Computers and Electronics in Agriculture 145 (2018) 282–288

Contents lists available at ScienceDirect

Computers and Electronics in Agriculture

journal homepage: www.elsevier.com/locate/compag

Self-tuning regulator for a tractor with varying speed and hitch forces

Benjamin Fernandez*, Pedro Javier Herrera, Jose Antonio Cerrada

Depto. de Ingeniería de Software y Sistemas Informáticos, ETSI Informática, UNED, C/Juan del Rosal, 16, 28040 Madrid, Spain

A R T I C L E I N F O

Keywords:
Tractor-implement system
Self-tuning regulator
Minimum-degree pole placement
Agricultural guidance
Control systems
Model-following
OpenGL
Openframeworks

A B S T R A C T

Due to the changes in the soil, speed and hitch forces, the dynamics of a farm tractor are constantly changing making the design of an autonomous lane-tracking controller a very complex task. To be able to react to those changes, this paper presents a new adaptive system based on a self-tuning regulator made up of a recursive least-squares parameter identification algorithm for the plant combined with a minimum-degree pole placement (MDPP) method for changing the parameters of a digital RST controller in real time. The MDPP is computed by solving the Diophantine equation for the desired closed-loop reference model. The results presented show how the system is able to adapt the control parameters for different speeds and changes in the hitch cornering stiffness. As future work, this method could also be applied and assessed as a general controller, covering different sizes and different types of steering systems for off-road vehicles.

1. Introduction

With an expected population growth of 30%, some sources (U.Nations, 2013) forecast an increase of up to 9.6 billion inhabitants by 2050 and agricultural processes will play a very important role in feeding this many people. There has been an increasing research interest in agricultural robotics and automation to help improve this processes and deal with the aforementioned problems. The goal is to use resources such as machinery and seeds, more efficiently, increasing yield without having to increase working area. For this, automating off-road vehicles will help tackle the problem (Moorehead et al., 2012). One of the main difficulties of automating an off-road vehicle for lane tracking, is that the dynamics of the system are changing constantly. This is due to different factors such as soil irregularities, changes in the driving speed and in the hitch load. This makes the design of a controller a very difficult and time consuming task, since finding a set of controller parameters for every single situation is nearly impossible. Therefore, the motivation for this research is to find a more efficient adaptive controller design that saves time in the implementation and parameter tuning and is also able to online cover the changes of the vehicle dynamics.

To deal with the aforementioned difficulties, the most widely used adaptive controller for lane tracking is a PID gain scheduler. This requires tuning the parameters for the look-up table in all different possible soils, covering a desired set of velocities with different combinations of front and rear implements. There has been main work performed in the area of adaptive control of off-road vehicles. For instance, Bevly and Gartley have done an analysis of a tractor-implement system (Gartley and Bevly, 2008) and proposed a Model Reference Adaptive System (MRAS) for yaw rate dynamics (Derrick and Bevly, 2008, 2009; Derrick et al., 2008) combined with gain scheduling for the lateral position. A non-linear adaptive and predictive controller is presented in Lenain et al. (2007) and a Fuzzy non-linear adaptive approach is also to be found in Zhang et al. (2013).

This article presents a new method for controlling the yaw rate dynamics of an off-road vehicle using a self-tuning regulator. This includes a minimum-degree pole placement design, based on an online identification of the vehicle dynamics plus an online computation of the control parameters. There are different improvements of the method presented compared with previous related work. One is that a linear second order system can be used as closed-loop reference model and due to this, the closed-loop yaw rate will tend to behave the same regardless of the changes in the soil conditions and hitch load. The linear design also makes the implementation of the controller in an embedded system less complex and less time consuming. This method will also need a gain scheduler for the lateral position. However, since the closed-loop yaw rate will tend to behave the same independently of the conditions, the look-up table of the lateral position will need fewer sets of PID parameters. Since the identification is based on a general second order system, this method can be applied to different steering systems such as skid-steering, 4 W-steering and articulated steering.

The paper is divided in the following way: The second section will analyze the vehicle dynamics of a tractor with hitch forces, as well as its poles and zeros to better understand the effects of the changes in the

* Corresponding author.
E-mail address: mail@benjaminfernandez.info (B. Fernandez).

https://doi.org/10.1016/j.compag.2017.12.027
Received 29 July 2017; Received in revised form 7 November 2017; Accepted 18 December 2017

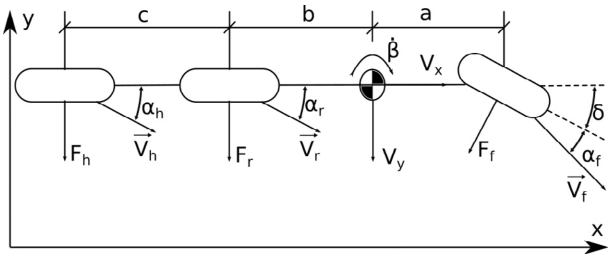

Fig. 1. Tractor-implement bicycle model.

soil and speed. Section three presents the self-tuning algorithm and the methodology. The results are presented in section four and finally section five presents the conclusions and future work.

2. Tractor-implement analysis

To be able to design an appropriate controller, an analysis of the vehicle dynamics, kinematics and its root-locus has to be performed. The dynamics and kinematics lead to the equation of motion of the system and its root-locus tells the response of the system according to the changes in the velocity, soil and hitch forces. This is essential for finding the model to be used as a reference for the controller design. If the dynamics of the reference model are too fast, the closed-loop system will experience steady state oscillations and even instability and if the dynamics are too slow, the controller will have a lot of room for improvement.

2.1. Bicycle model

Fig. 1 shows a 3-wheeled tractor-implement bicycle model. Here, $\dot{\beta}$ is the yaw rate around the center of gravity, δ is the steering angle and α_f,α_r and α_h are the front, rear and hitch slip angles, respectively. The distances from the front and rear axis to the center of gravity are a and b, respectively, and c is the distance from the rear axis to the hitch. Lateral forces at front, rear and hitch tires are represented by F_f,F_r and F_h, respectively, and by assuming constant longitudinal velocity (V_x), the longitudinal acceleration is null and the longitudinal forces are neglected. Therefore, the yaw rate dynamics of the model represented by Fig. 1, can be expressed by analysing the simplified lateral dynamics with Eq. (1).

$$\begin{aligned} \sum F_y &= m\cdot a_y \\ \sum M_{CG} &= I_z\cdot\ddot{\beta} \end{aligned} \tag{1}$$

From the kinematics point of view, and since the system has null longitudinal acceleration, the lateral acceleration is expressed in Eq. (2).

$$a_y = \dot{V}_y + \dot{\beta}\cdot V_x \tag{2}$$

Substituting into Eq. (1) and using the small angle approximation, we obtain the following simplified equation of motion:

$$\begin{aligned} m\cdot(\dot{V}_y + \dot{\beta}\cdot V_x) &= F_f + F_r + F_h \\ I_z\cdot\ddot{\beta} &= a\cdot F_f - b\cdot F_r - (c+b)\cdot F_h \end{aligned} \tag{3}$$

Assuming constant lateral forces, their relationship to the slip angles are given in Eq. (4) (Derrick and Bevly, 2009; Gillespie, 1992).

$$\begin{aligned} F_f &= -C_{\alpha f}\cdot\alpha_f \\ F_r &= -C_{\alpha_r}\cdot\alpha_r \\ F_h &= -C_{\alpha_h}\cdot\alpha_h \end{aligned} \tag{4}$$

where $C_{\alpha f}$,C_{α_r} and C_{α_h} are the front, rear and hitch cornering stiffness and vary depending on the conditions and types of soil. By substituting (4) into (3), the equation of motion looks as follows:

$$\begin{aligned} m\cdot(\dot{V}_y + \dot{\beta}\cdot V_x) &= -C_{\alpha f}\cdot\alpha_f - C_{\alpha_r}\cdot\alpha_r - C_{\alpha_h}\cdot\alpha_h \\ I_z\cdot\ddot{\beta} &= -a\cdot C_{\alpha f}\cdot\alpha_f + b\cdot C_{\alpha_r}\cdot\alpha_r + (c+b)\cdot C_{\alpha_h}\cdot\alpha_h \end{aligned} \tag{5}$$

Assuming a rigid body, we can say in general that the absolute lineal velocity at any of its points can be expressed as the lineal velocity of its center of gravity plus the velocity of the point with respect to its center of gravity and so wet get Eq. (6), where each velocity is represented by its x and y coordinates.

$$\begin{aligned} \frac{V_{f_y}}{V_{f_x}} &= \frac{V_y + \dot{\beta}\cdot a}{V_x} \\ \frac{V_{r_y}}{V_{r_x}} &= \frac{V_y - \dot{\beta}\cdot b}{V_x} \\ \frac{V_{h_y}}{V_{h_x}} &= \frac{V_y - \dot{\beta}\cdot(b+c)}{V_x} \end{aligned} \tag{6}$$

We can also observe from Fig. 1 that

$$\begin{aligned} \tan(\alpha_f + \delta) &= \frac{V_{f_y}}{V_{f_x}} \\ \tan(\alpha_r) &= \frac{V_{r_y}}{V_{r_x}} \\ \tan(\alpha_h) &= \frac{V_{h_y}}{V_{h_x}} \end{aligned} \tag{7}$$

By substituting Eq. (6) into Eq. (7), applying the small angle approximation and solving for α, the relationship between the slip angles and the longitudinal and lateral velocities of the center of gravity is found.

$$\begin{aligned} \alpha_f &= \frac{V_y + \dot{\beta}\cdot a}{V_x} - \delta \\ \alpha_r &= \frac{V_y - \dot{\beta}\cdot b}{V_x} \\ \alpha_h &= \frac{V_y - \dot{\beta}\cdot(b+c)}{V_x} \end{aligned} \tag{8}$$

Substituting Eq. (8) into Eq. (5) results into

$$\begin{aligned} m\cdot(\dot{V}_y + \dot{\beta}\cdot V_x) &= -C_{\alpha f}\cdot\left(\frac{V_y + \dot{\beta}\cdot a}{V_x} - \delta\right) - C_{\alpha_r}\cdot\frac{V_y - \dot{\beta}\cdot b}{V_x} - C_{\alpha_h}\cdot\frac{V_y - \dot{\beta}\cdot(b+c)}{V_x} \\ I_z\cdot\ddot{\beta} &= -a\cdot C_{\alpha f}\cdot\left(\frac{V_y + \dot{\beta}\cdot a}{V_x} - \delta\right) + b\cdot C_{\alpha_r}\cdot\frac{V_y - \dot{\beta}\cdot b}{V_x} + (c+b)\cdot C_{\alpha_h}\cdot\frac{V_y - \dot{\beta}\cdot(b+c)}{V_x} \end{aligned} \tag{9}$$

By introducing the following new variables C_1,C_2 and C_3 such as

$$\begin{aligned} C_1 &= -a\cdot C_{\alpha f} + b\cdot C_{\alpha_r} + (b+c)\cdot C_{\alpha_h} \\ C_2 &= C_{\alpha f} + C_{\alpha_r} + C_{\alpha_h} \\ C_3 &= a^2\cdot C_{\alpha f} + b^2\cdot C_{\alpha_r} + (b+c)^2\cdot C_{\alpha_h} \end{aligned} \tag{10}$$

Eq. (9) can be rewritten into the following state space equation of motion with state variables V_y and $\dot{\beta}$:

$$\begin{bmatrix} \dot{V}_y \\ \ddot{\beta} \end{bmatrix} = \begin{bmatrix} -\frac{C_2}{m\cdot V_x} & \frac{C_1}{m\cdot V_x} - V_x \\ \frac{C_1}{I_z\cdot V_x} & -\frac{C_3}{I_z\cdot V_x} \end{bmatrix}\cdot\begin{bmatrix} V_y \\ \dot{\beta} \end{bmatrix} + \begin{bmatrix} \frac{C_{\alpha f}}{m} \\ \frac{a\cdot C_{\alpha f}}{I_z} \end{bmatrix}\cdot\delta \tag{11}$$

Using Laplace transform, the transfer function between the steering angle and the yaw rate can be found by solving the following equation:

$$\frac{\dot{\beta}(s)}{\delta(s)} = \frac{C\cdot adj(s\cdot I - A)\cdot B}{det(s\cdot I - A)} \tag{12}$$

where $C = [0 \quad 1]$ since it is solved only for $\dot{\beta}$ and so one gets that

$$\frac{\dot{\beta}(s)}{\delta(s)} = \frac{b_2\cdot s + (b_1\cdot a_{21} - b_2\cdot a_{11})}{(s - a_{11})(s - a_{22}) - a_{12}\cdot a_{21}} \tag{13}$$

with