# Model-free control techniques for Stop & Go systems

Jorge Villagrá, Vicente Milanés, Joshué Pérez and Carlos González

***Abstract*— This paper presents a comparison of Stop & Go control algorithms, which deal with car following scenarios in urban environments. Since many vehicle/road interaction factors (road slope, aerodynamic forces) and actuator dynamics are very poorly known, two robust control strategies are proposed: an intelligent PID controller and a fuzzy controller. Both model-free techniques will be implemented and compared in simulation to show their suitability for demanding scenarios.**

## I. INTRODUCTION

Adaptive cruise control (ACC) and stop-and-go control systems have been deeply studied in recent years [1]. While ACC automatically accelerates or decelerates the vehicle to keep a quasi-constant target velocity and headway distance, stop-and-go deals with the vehicle in urban scenarios, with frequent and sometimes hard stops and accelerations.

Both situations present different comfort and safety constraints, and therefore, in most of the reported works, ACC and stop-and-go problems are treated separately.

Besides, Cooperative ACC (CACC) is a further development that adds vehicle-to-vehicle communication, providing the previous systems with more and better information about the vehicle it is following. With information of this type, the controllers will be able to better anticipate problems, enabling it to be safer and exhibit a smoother response.

The main idea of these control systems is to regulate the vehicle around the well-known two seconds distance rule, which attempts to respect a distance proportional to the human reaction time (approximately 2 s). Some approaches [2], [3] have tried to reproduce human behavior with deterministic models in order to achieve smooth control actions. Unfortunately, this kind of strategy may not necessarily lead to safe operation (see e.g. [1]).

Other authors (e.g. [4], [5]) have modeled reference inter-distances using different types of time polynomials, whose coefficients are obtained respecting safety acceleration and jerk constraints.

In general, these approaches produce acceptable results in an ACC scenario. However, during a sudden deceleration of the preceding car, the vehicles present a large transitory relative velocity and the actual inter-distance decreases abruptly. Hence, this dynamical scenario would not be suitably represented by static polynomial models, but by some kind of inter-distance dynamic model.

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In [6], the authors proposed a nonlinear reference model taking into account safe and comfort specification in an intuitive way. However, this work considers that the reference acceleration generated by the dynamic inter-distance model is instantaneously applied to the following vehicle. Since this assumption is hardly ever met in real urban situations, an advanced feedback controller should be introduced to cope with vehicle nonlinearities -specially in brake and engine dynamics at low speed- and environment disturbances -namely road slopes and wind gusts.

Different approaches have been proposed to tackle the actuators nonlinear dynamics. Input/output linearization [7], fuzzy logic (cf. [8], [9]) or sliding mode control (cf. [10] or [11]) have been used to deal with the engine control. Feedback linearization [12] and sliding modes [13] have also been implemented to control a nonlinear brake model. However, most of these approaches rely on precise models, so that any parameter variation during the life time of the vehicle may lead to a loss of performance, or even to an unstable behavior.

The main contribution of this paper consists in finding an engine/brake control algorithm that obtain the expected reference speeds and acceleration of the follower vehicle, while keeping a reference distance to the leader vehicle. Moreover, the control law will have to be robust to measurement noises, unmodeled dynamics -brake and engine dynamics- and disturbances -road inclination, aerodynamic forces or rolling resistance.

To that end, two model free control techniques will be implemented and compared with a realistic vehicle model: an intelligent PID controller (see [14] or [15] for previous works using this novel technique) and a fuzzy controller.

In Fig. 1 the dashed block represents the core of this work, which will use the dynamic inter-distance model obtained in [6] and the measurements provided by the follower CAN bus and the wireless communication system.

The rest of the paper is organized as follows. The vehicle model will be presented in Sec. II. The third section will be devoted to recall the dynamic inter-distance model. Sec. IV shows how an intelligent PID is adapted to this problem. The implementation of a fuzzy controller will be detailed in Sec. V. Both techniques will be evaluated under a simulation environment in VI, where disturbance robustness, comfort and safety will be compared and discussed. Finally, some concluding remarks and future work will be drawn in Section VII.

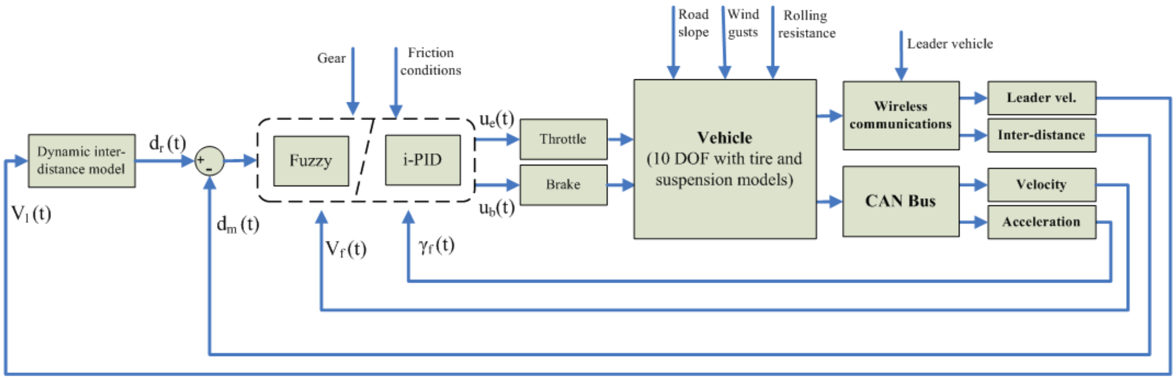


Fig. 1. Control scheme.

## II. VEHICLE MODEL

The balance of forces along the vehicle's longitudinal axis (cf. [16]) gives

$$M\dot{V}_x = F_{x_f} + F_{x_r} - F_a - R_{x_f} - R_{x_r} - Mg\sin\theta \qquad (1)$$

where $M$ is the mass of the vehicle, $V_x$ the longitudinal velocity, $F_{x_f}$ and $F_{x_r}$ the front and rear longitudinal tyre forces, respectively, $R_{x_f}$ and $R_{x_r}$ the front and rear tyre forces due to rolling resistance, $\theta$ the angle of inclination of the road, and $F_a$ the longitudinal aerodynamic drag force.

The rolling resistance forces are often modeled as a time-varying linear function of normal forces on each tyre, i.e., $R_x = k_r F_z$, with $k_r$ the rolling resistance coefficient and $F_z$ the vertical load of the vehicle.

The aerodynamic forces can be expressed as

$$F_a = \frac{1}{2}\rho C_d A_F (V_x + V_{wind})^2$$

with $\rho$ being the mass density of air, $C_d$ the aerodynamic drag coefficient, $A_F$ the frontal area of the vehicle (the projected area of the vehicle in the direction of travel), and $V_{wind}$ the wind speed.

Finally, the Pacejka model [17] is used for longitudinal tyre/road interaction forces, $F_x$. They depend on many factors, but essentially on longitudinal slip and normal forces. The normal forces will be computed as realistically as possible with a 10 degrees-of-freedom (d.o.f.) vehicle model – 6 d.o.f. for the chassis and an additional d.o.f. for each wheel.

The wheel rotation dynamics can be expressed as

$$I\dot{\omega} = -rF_x\tau_{e_i} - \tau_{b_i} \qquad (2)$$

where $I$ is the wheel's moment of inertia, $\dot{\omega}$ its angular velocity, $r$ the tyre radius, $\tau_{e_i}$ the applied engine torque, and $\tau_{b_i}$ the brake torque, both applied to each wheel's centre.

The engine torque $\tau_e$ can be expressed in terms of the throttle opening $u_e$ by the expression [18]

$$\tau_e = n u_e \tau_m \left(1 - \beta\left(\frac{\omega}{\omega_m} - 1\right)^2\right)$$

where $n$ is the gear ratio, $\beta$ is an engine torque parameter, and the maximum torque $\tau_m$ is obtained at engine speed $\omega_m$.

Finally, the dynamics between the braking control variable $u_b$ and braking torque $\tau_b$ can be modeled as a second-order linear system [19]

$$\tau_b(s) = \frac{K_b}{s^2 + 2\eta_b\omega_b s + \omega_b^2} u_b(s)$$

with $K_b$, $\eta_b$, and $\omega_b$ the static gain, damping factor, and natural frequency[1], respectively.

## III. DYNAMIC INTER-DISTANCE GENERATION

A reference model proposed by [6] will act as a feedforward term into the longitudinal control law. The basis of this model will be sketched in the next lines.

The inter-distance reference model describes a virtual vehicle dynamics which is positioned at a distance $d_r$ (the reference distance) from the leader vehicle. The reference model dynamics is given by

$$\ddot{d}_r = \ddot{x}_l - \ddot{x}_{f_r} \qquad (3)$$

where $\ddot{x}_l$ is the leader vehicle acceleration and

$$\ddot{x}_{f_r} = u_r(d_r, \dot{d}_r) \qquad (4)$$

is the follower acceleration, which is a nonlinear function of the inter-distance and of its time derivative.

Introducing $\tilde{d} = d_0 - d_r$ in (4), where $d_0$ is the safe nominal inter-distance, the control problem is then to find a suitable control feedforward control $u_r$, when $\tilde{d} \geqslant 0$, such that all the solutions of the dynamics (3) fulfill the following comfort and safety constraints:

- $d_r \geqslant d_c$, with $d_c$ the minimal inter-distance.
- $\|\ddot{x}_r\| \leqslant \gamma_{max}$, where $\gamma_{max}$ is the maximum attainable longitudinal acceleration.
- $\|\dddot{x}_r\| \leqslant J_{max}$, with $J_{max}$ a bound on the driver desired jerk.

[1] Since the braking dynamics is much faster than that of the vehicle, it can be replaced in the vehicle model by an algebraic expression, without loss of realism [20].

The authors of [6] propose to use a nonlinear damper/spring model $u_r = -c|\tilde{d}|\dot{\tilde{d}}$, which can be introduced in the dynamics Eq. (3) to give:

$$\ddot{\tilde{d}} = -c|\tilde{d}|\dot{\tilde{d}} - \ddot{x}_l.$$

The previous equation may be analytically integrated and expressed backwards in terms of $d_r$ as follows

$$\dot{d}_r = \frac{c}{2}(d_0 - d_r)^2 + \dot{x}_l(t) - \beta,\ \beta = \dot{x}_{f_r}(0) + \frac{c}{2}(d_0 - d_r(0))^2. \quad (5)$$

Note that this reference inter-distance depends upon the leader vehicle, distance $d_0$ and parameter $c$, which is, in turn, an algebraic function of safe and comfort parameters $d_c$, $V_{max}$, $\gamma_{max}$ and $J_{max}$ (cf. [6]). Fig. 2 shows how $\gamma_{max}$ influences the reference inter vehicular distance.

Finally, the feedforward control law -or follower acceleration- yields from (4)

$$\ddot{x}_{f_r} = u_r = -c|d_0 - d_r|\dot{d}_r \quad (6)$$

where the inter-distance evolution comes from the numerical integration of (5).

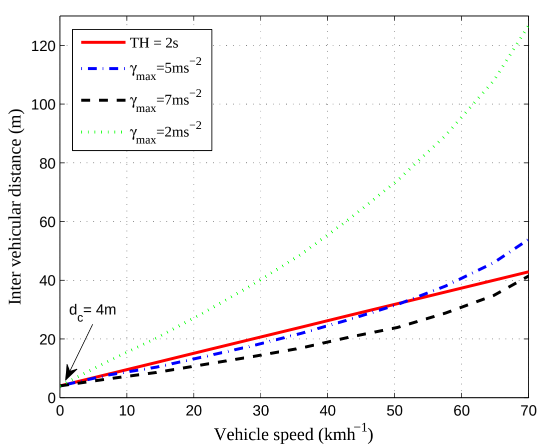


Fig. 2. Comparison of different distance policies: Constant time-headway rule (2 seconds) and the inter-distance model [6] with different maximum accelerations.

## IV. INTELLIGENT PID CONTROLLER[2]

As stated in [21], a finite dimension nonlinear system can be locally written as

$$y^{(\mu)} = F + \alpha u \quad (7)$$

where $\alpha \in \mathbb{R}$ and $\mu \in \mathbb{N}$ are two constant parameters, which do not necessarily represent a physical magnitude, and whose choice is based on the following guidelines

- $\mu$ is usually 1 o 2, and it may represent the system order, but not necessarily.
- $\alpha$ should allow $F$ and $\alpha u$ to be of the same order of magnitude.

The term $F$, which is a sort of non-linear black box identifier [21], is computed with the input value at the preceding sample time $u(t_{k-1})$ and with the $\mu$-th derivative estimation of the output $[y^{(\mu)}(t_k)]_e$ at the current sample time

$$F(t_k) = [y^{(\mu)}(t_k)]_e - \alpha u(t_{k-1}) \quad (8)$$

Using formalism introduced in (7) and Eq. (1) for both vehicles, the relative velocity dynamics can be expressed [15] as follows

$$\ddot{x}_{f_r}(t) = F(t) + \alpha u(t) \quad (9)$$

where $u = \{u_e, u_b\}$ are respectively the engine and brake control variables.

If (9) is inverted and merged with a PI controller [18], the resulting i-PI control law yields

$$u = \frac{1}{\alpha}\left(\ddot{x}_{f_r} - F\right) + K_P e + K_I \int edt, e = \dot{d}_r - (\dot{x}_l - \dot{x}_f) \quad (10)$$

where $K_P$ , $K_I \in \mathbb{R}^+$ are PI gains. Even though this is a nonlinear and varying parameter system, the linearized model of (1) can be useful to classically tune the PI controller (see [18]). Thereafter, two extra parameters $\alpha_e$ and $\alpha_b$ will be chosen (see table II) to enhance the dynamic behavior and disturbance rejection of the closed loop system.

Eq. (10) can be particularized in our case to control the throttle

$$\begin{aligned} u_e(t_k) &= \frac{1}{\alpha_e}\left(\ddot{x}_{f_r}(t_k) - F_e(t_k)\right) + K_{p_e} e(t_k) + \\ &\quad + K_{i_e}\int (e(t_k))\, dt \\ F_e(t_k) &= \hat{\ddot{x}}_f(t_k) - \alpha_e u_e(t_{k-1}) \end{aligned} \quad (11)$$

and to do likewise with the brake

$$\begin{aligned} u_b(t_k) &= \frac{1}{\alpha_b}\left(\ddot{x}_{f_r}(t_k) - F_b(t_k)\right) + K_{p_b} e(t_k) + \\ &\quad + K_{i_b}\int (e(t_k))\, dt \\ F_b(t_k) &= \hat{\ddot{x}}_f(t_k) - \alpha_b u_b(t_{k-1}) \end{aligned} \quad (12)$$

where $\hat{\ddot{x}}_f$ is a velocity derivative estimation.

Finally, a decision rule will be established to determine whether braking or throttle actions are needed. Control law (12) will be used if the reference acceleration is negative[3] and the inter-distance error is lower than a fixed value. In any other case, throttle control law (11) will be used.

TABLE I

I-PI CONTROLLER PARAMETERS

| | $\mathbf{K_p}$ | $\mathbf{K_i}$ | $\alpha$ |
|---|---|---|---|
| Brake (b) | 0.2 | 0.02 | 20 |
| Engine (e) | 0.2 | 0.1 | 20 |

[2]Note that this notation is not related to artificial intelligence techniques, but rather to the capacity to automatically complete what a standard linear controller are unable to do.

[3]Since the reference acceleration do not take into account disturbances such as road slope, this value will be slightly higher than zero to properly handle situations where vehicle is running downhill.

## V. FUZZY CONTROLLER

The second control technique used to be compared is based on soft computing. They are recognized as having a strong learning and cognition capability as well as good tolerance to uncertainty and imprecision. Among them, fuzzy logic - developed by Prof. Lofti A. Zadeh in 1965 - gives a good approximation to the human reasoning and is an intuitive control technique to be applied to autonomous vehicle control since they exhibit high nonlinear behavior.

This control technique consists of a rule base containing expert knowledge and a set of variables representing the considered linguistic values. Functionally, the fuzzy reasoning process can be divided in three stages named fuzzification - the stage in which a crisp input value is converted to a fuzzy value, inference engine - it simulates the human reasoning process by making fuzzy inference on the inputs and IF-THEN rules- and defuzzification - the fuzzy output values are converted to crisp values.

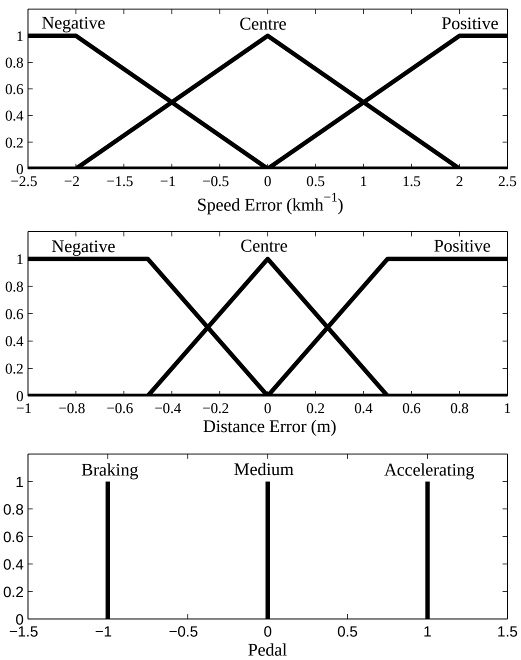


Fig. 3. Membership function definitions for the input (Speed Error and Distance Error) and output (Pedal) variables.

Figure 3 shows the membership functions for the input variables of our fuzzy controller. Two variables have been used to perform the control. On the one hand, the *Speed error* defined as the difference between the leading and trailing car in km/h. On the other hand, the *Distance error* that gives the difference between the distance obtained with the dynamic inter-distance model and the real distance between cars.

As output, the action over the longitudinal actuators - i.e. throttle and brake pedals - is generated. So, the output variable *Pedal* determines which actuator has to be pressed, and the magnitude of the action. The fuzzy output variables membership function shape is defined using Sugeno singletons which are based on monotonic functions. The possible output values are within the range [-1,1], where -1 indicates the brake pedal is completely depressed and 1 indicates the maximum action is applied to the accelerator pedal.

The control algorithm is represented in Fig. 4 as a control surface obtained by plotting the inferred control action Pedal for a grid of values of Speed Error and Distance Error. The appreciable smoothness in changes of slope indicates that the rules selected are appropriate.

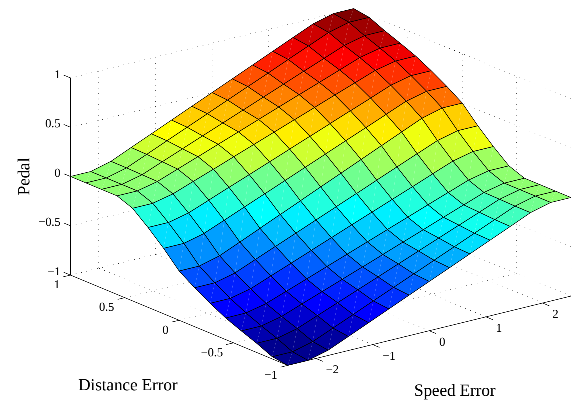


Fig. 4. Speed control surface.

## VI. SIMULATION RESULTS

To evaluate the closed-loop system behavior with both controllers, the vehicle dynamics will be simulated, as aforementioned, with a 10 d.o.f model, which takes into account tires, brakes and engine dynamics.

Measurement noises will be also considered in velocity and acceleration CAN based sensors. These corrupting noises will be modeled as additive white gaussian variables.

Wireless communications are based on a a peer-to-peer Wi-Fi (IEEE 802.11) network, which provides information about the position and velocity of the leader vehicle in the driving zone. A 25 Hz transmission rate will be considered in this work, where received data will be artificially noised to simulate transmission delays.

In Fig. 5 a velocity profile for the leader vehicle is plotted. This scenario has been conceived to evaluate the control algorithms in the wide range of operation, trying to cope with the most demanding maneuvers in urban scenarios. A robust longitudinal control algorithm [25] will be applied to the first vehicle to track the setpoints as precisely as possible. Furthermore, both leader and follower will have to accomplish their control goals while rejecting the disturbance induced by the road slope depicted in Fig. 5, where frequency progressively increases.

Leader and follower cars will be initially separated by a distance of 25 meters and running at the same speed, $\dot{x}_l = \dot{x}_f = 14$ ms$^{-1}$. Besides, the inter-distance dynamic model is parameterized to provide a maximum speed $V_{max} = 20$, a maximum acceleration $B_{max} = 5$ and a minimum inter-distance $d_c = 4$.

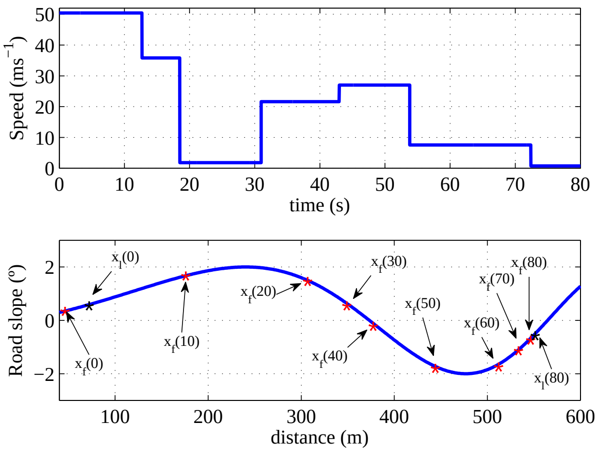


Fig. 5. Leader speed and road slope profiles

The vehicles' speed, the distance between them, the error with respect to the dynamic inter-distance model and the control action evolution are depicted in Fig. VI, which summarizes the most important aspects of the controllers behavior. They will be quantitatively evaluated with two different criteria: inter-distance tracking error $J_1$ and control action softness $J_2$. The former will be computed with integral absolute error

$$J_1 = \frac{1}{T}\int_0^T |d_r - (x_l - x_f)|dt$$

and the latter will be estimated with the mean value of the control actions[4] derivative

$$J_2 = \frac{1}{T}\int_0^T \left(\left|\frac{du_e(t)}{dt}\right| + \left|\frac{du_b(t)}{dt}\right|\right) dt$$

Even if the behavior of both controllers is at first sight satisfactory for the proposed scenario, some important differences can be highlighted in terms of safety, comfort and disturbance rejection.

In the next to bottom graph, one can see the inter-distance error with respect to the dynamic reference -that is plotted in second to top graph of Fig. VI. With both controllers there is an initial transient, that is corrected after 5 seconds approximately. Thereafter, i-PI provides a more precise tracking than fuzzy controller (see table II). Note that the loss of performance of fuzzy controller mainly occurs when a setpoint change in the leader velocity is combined with an important road slope variation. Finally, the comfort indicator of table II -which indicates than i-PI control action is much softer than fuzzy's- coincides with the fluctuating behavior of fuzzy controller in the bottom graph of fig. VI.

To sum up, even if both controllers seem adapted for this kind of application -and the fuzzy approach has a faster tuning process-, the i-PI controller obtains a better tracking quality in a disturbance environment and provides softer control actions.

[4]The sum of engine and brake control variables $u_e$ and $u_b$ is equivalent to *Pedal* variable in the fuzzy control implementation

TABLE II

PERFORMANCE CRITERIA

| Controlador | i-PID | Fuzzy |
|---|---|---|
| $\mathbf{J_1}$ | 0.0965 | 0.1465 |
| $\mathbf{J_2}$ | $2.91 \cdot 10^{-11}$ | $1.24 \cdot 10^{-10}$ |

## VII. CONCLUDING REMARKS

A comparison of two non-model based control approaches have been implemented and compared in simulation for a stop-and-go application. From a dynamic inter-distance model, a reference inter-distance is provided to an intelligent PID controller and a fuzzy controller, that are evaluated in terms of safety, comfort and disturbance rejection for a demanding urban situation. Simulations have shown very interesting results in both cases -specially using the i-PID approach- with no need of physical parameters knowledge and with a high degree of efficiency (performance, computational cost and calibration time). As a consequence, the implementation and test of these techniques on mass produced vehicles will be soon initiated.

## VIII. ACKNOWLEDGMENTS

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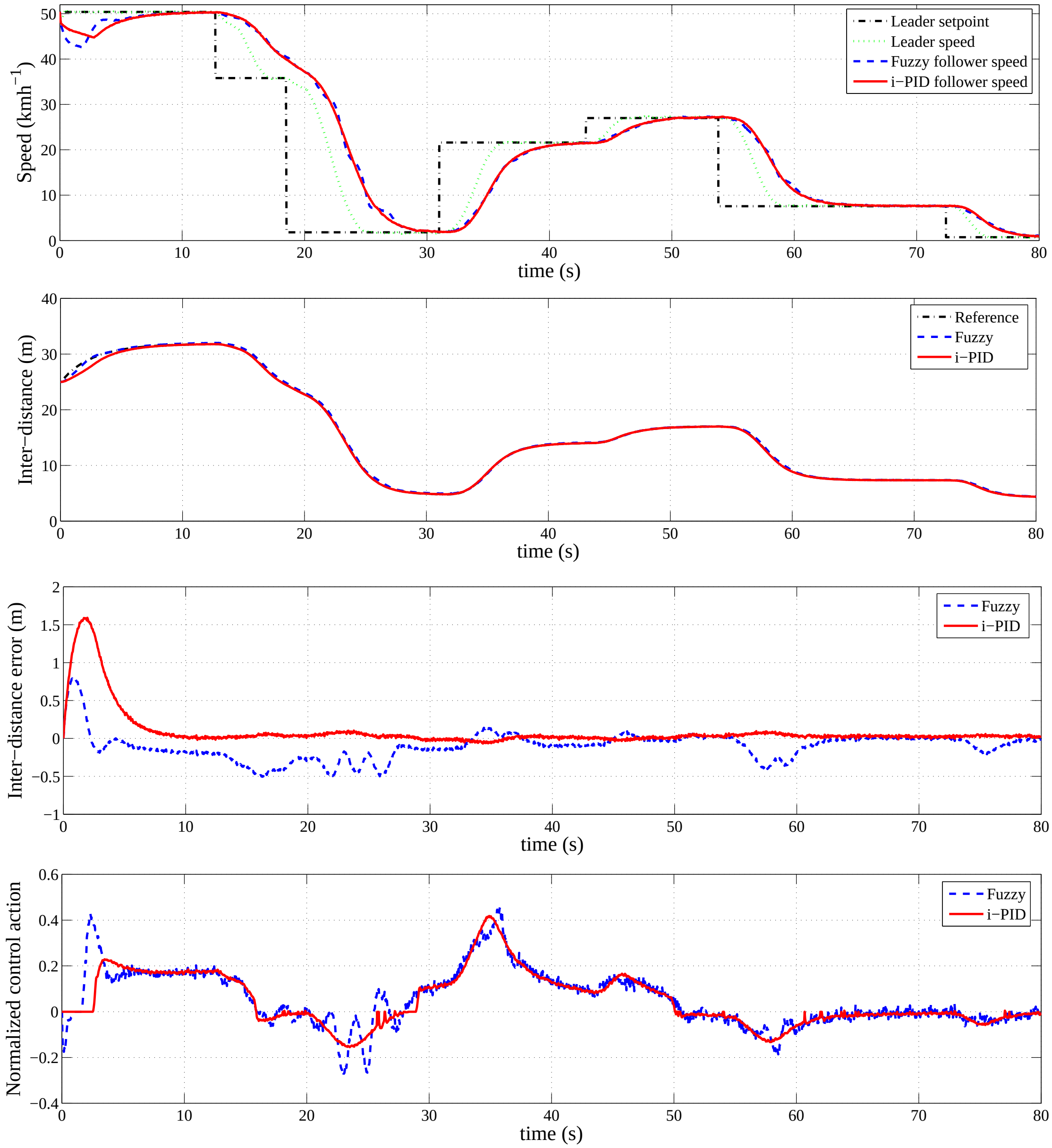


Fig. 6. (a) Leader and follower speed, (b) Inter-vehicular distance (c) Inter-distance error (d) Normalized control action.


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