; *XY* – the coordinate frame related to the HSTV's center of gravity; *X'Y'* – the coordinate frame related to ground; $X_i''Y_i''$ – the coordinate frame related to the *i*-th track roller; *L* – the HSTV's base; *B* – the HSTV's track; $x_{кi}$ – the *X*-axis coordinate of the *i*-th track roller in the *XY* coordinate frame; $y_{кi}$ – the *Y*-axis coordinate of the i-th track roller in the *XY* coordinate frame.

Рис. 3. Системы координат, используемые для описания динамики криволинейного движения БГМ: *C* – центр масс БГМ; *XY* – система координат, связанная с центром масс БГМ; *X'Y'* – система координат, связанная с опорным основанием; $X_i''Y_i''$ – система координат, связанная с *i*-ым опорным катком; *L* – база БГМ; *B* – колея БГМ; $x_{кi}$ – координата *i*-го катка по оси *X* в системе координат *XY*; $y_{кi}$ – координата *i*-го катка по оси *Y* в системе координат *XY*.

where a_x is the longitudinal acceleration of the HSTV center of mass; v_x is longitudinal velocity of the HSTV center of mass; v_y is transverse velocity of the HSTV center of mass; R_{xi} is the longitudinal reaction from the interaction of the *i*-th active section of the track with the supporting surface; m is the HSTV weight; a_y is the lateral acceleration of the HSTV center of mass; R_{yi} is the transverse reaction from the interaction of the *i*-th active section of the track with the supporting surface; J_z is the HSTV moment of inertia relative to the vertical axis passing through the center of mass; $M_{\text{PC}i}$ is the moment of resistance to the rotation of the *i*-th active section of the track; $M\left(\overrightarrow{R_i}\right)$ is the moment of the tangential reaction of the interaction of the *i*-th active section of the track with the support base relative to the center of mass of the HSTV; x', y' are the coordinates of the center of mass of the HSTV in the *X'Y'* coordinate system; and θ is the rotation angle of the *XY* coordinate system relative to the *X'Y'* coordinate system.

The moment of resistance to the rotation of the *i*-th active section of the track is determined using the following equation [12]:

$$M_{\text{PC}i} = 0{,}0375 \cdot R_{zi} \cdot \mu_{s\max i} \frac{\sqrt{S_{\text{ACT}i}}}{0{,}925 + \dfrac{0{,}15}{k_{fi} \cdot b_{\text{t}}}} . \left(-\operatorname{sign}\left(\omega_z\right)\right), \tag{6}$$

где R_{zi} is the vertical reaction acting on the active section of the track; $\mu_{s\max i} = \left(\mu_{sx\max i} + \mu_{sy\max i}\right)/2$ is the average value of the maximum interaction coefficients in the longitudinal and transverse directions of the active section of the track with the supporting surface; $\mu_{sx\max}$, $\mu_{sy\max}$ are the maximum values of the coefficients of interaction between the active section of the track and the supporting surface in the longitudinal and transverse directions, respectively; b_{t} is the track width; $S_{\text{ACT}i}$ is the area of the active section of the track; and k_{fi} is the actual curvature of the movement trajectory of the active section of the track determined at each modeling step.

In Ref. [13], it was established that the diagram of the distribution of normal reactions along the length of the support branch of a track when moving along a support base of the “dense soil” type is, in many cases, discontinuous. The bulk of the normal load is transmitted through the active sections of the track located under the track rollers. In this case, the links of the track chain located between the rollers do not participate in the transmission of the vertical reaction.

Thus, according to the method of implementing interaction with a support base of the “dense soil” type, a tracked propulsion unit can be investigated by analogy with a wheeled propulsion unit, in which the number of wheels is equal to the number of track rollers. This feature enables the transfer from considering the processes of interaction of the track contour with the support base to considering the interaction with only individual active sections of the tracks. The description of the specified interaction of the track active sections with the supporting base at the locations of the rollers is obtained using a system of equations containing the empirical equation of the friction ellipse to determine the coefficient of the interaction of the track active section with the supporting base following the direction of the sliding speed, the equation of connection between normal and tangential reactions of the interaction of the track sections with the support base, and the equation for determining the slip coefficient:

$$\begin{cases} v_{\text{sl}\,x}{}''_i = \left(v_x - \omega_z y_{ki}\right) - \omega_{\text{DW}} r_{\text{DW}}, \\ v_{\text{sl}\,y}{}''_i = v_y + \omega_z x_{ki}, \\ \sin\varphi_i = \dfrac{v_{\text{sl}\,y}{}''_i}{\sqrt{v^2_{\text{sl}\,x}{}''_i + v^2_{\text{sl}\,y}{}''_i}}, \\ \cos\varphi_i = \dfrac{v_{\text{sl}\,x}{}''_i}{\sqrt{v^2_{\text{sl}\,x}{}''_i + v^2_{\text{sl}\,y}{}''_i}}, \\ \mu_{si} = \dfrac{\mu_{sx\max} \cdot \mu_{sy\max}}{\sqrt{\mu^2_{sx\max} \cdot \sin^2\varphi_i + \mu^2_{sy\max} \cdot \cos^2\varphi_i}} \cdot \left(1 - e^{-\frac{S_{ki}}{S_0}}\right), \\ R_{xi} = -\mu_{si} \cdot R_{zi} \cdot \cos\varphi_i, \\ R_{yi} = -\mu_{si} \cdot R_{zi} \cdot \sin\varphi_i, \\ S_{ki} = \dfrac{\sqrt{v^2_{\text{ск}\,x}{}''_i + v^2_{\text{ск}\,y}{}''_i}}{\max\left(\sqrt{\left(v_y - \omega_z x_{ki}\right)^2 + \left(v_x + \omega_z y_{ki}\right)^2},\ \omega_{\text{DW}} r_{\text{DW}}\right)}, \end{cases} \tag{7}$$

where $v_{\text{sl}\,x}{}''_i$ and $v_{\text{sl}\,y}{}''_i$ are the projections of the sliding speed of the center of the *i*-th active section of the track on the coordinate axes *X''* and *Y''*, respectively; x_{ki} and y_{ki} are the longitudinal and transverse coordinates of the center of the *i*-th active section of the track, respectively; φ_i is the angle of rotation of the sliding speed vector of the *i*-th active section of the track relative to the *X''* coordinate axis; μ_{si} is the coefficient of the interaction of the *i*-th active section of the track with the supporting surface; S_{ki} is slip coefficient of the *i*-th active section of the track; the S_0 is a constant that determines the type of the interaction curve.

The system of equations expressed in Eq. (7) contains the dependencies for determining the projections of the sliding velocities of the corresponding active sections of the track according to Fig. 4.

In the mathematical simulation model, the following equations are used to describe the transmission:

$$\begin{cases} M_{\text{car1}} + M_{\text{car2}} + M_{\text{ICE}} i_{\text{GB}} i_{\text{FD}} \eta_{\text{GB}} \eta_{\text{FD}} = 0, \\ M_{\text{m}} - M_{\text{SCW1}} i_{\text{add}} \eta_{\text{add}} + M_{\text{SCW2}} i_{\text{add}} \eta_{\text{add}} = 0, \\ M_{\text{DW}i} = M_{\text{BCW}i} i_{\text{FD}} \eta_{\text{FD}}, \\ M_{\text{car}i} = \left(k_{\text{SPS}} \eta_i + 1\right) M_{\text{SCW}i}, \\ M_{\text{BCW}i} = k_{\text{SPS}} M_{\text{SCW}i} \eta_i, \\ k_{\text{SPS}} \omega_{\text{BCW1}} - \omega_{\text{m}} / i_{\text{add}} = \left(k_{\text{SPS}} + 1\right) \omega_{\text{car}}, \\ k_{\text{SPS}} \omega_{\text{BCW2}} + \omega_{\text{m}} / i_{\text{add}} = \left(k_{\text{SPS}} + 1\right) \omega_{\text{car}}, \end{cases} \tag{8}$$

where M_{car1} and M_{car2} are the moments on the carriers of the summing planetary gears; M_{ICE} is the moment of ICE; η_{GB} is the GB efficiency; η_{FD} is the FD efficiency; $M_{\text{м}}$ is the torque of the HSSM motor; M_{SCW1} and M_{SCW2} are the moments on the SCW of the summing planetary series; η_{add} is the efficiency of the cylindrical gear transmission of the SCW drive from the HSSM motor shaft; M_{DW1} and M_{DW2} are the moments on the HSTV DW; M_{BCW1} and M_{BCW2} are the moments on the BCW of the summing planetary series; i_{FD} is the FD gear ratio; η_{FD} is the FD efficiency; η_i is the efficiency of the summing planetary gears when the carrier is stopped, considering the direction of power flows; ω_{BCW1} and ω_{BCW2} are the angular velocities of the BCW of the summing planetary series; and ω_{car} is the angular velocity of the carrier of the summing planetary series.

Based on the diagram presented in Fig. 5, for the pressure and return hydraulic lines, the following equations can be derived [14, 15]:

$$\begin{aligned} & q_{\max\text{p}} e_{\text{p}} \omega_{\text{p}} - \operatorname{sign}(p_{\text{pr}} - p_{\text{ret}}) q_{\max\text{p}} \overline{N}_{\text{О н}} \omega_{\text{p}} - q_{\text{м}} \omega_{\text{м}} - \\ & - \operatorname{sign}(p_{\text{pr}} - p_{\text{ret}}) q_{\text{м}} \omega_{\text{м}} \overline{N}_{\text{О м}} + Q_{\text{п pr}} = \frac{V_{\text{f pr}}}{E_{\text{f}}} \frac{dp_{\text{pr}}}{dt}, \\ & -q_{\max\text{p}} e_{\text{p}} \omega_{\text{p}} + \operatorname{sign}(p_{\text{pr}} - p_{\text{ret}}) q_{\max\text{p}} \overline{N}_{\text{О н}} \omega_{\text{p}} + q_{\text{м}} \omega_{\text{м}} + \\ & + \operatorname{sign}(p_{\text{pr}} - p_{\text{ret}}) q_{\text{м}} \omega_{\text{м}} \overline{N}_{\text{О м}} + Q_{\text{п ret}} = \frac{V_{\text{f ret}}}{E_{\text{f}}} \frac{dp_{\text{ret}}}{dt}, \end{aligned} \tag{9}$$

where p_{pr} and p_{ret} are the pressures in the pressure and return hydraulic lines, respectively; ω_{p} is the angular speed of the pump shaft; $\overline{N}_{\text{О н}}$, $\overline{N}_{\text{О м}}$ are the relative volumetric losses in the pump and motor, respectively; $Q_{\text{п pr}}$, $Q_{\text{п ret}}$ are the flow rates through the feed valves of the pressure and return hydraulic lines, respectively; E_{f} is the reduced volume modulus of elasticity of the working fluid; and V_{f} is the volume of the working fluid in the hydraulic line.

The transition from the pressure difference in the hydraulic lines to the torque implemented on the shaft of the pump and the HSSM motor is made according to the following equation:

$$\begin{aligned} M_{\text{p}} &= q_{\max\text{p}} e_{\text{p}} \left(p_{\text{pr}} - p_{\text{ret}}\right) + q_{\max\text{p}} \left(p_{\text{pr}} - p_{\text{ret}}\right) \overline{N}_{\text{М p}}, \\ M_{\text{м}} &= q_{\text{м}} \left(p_{\text{pr}} - p_{\text{ret}}\right) - q_{\text{м}} \left(p_{\text{pr}} - p_{\text{ret}}\right) \overline{N}_{\text{М м}}, \end{aligned} \tag{11}$$

where $\overline{N}_{\text{М p}}$, $\overline{N}_{\text{М м}}$ are the relative mechanical losses in the pump and motor, respectively.

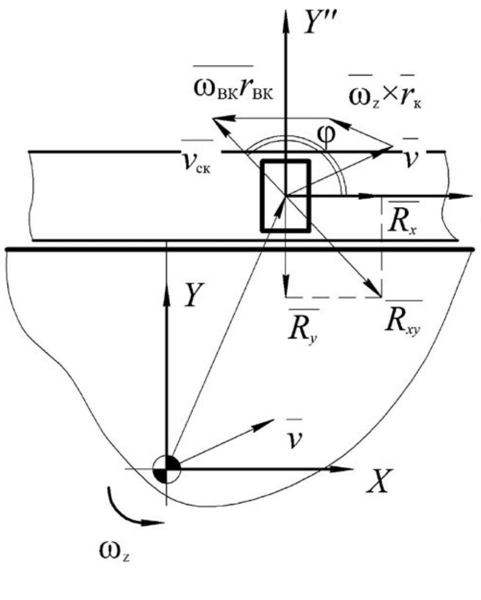

Fig. 4. The analytical model for defining of slip rate of the active part of a track: *XY* – the coordinate frame related to the HSTV's center of gravity; *X″Y″* – the coordinate frame related to the center of the active part of a track; v – velocity of the HSTV's center of gravity; $r_{\text{к}}$ – radius vector of the center of the active part of a track in the *XY* coordinate frame; ω_z – the HSTV's yaw rate relative to the center of gravity; $\omega_{\text{ВК}}$ – rotational velocity of the driving wheel; $r_{\text{ВК}}$ – the driving wheel's radius; $v_{\text{ск}}$ – slip rate of the active part of a track; φ – rotation angle of vector of slip rate of the active part of a track relative to *X″*-axis; R_{xy} – tangential reaction force of interaction between the active part of a track and ground; R_x, R_y – *X*- and *Y*-components of R_{xy} in the *X″Y″* coordinate frame.

Рис. 4. Схема определения скорости скольжения активного участка гусеницы: *XY* – система координат, связанная с центром масс БГМ; *X″Y″* – система координат, связанная с центром активного участка гусеницы; v – скорость центра масс БГМ; $r_{\text{к}}$ – радиус-вектор центра активного участка гусеницы в системе координат *XY*; ω_z – угловая скорость БГМ относительно вертикальной оси, проходящей через центр масс; $\omega_{\text{ВК}}$ – угловая скорость ВК; $r_{\text{ВК}}$ – радиус ВК; $v_{\text{ск}}$ – скорость скольжения активного участка гусеницы; φ – угол поворота вектора скорости скольжения активного участка гусеницы относительно оси координат *X″*; R_{xy} – касательная реакция взаимодействия активного участка гусеницы с опорной поверхностью; R_x, R_y – проекции R_{xy} на оси *X″* и *Y″* соответственно.

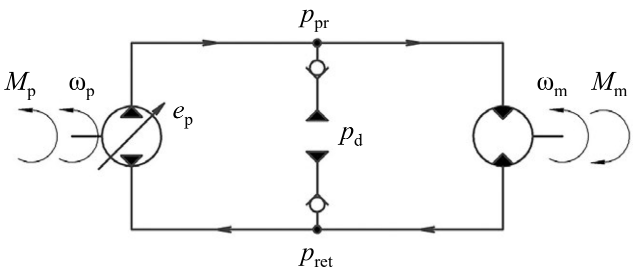

Fig. 5. Hydrostatic drive scheme.
Рис. 5. Схема гидростатического привода.

OPERATING ALGORITHM OF THE CURVILINEAR MOTION CS

The hydraulic pump washer tilt is controlled based on the steering wheel rotation angle (Fig. 6). For this purpose, the ECU recalculates the angle of rotation of the steering wheel into the required curvature of the HSTV trajectory and, further, into the required difference in the speed of the DW. Thus, a given position of the steering wheel is associated with a certain theoretical curvature of the trajectory, regardless of the engaged GB and the rotational speed of the ICE shaft, and the curvilinear movement is controlled like a car [16].

For a car, the characteristic relationship between the steering wheel angle and the theoretical steering curvature can be described using the steering gear ratio relationship (Fig. 7) [17].

To ensure control of the curvilinear movement of the HSTV as a car, a similar dependence of the calculated curvature of the trajectory on the angle of rotation of the steering wheel must be implemented, considering the permissible angles of the steering wheel rotation (approximately ±60°) with a dead zone relative to the central position. Given the limitation of the angle of rotation of the steering wheel by the dead zone relative to the central position, the required value of the trajectory curvature can be determined as follows:

$$\begin{cases} k_{req} = \dfrac{\alpha - \alpha_{dz}}{\alpha_{max} - \alpha_{dz}} k_{max}, \text{ by } \alpha > \alpha_{dz}, \\ k_{req} = 0, \text{ by } \alpha \le \alpha_{dz}, \end{cases} \quad (12)$$

where k_{req} is the required curvature of rotation of the HSTV; α is the current position of the HSTV steering wheel; α_{dz} is the size of the steering wheel rotation dead zone; α_{max}

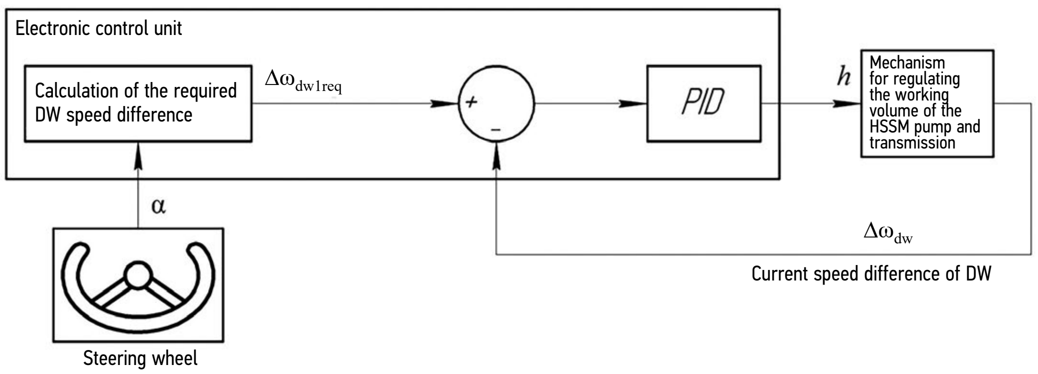

Fig. 6. Structural diagram of control of working volume of the HSSM pump: α – steering handwheel angle; $\Delta\omega_{dw1req}$ – demanded difference of driving wheels' rotation velocities; $\Delta\omega_{dw}$ – current difference of driving wheels' rotation velocities; h – control input by the HSSM pump swashplate.
Рис. 6. Структурная схема управления рабочим объемом насоса ГОМП: α – угол поворота штурвала; $\Delta\omega_{dw1req}$ – требуемая разность угловых скоростей ВК; $\Delta\omega_{dw}$ – текущая разность угловых скоростей ВК; h – управляющее воздействие наклоном шайбы насоса ГОМП.

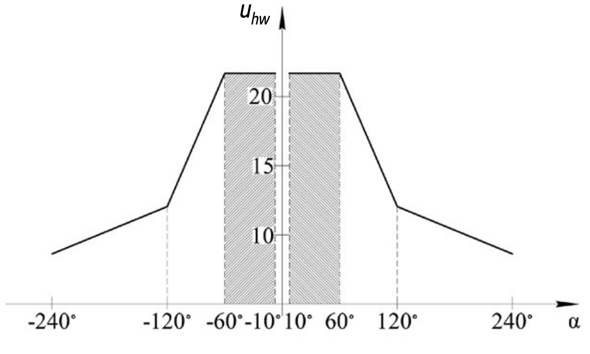

Fig. 7. Steering ratio depending on steering handwheel angle.
Рис. 7. Зависимость передаточного числа рулевого механизма от угла поворота рулевого колеса автомобиля.

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is the maximum angle of rotation of the steering wheel; and k_{max} is the maximum curvature of rotation of the HSTV.

To ensure curvilinear motion with the required curvature, the assessment of the reference values of the rotational speeds of the drive wheels needs to be performed considering the L/B ratio [18], which determines the reduction in the trajectory curvature due to sliding of the tracks. For the required value of the rotation curvature k_{req} the theoretical rotation curvature k_t is determined to be L/B times greater:

$$k_t = \frac{L}{B} k_{req}. \tag{13}$$

In this case, the actual rotation curvature of the HSTV $k_f \approx (B/L) k_t$ will correspond to the required rotation curvature k_{req}. In this case, the difference in the angular speeds of rotation of the tracks of the leading and lagging sides, which must be ensured for turning with curvature k_{req}, considering the Eqs. (1) and (13), is determined as follows:

$$\Delta\omega = k_{req} \frac{\omega_{DW1} + \omega_{DW2}}{2} L. \tag{14}$$

where $\Delta\omega$ is the required difference in the angular speeds of rewinding the side tracks.

The working volume of the HSSM pump, which is necessary to achieve the required curvature of movement and, accordingly, the required difference in the angular speeds of rewinding the tracks (determined considering slipping and skidding, as well as the advancing and lagging sides), can be determined using the following equation:

$$q_p = k_{req} \frac{(1 + k_{SPS}) i_{PTO} i_{add}}{i_{GB} i_{FD} \eta_{vol\,p} \eta_{vol\,m}} \cdot q_м \frac{L}{2}. \tag{15}$$

To ensure the specified working volume of the HSSM pump, the ECU implements the control action h on the mechanism for regulating the working volume of the HSSM pump:

$$h = \frac{q_p}{q_{p\,max}}, \tag{16}$$

where $q_{p\,max}$ is the maximum working volume of the HSSM pump.

Fig. 8 presents the “state machine” of the CS algorithm developed in the MATLAB Simulink environment without considering the processes of gear shifting in the GB.

The four states in the finite state machine of the CS are as follows:

- “straight,” a state corresponding to the HSTV straight-line movement while the steering wheel is in the neutral position;
- “trn_default,” HSTV rotation with a certain radius;
- “trn_b2,” HSTV rotation around the center of mass with the GB shaft braked;
- “trn_emergency,” emergency rotation mode used in case of sensor equipment failures, in which the pump flow is directly regulated by the degree of deviation of the steering control from the neutral position while an unambiguous relationship between the steering wheel rotation angles and the turning radius of the machine is not ensured.

At the initial moment of algorithm operation, the finite state machine is in the “straight” state. The condition for transition to the “trn_default” state is the deviation of the steering wheel from the neutral position. The condition for transition to the “trn_b2” state from the “straight” or “trn_default” state is the fulfillment of the speed limit for the HSTV movement, the activated neutral gear in the GB, and pressing the turn button in place. The return to the “straight” state from the “trn_b2” state occurs when the steering wheel is in the neutral position and one of the conditions for performing a turn in place is not met. The return to the “straight” state from the “trn_default” state occurs when the steering wheel is in the neutral position. A transition from the “trn_default” state to the “trn_b2” state also occurs; the transition condition is

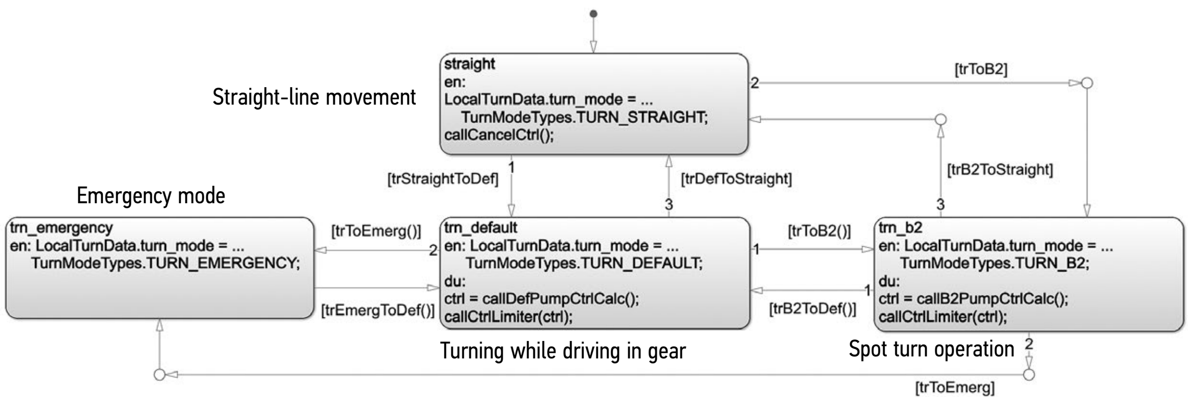

Fig. 8. The finite state machine of the CS in the MATLAB/Simulink software.
Рис. 8. Конечный автомат СУ в среде «MATLAB Simulink».

similar to the condition for exiting from the "trn_b2" state to the "straight" state; however, in this case, the steering wheel must deviate from the neutral position.

The implementation of the algorithm for the operation of the CS with curvilinear motion using the theory of finite state machines enables us to describe the structures of all possible states of the control object (which fully correspond to the HSTV physical states), determine for each state the driving influences and operating modes of the CS, and set the order of switching between the states. Only one state can be active at a time. To change the current operating mode of the control object, the state of the finite state machine needs to be changed. This order of the implementation of the control algorithm enables us to describe visually the control object for the CS and determine the CS reaction to the actions of the operator, which is not possible to do using traditional conditional operators. The implementation of the control algorithm also enables us to check and debug all operating modes and logical transitions of the CS on a virtual model, followed by generating code for the control program in an automated mode and avoiding errors in the ECU control program.

STAGE 1 COMPUTATIONAL EXPERIMENTS

Stage 1 of the method for developing an onboard system for controlling the curvilinear movement of an HSTV aims to verify and confirm the adequacy of the real-time mathematical simulation model of the HSTV curvilinear movement by comparing the results of real-time modeling with the results of modeling a reference model (with a small integration step) under identical control actions. To satisfy the requirement for the model to operate in real time, an integration method and a fixed integration step need to be selected. The integration step must ensure the necessary accuracy of calculations, as well as the speed necessary for working in real time. The computational experiment evaluates the performance of CS algorithms during an S-type maneuver.

Figs. 9 and 10 present the graphs of the changes in transverse and longitudinal speeds when simulating the HSTV motion based on the results of two calculations. The dotted line denotes the calculation with an integration step of 0.005 s, and the solid line denotes the calculation with an integration step of 0.01 s. In these calculations, an integration method based on Newton's method and an extrapolation method based on the current value of the variable and the value of its derivative in the current state were used to calculate the value of the variable at the next integration step.

In this case, a simulation model is considered a reference model, in which the calculation is performed in steps of 0.005 s (at the same time, real-time operation is not performed). By increasing the simulation step to 0.01 s, the requirement for real-time operation is met and the simulation model with the specified integration step needs to be verified.

Figs. 9 and 10 illustrate that the relative calculation error when working in real time (with an integration step of 0.01 s) compared with the reference model (with an integration step of 0.005 s) is insignificant (no more than 5%). The achieved accuracy is acceptable if calculations in real time are required.

STAGE 2 COMPUTATIONAL EXPERIMENTS

Stage 2 of the onboard system development method involves the implementation of control algorithms as part of a physical controller. In this case, the algorithm for controlling the HSTV curvilinear movement was transferred to the HYDAC TTC 580 control unit (ECU). Communication between the ECU and the real-time model operating in the MATLAB Simulink environment was implemented via a CAN 2.0 B interface using CAN-USB adapters.

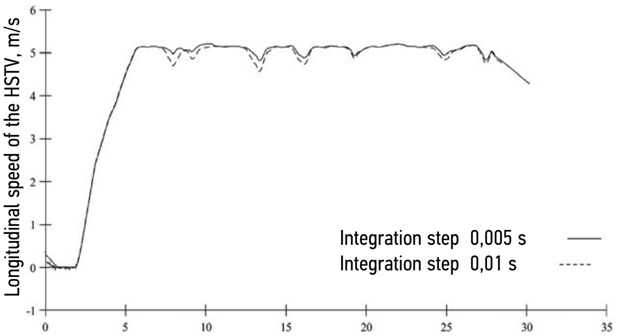

Fig. 9. Time-domain lateral velocity of the HSTV.
Рис. 9. Продольная скорость БГМ в зависимости от времени.

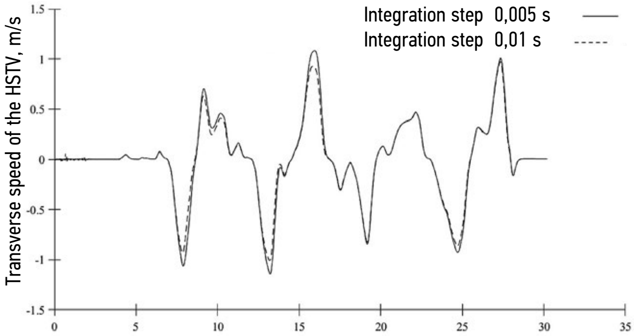

Fig. 10. Time-domain lateral velocity of the HSTV.
Рис. 10. Поперечная скорость БГМ в зависимости от времени.

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The workstation where the prototype CS and the communication between the mathematical simulation models of the dynamics of the curvilinear motion of the HSTV and ECU are implemented is presented in Fig. 11.

The experiment conducted evaluates the performance of CS algorithms on the ECU during the S-type maneuver. The results of the algorithm on the ECU with a mathematical simulation model of the dynamics of the HSTV curvilinear motion in the MATLAB Simulink environment are compared with the results of the algorithm and model in the MATLAB Simulink environment. The results of the experiment are presented in Figs. 12 and 13.

From the analysis of the calculations, we can determine the correlation of the results of the mathematical simulation modeling of the HSTV curvilinear motion when operating the CS algorithms in the MATLAB Simulink environment and directly in the controller.

Thus, based on the results of this work, two stages of the proposed method for developing a CS, which correspond to the operation of CS algorithms in a virtual environment and as part of a physical controller, were demonstrated. In Stage 3, the CS on a prototype HSTV needs to be tested and debugged.

CONCLUSIONS

A method for developing a CS, which enables the development and debugging of a CS in the absence of a prototype HSTV, has been implemented. When using the proposed method, the overall time for developing and debugging CS algorithms is reduced. The applicability of this method is proven by an example of CS development for the curvilinear movement of an HSTV with a dual-flow transmission.

Based on the results of the computational experiments, errors were obtained in determining the longitudinal and transverse speeds of the HSTV when operating algorithms in the controller relative to the operation of algorithms in a virtual environment. The relative error does not exceed 5% in the longitudinal speed of the HSTV and does not exceed 3% in the transverse speed of the HSTV, which is satisfactory for this task.

During the modeling process, when determining the HSTV coordinates, a calculation error accumulates when working in real time. The accumulation of error occurs because of the determination of the HSTV position

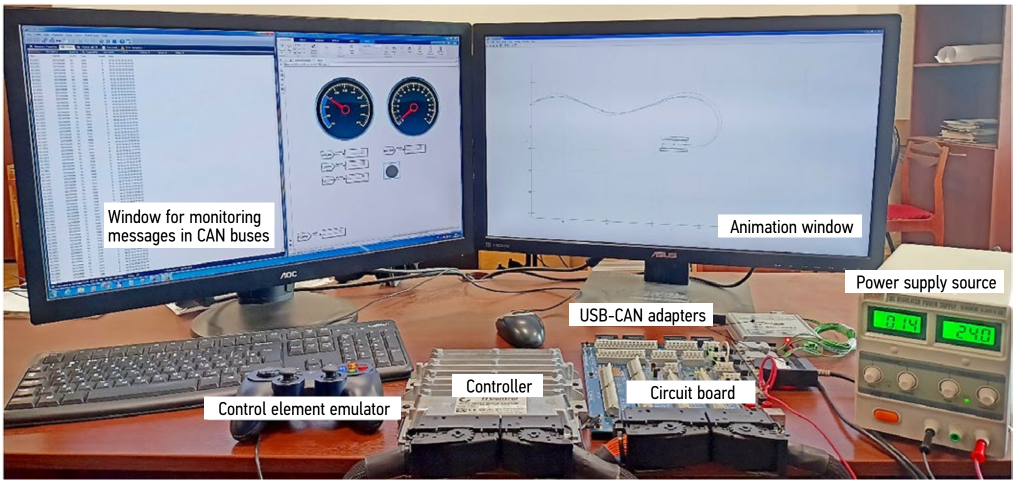

Fig. 11. The setup for working with the CS prototype.
Рис. 11. Установка для работы с опытным образцом СУ.

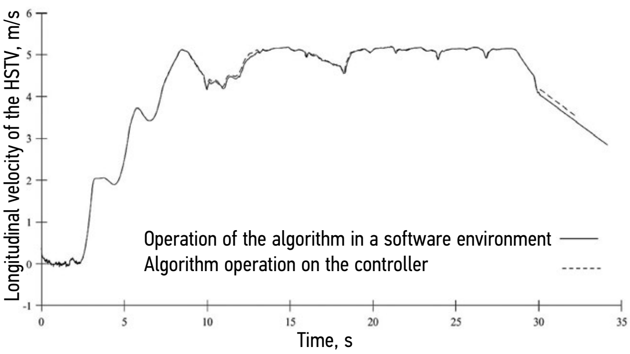

Fig. 12. Time-domain longitudinal velocity of the HSTV.
Рис. 12. Продольная скорость БГМ в зависимости от времени.

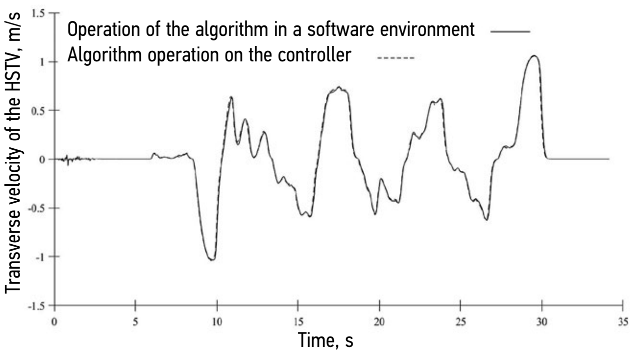

Fig. 13. Time-domain lateral velocity of the HSTV.
Рис. 13. Поперечная скорость БГМ в зависимости от времени.

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by integrating the speed although the specified error is not a criterion for the quality of operation when developing a CS with curvilinear motion. Notably, the HSTV position error, obtained as a result of the gradual accumulation of minor calculation errors, cannot be noticed by the test driver (operator) because the driver needs to make the appropriate compensation during movement. When assessing the HSTV controllability, the test driver visually assesses the speed and direction of the HSTV movement.

ADDITIONAL INFORMATION

Authors' contribution. N.V. Buzunov — search for publications on the topic of the article, writing and editing the text of the manuscript; V.V. Ivanenkov — editing the text of the manuscript; R.D. Pirozhkov — searching for publications on the topic of the article, writing the text of the manuscript, creating images; B.B. Kositsyn — development of the simulation model, expert opinion; G.O. Kotiev — expert opinion, approval of the final version. All authors made a substantial contribution to the conception of the work, acquisition, analysis, interpretation of data for the work, drafting and revising the work, final approval of the version to be published and agree to be accountable for all aspects of the work.

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Источник финансирования. Авторы заявляют об отсутствии внешнего финансирования при проведении исследования.

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