A theoretical study on the high-speed electric tracked vehicle mobility

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Abstract. The article deals with the problem of the characteristics selection for the electric motors of the drivetrains of the high-speed tracked vehicles with individually driven sprockets. The existing method of the tracked vehicle cornering dynamics analysis is not suitable for the analysis of the motion at the speeds near to the critical speeds in the skidding mode. The authors offer a method for improving the accuracy of the existing analytical dependencies by the use of neural network. The above-mentioned improvements provide estimation of the limit values of the required traction forces on the sides of the vehicle but they do not allow gathering statistical data on the electric motor operating conditions of short and long duration during the vehicle operation. The authors offer to solve this problem by means of a driving simulation complex. The complex provides simulation of the tracked vehicles driving along stochastic routes in real time mode and collection of the stochastic data on the electric motor loads.

Introduction

Today the electromechanical transmission for a tracked vehicle with individual driving sprockets is the most common choice for mobile robots and superlight tracked vehicles. Individual drive mode can be one of the operation modes for complex hybrid parallel electromechanical transmission. Individual drive is also a good option for articulated tracked vehicles with separately driven sections. Electric transmissions with individually driven wheels can also be used for wheeled vehicles to improve their agility.

The most heavy-duty working mode for a tracked vehicle with this type of transmission is turning. When the vehicle moves along a curved path, up to 100% of the overall thrust power could be used by the electric motor of one track, with the other electric motor switching into generating mode and creating the breaking force on the second track of the vehicle. To increase the average speed, one must provide steering of the tracked vehicle at the speeds close to the critical speeds at drifting. This demands high torque and power of the electric motor of the outside track, thus leading to increase in its mass and size. Besides, this power is not used fully during a straight-line motion.

Description of the mathematical model

Table 1 shows formulae for determination of needed power on outside track and inside track of the vehicle during a turn with radius $R_f$ at speed $V$. Fig. 1 shows the corresponding schematic diagram.
Figure 1. Forces and velocities of a tracked vehicle during a turn.

Table 1. Tracked vehicle thrust calculation formulae.

| Formula | Description |
|---------|-------------|
| 1. Theoretical turning radius $R_t$: |
| $R_t = R_f \frac{B}{k \cdot L}$, |
| where $B$ — track width; $L$ — wheelbase length; $k$ — proportional coefficient (individual for every vehicle) |
| 2. Nikitin’s formula for turning resistance $\mu$ [1]: |
| $\mu = \frac{\mu_{max}}{0.925 + 0.15 \frac{R_t}{B}}$, |
| where $\mu_{max}$ — maximum turning resistance coefficient for particular soil |
| 3. Critical speed before vehicle drift, $V_{crit}$: |
| $V_{crit} = \sqrt{\mu_{max} \cdot g \cdot R_f}$, |
| where $g$ — acceleration of gravity |
| 4. Outside track speed $V_2$ and inside track speed $V_1$: |
| $V_2 = V \left(1 + \frac{B}{2R_t}\right)$; $V_1 = V \left(1 - \frac{B}{2R_t}\right)$, |
| 5. Required thrusts on outside track $P_2$ and inside track $P_1$ [2]: |
| $P_2 = \frac{m \cdot g}{2} f_{gr} \left(1 + \frac{2V^2 \cdot H_z}{B \cdot R_f \cdot g} + \frac{\mu \cdot L}{2B} \left(1 - \left(\frac{V}{V_{crit}}\right)^4\right)\right) + \frac{m \cdot V^4 \cdot L}{4 \cdot R_f^3 \cdot \mu \cdot g}$; |

End of table 1
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\[ R_i = \frac{m \cdot g}{2} \left( f_{gr} \left(1 - \frac{2V^2 \cdot H_z}{B \cdot R_f \cdot g} \right) \frac{\mu \cdot L}{2B} \left(1 - \left( \frac{V}{V_{crit}} \right)^4 \right) \right) + \frac{m \cdot V^4 \cdot L}{4 \cdot R_f ^2 \cdot \mu \cdot g}, \]

where \( m \) – vehicle mass; \( f_{gr} \) – coefficient of resistance to straight-line motion over particular soil; \( H_z \) – vehicle center of mass height;

6. Track efficiency \( \eta = 0.95 - 0.005 \cdot 3.6 \cdot V \);

7. Required power for outside track \( N_2 \) and inside track \( N_1 \)

\[ \eta_2 = 0.95 - 0.005 \cdot 3.6 \cdot V_2; \]
\[ \eta_1 = 0.95 - 0.005 \cdot 3.6 \cdot V_1; \]
\[ N_2 = \frac{P_2 \cdot V_2}{\eta_2}; \]
\[ \left\{ \begin{array}{l} N_1 = \frac{P_1 \cdot V_1 \cdot \eta_1}{\eta_1}, \text{if } P_1 < 0 \\ N_1 = \frac{P_1 \cdot V_1}{\eta_1}, \text{if } P_1 > 0 \end{array} \right. \]

As one can see in the presented formulae it is crucial to know proportional coefficient \( k \) between theoretical and true turning radius, and current amount of resistance to turning – coefficient \( \mu \). Nikitin’s formula for \( \mu \) was derived from a number of experiments, and the proportional coefficient \( k \) is a function of speed. Due to this fact, these formulae for stationary modes of motion are not suitable for studying critical modes close to drifting with satisfying accuracy. Besides, with this formulae it is not possible to estimate stability and control of the tracked vehicle.

Solution to the problem of unsteady movement parameters calculation could be numerical simulation using mathematical models [4, 5, 6, 7, 8, 9, 10].

Figure 2. Planar motion of a tracked vehicle.

In this study, mathematical models [11, 12, 13, 14, 15, 16, 17] were developed where the un-steady motion of the tracked vehicle was simulated as the motion of a rigid body in horizontal plane on a
planar rigid surface. Translational and rotational motion of the vehicle center of mass was considered as shown in figure 2. Translational connection of the body and the track rollers was rigid (suspension stiffness is not considered), but redistribution of normal reactions was taken into account under all track rollers due to center mass acceleration, track rollers rolling resistance and air resistance forces.

Equations of motion for this vehicle model (1):

\[
\begin{align*}
    a_x &= \frac{dv_x}{dt} - \omega_z v_y = \frac{1}{m} \left( \sum_{i=1}^{n} R_{x_i} - P_w \right) \\
    a_y &= \frac{dv_y}{dt} + \omega_z v_x = \frac{1}{m} \left( \sum_{i=1}^{n} R_{y_i} \right) \\
    J_z \frac{d\omega_z}{dt} &= \sum_{i=1}^{n} M_{spi} - \sum_{i=1}^{n} M_{R_{x_i}} - \sum_{i=1}^{n} M_{R_{y_i}},
\end{align*}
\]

(1)

where \( m \) — vehicle mass; \( J_z \) — vehicle inertia around vertical axis \( z \); \( a_x, a_y \) — center of mass acceleration projections onto axes \( x-y \); \( v_x, v_y \) — center of mass velocity projections onto axes \( x-y \); \( v_{x'}, v_{y'} \) — center of mass velocity projections onto axes \( x'-y' \); \( \frac{dv_x}{dt}, \frac{dv_y}{dt} \) — center of mass relative derivative of velocity projections onto axes \( x-y \); \( \omega_z \) — center of mass vertical rotational velocity projection onto axis \( z \); \( \theta \) — rotation angle of vehicle in ground coordinate system; \( x', y' \) — center of mass coordinates in coordinate system \( x'-y' \); \( R_{x_i} \) — longitudinal reaction, acting on the active part of the track under the \( i \)-th roller; \( R_{y_i} \) — transversal reaction, acting on the active part of the track under the \( i \)-th roller; \( P_w \) — air resistance force, projected onto \( x \) axis of the coordinate system \( x-y \); \( M_{spi} \) — moment of resistance to turning for the part of the track under the \( i \)-th roller; \( n \) — number of rollers.

This system of equations is suitable for vehicle’s center of mass accelerations determination using forces and moments acting on the vehicle.

Forces under a track roller in the active part of the track are determined from the friction ellipse [18, 19, 20, 21]. Surface reaction force in the active part of the track is directed opposite to the slip velocity.

Shown mathematical model improves analytical steady mode formulae for a tracked vehicle in the case of unsteady critical motion close to drifting. Nikitin’s formula for resistance to turning and proportional coefficient \( k \) were replaced by a neural network that estimates modeling data for the tracked vehicle with different \( L/B \) range during a turn for a particular range of speeds and turning radii. Comparison of the analytical solution and numerical modelling is shown in figure 3. All calculation data are given for GM-569 tracked vehicle.
Resistance to turning in the mathematical model:

\[
\mu = \frac{4 \left( \sum_{i=1}^{n} M(R_{zi}) \right) + \sum_{i=1}^{n} M(R_{zi}f_{cp}) - \frac{mV^2H_{z}}{R_f f_{gr}}}{mg \cdot L \left( 1 - \left( \frac{V}{V_{crit}} \right)^4 \right)},
\]

where \( R_{zi} \) — normal reaction in the active part of the track under the i-th roller.

Simulation results estimate maximum required moments and power on driving wheels (figure 4), and allow estimation of the required thrust for maximum agility of the vehicle (turning at forward speed \( 0.8V_{crit} \)), as shown in figure 5.

However, presented model does not estimate stochastic data on the short-term and long term modes of electric motor duty cycles and therefore it is not suitable for modelling vehicles with individual drive of all rollers and wheels.
Figure 4. Estimation of required power and thrust/braking forces on each side of the vehicle

\[ \mu_{\text{max}} = 0.85; f_{\text{soil}} = 0.07; L/B = 1.6733 \]
Figure 5. Required vehicle thrust performance curve for obtaining agility when turning at forward speed of $0.8V_{crit}$.

Designing of the driving simulator

To solve this problem the authors designed a driving simulator based on the presented mathematical model. The simulator is capable of real-time simulation and allows studying vehicle reaction to human driver’s commands while simulating motion over different surfaces built on the base of the stochastic data on their properties, trajectory curvature, coefficient of friction, and coefficient of resistance to straight-line motion.

In order to maintain stable driving the authors developed a controller allowing connection of the driver’s commands (accelerator and brake pedal positions, steering wheel angle) with control parameters of the electric motor.

Figure 6. Driving simulator workspace.

Gathered vehicle motion statistics with information on electric motors work state allows to estimate the duty cycles of the motors (including short-term overloads), required zone of motor’s high efficiency, maximum efficiency curve for desired average vehicle speed, an also to formulate requirements for the cooling system of the motors.

Figure 7A shows a prototype tracked vehicle built at Bauman Moscow State Technical University. The prototype features diagonal driving wheels. Due to this fact, it is possible to study different types of electric transmissions. The prototype is equipped with complex measuring system capable to store data from:

- torques on the drive shafts on both sides of the vehicle;
- rotational speeds of the driving wheels;
- longitudinal and transversal vehicle accelerations;
• rotational speed of the vehicle about its vertical axis.
An example of the experimental measurements is shown in figure 8.

Figure 7. Prototype tracked vehicle (m=3000 kg; L/B=1,78).

Figure 8. Turning around with one track stopped. Experimental measurements.

Conclusion
As was determined in performed research, the transmission with individual drive of the driving wheels has significant disadvantages. At present, engineers of Bauman Moscow State Technical University
are studying different types of electromechanical transmissions (figure 9) including the ones with individual drive of all tracked wheels.

![Types of electromechanical transmissions](image)

**Figure 9.** Types of electromechanical transmissions:
- a) — two side electric motors;
- b) — dual way electromechanical transmission;
- c) — three way electromechanical transmission;
- d) — electromechanical transmission with central planetary gear.

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