Traction Control System Using a Fuzzy Representation of the Vehicle Model

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***Abstract*— In this paper, a traction control system has been designed in order to improve vehicle traction on slippery roads. First, a nonlinear model has been formulated from the vehicle longitudinal dynamics, the drivetrain model and the wheel motion. Then, a Takagi-Sugeno (*TS*) fuzzy representation has been proposed in order to take into account the nonlinearities of the obtained model and to simplify the controller design. Based on the obtained TS model, a proportional integral (PI) state feedback controller has been developed. The objective is to compute the optimal engine torque so that the longitudinal slip ratio does not exceed a desired value which is the limit of the optimal traction and the vehicle stability. Lypunov and H_∞ approaches have been used in the controller design and stability conditions are given in terms of Linear Matrix Inequalities (LMI). In order to show the effectiveness of the developed approaches, the vehicle model has been simulated on a slippery road with and without the traction control system.**

I. Introduction

Critical driving situations like a slippery or snowy road can lead to vehicle instability and reduce traction performance. This can be the consequence of the acceleration process which increases the wheel slip. In such situations, longitudinal forces which drive the vehicle are saturated and, moreover, the lateral forces decrease and approach to zero. At this point, the accident risk is great because the vehicle may spin and the front wheels cannot be correctly steered. Traction Control Systems (*TCS*) or Anti Slip Regulation Systems (*ASR*) came up since 1990 to improve the traction, stability and steer-ability of the vehicle. They are widely integrated into various vehicles nowadays.

The objective of a traction control system is to compute the optimal engine torque that can be transmitted to the wheels in order to ensure optimal traction and stability. Many research works on traction control systems have been performed during the past decade. Different control methods have been proposed, some of them are formulated for traction with a combustion engine using throttle valve control [1]-[3]. These approaches need to consider the behaviour of the combustion engine in the control loop [4]. In other studies, many authors have proposed traction control methods for electric vehicles [5]-[7]. Other papers have only considered the traction torque which is applied to the wheel as the control input, using several control methods, including Proportional-Integral-Derivative (*PID*) control, sliding-mode control and fuzzy control [8]-[11]. The advantage of the *PID* controller is the simplicity and its application without a mathematical model, but it is difficult to obtain the control parameters to achieve satisfactory results, especially when the road conditions change. Sliding mode control has good advantages to control nonlinear systems, however, problems stemming from chattering may have negative consequences on the *TCS*. The fuzzy control method requires many tests to find the optimal rules. In this work, we propose a new approach which has the advantage of simplicity despite nonlinearities of the system. The proposed method can be easily extended to consider parameter variations in the vehicle model and the unavailability of vehicle velocity measurement. A robust control method is proposed to avoid wheel slip on slippery roads by operating on the transmitted engine torque. Firstly, a nonlinear state model has been developed from the vehicle longitudinal dynamics, the wheel motion and the drivetrain model. Then, the obtained model is represented using a *TS* fuzzy model. Based on this representation, a fuzzy *PI* state feedback controller is designed and its main objective is to force the longitudinal slip ratio to track the desired value which is the limit of the stable region. An H_∞ tracking performance, which is related to the tracking error, is formulated. The proposed control system design is represented in terms of *LMI* which can be efficiently solved by using the existing *LMI* solvers. Simulation results have been obtained using two driving tests in order to illustrate the effectiveness of the developed techniques.

The paper is organized as follows: the description of the nonlinear vehicle model with the *TS* approximation is presented in Section II. The controller design procedure will be presented in Section III. In Section IV simulation results are given in order to show the effectiveness of the developed controller. Finally, some conclusions are given in section V.

II. Vehicle model analysis

The model used in this work describes the vehicle longitudinal dynamics and wheel dynamics during acceleration. The obtained nonlinear model has been approximated by a *TS* fuzzy representation in order to design the traction control system.

A. Vehicle and wheel motion

The equation of the vehicle longitudinal motion during acceleration in a plane road without considering the driving resistance and lateral dynamics is given as follows

$$M\dot{v}_x = F_{xfl} + F_{xfr} + F_{xrl} + F_{xrr} \qquad (1)$$

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where F_{xfl}, F_{xfr}, F_{xrl} and F_{xrr} are the longitudinal forces acting on the four wheels of the vehicle, M is the vehicle mass and v_x is the longitudinal speed.

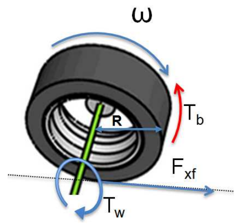

Fig. 1. Wheel motion model

The wheel motion is described by the following equation

$$J_w \dot{\omega}_w = T_w - T_b - RF_x \tag{2}$$

where J_w is wheel inertia moment, ω_w is the wheel angular velocity, T_w is the torque transmitted from the motor to the wheel axe, T_b is the torque generated by the braking, R is the wheel radius and F_x is the wheel traction force.

B. Drivetrain model

The drivetrain is the complete powertrain that propels the car except the engine. The precise components of the drivetrain vary according to the type of vehicle. We here consider a simple drivetrain including the clutch, the transmission, the driveshaft and the wheels [12][13].

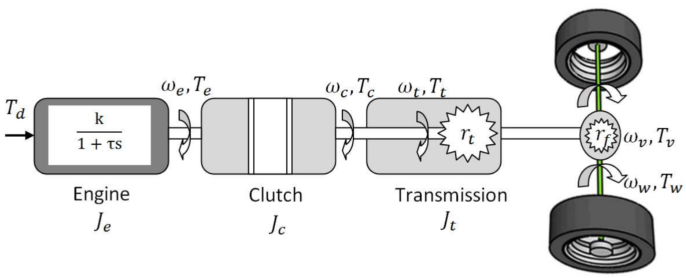

Fig. 2. Simple drivetrain model

For simplicity, the transient characteristics and the damping effects of clutch and transmission are ignored. Applying Newton's second law on each component of the drivetrain the following equations are obtained:

$$J_e \dot{\omega}_e = T_e - T_c \tag{3}$$

$$J_c \dot{\omega}_c = T_c - T_t \tag{4}$$

$$J_t \dot{\omega}_t = T_t - \frac{T_v}{r_t r_f} \tag{5}$$

$$J_v \dot{\omega}_v = T_v - 2T_b - 2RF_x \tag{6}$$

where, J_e, w_e and T_e are the engine inertia, angular speed and torque, respectively, J_c, w_c, and T_c the clutch inertia, angular speed and torque, respectively, J_t , w_t and T_t the transmission inertia, angular speed and torque, respectively. r_t is the transmission gear ratio, r_f the final gear ratio, and T_v the driveshaft torque.

The relation between the driveshaft and the wheel motion is given by

$$\omega_v = \omega_w, \quad J_v = 2J_w$$

It is also assumed that the relation between the different angular speeds are given as follows

$$\omega_e = \omega_c = \omega_t = r_t r_f \omega_v$$

The equivalent model from the engine to the wheel is obtained by substituting (3), (4) and (5) into (6)

$$J_{eq} \dot{\omega}_w = \frac{r_t r_f}{2} T_e - T_b - RF_x \tag{7}$$

where J_{eq} is the equivalent inertia given as follows

$$J_{eq} = \frac{(J_e + J_c + J_t)(r_t r_f)^2}{2} + J_w \tag{8}$$

C. State space model

In this work, only the front wheels are assumed to be drived. It is also assumed that the torque transmitted from the engine to each wheel is the same:

$$F_{xfl} = F_{xfr} = F_x \tag{9}$$

The engine has been considered as a simple actuators represented by a first order transfer function given as follows:

$$T_e = \frac{k}{1 + \tau s} T_d \tag{10}$$

where k and τ are the transfer function parameters which depend on the engine characteristics and T_d is the desired torque given by the driver or the traction controller.

Considering the longitudinal vehicle speed, the angular wheel speed and the engine torque as system states, the following state model can be obtained for the traction situation:

$$\begin{cases} M\dot{v}_x = 2F_x \\ J_{eq}\dot{\omega}_w = \frac{r_t r_f}{2} T_e - RF_x \\ \dot{T}_e = -\frac{1}{\tau} T_e + \frac{k}{\tau} T_d \end{cases} \tag{11}$$

The longitudinal force F_x is proportional to the vertical force F_z and the adhesion coefficient μ which is highly dependent on the road surface conditions [1]:

$$F_x = \mu(\lambda) F_z$$

Consider the effects of longitudinal load transfer during acceleration, the vertical force on the front wheels can be expressed as follows:

$$F_z = \frac{M}{2L}(gl_r - \dot{v}_x h)$$

where L is the distance between the front and rear axles, l_r is the distance from the rear axle to the vehicle Gravity Center (GC) and h is the height of CG.

The adhesion coefficient can be defined as a nonlinear function of the slip ratio between the two contact surfaces.

By definition, slip ratio is the normalized difference between the wheel speed and vehicle speed

$$\lambda = \frac{v_\omega - v_x}{v_\omega}, \qquad where \quad v_\omega = R\omega_w$$

Several formulations have been developed in the literature to aproximate the longitudinal adhesion coefficient (Pacejka, Kienke, Dugoff, etc.). Most of them are functions of the slip ratio using nonlinear relationships, however these models are hardly usable for model based control, since they leeds to very complex models. The adhesion coefficient is assumed in some research works to be proportional to the slip ratio [14], but this assumption leads to a linear model which is only valid for small values of the slip ratio. Hence, for large slip ratio, a nonlinear model must be considered. To overcome this problem, we propose to approximate the nonlinear behavior of the adhesion coefficient by a fuzzy *TS* model with two rules which describes all of the operating regions.

D. TS model description

In order to simplify the controller design, nonlinear model (11) is approximated in this paper by a TS fuzzy model. In the first step, the nonlinearity of the adhesion coefficient $\mu(\lambda)$ can be considered using the following rule based system:

$$\begin{array}{l} If \quad \lambda \quad is \quad M_1 \quad then \quad \mu(\lambda) = C_{\lambda 1}\lambda \\ If \quad \lambda \quad is \quad M_2 \quad then \quad \mu(\lambda) = C_{\lambda 2}\lambda \end{array} \tag{12}$$

where $C_{\lambda i}$ depend on the considered road conditions. $M_1(M_2)$ is a fuzzy set for small (large) slip angles, which fuzzy meaning is given by $h_{\lambda 1}(h_{\lambda 2})$.

Using the above described fuzzy rules, the Adhesion coefficient $\mu(\lambda)$ is written as follows:

$$\mu(\lambda) = h_{\lambda 1}C_{\lambda 1}\lambda + h_{\lambda 2}C_{\lambda 2}\lambda \tag{13}$$

Where membership functions $h_{\lambda i}$, $(i = 1,2)$ satisfy the following properties:

$$\begin{cases} \sum_{i=1}^{2} h_{\lambda i}(\lambda) = 1 \\ 0 \le h_{\lambda i}(\lambda) \ \le 1 \qquad i = 1,2 \end{cases} \tag{14}$$

and are given by the following expressions:

$$h_{\lambda i}(\lambda) = \frac{f_i(\lambda)}{\sum_{i=1}^{2} f_i(\lambda)}, \quad i = 1,2 \tag{15}$$

$$where \quad f_i(\lambda) = \frac{1}{\left(1 + \left|\frac{\lambda - c_i}{a_i}\right|\right)^{2b_i}} \tag{16}$$

Parameters of membership functions (a_i, b_i and c_i) are obtained using an identification method based on the Levenberg-Marquardt algorithm combined with the least square method [15].

Using the *TS* model of the adhesion coefficient given in (13), vehicle model (11) can be represented by the following *TS* fuzzy model:

$$\dot{x}(t) = \sum_{i=1}^{2} h_{\lambda i} A_i(x)x(t) + Bu(t) \tag{17}$$

$$A_i = \begin{bmatrix} -\frac{2C_{\lambda i}F_z}{Mv_\omega} & \frac{2C_{\lambda i}F_z}{Mv_\omega} & 0 \\ \frac{R^2C_{\lambda i}F_z}{J_{eq}v_\omega} & -\frac{R^2C_{\lambda i}F_z}{J_{eq}v_\omega} & \frac{r_t r_f}{2J_{eq}}R \\ 0 & 0 & -\frac{1}{\tau} \end{bmatrix}, \ B = \begin{bmatrix} 0 \\ 0 \\ \frac{k}{\tau} \end{bmatrix},$$

$x = \begin{bmatrix} v_x & v_\omega & T_e \end{bmatrix}^T$, $u = T_d$

Let us now consider the other nonlinearity of model (17) given by $z = F_z/v_\omega$ such that:

$$z_{min} < z < z_{max}$$

$$where \quad z_{max} = \frac{F_{zmax}}{v_{\omega min}}, \quad z_{min} = \frac{F_{zmin}}{v_{\omega max}}$$

Variable z can be written in the following form:

$$z = h_{z1}z_{max} + h_{z2}z_{min} \tag{18}$$

$$with \quad h_{z1} = \frac{z - z_{min}}{z_{max} - z_{min}}, \quad h_{z2} = \frac{z_{max} - z}{z_{max} - z_{min}}$$

Substituting (18) into (17) the TS fuzzy model is obtained in the following form

$$\dot{x}(t) = \sum_{i=1}^{4} h_i(\lambda, z)A_i x(t) + Bu(t) \tag{19}$$

$$where \quad h_i(\lambda, z) = \frac{g_i(\lambda, z)}{\sum_{i=1}^{4} g_i(\lambda, z)}$$

$$g_i(\lambda, z) = h_{\lambda j}h_{zk} \quad for \quad j = 1,2 \quad and \quad k = 1,2$$

Membership functions $h_i(\lambda, z)$, $(i = 1, ..4)$ also satisfy the following properties:

$$\begin{cases} \sum_{i=1}^{4} h_i(\lambda, z) = 1, \\ 0 \le h_i(\lambda, z) \ \le 1 \quad for \quad i = 1, ..4 \end{cases} \tag{20}$$

Matrices A_i are given as follows

$$A_1 = \begin{bmatrix} -\frac{2C_{\lambda 1}}{M}z_{max} & \frac{2C_{\lambda 1}}{M}z_{max} & 0 \\ \frac{R^2C_{\lambda 1}}{J_{eq}}z_{max} & -\frac{R^2C_{\lambda 1}}{J_{eq}}z_{max} & \frac{r_t r_f}{2J_{eq}}R \\ 0 & 0 & -\frac{1}{\tau} \end{bmatrix},$$

$$A_2 = \begin{bmatrix} -\frac{2C_{\lambda 1}}{M}z_{min} & \frac{2C_{\lambda 1}}{M}z_{min} & 0 \\ \frac{R^2C_{\lambda 1}}{J_{eq}}z_{min} & -\frac{R^2C_{\lambda 1}}{J_{eq}}z_{min} & \frac{r_t r_f}{2J_{eq}}R \\ 0 & 0 & -\frac{1}{\tau} \end{bmatrix},$$

$$A_3 = \begin{bmatrix} -\frac{2C_{\lambda 2}}{M} z_{max} & \frac{2C_{\lambda 2}}{M} z_{max} & 0 \\ \frac{R^2 C_{\lambda 2}}{J_{eq}} z_{max} & -\frac{R^2 C_{\lambda 2}}{J_{eq}} z_{max} & \frac{r_t r_f}{2J_{eq}} R \\ 0 & 0 & -\frac{1}{\tau} \end{bmatrix},$$

$$A_4 = \begin{bmatrix} -\frac{2C_{\lambda 2}}{M} z_{min} & \frac{2C_{\lambda 2}}{M} z_{min} & 0 \\ \frac{R^2 C_{\lambda 2}}{J_{eq}} z_{min} & -\frac{R^2 C_{\lambda 2}}{J_{eq}} z_{min} & \frac{r_t r_f}{2J_{eq}} R \\ 0 & 0 & -\frac{1}{\tau} \end{bmatrix}$$

III. CONTROL STRATEGY

The main goal of the control strategy is to provide the tracking of a reference value given by the desired slip ratio. The engine torque is considered as control inputs when the slip ratio exceeds a fixed threshold value (Fig. 3).

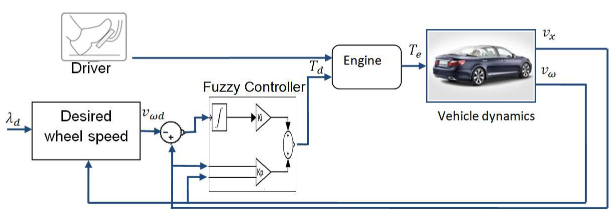

Fig. 3. Global structure of the controller

The idea is to force the wheel speed to track a desired value v_{ω_d} which depends on the desired slip ratio λ_d and the vehicle speed:

$$v_{\omega_d} = \frac{v_x}{1-\lambda_d}$$

The measurement of the vehicle speed and the wheel speed are necessary for the controller design and the membership function computation. The structure of the proposed fuzzy controller is given as follows

$$u(t) = -\sum_{i=1}^{4} h_i \left[K_{Pi} x + K_{Ii} \int_0^t \left(v_\omega - v_{\omega_d} \right) dt \right] \tag{21}$$

Let us denote the tracking error dynamic:

$$\dot{e}_c = v_\omega - v_{\omega_d}$$

Substituting (21) into (19) the following augmented system is obtained:

$$\begin{bmatrix} \dot{x} \\ \dot{e}_c \end{bmatrix} = \sum_{i=1}^{4} h_i \begin{bmatrix} A_i - BK_{Pi} & -BK_{Ii} \\ C & 0 \end{bmatrix} \begin{bmatrix} x \\ e_c \end{bmatrix} + \begin{bmatrix} \mathbb{O} \\ -1 \end{bmatrix} v_{\omega_d} \tag{22}$$

where $\mathbb{O}$ is the zero matrix of appropriate dimensions

In order to simplify the stability conditions of the closed loop system, augmented system (22) can be rewritten as follows

$$\dot{\bar{x}}(t) = \sum_{i=1}^{4} h_i \left[(\bar{A}_i - \bar{B} K_i) \bar{x}(t) + \bar{B}_w w(t) \right] \tag{23}$$

where

$$\bar{x} = \begin{bmatrix} x \\ e_c \end{bmatrix},\ \bar{A}_i = \begin{bmatrix} A_i & \mathbb{O} \\ C & 0 \end{bmatrix},\ \bar{B} = \begin{bmatrix} B \\ 0 \end{bmatrix},\ \bar{B}_w = \begin{bmatrix} \mathbb{O} \\ -1 \end{bmatrix},$$

$$C = \begin{bmatrix} 0 & 1 & 0 \end{bmatrix},\ K_i = \begin{bmatrix} K_{Pi} & K_{Ii} \end{bmatrix},\ w(t) = v_{\omega_d}(t)$$

In order to stabilize closed-loop system (23), H_∞ and Lyapunov approaches will be used to compute the controller gains.

Let us consider the H_∞ performance related to the tracking error e_c with null initial conditions as follows

$$\int_0^\infty e_c(t)^T e_c(t) dt \le \rho^2 \int_0^\infty w(t)^T w(t) dt$$

This H_∞ performance criterion can be rewritten in the following form:

$$\int_0^\infty \bar{x}^T(t) C_e^T C_e \bar{x}(t) dt \le \rho^2 \int_0^\infty w(t)^T w(t) dt \tag{24}$$

with $C_e = [0 \ \ 0 \ \ 1]$.

The objective is to design fuzzy controller (21) such that closed-loop system (23) satisfies the following specifications:

1) Augmented system (23) is globally stable with the controller (21).
2) H_∞ performance (24) for augmented system (23) is satisfied.

The following lemma presents the stabilization conditions of fuzzy system (23) under H_∞ performance (24).

Lemma 1: [16] If there exist a symmetric positive definite matrix P, a positive scalar ρ, solutions of the following inequalities:

$$\begin{gathered} P(\bar{A}_i - \bar{B}_i K_i) + (\bar{A}_i - \bar{B}_i K_i)^T P + \tfrac{1}{\rho^2} P \bar{B}_w \bar{B}_w^T P + C_e^T C_e < 0 \\ i = 1,4 \end{gathered} \tag{25}$$

then fuzzy closed loop system (23) is globaly asymptotically stable and the H_∞ control performance in (24) is guaranteed for a prescribed attenuation level ρ^2.

Corollary 1: If there exist a common symmetric positive definite matrix X, matrices V_i, K_i and scalars $\rho > 0$, solution of the following inequalities:

$$\begin{bmatrix} \Gamma_i & \bar{B}_w & XC_e^T \\ * & -\rho^2 \mathbb{I} & 0 \\ * & * & -\mathbb{I} \end{bmatrix} < 0, \quad i = 1,4 \tag{26}$$

where $\mathbb{I}$ is the identity matrix of appropriate dimensions

$$\Gamma_i = \bar{A}_i X + X \bar{A}_i^T - \bar{B} V_i - V_i^T \bar{B}^T$$
$$V_i = K_i X$$

Then fuzzy closed loop system (23) is asymptotically stable and the H_∞ control performance in (24) is guaranteed for a prescribed attenuation level ρ^2.

Proof 1: Pre- and post-multiply by X where $X = P^{-1}$ and using matrices $V_i = K_i X$ inequality (25) can be expressed in the following form:

$$\overline{\Omega}_i = \bar{A}_i X - \bar{B}_i V_i + X \bar{A}_i^T - V_i^T \bar{B}_i^T + \frac{1}{\rho^2} \bar{B}_w \bar{B}_w^T + X C_e^T C_e X < 0$$

Using Schur's complement, *LMI* conditions (26) are obtained.

Parameters X and V_i (thus $K_i = V_i X^{-1}$) can be easily obtained by solving (26).

IV. Simulation results

In order to illustrate the effectiveness of the designed controller, two tests have been conducted in Matlab/Simulink using nonlinear model (11) on a slippery road. Vehicle parameters are given in table (I).

TABLE I

Parameter values of the vehicle in simulation

M	Jw	Jc	Je	Jt	rt	rf	R
1370	1.2	0.015	0.2	0.04	13.5	3.5	0.33
$[kg]$	$[kgm^2]$	$[kgm^2]$	$[kgm^2]$	$[kgm^2]$	$[m]$	$[m]$	$[m]$

In the first test the input depicted in Fig. 4 has been used as the desired engine torque given by the driver. In the second test, the engine torque has been computed by the controller so that the longitudinal slip ratio does not exceed desired value $\lambda_d = 0.2$.

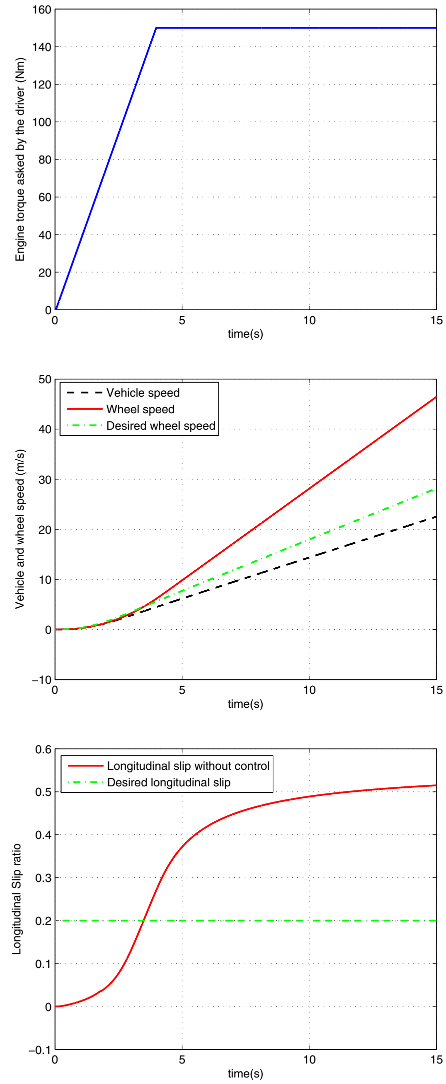

Fig. 4. Vehicle response using engine torque given by the driver

Simulation results in Fig. 4 show the vehicle response in the first test; the wheel speed has increased up to $45m/s$ while the vehicle speed has not exceeded $22m/s$. This implies that the wheel has slipped during this test. The figure also shows the longitudinal slip ratio in this test which exceeds 0.2 at $4s$. We can conclude that in this test the vehicle has lost its stability and the traction has not been at the optimal level. In the second test, the vehicle is equipped with the traction control which computes the optimal value of the engine torque. Simulation results of this test are depicted in Fig. 5. In this test, the wheel speed and the vehicle speed has been increased so that the slip ratio still stays at its desired value. This implies that the vehicle has accomplished the test with the maximum stability and optimal traction performance. In order to show the advantage of the traction control in optimizing energy consumption, the fuel consumption rate has been computed in this simulation in grams per second (g/s) as follows [17]:

$$r_f = S_{fc} P \tag{27}$$

where P is the produced power in watt whith $P = T_e \omega_e$ and S_{fc} is a constant of the specific fuel consumption which depends on the engine and fuel specification. The quantity of the fuel consumption is given in grams and deduced from (27) as follows:

$$q_f = \int_0^t S_{fc} P(t) dt \tag{28}$$

Figure 6 shows that the control traction system can also be a good way to save energy consumption in slipping situations.

V. Conclusion

In this paper we have developed a robust control method for the traction control systems. The objective of the control system is to compute the optimal engine torque in order to improve the vehicle traction and to avoid wheel slipping in critical situations such slippery roads. To achieve this goal, we have proposed an approach to design a fuzzy state feedback controller with integral term. The controller has been based on the Takagi-Sugeno (TS) fuzzy representation of the the vehicle longitudinal dynamics, the drivetrain model and the wheel motion. The proposed constraints can globally asymptotically stabilize the closed loop TS fuzzy system with the minimization of an H_∞ performance criterion in order to satisfy the tracking performances for any desired value of the slip ratio. Simulation results have verified and confirmed the effectiveness of the proposed approach to ensure the vehicle stability and the optimal traction.

VI. Acknowledgment

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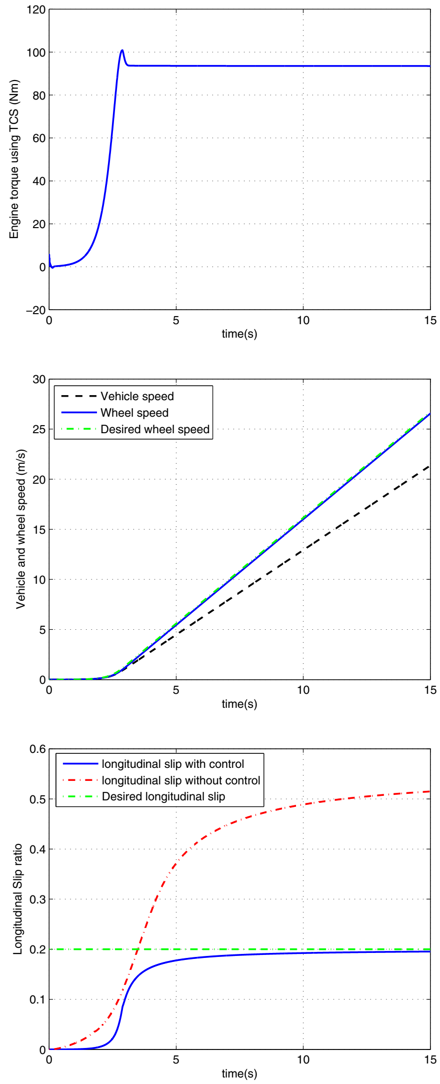

Fig. 5. Vehicle response using engine torque computed by the controller

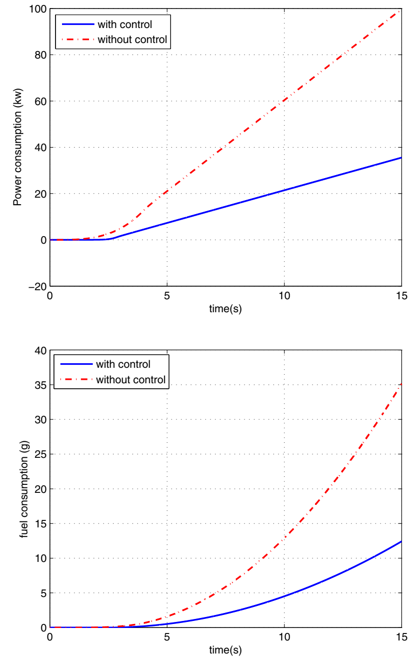

Fig. 6. Power and fuel consumption in test 1 and test 2

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