Design of the double-jointed multi-tracked vehicle steering control law providing its motion along a reference trajectory

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Abstract. Multi-tracked vehicles with two articulation joints is a good option for transportation of large indivisible cargoes in the Far North. Steering such vehicles is a difficult task for the operator and requires a high degree of skill. The authors propose automating traffic control systems or using unmanned vehicles that follow a reference trajectory, for example, in front of the moving vehicle (the leader), to provide safe cargo transportation. The paper considers the synthesis of the control law of the multi-tracked vehicle steering which would provide following a reference trajectory.

1. Introduction

The use of articulated tracked vehicles of various configurations is a rational solution for transportation of large heavy payloads in the Far North conditions. The use of multi-tracked vehicles with two saddle joints and cab over engine layout [1] (Figure 1) is a widespread option.

![Figure 1](image1.png)

Figure 1. Tracked off-road vehicles: a) Foremost Chieftain D; b) Foremost Husky 8; c) Ural – 5920

Steering of such large vehicles is a complex task and requires a high level of the operator skill. In this context, it is proposed to automate traffic control or to use unmanned multi-tracked vehicles that move along a reference trajectory, for example, along the trail of the leading vehicle. This paper focuses...
on the development of such a steering control law for vehicles of the presented type that would ensure their following a reference trajectory.

It is accepted for the purpose of solving the steering control problem of a double-jointed multi-tracked vehicle that the tractive capabilities of the vehicle are sufficiently provided, i.e. the maneuvering of the vehicle under insufficient power conditions is not addressed.

As an example, we shall consider a double-jointed multi-tracked vehicle with the following technical characteristics (Table 1).

Table 2.1 – Parameters of the multi-tracked vehicle

| Parameter                                | Value       |
|------------------------------------------|-------------|
| Gross weight \( m \), kg                 | 130000      |
| Distance between unit joints \( L \), m  | 9           |
| Track wheelbase \( L_m \), m             | 5.13        |
| Track width \( B \), m                   | 2.5         |
| Height of center of mass \( H_z \), m    | 2.5         |
| Number of bogie roller, \( n_k \)        | 7           |
| Bogie roller radius \( r_k \), m         | 0.35        |
| Driving wheel radius \( r_{gh} \), m     | 0.35        |
| Track unit maximum pivot angle \( \alpha_{max} \), degree | 45 |

2. Statement of the problem of the transport vehicle driving along a reference trajectory

One of the most typical tasks of unmanned driving is to ensure that the vehicle follows a reference trajectory.

The trajectory can be obtained by processing the coordinates of the lead vehicle [2], or it can be synthesized based on the location of the destination and obstacles detected by vision [3].

Vehicle accurate positioning on the reference trajectory and its plotting can be provided by the GPS and GLONASS navigation systems. It is known that the accuracy of the commercial GPS receivers is about 6 – 8 meters, which is obviously unacceptable. In this context, to solve the problem of the unmanned motion of the vehicle along a reference trajectory, it is possible to apply differential correction systems using data from alternative geostationary (base) sources, the reference coordinates of which are known with high accuracy, to improve the accuracy of satellite positioning. In this case, the position of the vehicle relative to the trajectory can be determined with the accuracy to 5 – 10 cm.

When the vehicle follows the reference trajectory, the position error \( e_r \) and trajectory angle error \( e_{\theta} \), are analyzed on the basis of the data obtained in this way, from which the control action for the drives of the transport vehicle is determined [4]. Calculation of these errors for a multi-tracked vehicle with two articulation joints is shown in Figure 2.
Figure 2. A multi-tracked vehicle with two articulation joints is driving along a reference trajectory

For implementing the vehicle control algorithm, we choose a platform point which is used for calculation of the distance error $e_r$ and the angle error $e_\theta$. As the results of computational experiments have shown, this point should be located at a distance $x_0$ from the geometric center of the multi-tracked vehicle platform. In this paper, the point was selected on the pivot axis of the front track unit, while varying its position on the platform in some cases can improve control results on the route.

The ability to rotate each of the transport vehicle's track unit individually makes the solution to the trajectory following problem ambiguous. Obviously, the additional degree of freedom offers several advantages during curvilinear motion – improved performance properties such as controllability and steerability, as well as improved energy efficiency.

3. Structure of the steering controller of a multi-tracked vehicle with two articulation joints

As a first approximation, steering of a multi-tracked vehicle with two articulation joints is analogous to the steering of a two-axle wheeled vehicle with the both steerable axles. Such vehicles can provide the necessary curvature of the trajectory at different positions of the steering pole. In this regard, in order to synthesize a steering control law, it is necessary to identify the characteristic driving modes of the vehicle type in question: turning with a minimum radius and parallel movement ("duck walk"). Thus, it is obvious that in case of significant deviation of the vehicle from the reference trajectory, it is convenient to use a "duck walk" for its elimination. In the case of a significant heading angle error, it is advisable to shift the steering pole to the center of the vehicle's base when steering to eliminate it as quickly as possible. Thus, depending on the position of the vehicle relative to the trajectory, the quickest (taking less time, distance or energy) return to the trajectory can be achieved by a combination of these driving modes. Obviously, the compensation of the position error by parallel movement of a double-jointed tracked train ("duck walk") is provided by turning the track units by the same angle in the direction of its reduction. The angular error is eliminated by turning the vehicle with the smallest possible radius by rotating the track units in opposite directions relative to the longitudinal axis of the vehicle. That is, the elimination of angular deviation prevents the elimination of linear deviation and vice versa. That is, the elimination of angular deviation prevents the elimination of linear deviation and vice versa.

In order to solve the above contradiction, the following algorithm for system operation is proposed – first the angular deviation is eliminated, and then the linear deviation is eliminated. In accordance with
the specified requirements, a steering controller for a double-jointed multi-tracked vehicle has been developed, the block diagram of which is shown in Fig. 3.

Figure 3. Block diagram of a double-jointed multi-tracked vehicle steering controller

In this scheme, if the control action of the PID controller \( PID_r \), which compensates for the angular deviation of the vehicle from the motion path, is present, the control action of the linear deviation controller \( PID_\theta \) is reduced (by multiplying by a factor less than unity). That is:

\[
\begin{align*}
\alpha_1^d &= (1 - |PID_\theta(e_\theta)|) \cdot \alpha_{\text{max}} \cdot PID_r(e_r) - \alpha_{\text{max}} \cdot PID_\theta(e_\theta); \\
\alpha_2^d &= (1 - |PID_\theta(e_\theta)|) \cdot \alpha_{\text{max}} \cdot PID_r(e_r) + \alpha_{\text{max}} \cdot PID_\theta(e_\theta),
\end{align*}
\]

where \( \alpha_1^d, \alpha_2^d \) are the required track units steering angles relative to the longitudinal axis of the vehicle to follow the reference trajectory; \( PID_\theta(e_\theta) \) is a controller that determines the control action based on the angular deviation from the trajectory \( e_\theta \); \( PID_r(e_r) \) is a controller that determines the control action based on the linear deviation from the trajectory \( e_r \); \( \alpha_{\text{max}} \) is the maximum track unit pivot angle. The resulting track unit turning angles \( \alpha_1^d, \alpha_2^d \) can be achieved either by driving the turntables which generate the turning moment between the bogie and the vehicle platform (kinematic steering method), or by creating different traction forces on the tracks (power steering method).

In the case of turning circles (as implemented in existing designs by means of hydraulic cylinders, Figure 1.1) PID controllers can also be used to realize the required angles. In case the turning moment is generated due to the difference of traction forces on the tracks, besides the above discussed controller (fig. 3.1) an additional one is required which would provide steering of each track unit by this difference of the traction forces.

At the same time, it is necessary to note, that traction control on the leading wheels of the track unit in this case determines not only its angular position relative to the platform (\( \alpha_1 \) and \( \alpha_2 \) in fig. 2.1), but also speed of the vehicle as a whole. In this case, the priority of the vehicle speed maintenance in relation to the implementation of the track unit angle should be the lowest, which will ensure the required curvature of the vehicle trajectory in conditions of insufficient power of the traction motors.

In such a formulation, the problem of controlling a single-track unit is similar to the problem of controlling a single tracked vehicle. Thus, in [5], the structure of the regulator designed to maintain a given speed and turning radius of the tracked vehicle was developed and investigated (taking into account the feature that when the power required to maintain a given mode of motion is lacking, the turning radius is maintained first of all).

Let us consider the variant of providing curvilinear movement of the vehicle solely by means of the difference of the corrective forces on the tracks of the bogies. In this regard, on the basis of the solution presented in the article [5], a control regulator for each track unit was developed, the block diagram of which is shown in Fig. 3.2.

This controller calculates the control actions based on the position of the vehicle speed control \( h_v \) and \( PID_\alpha \), which ensures that the desired \( \alpha_1^d \) and the actual \( \alpha_1 \) angle of rotation of the track unit relative to the vehicle's longitudinal axis are matched:

\[
h_R = h_v - PID_\alpha(\alpha_1^d - \alpha_1); \tag{2}
\]
The schematic diagram in Fig. 3.2 and expression (2) are shown for the front track unit controller. The steering angle $\alpha_2$ of the rear track unit is controlled in the same way.

Thus, the joint operation of the developed regulators will allow to realize the control law for turning of the double-jointed multi-tracked vehicle to ensure the following of the reference trajectory.

4. Determination of the parameters of the regulators of the developed steering control system

To determine the parameters of the presented controllers the mathematical simulation of the motion of the considered vehicle was used (dynamics simulation was performed in MATLAB Simulink [6], and synthesis of equations of motion and description of interaction of the running gear with the ground surface was done according to the approach similar to the one used in [7]).

Motion simulation was performed in the following computational cases (in all computational experiments, the following coefficients were used to describe the interaction of the running gear with the ground surface: slide friction coefficient 0.8, motion resistance coefficient 0.07).

The first case is the placement of the multi-tracked vehicle parallel to the trajectory with some linear deviation. Simulating the return of the vehicle to the trajectory with the parallel motion allows optimizing the parameters of the linear deviation controller. Fig. 4.1 and 4.2 show the results of the simulation of a parallel return to the trajectory. It can be seen that there are no angular deviations during the maneuver, which makes it possible to determine the parameters of the $PID_\alpha$ controller independently of $PID_\beta$.

During computational experiment the parameters of PID controllers were selected by method of successive approximations.

Figure 4.1 – Computational experiment for linear deviation controller optimization: a - 0 s; b - 3 s; c - 6 s; d - 9 s
Figure 4.2 – Trajectories of the computational experiment to optimize the linear deviation controller

When the tracked train is placed on a track with some angular deviation, it is possible to optimize the parameters of the angular deviation controller in a similar way (Figures 4.3 and 4.4).

Figure 4.3 – Computational experiment to optimize the angular deviation controller: a - 0 s; b - 1 s; c - 5 s; d - 10 s

Figure 4.4 – Trajectories of the computational experiment to optimize the angular deviation controller
5. Determination of the parameters of the regulators of the developed turning control system

In order to assess the effectiveness of the joint operation of the developed multi-tracked vehicle motion controllers, a simulation of a manoeuvre similar to a "transposition" was carried out. The scheme of the manoeuvre is shown in Fig. 5.1. The curvilinear section of the trajectory is a sinusoid with a curvature not exceeding the maximum provided by the turning kinematics of the vehicle in question.

![Figure 5.1 – Trajectories of the multi-tracked vehicle "transposition" manoeuvre](image)

According to the trajectory obtained, the controller ensures that the specified manoeuvre can be performed. At the same time, it should be noted that in order to minimize the width of the travel corridor, the developed regulators must be optimized.

Dependence of torques on the driving wheels of the vehicle when performing a maneuver is shown in Fig. 5.2.

![Figure 5.2 – Torques on the driving wheels of the multi-tracked vehicle: 1 - Front left; 2 - Rear left; 3 - Front right; 4 - Rear right](image)

Analysis of the presented results (Fig. 5.2) allows us to conclude that turning of track units by means of traction control on the tracks, although it allows to avoid additional drive of track units, but leads to the need to significantly increase the traction torque on the drive wheels of the faster running tracks.
(track units) and creation of braking torque on the lower running ones. Thus, it is obvious that to ensure energy-efficient motion of the multi-tracked vehicle, this feature will require special turning mechanisms [8, 9] (by analogy with a single tracked vehicle [10-15]), kinematic method of steering or a combination of force and kinematic methods of steering each track unit.

6. Conclusion
The paper presents a control law for a double-joint multi-tracked vehicle, which makes it possible to ensure unmanned movement along a given route, while effectively eliminating control errors in terms of both lateral displacement from the reference trajectory and course angle. The application of the developed control law of the vehicle under consideration allows ensuring unmanned movement of the vehicle or following the lead vehicle along the reference trajectory, which will simplify the work of the driver-operator (in this case, the lead vehicle operator), as well as increase the accuracy of the turn maneuvers and, accordingly, improve the safety of large-size cargo transportation.

In addition, it is found that in order to ensure energy-efficient movement of a double-joint tracked train within each track unit, special turning mechanisms (similar to a single tracked vehicle), a kinematic method of steering or a combination of force and kinematic methods should be used.

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