

Vehicle Longitudinal Control with the Sliding Mode Method Considering Uncertainty of the Model

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Abstract: In this paper, the control algorithm for longitudinal control of a passenger car that is moving on the highway and urban traffic is provided. Longitudinal dynamic equation including resisting forces acting on the vehicle presented and structural and non-structural uncertainties in the model are investigated. To control the speed of vehicles on the highway and vehicle distance control in city traffic, vehicle longitudinal control laws have been proposed. To control the speed of vehicles on the highway and in city traffic, vehicle longitudinal control laws have been proposed. In order to control the nonlinear systems, the sliding mode method which is a robust control method has been used for the design controller. The controller consists of two algorithms as speed control and distance control that their duty is done with the force generation to move the vehicle and overcome the disturbances and uncertainties using tracing of it by car. Finally, to ensure the ability of control and efficiency under different conditions, the control rules are applied to the vehicle dynamics equation.

Keywords: Intelligent transportation systems, Vehicle longitudinal control, Cruise control, Sliding mode, Uncertainty.

1. INTRODUCTION

Intelligent transportation systems mainly are used in computers, controllers, and automation technology to increase the performance of transportation and lead to reducing energy consumption and its environmental effects. Intelligent vehicles are also an integral part of intelligent transportation systems. These vehicles have the ability of perception, reasoning, and operator tools and are capable of driving tasks such as safe movement in a line, avoiding collisions with obstacles, overtaking in slow traffic, and chasing the car ahead automatically. Generally, the main motivation for the construction of intelligent vehicles is achieving safer, more comfortable, and more efficient driving.

The cruise control (CC) system is an auxiliary system that consists of maintaining the vehicle speed at a user (driver) pre-set speed. For instance, if the user wants to stabilize speed to 80 Km/h, he can activate the cruise controller and take his foot off the throttle pedal. In this case, an automated system is applied for the throttle control and fuel injection. As a result, the long journey will not be boring and harmful for drivers. The reasons that cruise control by which enhance vehicle safety and reduce human errors are as follows:

- Reducing driver tiredness and maintaining his performance by strengthening his detectability and judgment.

- The driver gets peace of mind from the vehicle's speed
- Pop up a warning sign in emergency situations
- Taking control of the vehicle and reducing the vehicle's speed when confronted with obstacles
- Not exceeding the speed limit on the road happens unconsciously and in turn is dangerous.

Lee *et al.* [1] presented an impact assessment of enhanced vehicle safety, as an example of automatic driving systems, by adaptive cruise control (ACC) systems. They estimated that 35% of the accidents could be reduced by the ACC system based on safety assessment. However, in the case of a cut-in of other vehicles, the system cannot be guaranteed safety as well as in other cases.

The first automated system that was applied by Watt and Boulton [2] as a speed-limiting for motor vehicles was the mechanical governor of locomotives. This governor maintained the vehicle speed almost uniformly. Other researchers worked on other types of the governor with various specifications and features which were reviewed by Maxwell [3] in 1868. In the following, Kammerhoff [4] invented a new and improved electrically-operated speed-limiting device for motor vehicles so that if the car attains a speed in excess of the predetermined limit the supply of air-fuel mixture to the engine cylinders is automatically cut down to decrease the speed of the car.

The conventional cruise control systems are only designed to maintain the vehicle speed at a pre-set limit, but they are not capable of avoiding collision with

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the vehicle ahead. In Iran, a cruise system combined with a PID controller is simulated to speed control a Samand vehicle [5]. In this regard, Ganji *et al.* [6] showed that using a PID controller cannot be reliable for speed control and the system output produces high overshoot and sometimes volatility. Choi *et al.* [7] presented a sliding mode control for a new anti-lock brake system of a passenger vehicle using electrorheological (ER) valves.

In the last decade of the 20th century, an advanced type of conventional cruise control system developed as the form of adaptive cruise control (ACC). ACC is similar to conventional cruise control that it keeps the vehicle pre-set speed. In addition, this new system can automatically adjust speed in order to maintain a proper headway distance (gap) between vehicles on the same line. It is Gredes and Hedrick [8] organized a combined engine and brake controller for automated highway vehicles based on the ideas of multiple-surface sliding control. The efficiency of the powertrain component in its longitudinal model was considered 100 percent and the Power loss was regardless. Hedrick continued his research in the field of longitudinal control [9], and eventually, in 2003, He developed his control algorithm for speed and distance control of heavy-duty trucks, in collaboration with Yun-Lu. Their experimental works showed that fuel control achieved good performance with only 2 meters distance tracking error. Simultaneously, Naranjo *et al.* [10] presented an ACC controller based on fuzzy logic and introduced specific fuzzy rules to control vehicles. According to their simulation results, speed tracking in higher ranges, has fewer errors and longer response time. Abdullah *et al.* [11] were other researchers who applied fuzzy logic to design their vehicle speed control. In 2012, Tanok and Parnichkun [12] presented a hybrid algorithm in order to longitudinal control in which nonlinear terms of dynamic equations were regardless. Using the first-order transfer function, they determined the brake command to maintain a proper gap based on the current speed of the linearized model. This controller was implemented based on sliding mode control of the throttle valve combined with fuzzy logic control of the brake pedal.

In addition to the researches listed above, the ACC method was the main research interest of other researchers such as Liang and Peng [13] who applied it to design an adaptive cruise control. The cost function of the higher-level controller is defined to minimize tracking errors. In this regard, the Z-transfer function was used to solve this optimization problem, and the acceleration produced by this controller was tracked by the lower-level controller. Dellnitz *et al.* in 2014 [14], developed an intelligent cruise control for an

electric vehicle based on the formulation of an optimal control problem. According to their results, this controller had the potential to decrease the overall energy consumption. A vehicle speed and (vehicle-to-vehicle) distance control algorithm were designed by Karami and Azadi [15] that was used for vehicle stop-and-go cruise control. The desired acceleration was obtained through a proportional controller and linear-quadratic optimal control theory for speed control and distance control, respectively.

Model imprecision is one of the main problems of practical work in the design of nonlinear systems which is called uncertainty. From the control point of view, modeling inaccuracies can be classified into two major kinds:

- Structured or parametric (comes from measurement errors)
- Unstructured uncertainties (come from the purposeful choice of simplified representation of the system's dynamic)

Modeling inaccuracies can have strong adverse effects on nonlinear control systems. Therefore, any practical design must address them explicitly [16]. As is evident in research conducted in the field of speed and distance control, optimal control theories, and fuzzy methods absorbed more interest rather than robust control which are suitable to overcome uncertainties. In this regard, Nemeth and Gaspar [17] investigated the speed control of a vehicle based on the H_∞ method. Considering uncertainty in the weight of the vehicle, they could do speed tracking as well. The first difference between their research and the aforementioned references is considering parametric uncertainty in the model. In addition, they considered the desirable driving force of wheels instead of the desirable accelerator as the controller output to better performance of speed tracking. They disregarded longitudinal wind turbulence applied to the vehicle. Furthermore, they did not examine the uncertainties caused by changes in the rolling resistance force. Also, their proposed controller is only applicable for speed control and does not include the stability of distance control.

There has been little attention paid to the system uncertainties despite their extensive use in a variety of industrial applications. Therefore, the novelty of this work is as: Both structural and non-structural uncertainties are considered in the proposed vehicle's model. In addition, in this study, the longitudinal wind turbulence is applied to the vehicle model; also, the vehicle speed and distance control are designed based on the sliding mode control, which is a powerful method

for controlling unknown nonlinear systems. Meanwhile, the losses in the power transmission line from the engine to the wheels and the needed torque to move the vehicle is calculated.

2. VEHICLE SYSTEM DYNAMIC EQUATIONS

2.1. Vehicle Dynamics

A motor vehicle is made up of many components distributed within its exterior envelope. Since the distance between the particles and vehicle components are always constant, we can assume the vehicle is a rigid body to do behavior analysis. In the rigid body motion analysis, all of the external forces and torques are applied to the center of mass. Therefore, any movement of the center of mass represents the movement of the entire collection. For many of the elementary analyses applied to the vehicle, all components move together. For example, under braking, the entire vehicle slows down as a unit; thus, it can be represented as one lumped mass located at its center of gravity (CG) with appropriate mass and inertia properties. For acceleration, braking, and most turning analyses this estimation is sufficient (Figure 1).

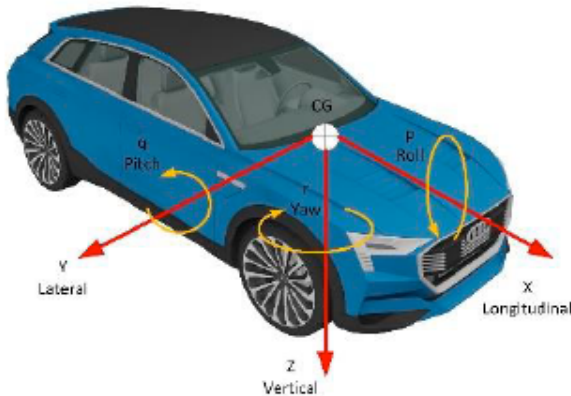


Figure 1: Degree of freedom of car as a rigid body.

As is evident in Figure 1, there are six degrees of freedom for a vehicle as a free rigid body, and consequently, six equation of motion is needed to express its position. It means that three equations for translation of the mass center and the other three equations for rotational movement around the mass center can be written based on Newton's Second Law. However, since the longitudinal control of the vehicle is the main goal of this study, the vehicle movement is considered only in the vehicle's longitudinal direction. Also, the assumed vehicle is earth-fixed and stable. Therefore, rotation in roll (around the X-axis), pitch (around the Y-axis), and yaw (around the Z-axis) and translating in the vertical and lateral direction are neglected. Considering the vehicle shown in Figure 2,

the equation of motion in the assumed vehicle can be written as Eq. (1) [18]:

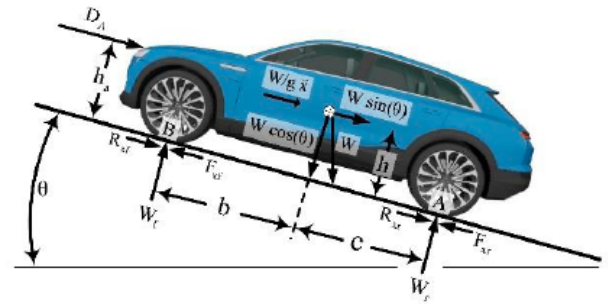


Figure 2: Longitudinal and vertical forces applied to the vehicle.

$$m\ddot{x} = \frac{W}{g}\ddot{x} = u - R_x - D_A - W \sin \theta \tag{1}$$

where m , \ddot{x} , W , u , D_A , R_x , and θ are respectively: the mass of the vehicle, longitudinal acceleration, the weight of the vehicle, the driving or braking force of the vehicle applied to the road (controller output), rolling resistance force, aerodynamic drag force acting on the body of the vehicle and the angle of road related to X-axis, respectively.

2.2. Model Uncertainties

From Eq. (1) we can rewrite the dynamic equation of the vehicle in its longitudinal direction as below:

$$m\ddot{x} = u + f(x) \tag{2}$$

where u is the applied force from the vehicle to the road and $f(x)$ is the total load of the road which can be driven from the combination of rolling force, aerodynamic drag force, and the weight of the vehicle.

$$f(x) = -C_r mg \cos \theta - \frac{1}{2} \rho C_d A \dot{x}^2 - mg \sin \theta \tag{3}$$

In Eq. (3), C_r , g , ρ , C_d , A , and \dot{x} are the coefficient of the rolling resistance force, gravitational acceleration, air density, the coefficient of drag force, frontal surface area, and vehicle speed, respectively. As the differential form is needed for the proposed controller, both sides of Eqs. (2) and (3) are divided by m as follows:

$$\ddot{x} = m^{-1}u + f(x) \tag{4}$$

$$f(x) = -C_r g \cos \theta - \frac{1}{2} m^{-1} \rho C_d A \dot{x}^2 - g \sin \theta \tag{5}$$

A cruise control system requires measuring various parameters including vehicle speed, vehicle acceleration, distance to the car ahead, and road ramp. In practice, these measurements are not precise and contain an error. Also, the weight of the vehicle is variable due to the number of passengers, fuel consumption, and tank discharge. Considering the curb

weight (nominal weight) for a small car, 1400 N is needed to overcome the load axle. However, if we consider passengers and additional loads, this force will rise to about 1800 N. This force difference has to be taken into account in the controller design; otherwise, the speed stabilization will not be done correctly and contains an error. So, the parameters m , \dot{x} , and θ are not clear in Eqs. (4) and (5) which creates uncertainties for the proposed system.

One of the resistive forces of the road against moving vehicles is tire rolling resistance. As there is a complicated relationship between the tire parameters and the coefficient of rolling resistance, presenting an analytical method to predict this force is so hard. This coefficient is dimensionless, and it changes nonlinearly according to vehicle speed, tire wind, tire temperature, road condition, etc. [18]; therefore, the exact amount of C_r is uncertain. As can be seen in most previous works in the context of longitudinal control, the researchers modeled aerodynamic drag force in Eq. (3) regardless of wind speed. In this paper, the longitudinal wind speed is applied as uncertainty to the model. After that, a controller will be designed to compensate for it.

Due to this fact, Eq. (4) shows a real model of the system in which the exact amount of its parameters and sentences are not clear. So, the system model is assumed as Eq. (6):

$$\ddot{x} = \hat{m}^{-1}u + \hat{f}(x) \tag{6}$$

where \hat{m} is not the exact amount of the vehicle mass which is known and $\hat{f}(x)$ includes the clear wording and parameters related to loading road which is approximate. Hence, the function $f(x)$ is defined as:

$$f(x) = \hat{f}(x) + \Delta f \tag{7}$$

However, the exact amount of model uncertainties, Δf is unknown, but it can be limited by its upper limit:

$$\Delta f \leq |\gamma| \tag{8}$$

Parameter m which defines the mass of the vehicle is unknown but of known bounds:

$$m_{\min} \leq m \leq m_{\max} \tag{9}$$

Since the control input enters multiplicatively in the dynamics, it is natural to choose our estimate \hat{m} of mass m as the geometric mean of the above bounds:

$$\hat{m} = (m_{\min} m_{\max})^{\frac{1}{2}} \tag{10}$$

Bounds Eqs. (9) and (10) can then be written in the form [16]:

$$\geq \frac{m}{\hat{m}} \geq \beta^{-1} \tag{11}$$

where β in the gain margin and is defined as:

$$\beta = (m_{\min}/m_{\max})^{\frac{1}{2}} \tag{12}$$

3. CONTROLLER DESIGN

The cruise control system for the vehicle is single-mode (speed control), while an ACC control has two longitudinal control modes: speed mode and gap mode. A vehicle equipped with ACC has additional radars and sensors which help it to measure the distance from the front vehicle on a highway.

An ACC system performs as follows [19]:

- To travel with the maximum speed set by the driver in cases where there are no leading vehicles in the range covered by the sensors. This is also referred to as the speed control mode.
- To maintain vehicle speed less than or equal to the speed of the leading vehicle at a specified distance, when the leading vehicle is in range and its speed is lower than the maximum speed set by the driver. In this case, the speed control switches and provides a safe distance from the leading vehicle by controlling the throttle and brake system. It is also referred to as the gap control mode. After the disappearance of the leading vehicle, the throttle control activates and returns to speed mode by increasing the acceleration (see Figure 3) [20].

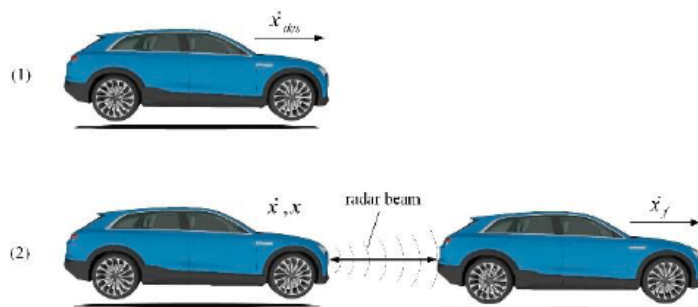


Figure 3: Different modes of a vehicle equipped with adaptive cruise control: 1) Speed control mode, 2) distance control mode.

The schematic block diagram of the speed and distance controller combined with system operators is depicted in Figure 4.

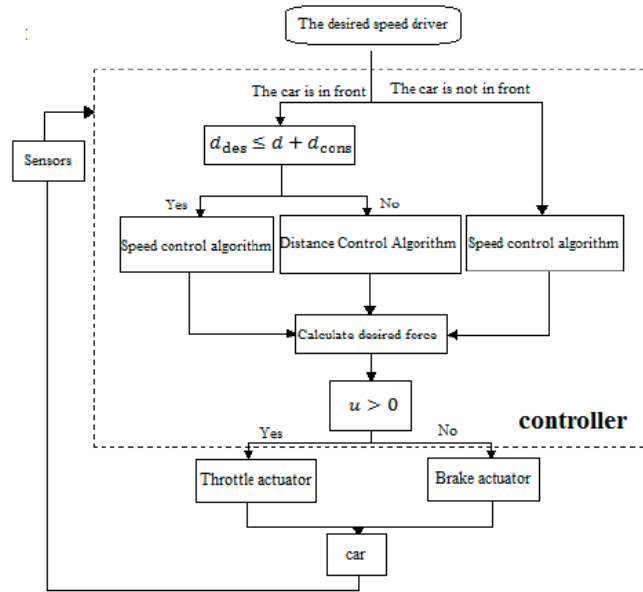


Figure 4: Longitudinal control algorithm of car.

Vehicle speed, the distance to the leading vehicle the driver’s desired speed are given as input to the proposed controller. This control scheme compares the desired and actual distance to the leading vehicle and chooses one of two control modes as follows:

If $d_{des} \leq d + d_{cons}$, speed control is appropriate, otherwise the distance mode will be chosen to avoid a collision with the vehicle ahead. d is the actual distance between the leader and follower and the controller detects it through a radar system. d_{cons} is the initial fixed distance which is defined to increase the reliability of the system and d_{des} is desired distance to the leading vehicle. d_{des} is the most important factor in ACC design which determines not only traffic flow characteristics but also vehicle safety. Three main selection policies have been proposed for it [19]: Constant Space-headway (CSH), Constant Time-Headway (CTH), and Variable Time-Headway (VTH).

In this study, the third policy is used to calculate d_{des} , and it suggests that the linear vehicle spacing should be a linear function of the vehicle’s speed. This spacing policy can be simply described as follows:

$$d_{des} = d_0 + \tau \dot{x} \tag{13}$$

where d_0 is the gap in meters between vehicles when completely stopped, τ is the desired time-gap setting in seconds which are considered 0.8 s and 5 m, respectively. \dot{x} is the speed of the vehicle in m/s that is calculated by sensors, continuously.

As mentioned above, the lack of accuracy in the modeling of nonlinear systems can have severe undesirable effects and should be considered in practical work. In such cases, robust control, adaptive methods, or a combination of both are appropriate solutions for this problem. Here, the authors suggest a non-linear spacing policy for speed and distance control, along with the related sliding mode controller.

3.1. Speed Control Mode

In this mode, the controller has to track the speed that is set by the driver. \dot{x}_{des} defines the driver’s desired speed and is entered as a control input. The velocity and acceleration of the vehicle are calculated at any moment by installed sensors. It is desirable that the vehicle speed reaches the preset speed.

$$\begin{cases} \dot{x} \rightarrow \dot{x}_{des} \\ \ddot{x} \rightarrow \ddot{x}_{des} \end{cases} \tag{14}$$

where \ddot{x}_{des} is changes in desirable speed which is determined by the vehicle’s driver. Tracking error and its derivatives are defined as Eqs. (15)-(17) [21, 22]:

$$e = x - x_{des} \tag{15}$$

$$\dot{e} = \dot{x} - \dot{x}_{des} \tag{16}$$

$$\ddot{e} = \ddot{x} - \ddot{x}_{des} \tag{17}$$

Based on sliding mode control, we use the Lyapunov theorem to define the controller and proof the stability of the system [16]. According to Eq. (18), the sliding surface is defined which contains the error dynamics. The controller’s objective is that the slip surface tends to zero. By considering $\lambda > 0$, the tracking error tends exponentially to zero.

$$s = \dot{e} + \lambda e \tag{18}$$

$$\dot{s} = \ddot{e} + \lambda \dot{e} \tag{19}$$

The Lyapunov function which contains the sliding surface is suggested as Eq. (20):

$$V(s) = \frac{1}{2} s^2 \tag{20}$$

which is a positive value and its derivative is:

$$\dot{V}(s) = s \dot{s} \tag{21}$$

If we choose $\dot{V}(s)$ in the manner that makes the negative function value, the system will be asymptotic stable [16]; therefore, by choosing $\eta > 0$, the Eq. (21) is as follows:

$$\dot{V} \leq -\eta |s| \tag{22}$$

Our aim is to design the controller (u) in such a way that overcomes the total road load and realizes desired

speed. By substituting Eqs. (17) and (4) in Eq. (19), we have:

$$\dot{s} = f(x) + m^{-1}u - \ddot{x}_{des} + \lambda\dot{e} \quad (23)$$

Equation (24) defines a controller in which the slip surface tends to zero:

$$u = \hat{m}(-\hat{f}(x) + \ddot{x}_{des} - \lambda\dot{e} - k \operatorname{sgn}(s)) \quad (24)$$

In the above equation, $k \operatorname{sgn}(s)$ is added to the controller to compensate for uncertainties. In order to determine k , we have to replace Eq. (24) in Eq. (23):

$$\dot{s} = f(x) + m^{-1}\hat{m}(-\hat{f}(x) + \ddot{x}_{des} - \lambda\dot{e} - k \operatorname{sgn}(s)) - \ddot{x}_{des} + \lambda\dot{e} \quad (25)$$

Substituting Eqs. (21) and (22) in Eq. (25), the following equation for k is obtained:

$$k = \beta(\eta + \gamma) + (\beta - 1)|\hat{f}(x) - \ddot{x}_{des} + \lambda\dot{e}| \quad (26)$$

Distance control mode

As indicated in Figure 5, to avoid a collision with the leading vehicle, the controller has to switch to distance control. It is desirable to reach the distance obtained from Eq. (13):

$$\begin{cases} d \rightarrow d_{des} \\ \dot{d} \rightarrow \dot{d}_{des} \end{cases} \quad (27)$$

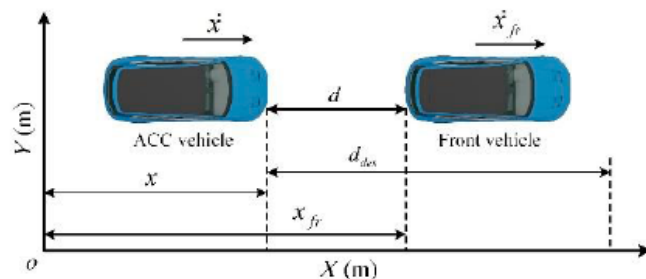


Figure 5: The relationship between the x coordinates of two cars and the distance between them in distance control mode.

As can be seen in Figure 5, the relationship between the x coordinates of two vehicles and the distance between them (obtained from radar) and the rate of changes are as follows:

$$d = x_{fr} - x \quad (28)$$

$$\dot{d} = \dot{x}_{fr} - \dot{x} \quad (29)$$

$$\ddot{d} = \ddot{x}_{fr} - \ddot{x} \quad (30)$$

The distance tracking error and its derivatives are given by:

$$e = d - d_{des} \quad (31)$$

$$\dot{e} = \dot{d} - \dot{d}_{des} \quad (32)$$

$$\ddot{e} = \ddot{d} - \ddot{d}_{des} \quad (33)$$

Like speed control mode, we use the sliding mode and sliding surface which is defined in Eq. (18) to determine distance control law. Its stability is proven by the Lyapunov function that was mentioned in Eq. (20). By substituting Eqs. (30) and (33) and the equation of motion (Eq. (4)) in Eq. (19), \dot{s} can be rewritten as:

$$\dot{s} = \ddot{x}_{fr} - f(x) - m^{-1}u - \ddot{d}_{des} + \lambda\dot{e} \quad (34)$$

Continuously, the control law is defined as Eq. (35) to tend the slip surface to zero.

$$u = \hat{m}(\ddot{x}_{fr} - \hat{f}(x) - \ddot{d}_{des} + \lambda\dot{e} - k \operatorname{sgn}(s)) \quad (35)$$

To determine the k value, the same procedure as speed control is used. In this regard, we substitute Eq. (35) in Eq. (34):

$$k = -\beta(\eta + \gamma) - |(1 - \beta)(\ddot{x}_{fr} - \hat{f}(x) - \ddot{d}_{des} + \lambda\dot{e})| \quad (36)$$

A simplification of k can be obtained from Eqs. (21), (22), and (36):

$$k = -\beta(-\eta + \gamma) - (1 - \beta)(\ddot{x}_{fr} - \hat{f}(x) - \ddot{d}_{des} + \lambda\dot{e}) \quad (37)$$

3.2. System Operators

The controller output for both speed and the distance control mode is applied force of the wheels to the road and must be fulfilled by the car. System operators of a vehicle including engine and brake system. There is a direct relationship between choosing a proper operator and the calculated force by the controller. If $u > 0$ it means that the driving force has to be applied from the vehicle to the road surface. So, the controller has to send a command to the engine to generate the required force. And conversely, if $u < 0$, the opposing force may be reached by sending a related command to the braking system.

In order to apply driving force, we need to determine the required torque by Eq. (38) [18]:

$$T_e = \frac{r_{wh}u}{N_{tf}\eta_{tf}} + \frac{[(I_e + I_t)N_{tf}^2 + I_f N_f^2 + I_{wh}]\ddot{x}}{r_{wh}N_{tf}\eta_{tf}} \quad (38)$$

where r_{wh} , I_e , I_t , I_f , and I_{wh} are the radius of the wheel, the rotary inertia of the engine, the gearbox rotary inertia, the rotary inertia of the final gear (differential), the rotary inertia of wheels and the axles, respectively. Parameters N_{tf} , N_f , and η_{tf} define the ratio of combined gearbox and differential, respectively. Equation 38 introduces a perfect formula to calculate the required torque of the vehicle and consists of two parts:

The first algebraic expression on the right side of Eq. (38), produces the driving force to neutralize load road

and accelerate the vehicle. In this expression, the mechanical losses are taken into account by adding the efficiency of powertrain components. Since the engine torque decreases to speed up the inertial components of the moving power chain, thus the second expression is added to Eq. (38).

The torque is a function of engine speed. In conventional internal combustion engines, by changing the angle of the throttle, the amount of air entering the engine and consequently the amount of fuel injection is changed. This process changes the speed and torque produced by the engine. According to different torques that can be obtained based on the engine speed and throttle angle, the engine specification graph is depicted and stored in the electronic control unit of the engine. Therefore, after calculating the required torque from Eq. (38), the corresponding throttle angle can be obtained from the specification graph. After that, a suitable command of the throttle should be sent to its operator.

If the braking force is needed, firstly the braking force between the front and rear axles has to be calculated. Distribution of braking forces between the front and rear axles -when all of the tires are unlocked- is a function of braking system design. The intended braking system is a conventional hydraulic braking system, and the braking force distribution is according to Eq. (39) [17]:

$$F_r = -F_{fr} - \frac{mgc}{2h} + \sqrt{\left(\frac{mgb}{2h}\right)^2 + \frac{F_{fr}(b+c)mg}{h}} \tag{39}$$

Where F_r and F_{fr} are the braking forces of the front and rear axles, respectively. h is the height of the vehicle from the ground. b and c are the distance between the front and rear axle to the center of mass. The braking force is evenly divided between the right and left wheels. The hydraulic pressure of brake cylinders is calculated by Eq. (40) for each wheel. Accordingly, the necessary command is sent to the braking operator who leads to increasing pressure.

$$p_i = \frac{F_i r_{wh}}{\mu} \tag{40}$$

where p_i is the pressure of the braking cylinder for each wheel and μ is the braking constant which depends on each wheel's braking structure.

4. RESULTS AND DISCUSSION

In order to assess the performance of the proposed controller, a simulation of different modes of the ACC system, *i.e.*, speed control and distance control, is done. The technical specification of the applied vehicle is shown in Table 1.

4.1. Speed Control Simulation

This simulation includes two maneuvers. In the first one, it is assumed that the vehicle is moving on the highway without the existence of a leading vehicle. The actual speed is $25 \frac{m}{s}$ which is desirable for the driver to reach $35 \frac{m}{s}$. Uncertainties involved in vehicle weight and the road surface are assumed to be wet and slippery which causes changes in the coefficient of rolling resistance force. As is clear from Figure 6, the longitudinal wind disturbance is applied to the vehicle accidentally as long as the vehicle starts moving.

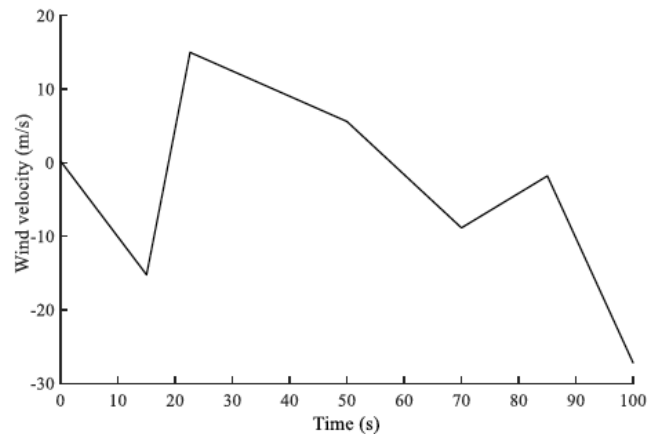


Figure 6: Longitudinal wind disturbance applied to the car.

The tracking speed curve, speed tracking error, the changes in controller output, and the engine torque

Table 1: Technical Specifications of the Vehicle used in the Simulation

Quantity	Size	Unit
Net weight of the unmanned vehicle	1250	kg
Maximum weight of the vehicle with passengers and full fuel tank	1600	kg
initial coefficient of rolling resistance force	0.015	Dimensionless
aerodynamic drag coefficient	0.42	Dimensionless
vehicle frontal area	2	m ²
radius of wheel	0.32	m

changes over time can be clearly seen in Figures 7, 8, 9, and 10, respectively.

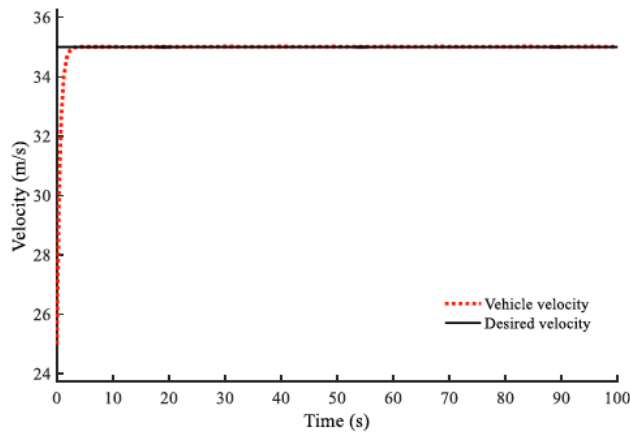


Figure 7: Vehicle speed curve by considering various uncertainties.

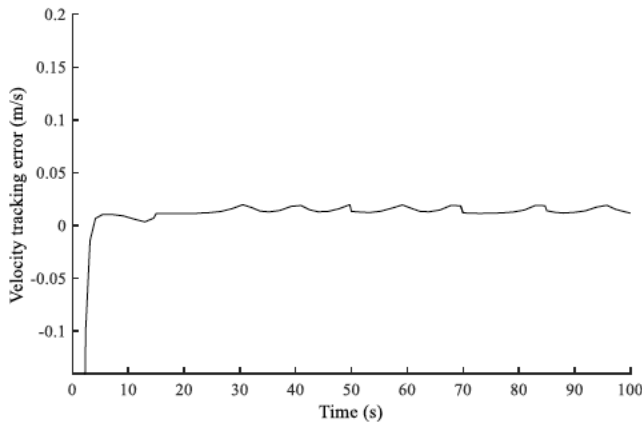


Figure 8: Error of velocity tracking.

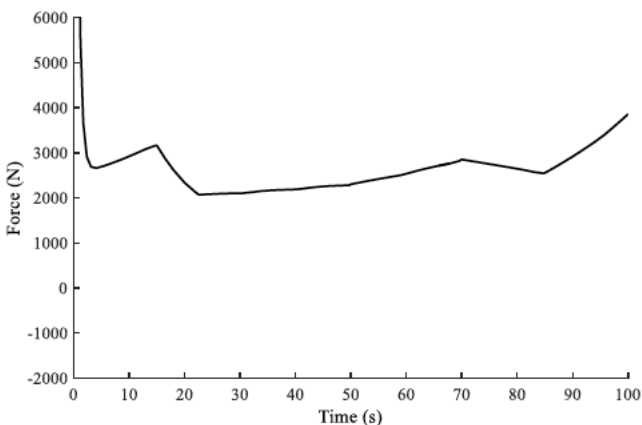


Figure 9: Controller output force.

4.1.1. Discussion on Speed Control Results

As can be seen in Figures 7 and 8, the sliding mode controller is capable of tracking desired speed with good accuracy, especially in the presence of uncertainties. Adding $k \operatorname{sgn}(s)$ expression to the control law, make it possible for the controller to compensate for uncertainties and disturbances.

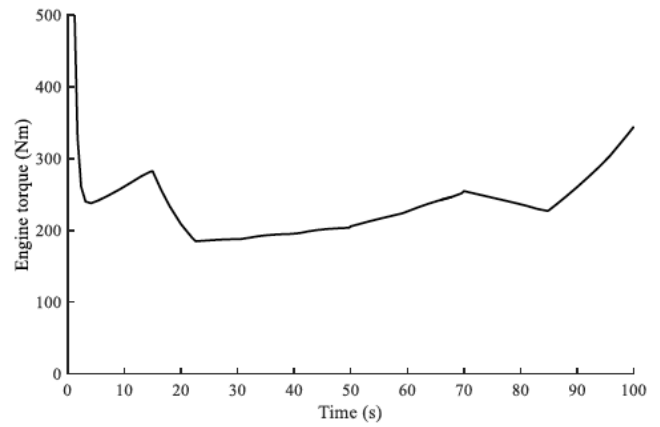


Figure 10: Car engine generative torque.

As Figure 8 shows, the maximum tracking error is less than $0.05 \frac{m}{s}$, which is negligible. In addition, there is no instability in the simulation results. According to Figure 6 and Figure 9 and the model of the vehicle, when the wind is blowing in the same direction of the vehicle, the relative speed of the vehicle will be reduced. Moreover, less force is needed to overcome aerodynamic drag force. Adversely, when the blowing is against vehicle movement, more power is needed to conquer that. The accomplishment of this structure can be seen in Figure 9, where the proposed sliding mode controller is able to compensate for the effects of longitudinal wind disturbance by increasing the force of control in times of 15, 70, and 100 seconds and decreasing the force of the control in times of 22 and 85 seconds. The required torque to produce this force is depicted in Figure 10. By comparing Figure 9 and 10 it can be deduced that there is a linear relationship between produced engine torque and the required driving force and the majority of engine force is needed to move the vehicle. Also, the engine does not allocate much torque to accelerate the moving parts of the power train. Due to the constant speed, the acceleration is close to zero which causes the second algebraic expression of Eq. (31) being negligible. Consequently, the engine torque will be reduced. Thus, the cruise control system results in reducing the required power and consequently reducing fuel consumption.

4.2. Distance Control Simulation

In the second part of the simulation, the vehicle is assumed to move in urban traffic. At first, the vehicle speed is $25 \frac{m}{s}$ and the leading vehicle has the same speed. Over time, the leading vehicle reduces its speed with constant acceleration until the 55 seconds after motion passes. After that, the leading car again accelerates and moves more quickly than the initial speed. So, the controller should perform vehicle deceleration and track the leading vehicle by considering the safe distance. Similar to the previous model, the aforementioned uncertainties are included in the model, and the longitudinal wind disturbance is applied to the vehicle. In Figure 11, the speed of the proposed vehicle and the leading vehicle are shown.

Figure 12, 13, and 14 represent the changes in controller output, engine torque, and the hydraulic pressure of braking cylinders for each wheel with respect to time, respectively.

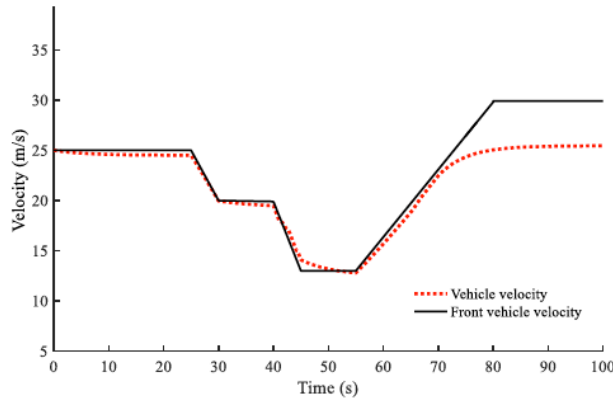


Figure 11: Curves of front vehicle velocity and tracked velocity by considering various uncertainties.

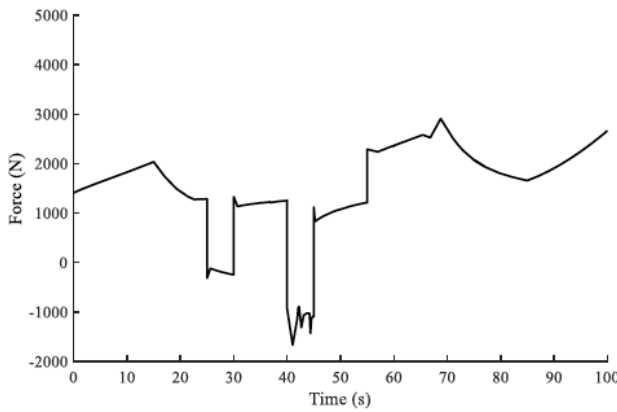


Figure 12: Controller output force.

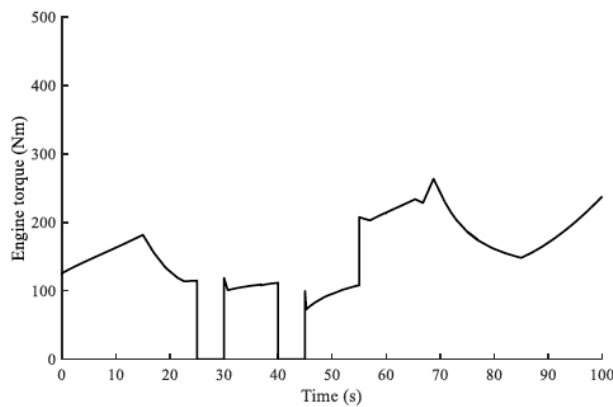


Figure 13: Car engine generative torque.

4.2.1. Discussion of Distance Control Results

In this case, the controller is well-managed and decreases the vehicle speed as the speed of the leading vehicle reduces. Furthermore, it tracks the speed of the car ahead by observing a safe distance. As can be seen in Figure 11, after 72 seconds of simulation, the front car accelerated and had gotten far from it, at this time, the controller changes to speed mode and start tracking the initial speed ($25 \frac{m}{s}$).

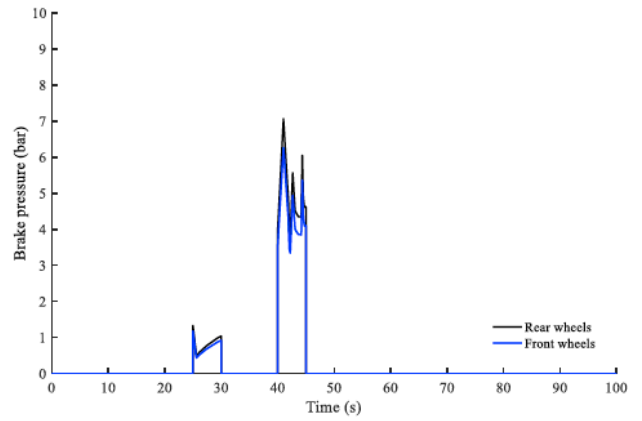


Figure 14: Hydraulic pressure brake cylinder of front and rear wheels.

In addition, the proposed controller is resistant to changes in the mass of the vehicle, rolling resistance force, and longitudinal wind disturbance and has been able to retain its stability over the simulation time.

Figure 12 shows the controller output over time. At times 25 and 40 seconds, when the leading vehicle moves with negative acceleration, the controller output force has to decrease to avoid the collision and apply the braking force to the vehicle. The hydraulic pressure brake cylinder of the front and rear wheels is shown in Figure 14 to achieve the negative force of the controller. There is a difference between the pressure of the front and rear wheels which is created because of the intended model for the braking system (Eq. (32)) and the geometry of the vehicle. This difference causes the braking force to be divided between the front and rear wheels with values of 47 and 53 percent, respectively. The difference between the brake cylinder pressure in the two different braking periods is produced by the different accelerations of the leading vehicle. In a period of 40 to 45 seconds, the acceleration of the car ahead decreases more quickly than the leading vehicle. So more braking force and consequently greater pressure are needed to decelerate.

In addition, the engine torque changes with respect to time are depicted in Figure 13. As expected, in the range of 0 to 25 seconds when the vehicle speed is constant, the amount of engine torque is changed to overcome longitudinal wind disturbance. Between the 25 to 55 seconds of the movement, the engine torque is decreased due to the reduced acceleration of the leading vehicle. Also, in the range of 55 to 72 seconds, more torque is needed to overcome the longitudinal wind disturbances that blow in opposite directions and tracking increasing acceleration. After that, the engine torque encountered a negative and positive trend, respectively, to overcome longitudinal wind disturbance inline or the opposite of the route direction.

5. CONCLUSION

In this paper, a new controller based on the sliding mode method is designed to control an adaptive cruise control system. The simulation is done for both speed and distance control modes. As is evident from the sample simulation results, the following results could be carried out in the presence of uncertainties and disturbances:

- According to this research and the conducted studies, applying cruise control systems in vehicles leads to reducing fuel consumption and increasing vehicle safety.
- The available cruise control systems are designed in order to increase productivity. In this regard, the accurate measurement of various parameters is needed which is costly and challenging.
- Using the sliding mode controller and proposed algorithm for the ACC system, increase the system performance and covers the disadvantages of conventional controllers. This controller is capable of speed tracking in the presence of uncertainty in vehicle mass, longitudinal wind disturbance, and inclement weather. More importantly, the proposed controller avoids collision with the leading vehicle.

NOMENCLATURE

c	vehicle frontal area (m^2)
C_d	aerodynamic drag coefficient
C_r	the coefficient of rolling resistance force
D_A	aerodynamic drag force (N)
d	distance between vehicles (m)
e	control error
F	braking force (N)
$f(x)$	Total load road (N)
g	gravitational acceleration (ms^{-2})
I	rotary inertia (Nms^2)
k	controller discontinuity
m	vehicle mass (kg)
s	sliding surface
T	engine torque (Nm)
u	controller output force (N)
$V(s)$	Lyapunov function
W	weight of vehicle ($kgms^{-2}$)
x	vehicle position (m)

\dot{x}	vehicle speed (ms^{-1})
\ddot{x}	vehicle acceleration (ms^{-2})
θ	road angle to the horizontal direction (degree)
ρ	air density (kgm^{-3})
τ	leading time of vehicle (s)
μ	braking system constant (m^3)
γ	system uncertainties
β	gain margin
η	constant coefficient of the controller
λ	constant coefficient of the controller
cons	constant
des	desired value
e	engine
fr	front
i	the counter of each wheel
r	rear
t	gearbox
wh	wheel
f	the final gear

LIST OF ABBREVIATIONS

ACC	Adaptive cruise control
CC	Cruise control
CG	Center of gravity
CSH	Constant Space-headway
CTH	Constant Time-Headway
VTH	Variable Time-Headway

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