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Design and experimental evaluation of a novel sliding mode controller for an articulated vehicle

T. Nayl, G. Nikolakopoulos *, T. Gustafsson, D. Kominiak, R. Nyberg

Control Engineering Group, Department of Computer Science, Electrical and Space Engineering, Luleå University of Technology, Luleå, Sweden

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ABSTRACT

This article presents the design and experimental evaluation of a novel sliding mode control scheme, being applied to the case of an articulated vehicle. The proposed sliding mode controller is based on a novel continuous sliding surface, being introduced for reducing the chattering phenomenon, while achieving a better tracking performance and a fast minimization of the corresponding tracking error. The derivation of the sliding mode controller relies on the fully nonlinear kinematic model of the articulated vehicle, while the overall stability of the control scheme is proven based on the Lyapunov's stability condition. The performance of the established control scheme is being experimentally evaluated through multiple path tracking scenarios on a small scale and fully realistic articulated vehicle.

1. Introduction

Recently, there has been a significant focus in designing automated vehicles for their utilization in the mining industry, with the general focus to be on improving the mining methods and increasing the technological readiness in autonomous operations. Among the current vehicle types utilized in a mining field, articulated ones are the most characteristic type that can be found most frequently, such as the Load Haul Dump (LHD) vehicles. In general the articulated vehicles consist of two parts, a tractor and a trailer, linked with a rigid free joint. Each body has a single axle and the wheels are all non-steerable, while the steering action is performed on the joint, by changing the corresponding articulated angle, between the front and the rear part of the vehicle.

In the related literature there have been several research approaches for the problem of modeling articulated vehicles as in [1–4], either by considering point kinematic properties or based on the theory of multiple body dynamics. Furthermore, from a control point of view, there have been proposed many traditional techniques for non-holonomic vehicles as the articulated vehicles in [5], where a linear control feedback has been applied, while in [6], a Lyapunov based approach has been presented. In [7] a control scheme based on LMIs has been evaluated and in [8] a pole placement technique has been applied. Moreover, in [9] a path tracking controller based on error dynamics, and in [10] a Switching Model Predictive Control framework has been established. However, to the authors' best knowledge, the application of sliding mode control scheme and its merits, based on the nonlinear error dynamics modeling of the articulated vehicle, has not appeared yet in the related literature.

In general, Sliding Mode Control (SMC) is a robust control scheme based on the concept of changing the structure of the controller, with respect to the changing state of the system in order to obtain a desired response [11]. The biggest advantage of the SMC is its insensitivity to variations in system parameters, external disturbances and modeling errors [12], a fact that can be achieved by forcing the state trajectory of the system to follow the desired sliding surface as fast as possible and in a minimum tracking error. Among the general advantages of utilizing the sliding mode control, it should be mentioned the ability of achieving a fast response, a good transient performance and an overall robustness regarding parameters' variations [13].

The aim of this article is to present a novel SMC scheme, being tuned and developed based on the nonlinear kinematic equations of an articulated vehicle model. Following this aim, the main contributions of the article are three. Firstly, a novel continuous sliding control surface for controlling the articulated vehicle, while reducing the chattering effects, and improving the reference tracking is being presented. Secondly, the overall stability of the proposed novel SMC is being proven based on Lyapunov's theory. Thirdly, the SMC's overall performance is being extensively evaluated by the utilization of a realistic small scale articulated vehicle.

The rest of the article is structured as it follows. In Section 2 the non-linear full kinematic model of the articulated vehicle will be presented. The novel sliding mode control scheme, based on the error dynamic model of the articulated vehicle, is extensively presented in Section 3. In Section 4, the architecture of the overall experimental platform is being presented, followed by multiple

* Corresponding author.
E-mail address: geonik@ltu.se (G. Nikolakopoulos).

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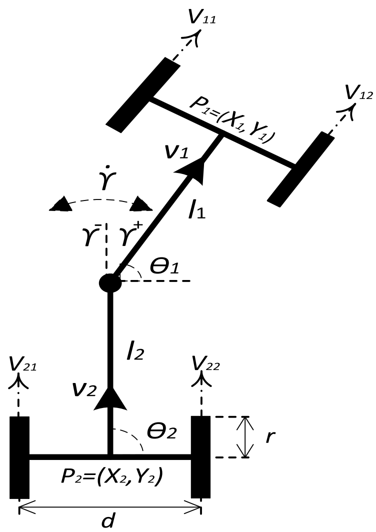

Fig. 1. The articulated vehicle model.

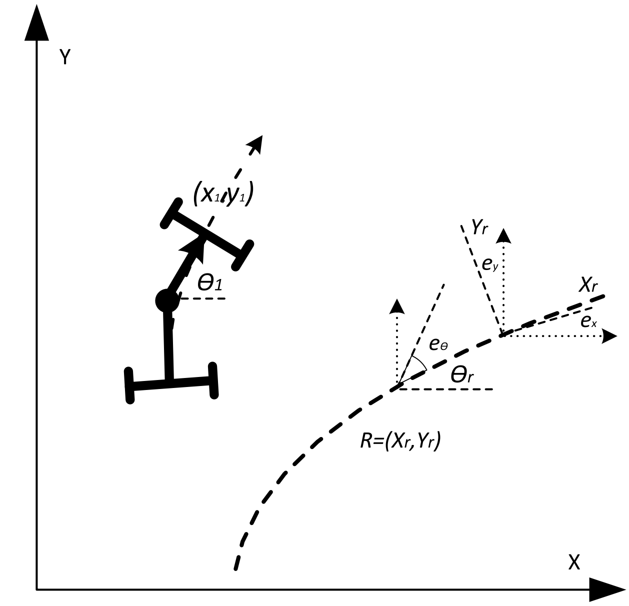

Fig. 2. A path view of the error coordinates of the articulated vehicle model.

experimental results that prove the applicability and the overall performance of the proposed scheme in Section 5. Finally, concluding remarks are mentioned in Section 6.

2. Articulated vehicle model

The articulated vehicle, considered in this article, can be described by two parts, linked with a rigid free joint, with lengths l_1 and l_2 respectively, while each part has a single axle and all wheels are non-steerable. The steering action is being performed on the middle joint, by changing the corresponding articulated angle γ, in the middle of the vehicle and $\dot{\gamma}$ is the rate of change for this articulated angle. The velocities v_1 and v_2 are considered to have the same changing with respect to the velocity of the rigid free joint of the vehicle, indicated by $\dot{\theta}_1$, $\dot{\theta}_2$, which are the angular velocities of the front and rear parts of the vehicle respectively. Overall, the articulated vehicle's geometry is depicted in Fig. 1.

Based on the modeling approach presented in [14], the full kinematic model of the articulated vehicle, under the non-holonomic constraints, can be provided as it follows:

$$\begin{aligned} \dot{x}_1 &= v_1 \cos\theta_1 \\ \dot{y}_1 &= v_1 \sin\theta_1 \\ \dot{\theta}_1 &= \frac{v_1 \sin\gamma + l_2\dot{\gamma}}{l_1\cos\gamma + l_2} \end{aligned} \quad (1)$$

where x_1 and y_1 are the position variables and θ_1 is the front orientation angle. In the presented approach, the longitudinal velocity v_1 and the steering angle $\dot{\gamma}$ are the control signals, while the kinematic equations in a state space formulation can be given as:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{y}_1 \\ \dot{\theta}_1 \\ \dot{\gamma} \end{bmatrix} = \begin{bmatrix} \cos\theta_1 & 0 \\ \sin\theta_1 & 0 \\ \dfrac{\sin\gamma}{l_1\cos\gamma + l_2} & \dfrac{l_2}{l_1\cos\gamma + l_2} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} v_1 \\ \dot{\gamma} \end{bmatrix} \quad (2)$$

The articulated vehicle has five DC-motors, four of them are dedicated on each wheel and one for controlling the articulated angle in the middle joint. The velocities of the motor wheels are being controlled individually, but in a coordinated manner and in fully symphony with the overall dimensions of the articulated vehicle and the articulation angle (γ and $\dot{\gamma}$), as depicted in the following geometrically derived equations [15]:

$$\begin{aligned} v_{11} &= v_1 + \frac{v_1 \sin\gamma}{4l(1+\cos\gamma)} - \frac{v_1\,\dot{\gamma}\,d}{2(1+\cos\gamma)} \\ v_{12} &= v_1 - \frac{v_1 \sin\gamma}{4l(1+\cos\gamma)} + \frac{v_1\,\dot{\gamma}\,d}{2(1+\cos\gamma)} \\ v_{21} &= v_1\cos\gamma + \frac{v_1 \sin^2\gamma}{2(1+\cos\gamma)} + \frac{d\sin\gamma}{2l(1+\cos\gamma)} \\ &\quad + \frac{\dot{\gamma}(2l\sin\gamma - d\cos\gamma)}{2(1+\cos\gamma)} \\ v_{22} &= v_1\cos\gamma + \frac{v_1 \sin^2\gamma}{2(1+\cos\gamma)} - \frac{d\sin\gamma}{2l(1+\cos\gamma)} \\ &\quad + \frac{\dot{\gamma}(2l\sin\gamma + d\cos\gamma)}{2(1+\cos\gamma)} \end{aligned} \quad (3)$$

where v_{11} and v_{12} denote the right and left front wheels linear speeds, and v_{21}, v_{22} denote the right and left rear wheels' linear speeds, respectively. The speeds of the four wheels can be measured from the encoders of the motors. Furthermore, it is considered that the reference path coordinates are being denoted by $\mathbf{x}_r = [x_r\ y_r\ \theta_r]^T$ and that the vehicle is moving with a constant velocity of v_r. The overall geometry of the articulated vehicle, including the corresponding error definitions, is depicted at Fig. 2.

In the following derivation, the three error coordinates, expressing the deviation of the vehicle from the desired position and heading, are denoted as $\mathbf{x}_e = [e_x\ e_y\ e_\theta]^T$, while in a state space formulation this error dynamics modeling, with respect to the reference path, can be formulated as it follows:

$$\begin{bmatrix} e_x \\ e_y \\ e_\theta \end{bmatrix} = \begin{bmatrix} \cos\theta_1 & \sin\theta_1 & 0 \\ -\sin\theta_1 & \cos\theta_1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x_1 - x_r \\ y_1 - y_r \\ \theta_1 - \theta_r \end{bmatrix} \quad (4)$$

In the sequel it is being assumed that the vehicle is driven without lateral slip angles in all wheels and that the lengths of the front and rear parts are of the same length l. Moreover, it is assumed that the vehicle is moving on a flat surface and the curvature of the reference path is denoted by $c_r = \frac{1}{R}$, where R is the radius of this path. Differentiation of the error dynamic equations

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