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Estimating operation modes for the individual wheel electric drive of the all-wheel drive vehicle with the use of the driving simulator

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Abstract. With the development of the electric drive systems and vehicle on-board power sources the electric transport is becoming more popular, in particular wheeled vehicles with individually driven wheels, while the question of choosing power plants for this type of transport remains unresolved. In the presented article, the authors illustrate an approach to determining the requirements for power plants of such vehicles by an example of a four-wheel drive car equipped with an individual electric wheel drive. The approach is based on the use of the real-time driving simulator allowing the operator to drive the vehicle along typical routes. The typical routes for the driving simulator are generated from the known statistical data on the excitation inputs during the vehicle motion. Thus, the developed driving simulator provides analysis of the vehicle traction motors operation in the conditions most close to the actual ones and collection of the necessary statistical data which can be used for statement of the requirements for new electric machines which would provide their most effective use on the vehicle.

Currently, the development of the wheeled vehicles with individual drive wheels is getting substantial boost. The increased interest in this type of drive can be explained by the fact that the use of the drive motor provides a number of advantages:

- increased cross-country ability of the wheeled vehicle;
- stability and control improvement;
- better dynamic properties of the wheeled vehicle;
- better comfort for the driver and passengers;
- simplified gearing system as compared to the mechanical drivetrain.

Nowadays, the main disadvantage of this transmission layout is the high weight and cost of the fuel and energy sources, which reduces its competitiveness.

For the high-speed wheeled vehicles the most difficult mode is the upslope motion. In this mode, the normal reactions in the contact patches of the front axle wheels reduce, thus reducing the maximum longitudinal tractive forces in the contact patch. To increase the average speeds it is necessary that the drive motor of the more loaded wheels of the rear axles be able to realize higher torque and power in a wide range of rotation frequencies. Such requirements for the drive motor lead to a significant increase in their size and weight, which means, for the case of rectilinear motion of the transport vehicle, that the transmission developed on this principle will be oversized.

Since the transport wheeled vehicles work in the heavy duty modes only a small amount of their operating time, it is advisable to drive in such modes due to the operation of the drive motor in the short-term operation mode. Thus, in order to state the requirements for the electric machines and to ensure a minimum weight of the power drive it is necessary to collect statistical information about the short-term and long-term operation of the drive motor on the vehicle.

To solve this problem the authors have developed a driving simulator on the basis of the simulation mathematical model of the dynamics of a horizontal curvilinear and rectilinear upslope/downslope motion of a wheeled vehicle [1,2,3,4,5,6].

In the mathematical model [7], the authors consider the dynamics of the wheeled vehicle as the motion of a solid body in a horizontal plane on a planar undeformed ground surface. The dynamics of the wheeled vehicle also consists of the translational motion of the center of mass and the rotational motion around the center of mass (figure 1). The connection of the wheels with the vehicle frame in the vertical direction is considered to be rigid, that is the elastic properties of the suspension are neglected. At the same time, the model takes into account the redistribution of the wheel loads.

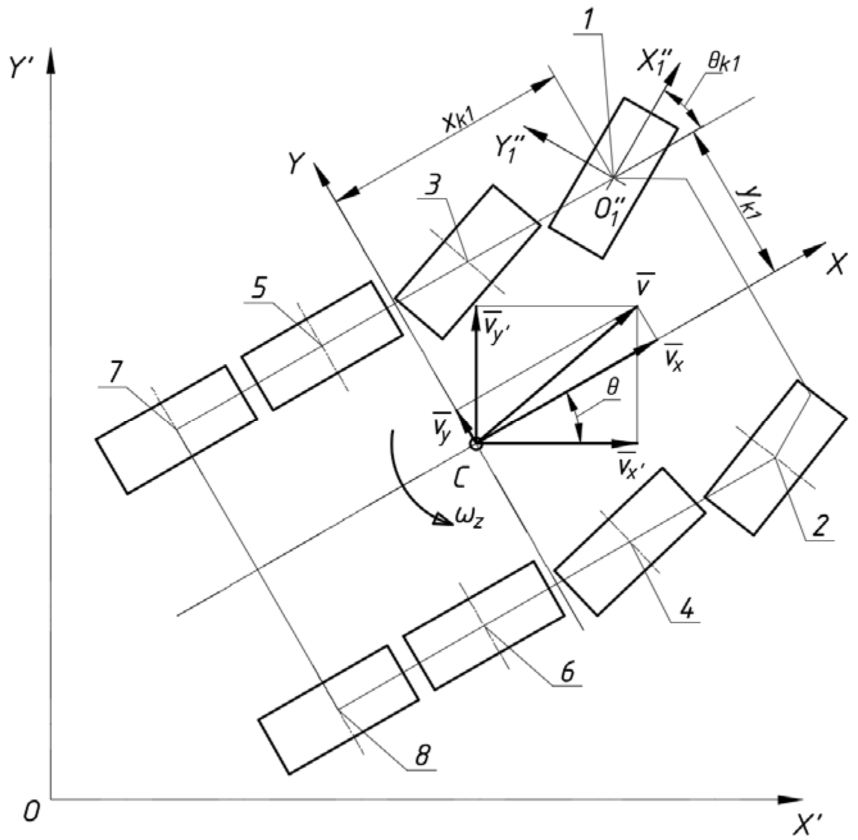

Figure 1. Planar motion of the wheeled vehicle.

According to the presented scheme of the vehicle motion (figure 1), the system of equations describing the planar motion of the wheeled vehicle has form (1). This system allows calculation of the current acceleration of the center of mass from the values of the effective forces and moments acting on the vehicle.

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$$\begin{cases} a_x = \dfrac{dv_x}{dt} - \omega_z v_y = \dfrac{1}{m}\left(\sum_{i=1}^{n} R_{x_i} - P_w\right) \\ a_y = \dfrac{dv_y}{dt} + \omega_z v_x = \dfrac{1}{m}\left(\sum_{i=1}^{\mathrm{n}} R_{y_i}\right) \\ J_z \dfrac{d\omega_z}{dt} = \sum_{i=1}^{\mathrm{n}} M\left(R_{y_i}\right) - \sum_{i=1}^{\mathrm{n}} M\left(R_{x_i}\right) - \sum_{i=1}^{\mathrm{n}} M_{\text{сп}_i}, \\ v_{x'} = \dfrac{dx'}{dt} = v_x \cos(\theta) - v_y \sin(\theta) \\ v_{y'} = \dfrac{dy'}{dt} = v_x \sin(\theta) + v_y \cos(\theta) \\ \omega_z = \dfrac{d\theta}{dt} \end{cases} \tag{1}$$

where

m –vehicle mass; J_z is the vehicle mass moment of inertia relative to the z-axis passing through the center of mass; a_x, a_y – projections of the vehicle center of mass acceleration onto the x - and y-axes; v_x, v_y – projections of the vehicle center of mass velocity onto the x - and y-axes; $v_{x'}$, $v_{y'}$ – projections of the acceleration of the vehicle center of mass onto the x' - and y'-axes; $\frac{dv_x}{dt}, \frac{dv_y}{dt}$ - projections of the relative derivative of the velocity vector of the vehicle center of mass onto the x - and y-axes; ω_z – projection of the angular velocity vector of rotation of the vehicle onto the vertical z-axis; θ - the rotation angle, in the fixed coordinate system; x', y' – coordinates of the vehicle mass center in the coordinate system $x' - y'$; R_{x_i}, R_{y_i}– longitudinal and lateral components of the reaction acting at the contact patch of the *i*-th wheel; P_w– projection of the air resistance force onto the x -axis; $M_{\text{сп}_i}$ – moment of resistance to rotation of the contact patch of the *i*-th wheel around a vertical axis passing through the patch center; n – number of the vehicle wheels.

The force acting between the wheel and the ground in the plane of the road is determined with the use of the approach based on the friction ellipse, according to which the force of interaction of the tire contact patch with the ground is directed opposite to the slip velocity [8, 9, 10, 11, 12].

The steering wheel angle is connected with the angles of the steered wheels according to Ackerman geometry [10] and the steering gear ratio.

The following system of equations describes the connection of the drive motor torque of each wheel with the value of the control parameter and the rotor speed:

$$M_d(\omega_{id}, h) = \begin{cases} M_{id}^{max} \cdot h\,,\ \text{если}\ \omega_{id} < \dfrac{N_{id}^{max}}{M_{id}^{max}}\ \text{и}\ \omega_{id} \geq 0 \\ \dfrac{N_{id}^{max} \cdot h}{\omega_{id}}\ ,\ \text{если}\ \omega_{id} \geq \dfrac{N_{id}^{max}}{M_{id}^{max}}\ \text{и}\ \omega_{id} < \omega_{id}^{max} \\ 0 \quad ,\text{если}\ \omega_{id} \geq \omega_{id}^{max} \end{cases}, \tag{2}$$

where

M_{id}^{max}– maximum traction/braking torque generated by the motor; N_{id}^{max}– maximum traction/braking power of the motor; ω_{id}^{max}– limit rotation frequency of the rotor; ω_{id}– current rotation frequency of the rotor; h – the value of the control parameter ($h \in [-1,1]$).

The presented simulation mathematical model has been used for the development of the real-time driving simulator. The simulator software uses the 4-5 order Runge–Kutta–Fehlberg integrator [11] and can work under conventional MS Windows operating system providing synchronization of computation time and real time by waiting for synchronization [15].

The driving simulator includes:

- a computer which provides operation of the wheeled vehicle model in the real-time mode – determination of the coordinates of the vehicle on the route, calculation of the current speeds and acceleration of the vehicle under the influence of external forces;
- controls which transmit the driver inputs into the simulation model and provide feedback;
- a graphical visual display that allows the operator to estimate the curvature of the upcoming turn, the angle of the slope, driving conditions (the type of the road) in the same way as it is done by the driver of a real vehicle, as well as to receive information about the current parameters of the vehicle motion (speed, traveled distance, etc.).

Figure 2 shows the layout of the driving simulator.

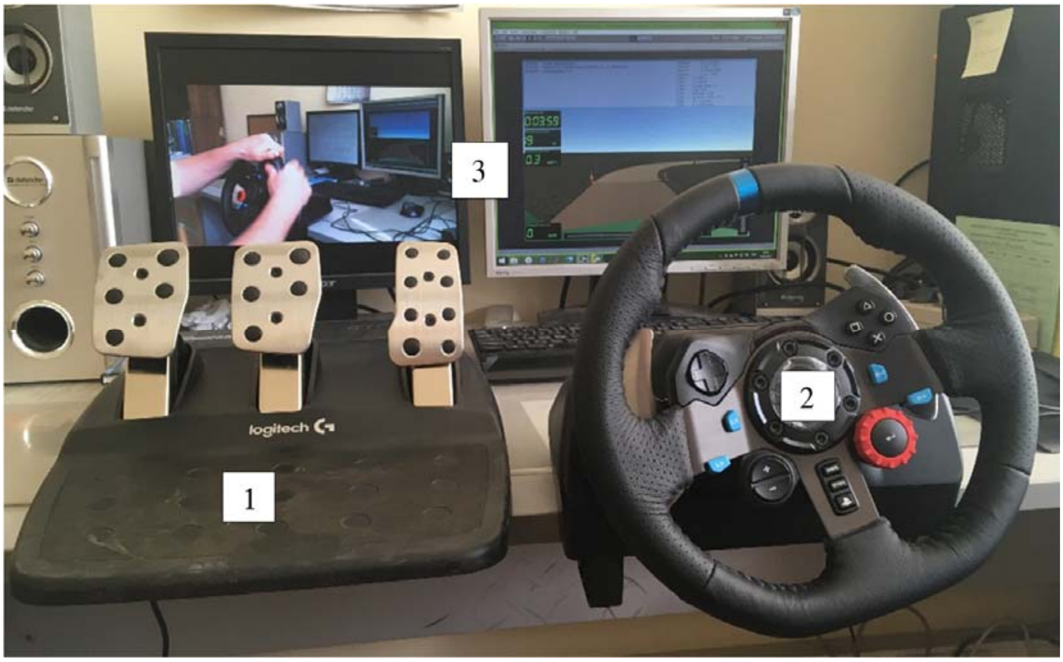

Figure 2. Driving simulator layout:

1,2 – controls; 3 – graphic information output station.

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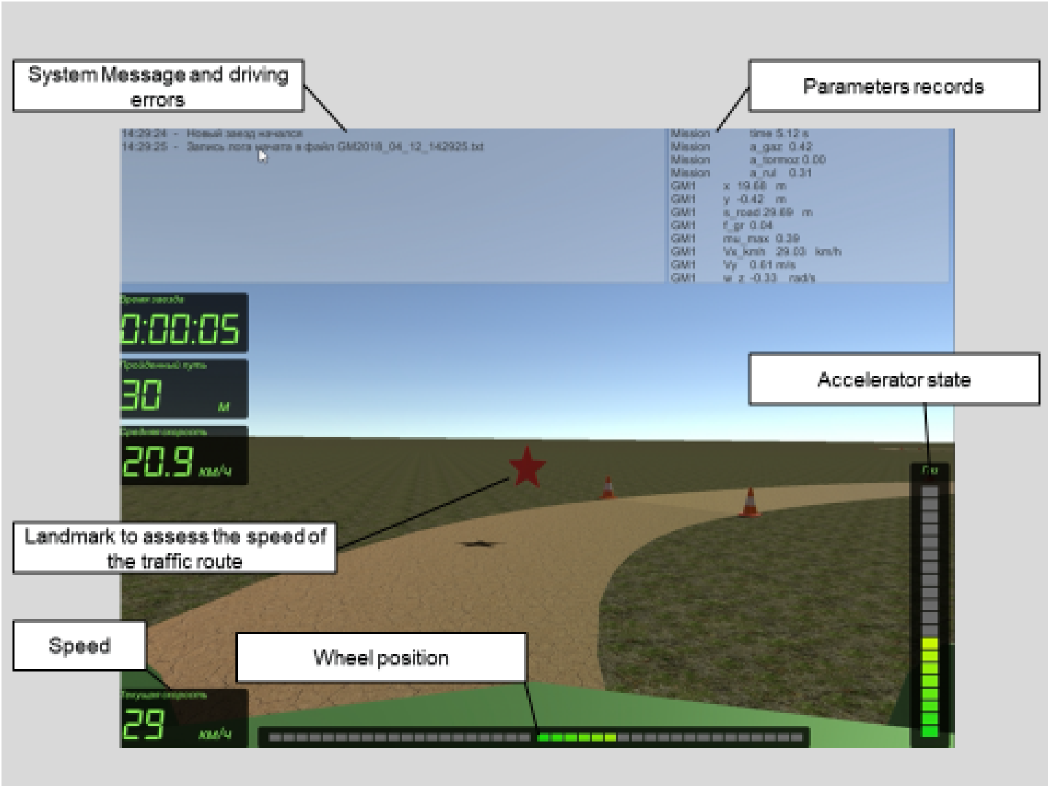

Figure 3. Graphical information output of the driving simulator.

The principle of operation of the presented driving simulator is as follows. The origin of the model fixed coordinate system is superposed with the start point of the route and the center of mass of the vehicle. The operator provides control signals which, by means of the sensors installed on the controls, are transmitted from the operator to the computer and used as the input parameters for the simulation model. During the simulation, according to the control actions and road conditions under each wheel of the vehicle, the model calculates the kinematic parameters of the vehicle (coordinates of the center of mass, wheel speeds, etc.) at a given step on the real-time scale by numerical integration. At the end of the time step the vehicle is transferred in the fixed coordinate system (relative to the start point of the route) to the position calculated during the integration process. The model parameters of interest to the operator (speed, traveled distance, etc.), and the position of the vehicle relative to the typical route are displayed on the simulator screen in a virtual three-dimensional space and on the dials of the virtual dashboard of the simulator. The driving simulator operator, visually receiving information from the screen and feeling the force on the controls, adjusts the control action based on the experience of operating real vehicles and the simulator again starts the process of calculating the kinematic parameters of the vehicle for the next time step.

Figure 4 shows the block diagram of the driving simulator elements interaction.

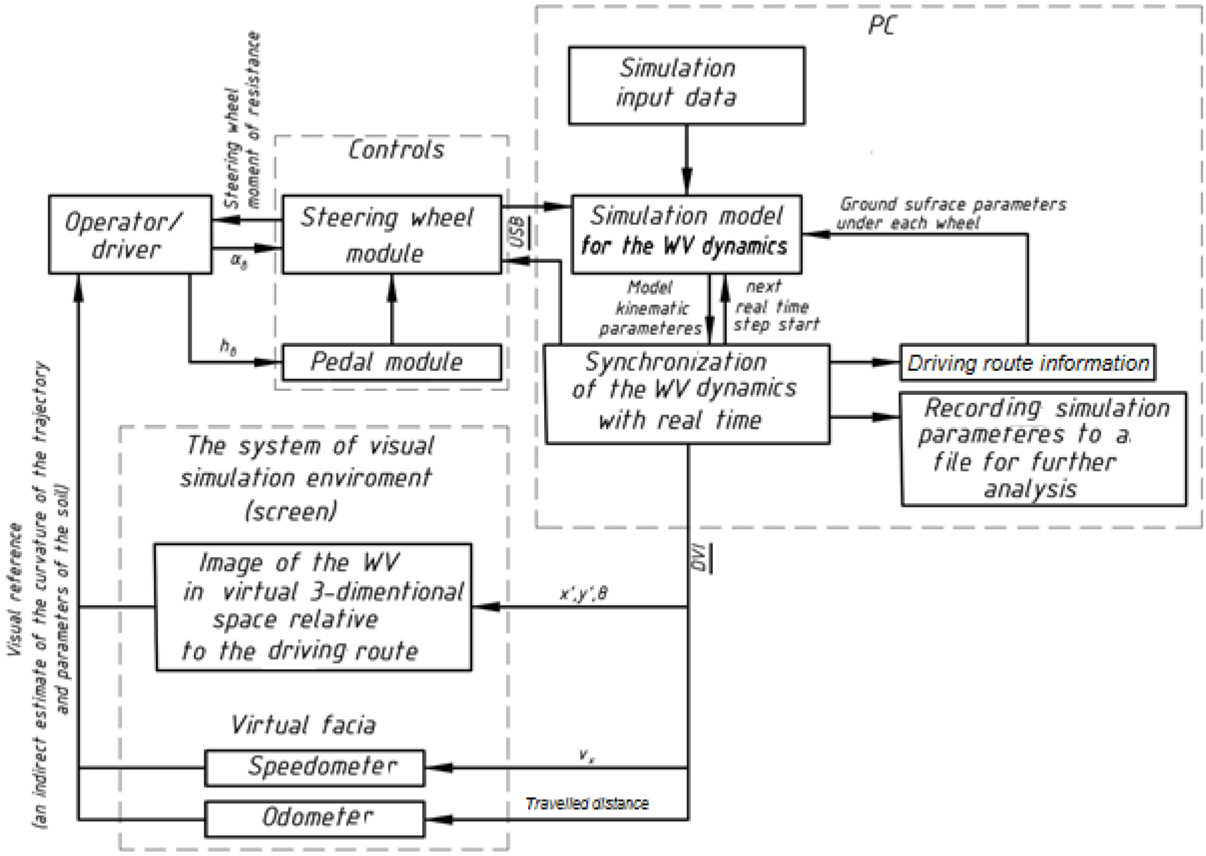

Figure 4. Block diagram of the driving simulator elements interaction (WV – wheeled vehicle).

Typical driving routes were generated with the use of the mathematical statistics methods and well-known probabilistic characteristics of excitation inputs affecting the dynamics of the transport wheeled vehicle. The values of the road curvature k_r, maximum road – tire friction coefficient μ_s_max, and total motion resistance coefficient f_sum were randomly distributed along the driving route with the use of the statistical data on these parameters by the non-canonical representations method [16, 17, 18, 19, 20]. Figure 5 shows an example of the driving route obtained from the realization of the random function of the road curvature $k_d(s)$.

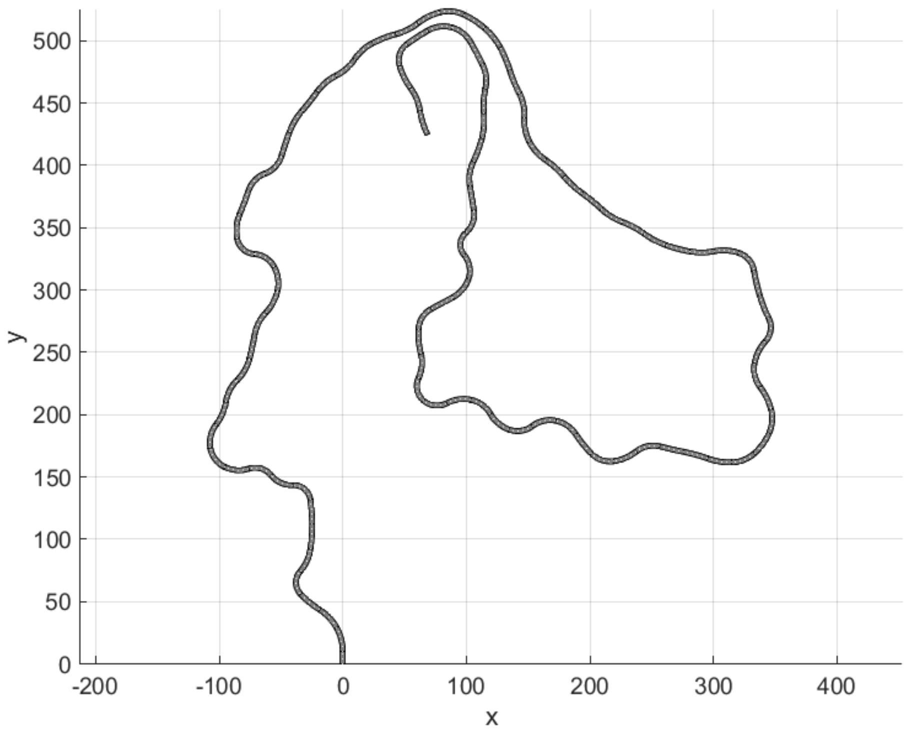

Figure 5. An example of the generated driving route.

To evaluate the required parameters of the drive motor cooling system in the vehicle operation conditions on the route, the corresponding mathematical model was developed (see Figure 6).

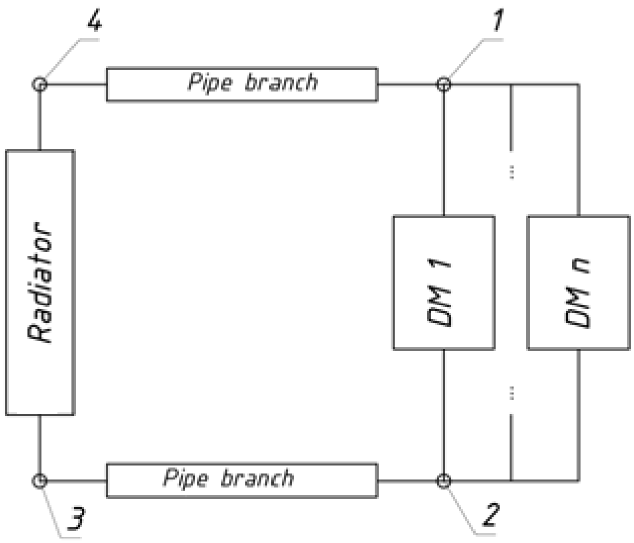

Figure 6. Vehicle cooling system schematic.

The following assumptions for the cooling system model have been made:

- Fluid flow in the cooling system is constant and does not depend on the rotor speed of the drive motor;

- In node 1, the coolant flow is distributed equally between the motors;
- Heat exchange between the cooling liquid and the external environment is carried out only by means of the radiator (there is no heat exchange through the pipe branches);
- The heat capacity and density of the cooling liquid are independent from the temperature;
- All the heat power generated by the drive motor is used to heat the cooling liquid.

The following system of equations describes the cooling system of the vehicle:

$$\begin{cases} c_j \rho_j V_{patr_{41}} \dfrac{dT_1}{dt} = \Phi_4 - \sum_{i=1}^{n} \Phi_{1_i} \\ c_j \rho_j V_{DM} \dfrac{dT_{2_i}}{dt} = \Phi_{1_i} - \Phi_{2_i} + N_{Q_i} \\ c_j \rho_j V_{patr_{23}} \dfrac{dT_3}{dt} = \sum_{i=1}^{n} \Phi_{2_i} - \Phi_3 \\ c_j \rho_j V_{rad} \dfrac{dT_4}{dt} = \Phi_3 - \Phi_4 - N_{rad} \end{cases}, \quad (3)$$

where
c_j – heat capacity of the cooling liquid; ρ_j – density of the cooling liquid; T_1, T_3, T_4– the temperature of the liquid in the respective nodes of the cooling system; T_{2_i} – the temperature the liquid coming from the i-th drive motor in the 2nd node of the cooling system; $V_{patr_{41}}$ – volume of the liquid in the pipe at the segment 4 to 1 of the cooling system; $V_{patr_{23}}$ – volume of the liquid in the pipe at the segment 2 to 3 of the cooling system; V_{DM} – liquid volume in the drive motor; V_{rad}–liquid volume in the radiator; Φ_3, Φ_4 – thermal energy flux in the 3rd and 4th node of the cooling system; Φ_{1_i} – flux of the thermal energy entering the segment of the cooling system containing the i-th drive motor from node 1; Φ_{2_i} – flux of the thermal energy entering the segment of the cooling system containing the i-th drive motor from node 2; N_{rad}– power of the heat exchange between the cooling liquid in the radiator and the environment; N_{Q_i} – thermal power transmitted to the coolant of the i-th drive motor; n – number of used drive motors.

Thus, riding along typical routes it is possible to register parameters of the drive motors and use these parameters for formulation of the requirements for their long-term and short-term operation modes and their cooling system in order to ensure effective use of the electric motors.
An example simulation has been performed for an 8x8 wheeled vehicle riding along a typical 2 km route (Figure 5) on the traffic-compacted soil road (*f_sum*=0,015…0,14, *μ_s_max*=0,31…0,85).

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Table 1. Vehicle specifications.

Gross weight, kg	25000
Wheelbase, mm	4625
Track width, mm	2500
Height of the center of mass, mm	1400
Moment of inertia around the y-axis passing through the center of mass, kg. m2	57580
Tire radius, mm	630
Steering type	four-wheel steering on the first an the second axes
Tire width, mm	500
Distance from the rear axle to the center of mass, mm	2312
Distance from the rear axis to axis 1, mm	4625
Distance from the rear axis to axis 2, mm	3083
Distance from the rear axis to axis 3, mm	1542
Type of transmission layout	Individual electric drive of all wheels.
Steering gear ratio	10
Drive motor maximum power , kW	62,5 x 8
Drive motor maximum torque, Nm	1000
Drive motor speed at the maximum torque, rpm	4000

Figure 7 shows the time histories of the actual and average speed of the vehicle driving on the simulated route.

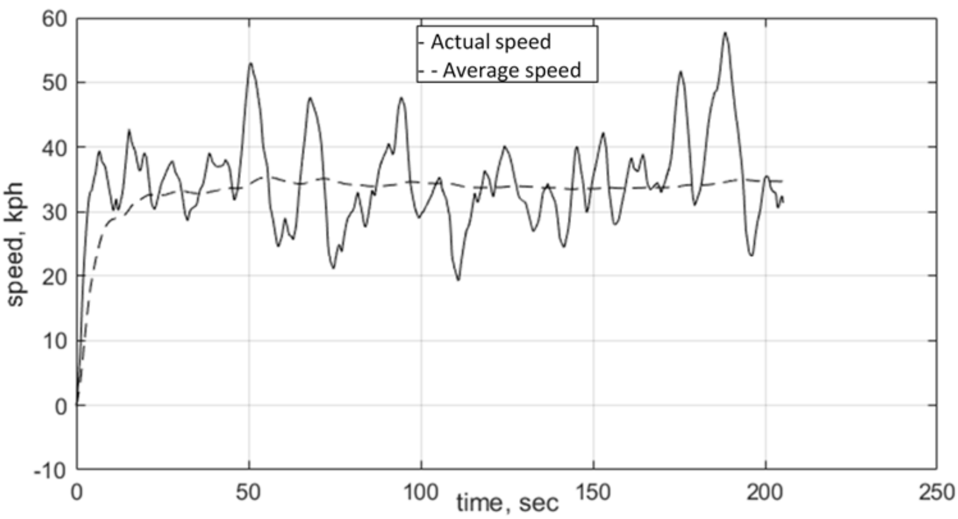

Figure 7. Actual and average speed of the vehicle on the simulated route.

It can be seen from the simulation results that the vehicle average speed for this type of the road is close to the steady-state value of 34.7 km/h. For this reason it can be concluded that the length of the route is sufficient and the simulation can be stopped since further statistics collection will not have an impact on the result.

Figures 8 and 9 show the time histories of the torques and rotor speeds of the drive motors for a typical route.

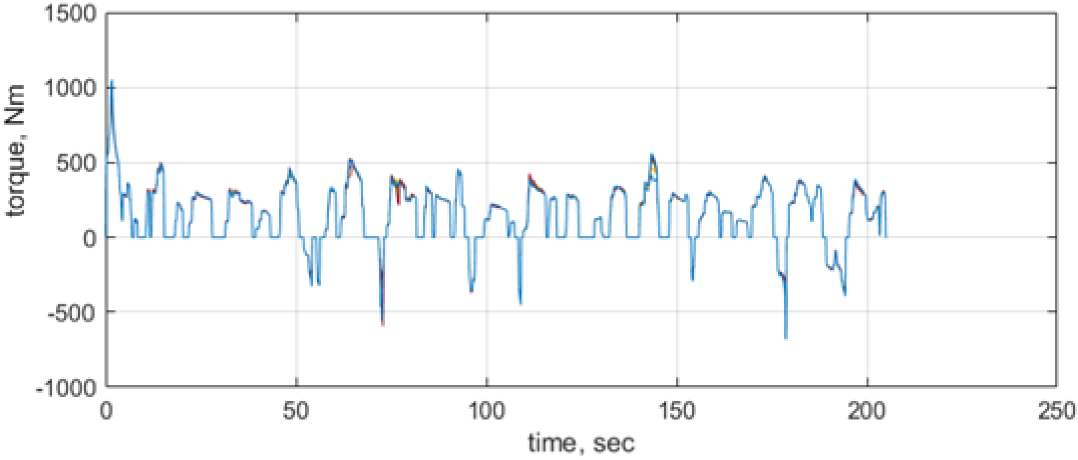

Figure 8. Torques of the drive motors on a typical route.

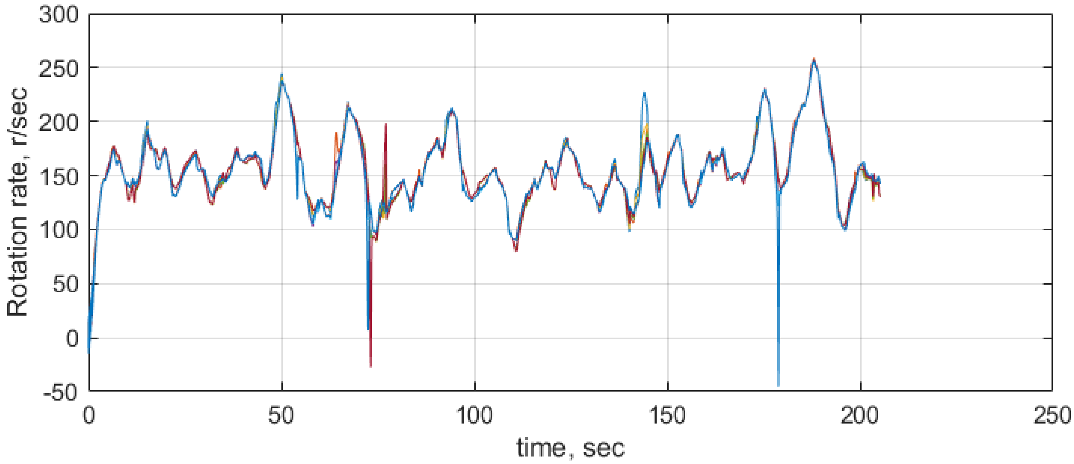

Figure 9. Rotor speeds of the drive motors on a typical route.

Figure 10 shows the time history of the coolant temperature in the cooling system nodes when driving on a typical route. The liquid flow rate in the system is taken to be 40 l/ min, the volume of liquid inside the drive motor $V_{DM} = 1$ l, the pipes ${V_{patr}}_{41} = 0{,}5$ l, ${V_{patr}}_{23} = 0{,}5$ l and the radiator $V_{\rm rad} = 3$ l.

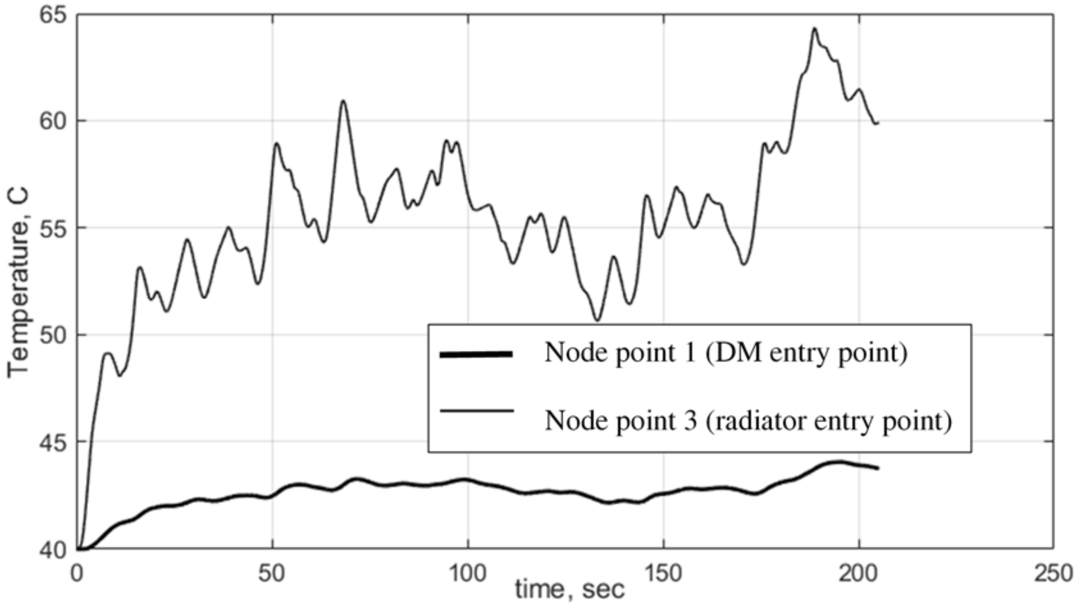

Figure 10. Coolant temperature for two node points of the cooling system.

The simulation results show that the drive motors used on the tested vehicle provide average speed 34.7 km/h on a typical route, and the cooling system provides the temperature of the cooling liquid exiting the drive motors no more than 65°C.

The proposed approach to the determination of the required parameters of the fuel and energy of transport vehicles with individual wheel drive allows to take into account the design features of the vehicle, road and ground conditions and qualification of the drivers. The data from the developed driving simulator make it possible to provide the desired torque – speed characteristics of the drive motor, rational gear ratio of the transmission, coolant consumption, the area of the radiator, etc at the early stages of the wheeled vehicle development.

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