DYNAMIC MODELS OF AN AGV BASED ON EXPERIMENTAL RESULTS [1]

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Abstract: Based on experimental results, longitudinal and steering system models of an Autonomous Guided Vehicle (AGV) are determined. The longitudinal model has been determined by using a method based on the velocity step response of the vehicle and the steering model has been obtained by means of the simulation of an approximated dynamic model of the steering system. The simulated results are compared with the real experiments. Finally, the determined dynamic model are used in simulation with several control schemes to see the robutness of the controllers.

Keywords: Dynamic model, AGV, steering and longitudinal model

1. INTRODUCTION

A longitudinal and steering system models of an industrial electrical vehicle are presented here. These models, together with the lateral dynamics of the vehicle, form a complete model. Up to now (Suárez *et al.*, 2003*a*; Suárez *et al.*, 2003*b*) several control schemes (fuzzy, pure-pursuit, fractional,...) were tested with a model composed of the lateral dynamics and an approximated first order model of the steering system. No longitudinal model was considered. The aim of this work is to obtain a complete dynamic model of the vehicle, nearer to reality, that allows to simulate several path-tracking controllers. This will permit, before any real implementation was carried out, to choose the most appropriate controller and to reject those ones with worse performances. Furthermore, it will permit a fast an easy comparative study of several control techniques without the need for a real implementation on the vehicle instrumentation system, what would imply a great waste of time.

These models have been obtained from some experimental results of several test done on a private circuit. An industrial vehicle equipped with some instrumentation and control hardware and software has been used.

The longitudinal model of the vehicle is difficult to characterize. Some authors present some approximation to a simple first-order model (Rodriguez-Castaño *et al.*, 2003) or even to a more complex model (Kodagoda *et al.*, 2002; Abichou *et al.*, 2003) taking into account some difficult to obtain parameters such as coefficients of air drag force. Here a simple method, based on step responses, is presented.

The rest of the paper is organized as follows. In the section 2 the vehicle and the test circuit are described. In section 3 the velocity tests are presented together with a method of obtaining the

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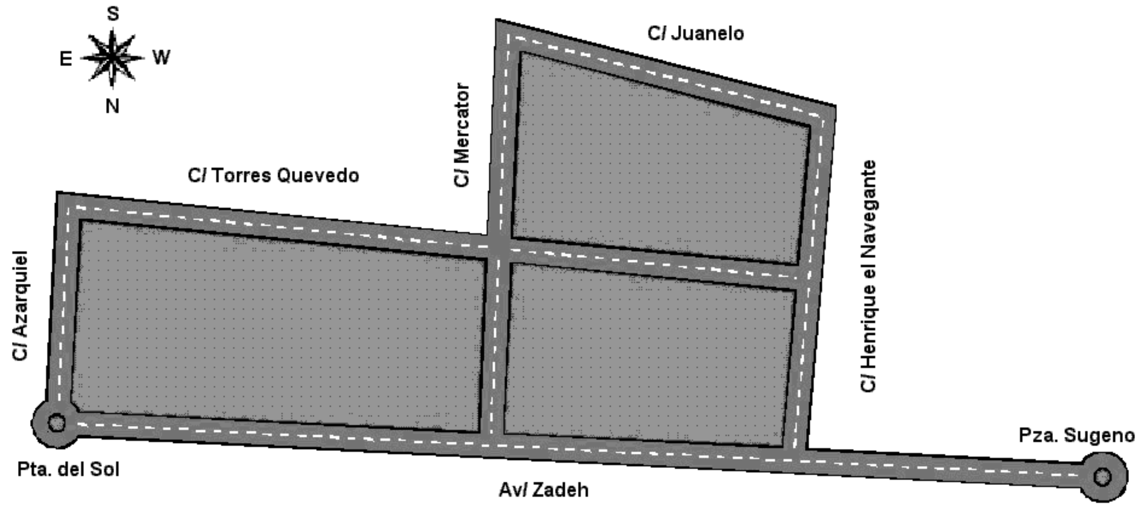

Fig. 1. Layout of the private circuits (ZOCO) where the experiments were carried out.

longitudinal model. The section 4 describes the steering tests, the steering system and its model and also the simulations results are compared with the experimental tests in this same section. In the section 5 some simulation with several controller are presented. Finally, the conclusions and the future work.

2. SYSTEM DESCRIPTION

2.1 Test zone

The experiments were carried out on private circuit located at the Institute of Industrial Automation in Arganda del Rey (Madrid). This circuits is known as ZOCO (*ZOna de COnducción*, driving zone) and it is composed of several streets with some intersections and roundabouts (figure 1).

2.2 Vehicle description

The vehicle is an electrical car, a Citroën Berlingo, with an Ackerman steering system. The test system consists of two computers: an industrial PC with the necessary software to govern the car and to communicate with the external world, an another one (laptop) with the desired tests that communicates with the first one serially.

The industrial computer has a software that permits to read the commands from the laptop and sends velocity orders to the traction motor and position orders to the flywheel motor. Also it is able to obtain some data (velocity and position of the vehicle and position of the flywheel) from the sensors. For obtaining the velocity data, the vehicle has an internal electronic calculator that communicates serially with the industrial PC. The position data of the flywheel are gotten from a PID control card connected to the computer.

Fig. 2. On board industrial computer

The test computer is a laptop with a test software, developed in *LabView*, that communicates serially with the industrial PC for sending test commands and for receiving data of the vehicle state. A simple serial protocol was establish for communicating both computers. Some parameters of the software are user selectable, such as the communication speed (we chose 19600 bauds), or the sample period.

3. LONGITUDINAL MODEL

3.1 Velocity tests

Fourteen different test were done with the aim of obtaining a longitudinal dynamic model of the vehicle. Each test consists of introducing a velocity step command and capturing the response of the vehicle. The velocity data were acquired and stored in different files for later treatments in the simulation software (*Matlab*). An important problem in the realization of the experiments was the difficulty for finding a flat and long enough track. Due to this reason, the experiments were carried out at low speeds, and a sufficient flat but no long track was selected. The experiments at low speed were satisfactory and the vehicle velocity settles, but at faster speed the vehicle reaches to

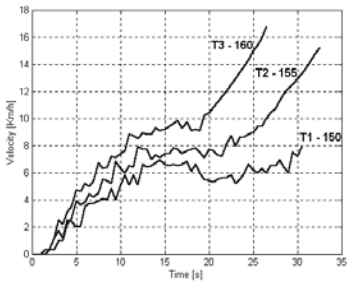

Fig. 3. Results of three differents velocity tests

the downward slope of the track (that acts as a disturbance) and hence the vehicle starts to gather speed.

In the figure 3, three different tests are depicted (T1, T2 and T3). In the test T1, the velocity of the car settles, but in the tests T2 and T3 test we can observe how the velocity of the vehicle establishes, but later it increases due to the arriving of the car to the downward section of the track. These three results are good enough to obtain a longitudinal dynamic model of the vehicle.

3.2 Method for obtaining the longitudinal model

With the aim of obtaining an approximated longitudinal model of the vehicle dynamics the method of *measuring the percentage values* were used (Puente, 1997). This method, proposed by Schwarze, try to find, in the step response of the system, a transfer function like this:

$$G(s) = \frac{K}{(1+Ts)^n} \tag{1}$$

where K is the system gain and T is the time constant of a n-order pole.

Other methods, such as those based on measuring the slope on the inflection point of the S-shape response, are also possible but difficult of measuring. The Schwarze's method is applied to those systems with an S-shape step response, as our case (figure 3). On the experimental curve several times must be measured, in particular, the times at 10%, 30%, 50%, 70% and 90% of the final value. Then the relationships t_{10}/t_{90}, t_{10}/t_{70}, t_{10}/t_{50}, t_{10}/t_{30}, t_{30}/t_{50} and t_{30}/t_{70} are obtained. With these values the order of the system n can be obtained graphycally. Once n is obtained and with another graphic, we can get the value of the time constant T. From the step response the value of the system gain K is calculated.

From the Schwarze's method we have obtained $n = 3$, $K = 1$ and $T = 2$, approximately. Then the transfer function and hence the longitudinal dynamic model of the vehicle is:

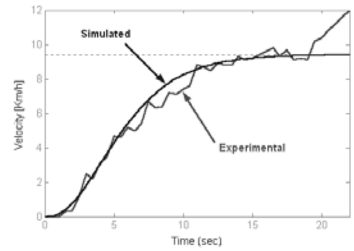

Fig. 4. Simulated response of the vehicle longitudinal model versus the experimental results

$$G(s) = \frac{1}{(1+2s)^3} = \frac{1}{8s^3 + 12s^2 + 6s + 1} \tag{2}$$

As shown in the figure 4, this transfer function is a good approximation to the experimental values.

4. STEERING SYSTEM MODEL

For obtaining an approximate model of the steering system of the Citroën Berlingo Vehicle several experiments have been carried out using the same communication software described above. However, in this case new communications commands were established.

4.1 Steering tests

In the seven experiments carried out, several position steps with different amplitudes each one were introduced to the vehicle. The flywheel positions were registered with a sample time of 50 ms and later they were analyzed in *Matlab* software. Some of these results are depicted in the figure 5.

4.2 Description of the steering system

The Citroën Berlingo vehicle has an hydraulic powered steering system. An electric DC geared motor has been arranged to move the flywheel. Its main characteristics are shown in table 1.

Table 1. Steering motor technical data

Parameter	Value	Description
V_n	$12\ V$	Nominal voltage
R	$0.16\ \Omega$	Terminal resistance
L	$30\ \mu H$	Rotor inductance
k_e	$1.76\ mV/rpm$	Back-EMF constant
k_t	$2.379\ oz\text{-}in/A$	Torque constant
r_g	$99:1$	gearbox reduction ratio

The motion of the flywheel motoris controlled by a hardware card with a LM629 motion controller. This controller, from National Semiconductor, has several functional blocks as shown in figure 6.

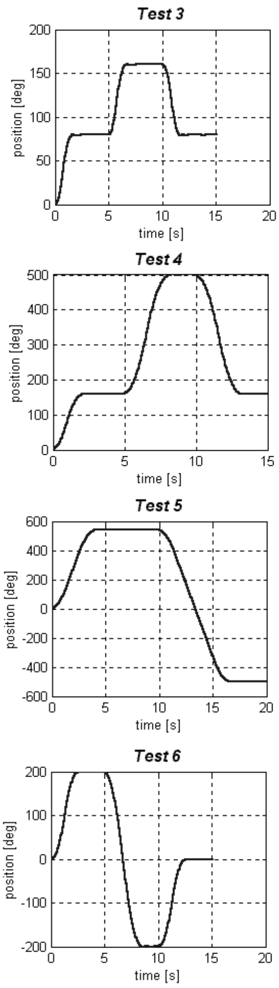

Fig. 5. Some experimental results of the flywheel position tests.

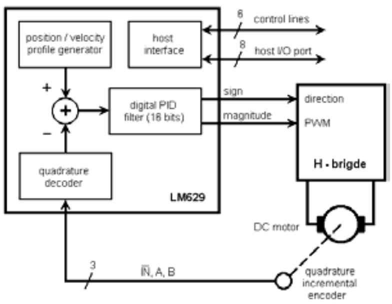

Fig. 6. Functional blocks of the motion controller circuit LM629 from National Semiconductor

The digital controller is a 16 bits PID filter, whose outputs (sign and magnitude) are used to drive a DC motor by means of an H-bridge driver using the PWM technique. The H-bridge driver allows the motor to be driven in either direction.

The function of the quadrature decoder block is to read the position, velocity, acceleration and direction of the motor motion by using an optical incremental encoder.

To change the position of the motor, the profile generator block generates a trapezoidal velocity profile as shown in the figure 7. The flywheel

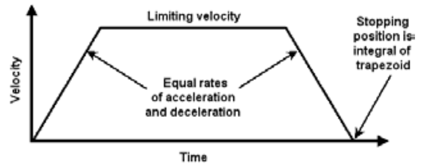

Fig. 7. Trapezoidal velocity profile of the LM629 profile generator block

velocity was limited to the maximum value of 200 degrees per second and its acceleration to 300 deg $/s^2$.

4.3 Modeling of the steering system

For obtaining a dynamic model of the steering system, the behavior of the digital PID controller, the electric motor and the steering system load has been simulated for trying to approximate to the experimental results. Therefore, the steering model is composed of three different parts (figure 8):

- the digital PID controller model
- the motor model
- and the load model (the steering system itself).

4.3.1. Electric motor model. Back-emf of a permanent magnet DC motor is given by the expression (Mohan *et al.*, 1995):

$$e = k_e n \tag{3}$$

where n is the motor speed in rpm and k_e is back-emf constant. The velocity control is accomplished by applying a voltage u_n to the armature terminals which causes a current across the motor windings. Hence, the current i in the armature circuits is determined by:

$$u_n = L\frac{di}{dt} + Ri + e \tag{4}$$

where L and R are the armature-winding inductance and resistance, respectively. Then, the motor torque is given by:

$$\tau = k_t i \tag{5}$$

where k_t is the torque constant.

The figure 8 shows the *Simulink* diagram of the motor model. A saturation block is included because the voltage is limited to $12V$ and moreover the control voltage is a PWM signal comprised between 0 and $12V$.

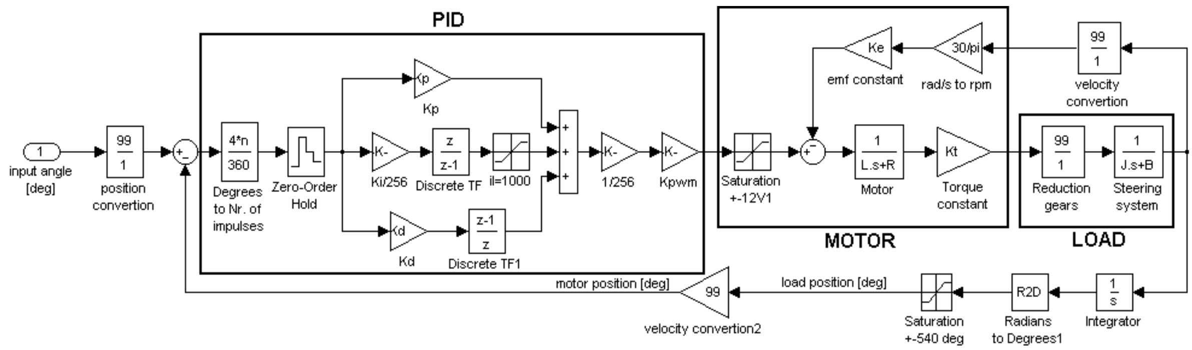

Fig. 8. Simulink diagram of the steering system dynamic model

4.3.2. Flywheel load model. A first order model has been supposed for the load which consists of the reduction gear (with a reduction ratio of 99:1), the powered steering and the mechanical transmission to the front wheels (figure 8). The total equivalent inertia J and viscous friction B are unknown and must be determined. An angular limitation of $\pm 540°$ has been imposed to the flywheel system which involves a $\pm 30°$ limitation of the front wheels steering angle.

4.3.3. Digital PID controller model. The position sensor is an optical quadrature encoder with 500*ppr*. The motion controller device LM629 use the decodification method that increases the encoder resolution by four. Every sample period ($T_S = 256\mu s$) a new motor position is acquired. The PID algorithm is given by the following equation:

$$u(n) = K_p e(n) + K_i T_s \sum_{n=0}^{N} e(n) + \qquad (6)$$
$$+ \frac{K_d}{T'} \left[e(n) - e(n-1) \right]$$

where T' is the derivative sample interval ($T' = 7T_S$). In terms of LM629 coefficients the equation 6 becomes:

$$u(n) = k_p e(n) + k_i \sum_{n=0}^{N} e(n) + \qquad (7)$$
$$+ k_d \left[e(n') - e(n' - 1) \right]$$

where k_p, k_i and k_d are the discrete-time coefficients, $e(n)$ the position error at sample time n and n' is the sampling at the derivative sampling rate.

In our case the PID coefficients are $K_p = 20$, $K_i = 50$ and $K_d = 1000$. The integral term is limited to ± 1000 and all the coefficients are computed as 16 bits terms.

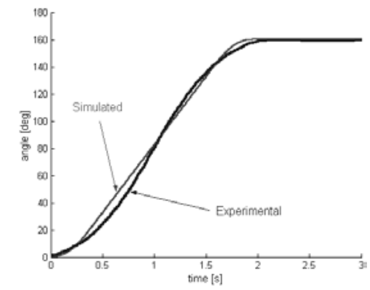

Fig. 9. Simulated result vs experimental response of the steering system when the input signal is 160°

Finally, the PID output is converted to a *PWM* and *sign* signal.

4.4 Simulation results

The whole model, where J and B are unknown, is simulated and these parameters are adjusted, by trial and error, to obtain an approximation to the experiments response. It is not important the J and B absolute values, but the B/J ratio. The steering system response has been simulated when the flywheel command signal changes from 0° to 160°. The trapezoidal velocity profile is generated and the flywheel commanded position is calculated by integration. In the figure 9 the simulated and the experimental results are compared. We have obtained a ratio $B/J = 3$. We have chosen $J = 1$ and $B = 3$. The simulated response is quite similar to the experimental one.

5. CONTROLLERS SIMULATIONS

The old model was simulated with several path-tracking controllers and the same control schemes has been used to simulate the new and complete model (lateral and longitudinal dynamic and the new steering system model). The control laws remain unchaged to test the controllers robustness with the new model. Four different control techniques have been tested: fuzzy (figure 10), pure-pursuit (figure 11), ϵ-controller (Davidson and

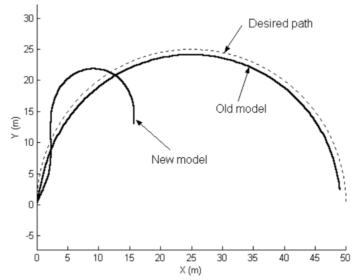

Fig. 10. Fuzzy controllers

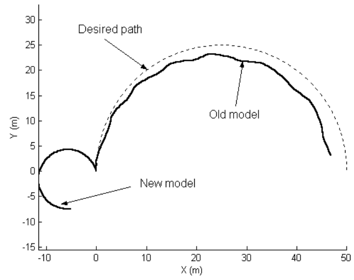

Fig. 11. Pure-pursuit controllers

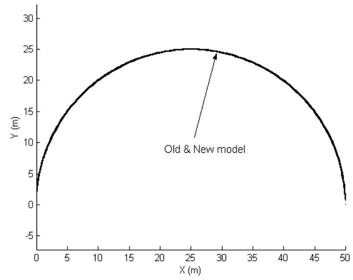

Fig. 12. ϵ-Controller with PI regulator

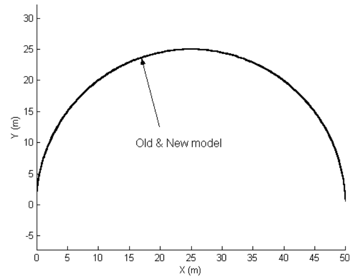

Fig. 13. ϵ-Controller with PI^{α} regulator

Bahl, 2001) with PI (figure 12) and ϵ-controller with PI^{α} (figure 13). The vehicle must travel a semicircular path with a radius of $25m$.

The worst behaviors are for the pure-pursuit and fuzzy controllers. However, the ϵ-controller (with PI or PI^{α} regulator) is very robust. The old and new model responses are almost undistiguishable.

6. CONCLUSIONS

Some experimental results carried out with a real industrial vehicle have been presented here with the aim of showing a simple method for determining a longitudinal model and a steering system model. The obtained models are satisfactory and they have presented results very similar to the experimental responses. Also we have tested the new model and compared with the old one to see the robustness of the controllers.

In the future, we will use these models to simulate path-tracking algorithms and low level controllers, with special attention to the application of the fractional order controller (FOC) to these issues. If needed, these models would be improved.

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