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A Comprehensive Strategy for Longitudinal Vehicle Control with Fuzzy Supervisory Expert System

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Abstract

The main objectives of vehicle motion control on an automated highway system are stable and safe automatic longitudinal and/or lateral path following in a platoon of vehicles. Various controllers can be used to satisfy the same objectives, but they may require different variables to be sensed or different conditions to be met. Supervision can select a controller and switch to a different controller depending on the conditions. Specifically, a fuzzy supervisory expert system checks for various system conditions and chooses a controller from PID, PI, sliding mode and a fuzzy controller or gives a distress signal. The choice between PID and PI controllers is based on the availability of the error derivative. Robust and complex sliding mode control can counter the external disturbances which the PID and PI controllers cannot handle. The fuzzy controller is used when the sensors are not working perfectly but the sensor values are still reliable enough to define corresponding fuzzy linguistic variables.

1. Introduction

Fuzzy set theory [1] is a useful tool in representing and manipulating subjective linguistic variables, especially for computer-based decision making in the presence of uncertainty [2] and [3]. Fuzzy logic has been used successfully to solve practical control problems [4], [5], [6], and [7]. Probability theory can also deal with uncertainties in decision making [8-11], but there are serious limitations in obtaining numerical probabilities for events for decision making [13] and [14]. We will, instead, use fuzzy logic as a human-like decision maker for a specific application of longitudinal vehicle control.

Longitudinal vehicle motion control is important as a part of an automated highway system [1], [2]; the objective of the control is to maintain a specified headway between vehicles. Various controllers could do this job e.g. PID [1], PI, sliding mode [1] and [2] or fuzzy controllers [3]; each works best when some conditions are met. Supervisory control would be ideal to select one of them [4]. We shall use fuzzy rules to match conditions and decide which controller to be used, or give a distress signal when something goes seriously wrong.

The expert system detects sensor faults using the redundancy in measurement and external disturbances are detected by inferences made from the measured and calculated system and control variables. The default control is PID. If the velocity error signal becomes unreliable, the control is changed to PI. A traction based sliding mode control is used when external disturbances cannot be handled by the PI or the PID controllers, since a traction based sliding mode control is robust to parametric uncertainties and disturbances [1], [2] and [5]. If the sensors are not accurate but the sensor reliability is high enough for the sensor outputs to be transformed into meaningful fuzzy linguistic variables, the expert system gives the control to the fuzzy controller.

We shall next develop system dynamics for simulation and the design of the sliding control. The four types of controllers and the supervisory control are designed next, and finally simulation results are shown.

2. Background

To design an analytic controller, a representative mathematical model of the system is needed. In this section, a mathematical model for vehicle traction control is described [5, 6, 17, 18, 19] for analysis of the system, design of control laws, and computer simulations. Although, the model considered here is relatively simple, it retains the essential dynamic elements of the system.

2.1 System Dynamics

A vehicle model, which is appropriate for both acceleration and deceleration, is described in this sub-section. The model identifies wheel speed and vehicle speed as state variables and wheel torque as the input variable. The two state variables in this model are associated with one-wheel rotational dynamics and linear vehicle dynamics. The wheel dynamics and vehicle dynamics are derived by applying Newton's law.

2.1.1 Wheel Dynamics

The wheel angular dynamic equation is

$$\dot{\omega}_w = [T_e - T_b - R_w F_t - R_w F_w]/J_w \quad (1)$$

where J_w is the moment of inertia of the wheel, ω_w is the angular velocity of the wheel, and the other quantities are as defined in Table 1. The total torque consists of shaft torque from the engine, which is opposed by the brake torque and the torque components due to the tire tractive force and the wheel viscous friction force. The tractive force developed on the tire-road contact surface is dependent on the wheel slip, the difference between the vehicle speed and the wheel speed, normalized by the vehicle speed for braking and the wheel speed for acceleration. Engine torque and the driving wheel effective moment of inertia depend on the transmission gear shifts.

R_w	Radius of the wheel
N_v	Normal reaction force from the ground
T_e	Shaft torque from the engine
T_b	Brake torque
F_t	Tractive force
F_w	Wheel viscous friction

Table 1 Wheel Parameters

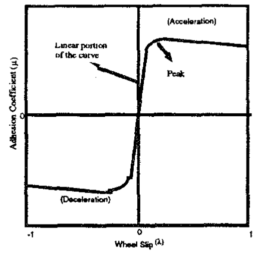

Figure 1 A typical μ–λ Curve

Applying a driving torque or a braking torque to a pneumatic tire produces tractive force at the tire-ground contact patch [14, 20]. The phenomenon referred to as wheel slip [14, 17, 18, 19, 20] is responsible for producing traction. The adhesion coefficient $\mu(\lambda)$ is a function of wheel slip λ. Figure 1 shows a typical $\mu-\lambda$ curve. References [17, 18 and 19] are the sources for the typical curve and [8] gives a more detailed mathematical description of the tire model. Mathematically, wheel slip is defined as

$$\lambda = (\omega_w - \omega_v)/\omega, \ \omega \neq 0 \qquad (2)$$

where, ω_v is vehicle angular velocity defined as

$$\omega_v = \frac{V}{R_w} \qquad (3)$$

which is equal to the linear vehicle velocity, V, divided by the radius of the wheel. The variable ω is defined as

$$\omega = \max(\omega_w, \omega_v) = \begin{cases} \omega_w \text{ for } \omega_w \geq \omega_v \\ \omega_v \text{ for } \omega_w < \omega_v \end{cases} \qquad (4)$$

which is the maximum of vehicle angular velocity and wheel angular velocity.

The tire tractive force is given by

$$F_t = \mu(\lambda) N_v \qquad (5)$$

where the normal tire force, N_v, depends on vehicle parameters such as the mass of the vehicle, location of the center of gravity of the vehicle, and the steering and suspension dynamics. The adhesion coefficient, which is the ratio between the tractive force and the normal load, depends on the road-tire conditions and the value of the wheel slip [5, 17]. For various road conditions, the curves have different peak values and slopes, as shown in Figure 2. The adhesion coefficient-slip characteristics are influenced by operational parameters like speed and vertical load. The average peak values for various road surface conditions are shown in Table 2 [14].

Surface	Av. Peak
Asphalt and concrete (dry)	0.8-0.9
Asphalt (wet)	0.5-0.6
Concrete (wet)	0.8
Earth road (dry)	0.68
Earth road (wet)	0.55
Gravel	0.6
Ice	0.1
Snow (hard packed)	0.2

Table 2 Average peak values for friction coefficient.

The model for wheel dynamics is shown in Figure 3. The parameters in this figure are defined in Table 1. The figure shows the acceleration case for which the tractive force and wheel viscous friction force are directed toward the motion. The wheel is rotating in the clockwise motion and slipping against the ground, i.e. $\omega_w > \omega_v$. The slipping produces the tractive force towards right causing the vehicle to accelerate towards right. In the case of deceleration, the wheel still rotates in the clockwise motion but skids against the ground, i.e. $\omega_w < \omega_v$. The skidding produces the tractive force towards left causing the vehicle to decelerate.

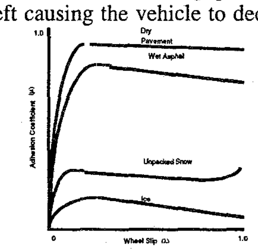

Figure 2 μ–λ Curves for Different Road Conditions

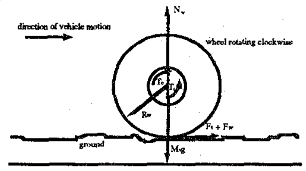

Figure 3 Wheel Dynamics

2.1.2 Vehicle Dynamics

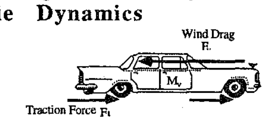

Figure 4 Vehicle Dynamics

The vehicle model considered for the system dynamics is shown in Figure 4. The parameters in the figure are defined as: F_v : Wind drag force (function of vehicle velocity); M_v : Vehicle mass; N_w : Number of driving wheels (during acceleration) or the total number of wheels (during braking).

The linear acceleration of the vehicle is governed by the tractive forces from the wheels and the aerodynamic friction force. The tractive force, F_t, is the average friction force of the driving wheels for acceleration and the average friction force of all wheels for deceleration. The dynamic equation for the vehicle motion is

$$\dot{V} = [N_w F_t - F_v]/M_v \qquad (6)$$

The total tractive force is equal to the product of the average friction force, F_t and the number of relevant wheels, N_w. The aerodynamic drag is a nonlinear function of the vehicle velocity and is highly dependent on weather conditions. It is usually proportional to the square of the vehicle velocity.

2.1.3 Combined System

The dynamic equation of the whole system can be written in state variable form by defining convenient state variables. By defining the state variables as

$$x_1 = \frac{V}{R_w} \qquad (7)$$

$$x_2 = \omega_w \qquad (8)$$

and denoting $x = \max(x_1, x_2)$, we can rewrite Equations (1) and (6) as

$$\dot{x}_1 = -f_1(x_1) + b_{1N}\,\mu(\lambda) \qquad (9)$$

$$\dot{x}_2 = -f_2(x_2) - b_{2N}\,\mu(\lambda) + b_3 T \qquad (10)$$

where

$$T = T_e - T_b$$

$$\lambda = (x_2 - x_1)/x)$$

$$f_1(x_1) = [F_v(R_w x_1)]/(M_v R_w)$$

$$b_{1N} = N_v N_w/(M_v R_w)$$
$$f_2(x_2) = F_w(x_2)/J_w$$
$$b_{2N} = R_w N_v/J_w$$
$$b_3 = 1/J_w \quad (11)$$

The combined dynamic system can be represented as shown in the Figure 5. The control input is the applied torque at the wheels, which is equal to the difference between the shaft torque from the engine and the braking torque. During acceleration, engine torque is the primary input where as during deceleration, the braking torque is the primary input.

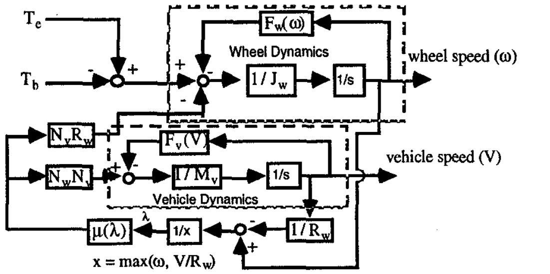

Figure 5 Vehicle/Brake/Road Dynamics: One-Wheel Model

2.1.4 System Dynamics In Terms Of Slip

Wheel slip is chosen as the controlled variable for traction control algorithms because of its strong influence on the tractive force between the tire and the road. Wheel slip is calculated from Equation (3) by using the measurements of wheel angular velocity and the estimated value of the vehicle velocity from either the accelerometer data or the magnetic marker data, as explained in the Introduction. By controlling the wheel slip, we control the tractive force to obtain desired output from the system. In order to control the wheel slip, it is convenient to have system dynamic equations in terms of wheel slip. Since the functional relationship between the wheel slip and the state variables is different for acceleration and deceleration, the equations for the two cases are described separately in the following sub-sections.

Acceleration Case

During acceleration, condition $x_2 > x_1, (x_2 \neq 0)$ is satisfied and therefore

$$\lambda = (x_2 - x_1)/x_2 \quad (12)$$

Differentiating this equation, we obtain

$$\dot{\lambda} = [(1 - \lambda)\dot{x}_2 - \dot{x}_1]/x_2 \quad (13)$$

Substituting Equations (9), (10) and (12) into Equation (13), we obtain

$$\dot{\lambda} = [\,[f_1(x_1) - (1 - \lambda)f_2(x_2)] - [(1 - \lambda)b_{2N} + b_{1N}]\mu + (1 - \lambda)b_3T]/x_2 \quad (14)$$

which is the wheel slip dynamic equation during acceleration. This equation is highly nonlinear and involves uncertainties in its parameters. The non linearity of the equation is caused by the following factors: (1) the relationship of wheel slip with wheel velocity and vehicle velocity is nonlinear, (2) the $\mu-\lambda$ relationship is nonlinear, (3) there are multiplicative terms like $(1 - \lambda)b_{2N}\mu/x_2$ and $(1 - \lambda)b_3T/x_2$ in the equation, and (4) functions $f_1(x_1)$ and $f_2(x_2)$ are nonlinear.

Deceleration Case

During deceleration, condition $x_2 \leq x_1, (x_1 \neq 0)$ is satisfied, and therefore wheel slip is defined as:

$$\lambda = (x_2 - x_1)/x_1 \quad (15)$$

Differentiating this equation, we obtain

$$\dot{\lambda} = [\dot{x}_2 - (1 + \lambda)\,\dot{x}_1]/x_1 \quad (16)$$

Substituting Equations (9), (10) and (15) into Equation (16), we obtain

$$\dot{\lambda} = [\,[(1 + \lambda)f_1(x_1) - f_2(x_2)] - [b_{2N} + (1 + \lambda)\,b_{1N}]\mu + b_3T]/x_1 \quad (17)$$

This gives the wheel slip dynamic equation for deceleration case. This equation is also highly nonlinear and involves uncertainties like Equation (14).

3. Control Design

3.1 PID Control

For the PID control, the control law is given by

$$T = -k_1\dot{\varepsilon} - k_2\varepsilon - k_3\int_0^t \varepsilon\, dt \quad (18)$$

Here ε is the spacing error between the vehicle being controlled and the vehicle in front, i.e the difference between the actual and desired distances between the two vehicles. When the desired distance between the two vehicles is a constant $\dot{\varepsilon}$ becomes the velocity error.

This control is fast and easy to calculate. It requires the spacing error and the velocity error. The integral of the spacing error can be calculated by accumulating the error.

3.2 PI Control

For the PI control, the control law has only the proportional and the integral term and lacks the derivative term of the PID control law:

$$T = -k_2\varepsilon - k_3\int_0^t \varepsilon\, dt \quad (19)$$

This law is also fast and easy and requires only the spacing error signal. This law, however has a longer transient behavior.

3.3 Sliding Mode Control (SMC)

A multiple surface adaptive sliding adaptive SMC is designed next. First a slip control is designed which can maintain the wheel slip at any value which in turn controls the available traction due to the slip-traction relationship shown by the $\lambda-\mu$ curve. Then the sliding surface with spacing errors is used to calculate the desired traction to be used by the slip control. The slip dynamic equation for acceleration can be written as

$$\dot{\lambda} = f + bu \quad (20)$$

with

$$f = \frac{1}{x_2}[f_1(x_1) - (1 - \lambda)(f_2(x_2) + b_{2N}\mu(\lambda)) - b_{1N}\mu(\lambda)] \quad (21)$$

$$u = \frac{(1 - \lambda)}{x_2}T \quad \text{and} \quad b = b_3 \quad (22)$$

Define the switching surface as

$$s = \lambda_e + c\int_0^t \lambda_e dt \quad (23)$$

where, $\lambda_e = \lambda - \lambda_d$; d stands for desired.

We estimate f as $\hat{f}$, so that $|f - \hat{f}| \leq F$. There is an upper bound and a lower bound to b as $0 < b_{min} < b < b_{max}$. Define $\beta = \sqrt{b_{max}/b_{min}}$ and take $\hat{b} = \sqrt{b_{max}b_{min}}$. The controller is designed as

$$T = \frac{x_2}{(1-\lambda)}u \qquad (24)$$

$$u = \hat{b}^{-1}[\hat{u} - k\,\mathrm{sgn}(s)] \qquad (25)$$

$$\hat{u} = -\hat{f} + \dot{\lambda}_d - c\lambda_e \qquad (26)$$

The stability of the system is guaranteed with a finite reaching time towards the switching surface if we take

$$k \geq \beta(F+\eta) + (\beta - 1)|\hat{u}| \qquad (27)$$

To reduce chattering and make s convergent towards zero, we define a boundary of a fixed width ϕ and define a function ipush(.) as [20]

$$\mathrm{ipush}(a,p,s,\phi) = as/\phi + \frac{p}{\phi}\int_0^t s\,dt \quad \text{for } |s| \leq \phi \qquad (28)$$

$$\mathrm{ipush}(a,p,s/F) = \mathrm{sgn}(s) \qquad \text{otherwise}$$

take $a = 2b\phi/k(x_d)$ and $p = b/2$ and change u to

$$u = \hat{b}^{-1}[\hat{u} - k\,\mathrm{ipush}(a,p,s,\phi)] \qquad (29)$$

Similarly, for deceleration we can obtain the system in the form of equation(15) by substituting

$$f = [(1+\lambda)f_1(x_1) - f_2(x_2)] \qquad (30)$$
$$-[b_{2N} + (1+\lambda)\, b_{1N}|\mu]/x_1$$

$$u = T/x_1 \qquad (31)$$

$$\text{and } b = b_3 \qquad (32)$$

and therefore the control design steps are the same.

Adaptive Sliding Mode :

By making the above described scheme adaptive, we can reduce the amount of control discontinuity due to the uncertainty on b. Adaptation is done only outside the boundary layer so that the effect of noise on adaptation is minimal. For the same system as mentioned above, we define

$$\Delta s = s - \alpha(.)\mathrm{ipush}(a,p,s,\phi) \qquad (33)$$

$$\alpha(.) = \phi/[a + \frac{p}{s}\int_0^t s\,dt] \qquad (34)$$

Notice that within the boundary $\Delta s = 0$ and outside the boundary $\Delta s = s - [\phi/a]\mathrm{sgn}(s)$, because the integral is zero outside the boundary. Define a Lyapunov function as

$$V = \frac{(\Delta s)^2}{2} + \frac{(b - \hat{b})^2}{2} \qquad (35)$$

for outside the boundary. Differentiating and rearranging, we obtain

$$\dot{V} \leq -\eta|s| \qquad (36)$$

by taking

$$k \geq \beta(F+\eta) \qquad (37)$$

and the adaptation law as

$$\dot{\hat{b}} = -\frac{\hat{u}}{\hat{b}}\Delta s \qquad (38)$$

The block diagram of the control is shown in fig. 6. In this figure, subscript e refers to the estimate of the variables and D refers to the Laplace operator. The dashed line shows the adaptation of b estimate.

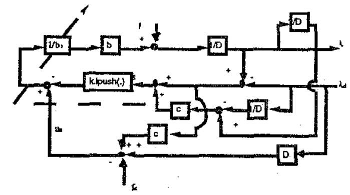

Figure 6

The actual control is obtained by using two sliding surfaces, one for tracking the vehicle velocity and the second to obtain the torque control (described above). The overall technique is discussed below.

Let s_1 be the first sliding surface.

$$s_1 = \dot{\varepsilon} + c_1\varepsilon + c_2\int_0^t \varepsilon\,dt \qquad (39)$$

Here ε is the spacing error between the vehicle being controlled and the vehicle in front as previously defined. When the desired distance between the two vehicles is a constant $\dot{\varepsilon}$ becomes the velocity error.

$$\dot{s}_1 = \ddot{\varepsilon} + c_1\dot{\varepsilon} + c_2\varepsilon = 0 \qquad (40)$$

Here $\ddot{\varepsilon} = \ddot{x}_1 - \ddot{x}_{1des}$ and x_{1des} is the desired x_1. For the first sliding surface $\mu(\lambda)$ is considered a pseudo control input, which is further controlled by the second sliding surface using the system control input T. Using the above mentioned substitutions and introducing - ksgn(s_1) term for robustness, we get

$$\ddot{x}_1 = \ddot{x}_{1des} - c_1\dot{\varepsilon} - c_2\varepsilon - k\mathrm{sgn}(s_1) \qquad (41)$$

$$-\hat{f}_1(x_1) + \hat{b}_{1N}\hat{m}(l) = \ddot{x}_{1des} - c_1\dot{e} - c_2 e - k\mathrm{sgn}(s_1) \qquad (42)$$

$$\hat{\mu}(\lambda) = \frac{1}{\hat{b}_{1N}}\left[\ddot{x}_{1des} + \hat{f}_1(x_1) - c_1\dot{\varepsilon} - c_2\varepsilon - k\mathrm{sgn}(s_1)\right] \qquad (43)$$

$\hat{\mu}(\lambda)$ is the desired friction value from which the desired slip λ_{des} can be obtained by using the estimated $\lambda-\mu$ curve. $\hat{f}_1(x_1)$ is the estimated value of $f_1(x_1)$. For chattering reduction the function sgn(.) is replaced by ipush(.), and the law can be made adaptive. To obtain the desired slip, we try to control the wheel slip directly at the desired value using the algorithm of the slip-control.

3.4 Fuzzy Control (FC)

Fuzzy control is based on fuzzy set theory [8]. The plant output which belongs to the crisp set of real numbers is fuzzified so as to be able to be used by the FC. The FC provides the control input for the plant in fuzzy terms, which is passed through a defuzzifier to transform it to a crisp number to use it as the control input. The process is outlined in fig. 7.

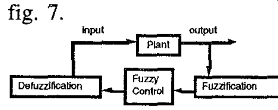

Figure 7

The FC process can be divided into the following three parts:

Fuzzification: The plant variables which are measured by the sensors and are changed into fuzzy variables by using membership functions for linguistic categories in which they could fall. The variables which are fuzzified are vehicle velocity, spacing error, velocity error and wheel slip. For the spacing error and the velocity error there are only two categories, positive(P) and negative(N). The wheel slip has six categories, positive small(PS), positive medium(PM), positive big(PB), negative small(NS), negative medium(NM) and negative big(NB). The change in torque also has the same six categories. The membership functions, which can have values from zero to one are monotonic functions or triangular as shown in fig. 8. Notice that the membership function of PB and NB for change of torque is a point.

Fuzzy control: Fuzzy control rules are written in terms of "if then" rules. The membership function of a fuzzy conjunction "A and B" is given by

$$\mu(x,y) = \min[\mu_A(x),\mu_B(y)] \quad (44)$$

where m denotes the membership function. The membership function of disjunction "A or B" is

$$\mu(x,y) = \max[\mu_A(x),\mu_B(y)] \quad (45)$$

The membership function of the outcome of a rule is determined by the membership function of the variable it is dependent on. The outcome in this case is the change in torque. For instance if the spacing error and the velocity errors both come out to be positive big and there is a rule that "if spacing error PB and velocity error PB then change in torque NB", then the membership function of the change in torque will be the minimum of the membership values of spacing error and the velocity error in the class PB.

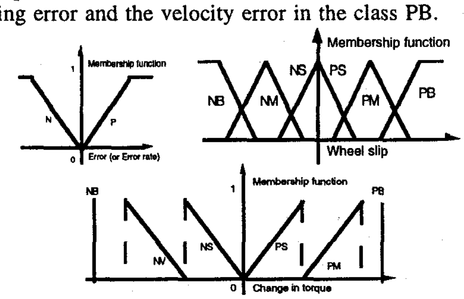

Figure 8

Defuzzification: The membership function of the control variable "change in torque" is also kept monotonic so that the defuzzification process is just the corresponding value of the variable to the membership value as shown in fig. 9. In the figure DT refers to "change in torque".

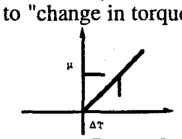

Figure 9

4. Fuzzy Supervisory Expert System (FSES)

The fuzzy supervisory expert system is a macro controller used to select the lower level control strategy, (PID, PI, SMC, or FC) at various times during vehicle motion. FSES also decides when none of the controllers is sufficient; in that case it gives a distress signal.

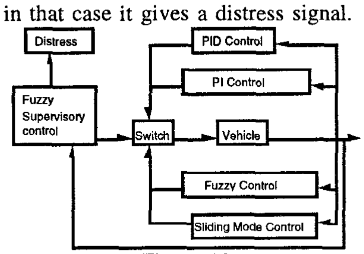

Figure 10

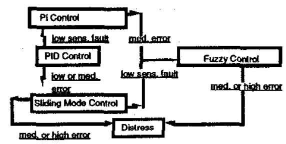

Figure 11

The FSES makes decisions based on various sensor readings. For the simulation study discussed later in this paper, FSES employs two variables, namely spacing error and sensor fault. The spacing error is obtained directly from the sensor, and the sensor fault is detected using redundancy of sensors. The difference in the corresponding outputs measures the sensor fault.

The crisp values of spacing error and sensor fault are converted to fuzzy linguistic variables using triangular membership functions. The sensor fault is divided into zero, low, medium, or high values, and spacing error into PS, PM, PB, NS, NM, or NB, as was done for wheel slip in the design of FC.

The database of FSES consists of all the fuzzy rules written in "if then" form. The "if" part of the rules checks which controller the vehicle is using at that time and the fuzzy values of the spacing error and sensor fault. The "then" part of the rule conveys which controller to switch to after deciding if a switch has to be made. The "then" part of the rule also gives a distress signal for appropriate conditions.

The FSES algorithm starts by obtaining the two input variables. Then it changes the variables into fuzzy linguistic terms e.g. low error, medium velocity, etc. and compares the condition to the conditions in the database and gives a command according to the matched rule. The command is then to continue with the controller, switch to a different controller or to give a distress signal. The overall architecture is shown in fig. 10. Some of the rules of the supervisor are shown in the form of a flowchart in fig. 11. A rule from the flowchart is for instance : if a low sensor fault is detected while using SMC, then switch to FC.

5. Simulations

The simulation is performed for the acceleration case . A typical μ-λ curve is taken and considered fixed. We assume a level (flat) road. During acceleration, only the two wheels attached to the engine are considered because the inertia of the other two becomes negligible. The vehicle parameters for simulation during acceleration are given in Table 1.

Parameters	Values
mass of the vehicle (M_v)	1000 Kg
radius of the wheel (R_w)	0.31 m
wheel inertia (I_w)	0.65 Kg/m
engine inertia (I_e)	0.429 Kg/m
overall gear-ratio (r)	9.5285
normal tire force (N_v)	2287 N
road elevation (θ)	0 deg
number of wheels (N_w)	2
Equivalent wheel inertia	$I_w + I_e r^2/2$ Kg/m
$f_1(x_1)$	cx_1^2 N, c = 0.595
$f_2(x_2)$	0 N

Table 1

The μ-λ curve is shown in fig. 2 with maximum μ value to be 0.2 and the corresponding λ value to be 0.165.

Maximum allowable torque was taken as 571.71 Nm/sec and the minimum as -1000 Nm/sec. Using Table 1, following parameters are calculated to be: $b_{1N} = 14.7548$, $b_{2N} = 35.2284$, $b_3 = 0.0497$. Parametric uncertainty is taken to be about 25% for all parameters while using SMC. The sampling rate in the simulation is 0.5 KHz.

In the simulation, a small sensor disturbance is introduced from 0.2 to 0.3 seconds and from 1.5 to 1.6 seconds so that the control changes from PID to PI and from 0.5 to 0.55 seconds a high wind disturbance is introduced so that the control is changed to sliding mode after the supervisor detects the disturbance. Also from 1.6 to 1.7 seconds, a medium sensor disturbance is introduced, so that the FC comes into effect.

The simulation results are shown in fig. 12 and fig. 13. The control numbers refer to different controls as: (1) PID control, (2) PI control, (3) SMC, (4) FC, and (5) Distress.

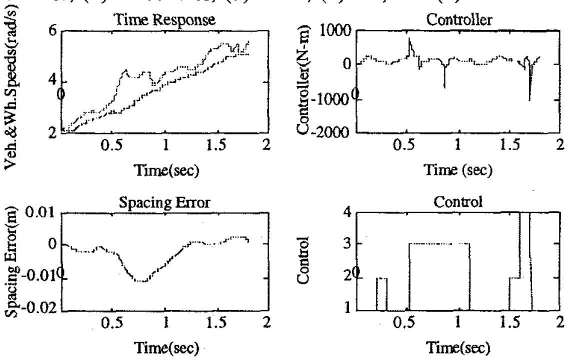

Figure 12

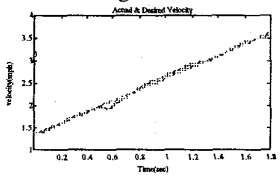

Figure 13

6. Conclusions

Various controllers were designed for longitudinal vehicle control and a fuzzy supervisory expert system was built to decide which control to be used and when to switch from one control law to another. A simulation study showed promising results for the high level complex control strategy. This strategy can increase the range of operation of various controllers and can improve their performance in the presence of disturbances and faults.

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