DEVELOPMENT OF ACTIVE IDLE STOP SYSTEM FOR AUTOMOTIVE VEHICLE DURING UPHILL DRIVING

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ABSTRACT

This manuscript discusses the Active Idle Stop (AIS) system for a passenger vehicle system which is used to improve the dynamic performance of the vehicle when traveling uphill. The AIS function is developed mainly to improve the drawback in the existing vehicle system when driving uphill. Vehicles face unwanted deceleration and rollback when they are started on an incline. In this study, a control strategy using a Proportional-Integral-Derivative controller is used to improve the deceleration and rollback conditions during an idle stop on an uphill road gradient. A nonlinear vehicle longitudinal model has been used as the testing platform for the AIS function. Meanwhile, an optimization tool known as the Genetic Algorithm is used to improve the controller parameters according to the desired response of the vehicle. Based on the simulation results, it is possible to improve the vehicle's performance using the AIS system to improve the rollback effect where the deceleration effect on the vehicle is reduced significantly.

KEYWORDS: *Active idle stop; Proportional-Integral-Derivative controller; road gradient; nonlinear vehicle longitudinal model; genetic algorithm; deceleration*

1.0 INTRODUCTION

Most hybrid and conventional vehicles commonly include an auto idle stop system. This system is mainly used to automatically stop and restart the engine in order to maximize the fuel efficiency depending on environmental and vehicle conditions (Bishop, Nedungadi, Ostrowski & Surampudi, 2007). The auto idle stop system is activated once the vehicle accelerates from a static condition at any intersection point after a stopping point. The car's engine automatically switches off, and restarts when the driver wants to drive from the initial location (Cho, Kim & Choi, 2012). The auto idling stop system generally uses an electric motor to restart the engine from the halt condition. It has the capability to minimize CO_2 emissions by up to 8% compared to conventional vehicles without an auto idle stop function (Wishart & Shirk, 2012).

Stopping and starting an engine is a simple principle, but doing so whilst retaining the vehicle attributes, drivability, and ensuring that the behavior of the idle stop system is natural to the driver makes it a complex task (Arunkumar, Muthumani & Balasubramani, 2015).

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Most of the current research is focusing on the auto idle stop to improve the fuel efficiency and emission rate. However, most researchers have neglected the performance of the vehicle's system while stopping and driving uphill. Vehicle drivers are experiencing difficulties when stopping and restarting on hills and vehicles driven by inexperienced drivers may even slip (Arunkumar et al., 2015). A similar problem exists in vehicles equipped with the auto idle stop, whereby they need to adapt to the changes in external driving conditions. Even though the auto idle stop has the capability to restart the vehicle from the halt condition, the issue with the rollback condition has still not been solved in most research analysis. Therefore, a control strategy is greatly required in order to overcome the shortcomings in current passenger vehicles. The developed system should have the capacity to reduce the deceleration and at the same time minimize the rollback while driving on a road with an incline.

One of the automotive research works is to improve uphill driving condition which falls under safety aspects. In studies about the starting control of electric vehicles, an assist system known as Hill Start Assistance has been developed in commercial vehicle applications (Delvecchio, Savaresi & Spelta, 2009). Meanwhile, Okuda, Komatsu and Nakahara (2006) developed a simulation model of combined vehicle and motor in the acceleration process of electric vehicle (EV). In this research, the authors have shown that good mechanical properties of the electric motor could obtained by using double closed-loop control strategies. Besides, Haifeng, Zhaodu and Guoye (2007) developed slope recognition by simplifying the vehicle longitudinal dynamic equation, and then a *Luenberger* observer was designed to recognize the road slope online in order to meet the requirements of safety and driving comfort in the hill start process.

Nevertheless, other researchers such as Wang, Li, Gao and Dai (2014) conducted a study on the hill start auto control technology of medium and heavy duty vehicles. The brake cylinder spring and pneumatic auto parking brake were analyzed in this study. Furthermore, the relationship between the driving force and braking force and the grade resistance in the hill start process were analyzed and a control strategy of a pneumatic auto parking brake system was proposed. Similarly, Jiang, Xu, Jin and Chen (2012) conducted research on a heavy truck with AMT for a hill start control strategy. The model was built with MATLAB Simulink and a simulation study was carried out to study the performance of the vehicle. A hill start control strategy was achieved by a brake system which is controlled by a push button to change the vehicle into a hill start mode. In this mode, the vehicle's electronic control unit changes the gear ratio into the appropriate gear ratio by a pneumatic actuator, and at the same time the brake system will be lock by the ABS valve to prevent the wheels from rolling back.

The greatest disadvantage of the hill holder is that the vehicle can be locked for three to four seconds only when on a sloping road, and besides this the driver still needs to be aware and activate the hill holder mechanism. Furthermore, it is expensive to apply such a design into an actual vehicle. Moreover, the hill holder mechanism has only being studied and research conducted for heavy vehicles, hybrid and electric vehicles. However, the effects of road slope on the auto idle stop equipped conventional vehicle have been neglected. These shortcomings have been identified on the hill holder mechanism and the ensuing problems faced by drivers when driving on hills. In order to overcome these shortcomings, the AIS is developed in this study. It is designed in such a way to control the throttling input of the powertrain system according to the gradient level, θ. The vehicle driving conditions on a level road is used as the reference point to control the throttle input.

2.0 MODEL DEVELOPMENT

The development of the passenger type of vehicle model mainly concerns three types of dynamics system, known as lateral, longitudinal and vertical dynamics systems. However, this study considers the performance of the vehicle model in the longitudinal direction only since the vehicle is assumed to travel in a straight direction without any steering input. Besides, the vehicle is also assumed to travel on a flat road surface without any irregularities on its surface. The vehicle longitudinal model is developed based on a single sprung mass which is connected with four unsprung masses. The sprung mass or the vehicle body is allowed to pitch without any rolling condition; meanwhile, the unsprung mass is allowed to rotate and displace in the longitudinal direction only. The vehicle model is developed using a powertrain and brake dynamics system to represents vehicle acceleration and deceleration while traveling on the road. Additionally, the Pacejka Magic Tire model has been used to represent the behavior of the tire model in the vehicle longitudinal model (Short, Pont, & Huang, Q, 2004). A complete diagram of the vehicle longitudinal model is shown in Figure 1.

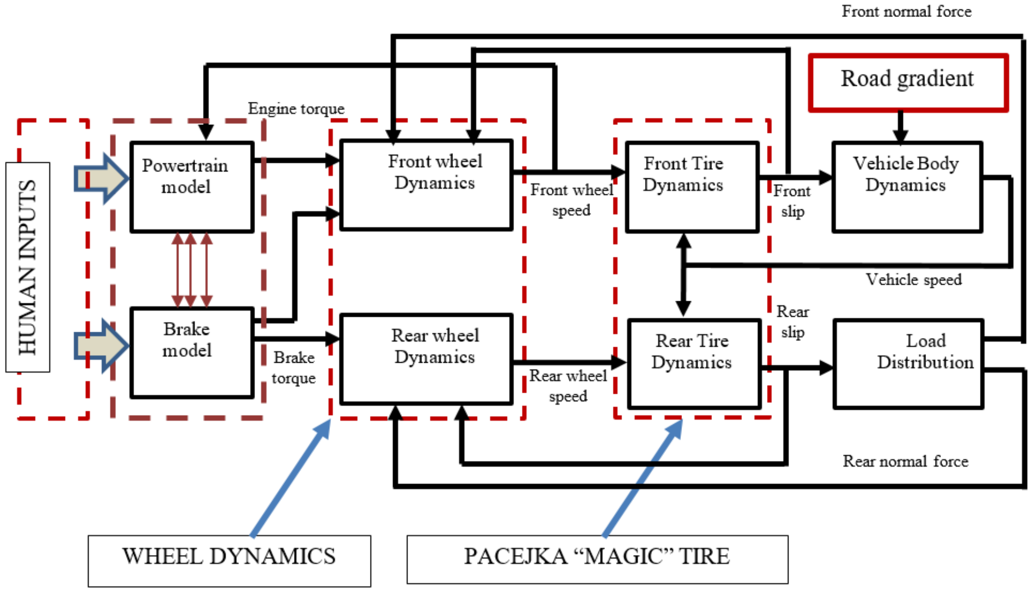


Figure 1. Vehicle Longitudinal Model Block Diagram

The vehicle longitudinal model can be categorized as a combination of load distribution model, vehicle dynamics model, wheel dynamics model, tire traction model and, finally, as the powertrain and brake model. The inputs used in the vehicle longitudinal model are the brake and throttle setting inputs. These inputs are given by the driver him/herself to control the motion of the vehicle model in the longitudinal direction. Both engine and brake dynamics systems produce engine torque and brake torque to the wheel dynamics to control the vehicle movement as required by the driver. The output of the wheel dynamics block produces the longitudinal slip of the vehicle while accelerating and also when braking. The slip ratio is used to study the condition of the vehicle during acceleration and braking. Additional road gradient input is given in the vehicle dynamics model to measure the performance of the vehicle velocity and acceleration when on an incline.

2.1 Modeling Assumptions

In the development of the vehicle longitudinal model, the vehicle speed is assumed to be equal to the speed of the wheels scaled by the current gear and final drive axle transmission system. Generally, the assumption is made for both incline and level road surface conditions. Besides, the external disturbance, which can be described as rolling resistance and aerodynamic force, is also included in the development of the vehicle longitudinal model during simulation procedures (Short et al., 2004). The throttle and brake input settings are indicated in term of percentages where a zero percentage shows no input from the driver whilst 100% shows maximum input by the driver. The detailed derivation of the vehicle longitudinal model has been described in Aparow, Hudha, Ahmad and Jamaluddin (2014). In this study, the verification of the vehicle longitudinal model is discussed where the model's responses are compared with CarSim validation software. Two types of condition, accelerating and braking mode, are used to evaluate the behavior of the developed vehicle model.

2.2 Verification of Model

The 5 DOF vehicle model has been developed using MATLAB Simulink software using the sedan vehicle as shown in Figure 2. The model is tested using two types of vehicle conditions: sudden acceleration and braking.

Figure 2. Vehicle Model from CarSim software

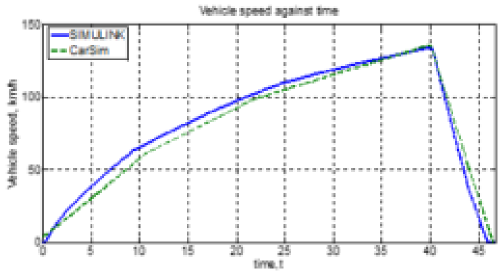
Two types of throttle inputs, throttling and braking inputs, are used to produce full speed for the vehicle. A step input function is used to represent the input from the driver and the input is indicated in percentages (%). Two levels of throttling inputs, 50% and 100%, are applied to the vehicle until the vehicle approaches 40 seconds from the initial testing period. Then, a sudden braking (100%) input is given to the vehicle until the vehicle halts. The behavior of the vehicle model is verified with the validated CarSim software in terms of vehicle speed, front and rear wheel speed, distance traveled by the vehicle after sudden braking, and lastly the wheel slips at front and rear tires. The vehicle model parameters used in the simulation are obtained from the simulator and listed in Table 1.

Table 1. Vehicle model parameters

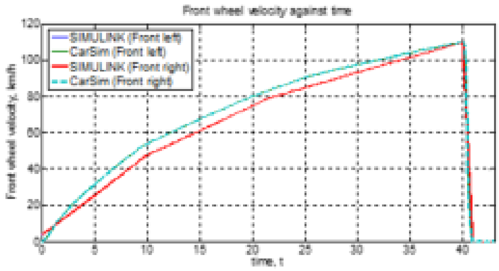
Parameters	Symbols	Value
Total wheel base	L	3 m
Front wheel base	B	1.5 m
Rear wheel base	C	1.5 m
Height of vehicle	H	0.6 m
Vehicle mass	m	1626
Tire radius	R_j	0.3 m
Aerodynamic drag coefficient	C_d	0.29
Rolling resistance	C_r	0.01
Viscous friction coefficient	C_{fj}	0.1 Nm/$rads^{-1}$
Single equivalent lag for throttle	τ_{es}	0.2 s
Single equivalent lag for brake	τ_{bs}	0.3 s
Front brake constant	K_{bf}	13.33 Nm/Bar
Rear brake constant	K_{br}	6.666 Nm/Bar
Inertia of wheel	I_ω	4.5 kg/m^2

The verification results of the vehicle longitudinal model during 50% of throttling input and sudden braking are shown in Figures 3(a) to 3(f). During this condition, the vehicle reached maximum velocity of 80 km/h at the 40th second, as shown in Figure 3(a). Similarly, the model's wheels also approached maximum velocity up to 80 km/h, as shown in 3(b). Sudden braking is applied to the vehicle until the vehicle decelerates and finally stops. All four wheels of the vehicle started to lock once 100% braking force was applied and a slip moment occurred before the vehicle was fully stopped, as shown in Figures 3(a) and 3 (b). The wheel slippage increased the vehicle's stopping distance, as shown in Figures 3(d) and 3(e). It shows that a conventional vehicle without any active safety system generates unwanted slip motion due to wheel locking and this causes the vehicle to slip.

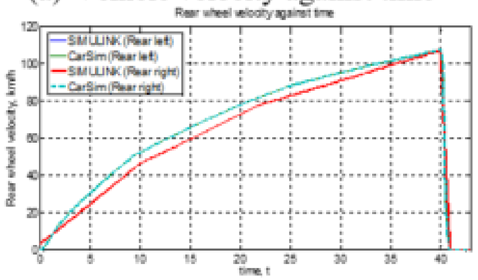
Based on the comparison results, it can be observed that the developed vehicle longitudinal model is able to follow the response of the CarSim vehicle model with minor deviations. Similarly, the distance traveled for both the simulation model and the CarSim model shown in Figure 3(f) shows a satisfactory response with negligible deviation. The differences in both simulation and CarSim have been observed in the verification results for the effect of pitch moment during braking condition. This moment is not included in the vehicle longitudinal model but the model's response is still in the acceptable range, which is less than 5%.



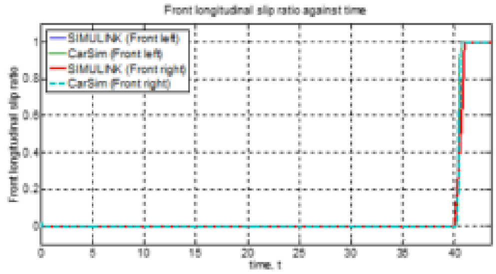
(a) Vehicle velocity against time



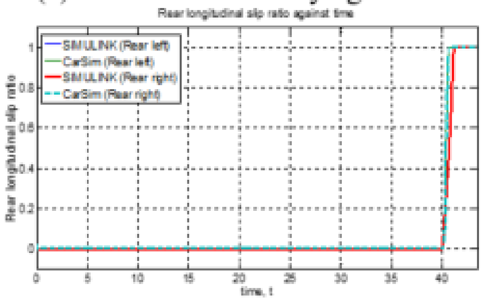
(b) Front wheel velocity against time



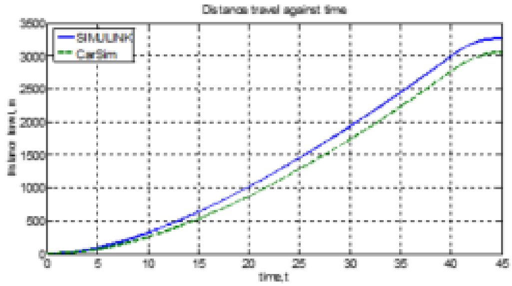
(c) Rear wheel velocity against time



(d) Front longitudinal slip against time



(e) Rear longitudinal slip against time

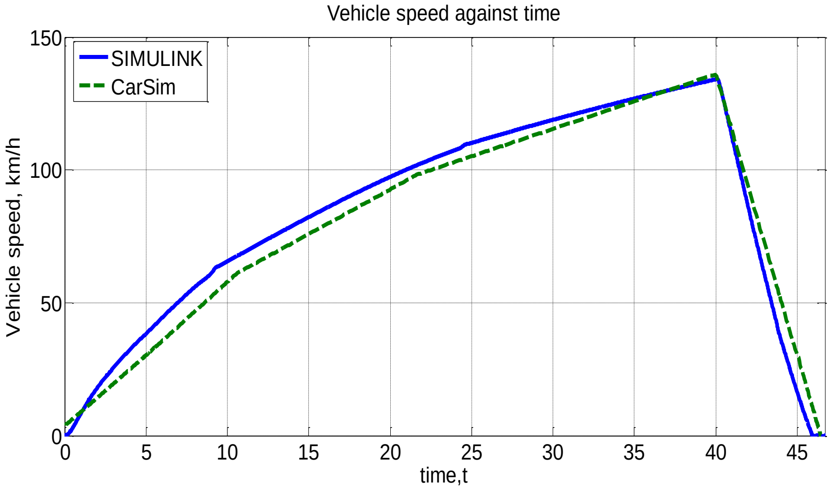


(f) Distance traveled against time

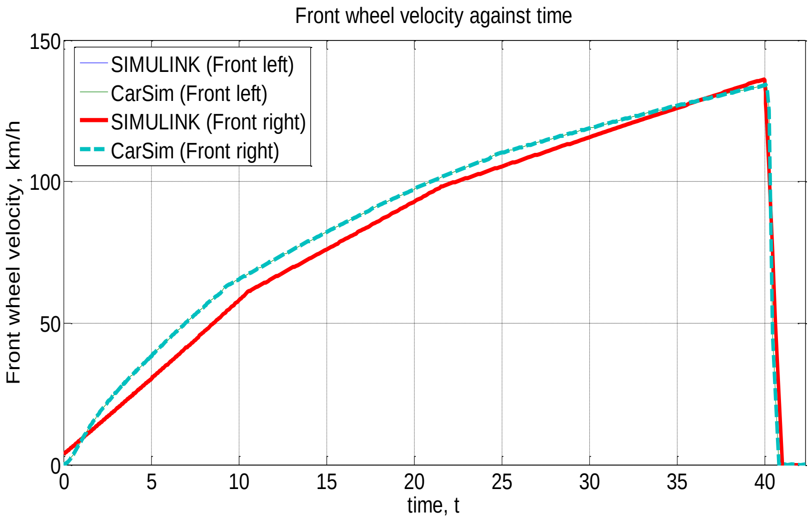
Figure 3. Vehicle response at throttle input of 50% and sudden braking

Similarly, the vehicle longitudinal model is compared with the validated model with full throttle speed (100%) and sudden braking input is applied at 40 seconds. The vehicle model is able to a reach maximum speed of up to 137 km/h before the vehicle deaccelerates to 0 km/h, as shown in Figures 4(b) to 4(c). As discussed in the earlier condition, the vehicle speed approaches zero speed at t = 46.5 s.

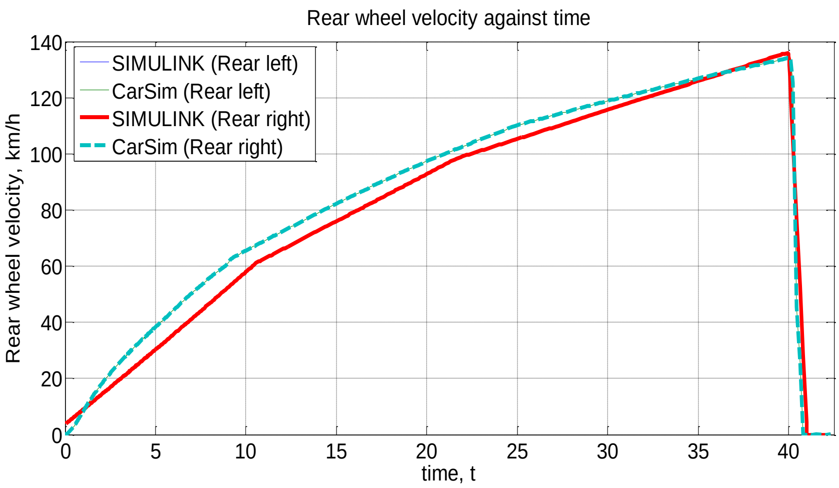
However, the wheels start to lock up and drag the vehicle until t = 46.5 s. The distance traveled by the vehicle has increased compared to the previous condition. From Figure 4(f), it can be observed that the vehicle model is able to follow the behavior of the validated model with minor differences. Similar responses can be observed for both front and rear wheel slips where the slips produce maximum ratio, 1, once the wheel starts to lock up, as shown in Figures 4(d) and 4(e). This explains that the vehicle starts to drag before reaching the halt condition due to the wheel locking condition.



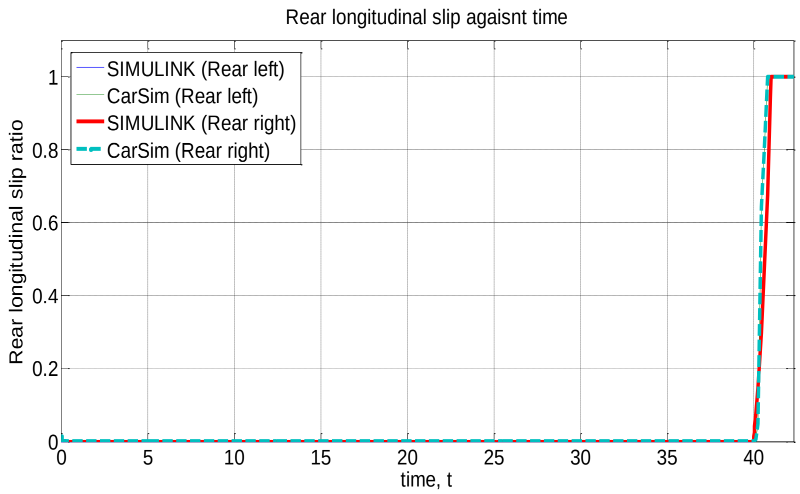
(a) Vehicle velocity against time



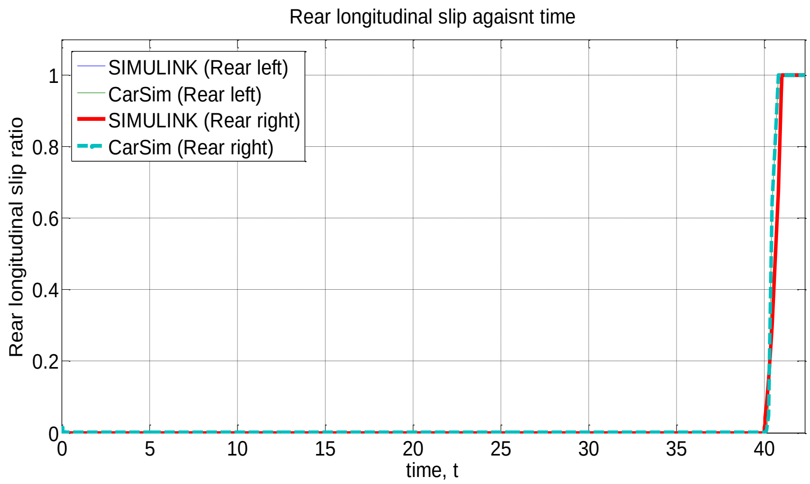
(b) Front wheel velocity against time



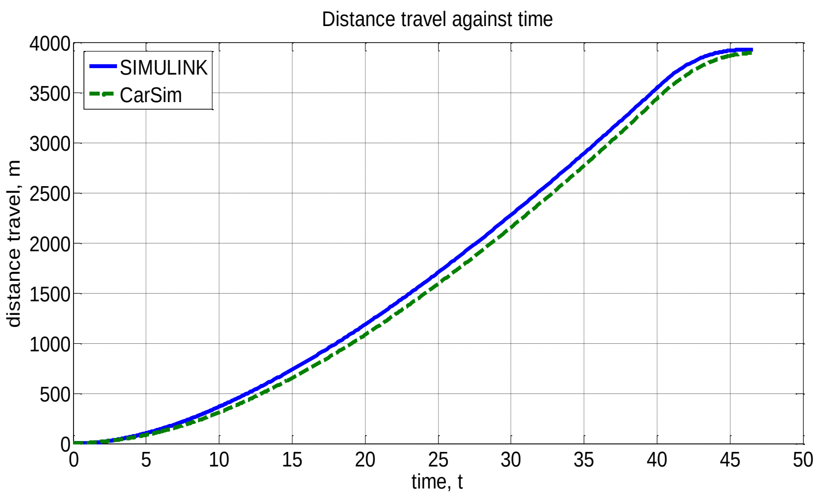
(c) Rear wheel velocity against time



(d) Front longitudinal slip against time



(e) Rear longitudinal slip against time



(f) Distance traveled against time

Figure 4. Vehicle response at throttle input of 100% and sudden braking

Based on the results, it can be concluded that the Simulink vehicle model developed has similar behavior to the CarSim vehicle. The verification has been performed using two types of input conditions, as mentioned earlier in this study. Even though there is minor deviation in the verification results, the vehicle model is still able to represent the actual behavior of the vehicle. The minor deviation mainly occurred because the vehicle's ride and handling performance is neglected in this study since the vehicle is traveling in a longitudinal direction. The effects of suspension systems are assumed to be in ideal condition since the vehicle is traveling on a flat road without any irregularities in its surface. Thus, this vehicle longitudinal model can be used for further development of an AIS system in the longitudinal direction.

3.0 ELECTRONIC THROTTLE BODY

The electronic throttle body is the major component in the vehicle's powertrain system controlling air flow mixture into the combustion chamber. This air flow mixture is very important to enhance the dynamic performance of a vehicle when it is being driven. Most passenger vehicles use the ETB mechanism to control air fuel mixture to the combustion chamber.

3.1 Description of the Electronic Throttle Body

Figure 5 shows the current design of electronic throttle body (ETB) which is used in Honda vehicles (Hashimoto, Ishiguro, Yasui & Akazaki, 2013). This ETB mechanism consists of an electric DC motor which is used to actuate the throttle plate valve, as shown in the figure below.

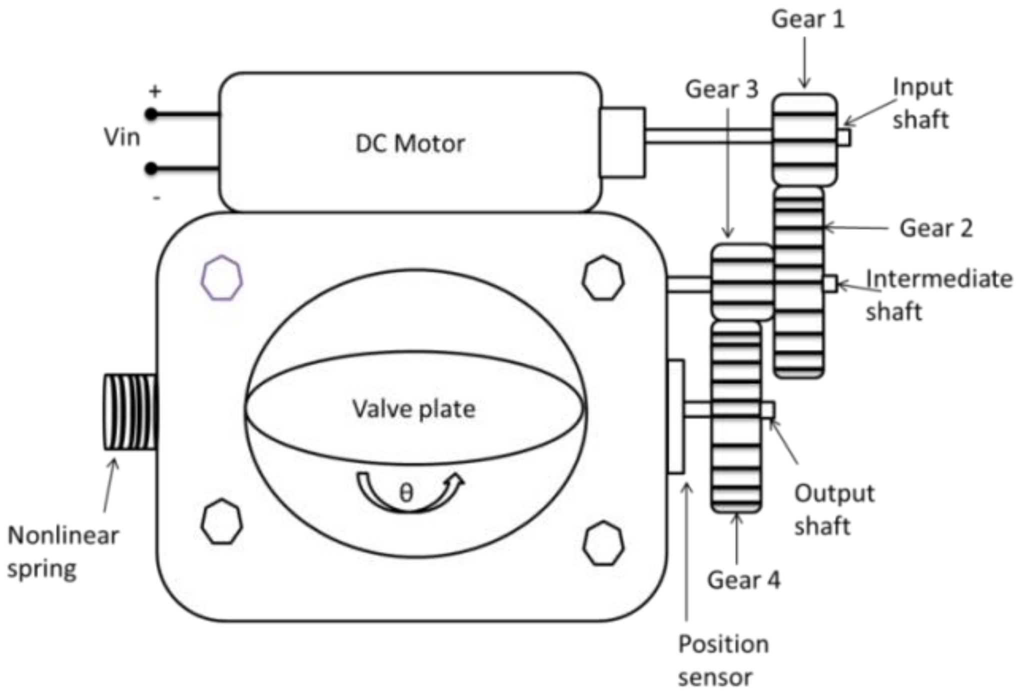


Figure 5. Electronic throttle body

The ETB mechanism consists of mechanical systems such as gears, intermediate shaft, nonlinear spring, and throttle valve plate, input and output shafts. The rotation of the throttle valve plate is sensed by a position sensor which is installed internally in the ETB mechanism. The set of reduction gears and a nonlinear spring which is attached together with ETB mechanism are used to transfer the rotation motion of the DC motor to the throttle valve plate. An angle limiter is attached to the ETB mechanism to halt the rotation motion of the throttle valve once it approaches a 90 degree angle. Meanwhile, the lower opening throttle valve or fully closed position of the throttle valve is initiated from a 7.5 degree angles. During this position, a partial amount of air flow is required in order to maintain the vehicle in the idle speed condition.

3.2 Mathematical Modeling

The governing equations for the electronic throttle body are discussed in the following three sections, which are nonlinear frictional torque, nonlinear spring system and electric DC motor. The throttle body, which is coupled with the gear reduction system, is as shown in Figure 5.

a) The nonlinear frictional torque

The nonlinear factor of the throttle body includes friction, airflow disturbance and motor rotation angular velocity. These occur during low speed operation of the vehicle where the opening of the throttle valve is on a small angle. These nonlinear factors have a great impact on the system performance of a vehicle in the driving condition. The friction torque model is developed based on a static coulomb model, as shown in Figure 6 and described in Equation (1), where T_f represents all friction torque, and is simplified as in Equation (2), F_s is the positive constant representing friction in the system and ω_m is the motor rotation angular velocity.

$$T_f = \begin{cases} F_s\ , \ \omega_m & > 0 \\ \ 0 \quad \omega_m & = 0 \\ -F_s\, \omega_m & < 0 \end{cases} \tag{1}$$

or it can be simplified as,

$$T_f = F_s\, sgn(\omega_m) \tag{2}$$

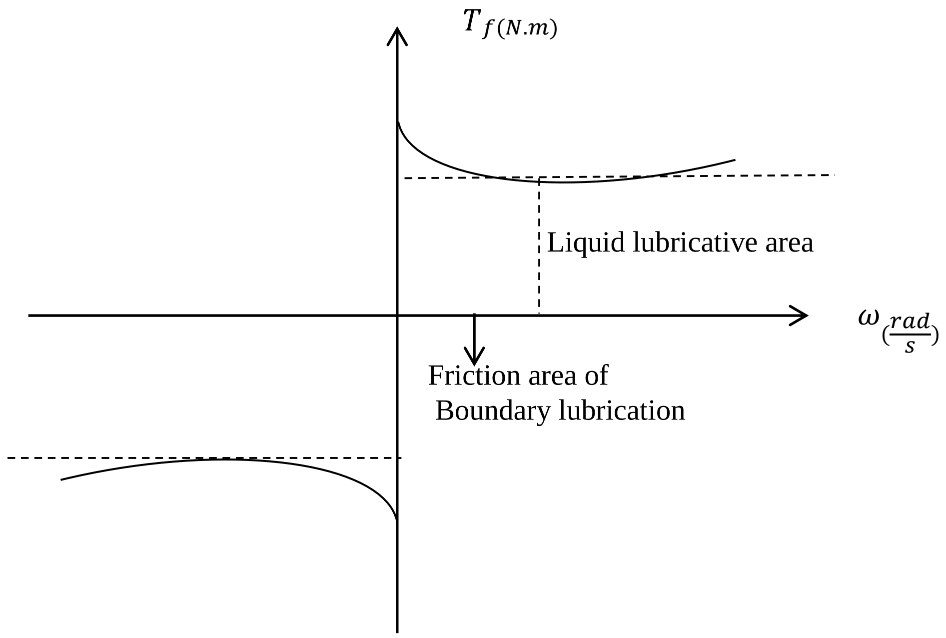


Figure 6. Frictional effect

b) The nonlinear spring system

A return spring in this system acts as the failure protection measure, which ensures a safe return of the throttle plate to its closed position when there is no driving torque generated by the DC motor. The stiff spring is a standard feature attached to the electronic throttle valve. It acts as a fail-safe mechanism. In situations where there is no power supply, this spring will force the valve plate to return to its original position of about 7.5 degrees, which is slightly above the closed position. This is to allow a slight amount of air to be supplied to the engine when there is no input control available in order to prevent a sudden lock of engine revolution while the vehicle is in motion.

In addition, the movement of the valve plate is limited to the maximum of 90 degrees and the minimum of 7.5 degrees of angle. These limited stop are realized by a highly stiff spring, ideally with infinite gain. Figure 7 describes the characteristics of the nonlinear spring model (AL-Samarraie & Abbas, 2012).

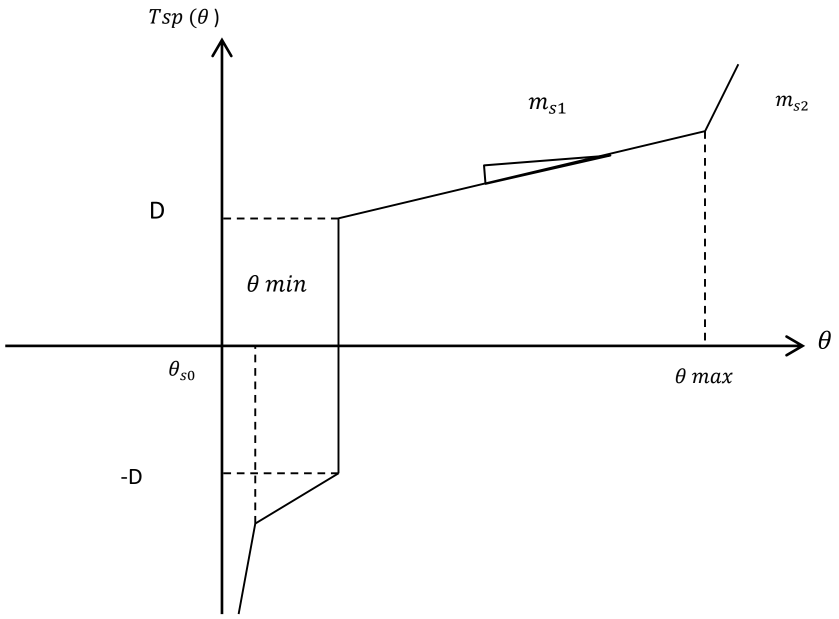


Figure 7. Nonlinear spring

The torque contributed by the spring force and its components, T_s, can be written in the following form as in Equation (3) and it can be simplified as in Equation (4), whereby the D is the initial position of the spring torque, θ_{max} is the maximum opening throttle valve, θ_{s0} is the throttle initial position and m_{s1} is the spring coefficient respectively.

$$T_s = \begin{cases} D + m_{s1}(\theta_s - \theta_{s0}), if\ \theta_{s0} < \theta_s < \theta_{max} \\ D - m_{s1}(\theta_{s0} - \theta_s), if\ \theta_{min} < \theta_s < \theta_{s0} \end{cases} \tag{3}$$

Or it can be simplified as,

$$T_s = m_{s1}\ (\theta_s - \theta_{s0}) + Dsgn(\theta_s - \theta_{s0}) \tag{4}$$

Another nonlinearity source besides the nonlinear spring and friction is the clearance in between a pair of mounted gears, which is called a gear backlash. In this work, the gear backlash is ignored during the mathematical model development of the throttle valve.

(ii) Electric DC motor with gearing mechanism

The model of the electronic throttle body can be described using the three main equations shown below, where Equation (5) and Equation (6) show the relationship between the input voltage of the motor, U_m, with z in the current across the DC motor winding, Rz motor resistance and to K_v motor back emf constant to the valve plate position angle θ_s to rotor angular velocity ω_m. Considering the sum of torque acting on the system which is shown in Equation (7) are torque supply by motor T_m, torque contributed from inertia force T_i, damping force T_d , frictional of the gear train T_f and torque contributed from spring force T_s (Loh et al., 2013).

$$U_m = L\dot{z} + K_v\omega_m + Rz \tag{5}$$

$$\dot{\theta}_s = (\, n_{1/2}n_{3/4})\omega_m \tag{6}$$

$$T_m = T_i + T_d \ + T_f + T_s \tag{7}$$

$$T_m = K_t\text{z} \tag{8}$$

$$T_i \ = J_{tot}\omega_m \tag{9}$$

$$T_d = B_{tot}\omega_m \tag{10}$$

The electrical part of the electronic throttle body is considered using z, current through the DC motor armature coil winding, and U_m voltage to the DC motor, J_{tot} and B_{tot} is total inertia and total damping acting on the system respectively, as described in Equations (11) and (12).

$$\dot{\omega} = -\ \frac{B_{tot}}{J_{tot}}\ \omega_m - \frac{1}{J_{tot}}F_s\, sgn(\omega_m) - \frac{1}{J_{tot}}\ [m_{s1}(\theta_s - \theta_{s0}) +\ Dsgn(\theta_s - \theta_{s0}) + \frac{K_t}{J_{tot}}z] \tag{11}$$

$$\dot{z} = -\frac{Kv}{L}\ \omega_m - \frac{R}{L}z +\ \frac{1}{L}u_m \tag{12}$$

The nonlinear functions of T_f and T_s are included as described in Equations (2) and (4). These equations can be assigned to three main equations of $x_1 = \theta_s$ and $x_2 = (n_{1/2}n_{3/4})\ \omega_m$, so that the state of the Equations (13), (14) and (15) according to (Akmar et al., 2008) can be simplified and written as;

$$\dot{x}_1 = x_2 \tag{13}$$

$$\dot{x}_2 = a_{10}(\, x_1 -\ x_{10}) \ +\ a_{11}\, x_2 \ +\ a_{12}z \ -\ a_{13}sgn(x_2) - a_{14}sgn(x_1 -\ x_{10}) \tag{14}$$

$$\dot{z} = a_{15}x_2 + a_{16}\, z +\ a_{17}u_m \tag{15}$$

The nominal parameter values considered in the model are given in Tables 2 and 3. The throttle model as given in Equation (15) will appear in the electronic throttle body modeling and be integrated into a conventional PID controller that forces the state to track a certain desired throttle valve angle.

Table 2. Electric throttle body parameters

Parameters	Symbols	Value	Unit
Inertia of rotor	J_m	$\geq 2 \times 10^{-6}$	kg.m/s^2
Viscous damping constant of rotor	B_m	2.05×10^{-5}	N·m·s/rad
Viscous damping constant of intermediate gear	B_{int}	0.0088	N·m·s/rad
Motor induction	L_m	$\leq 3 \times 10^{-3}$	H
Motor resistance	R_m	1-5	ohm
Motor torque constant	K_t	0.0185	Vsec/rad
Motor back emf constant	K_{emf}	0.01-0.08	Vsec/rad
Spring default position	θ_{s0}	$1 \pm 2.32 \times 10^{-3}$	rad
Spring maximum position	$\theta\ max$	$1 \pm 4.03 \times 10^{-2}$	Nm/rad
Spring minimum position	$\theta\ min$	$1 \pm 1.33 \times 10^{-2}$	Nm/rad
Gear ratio	n	20.68	unitless

Table 3. Parameter values of $a_{10}\ to\ a_{17}$

Parameters	Value
a_{10}	-1.6802×10^{-3}
a_{11}	-32.9820
a_{12}	4.2941×10^{-3}
a_{13}	2.1073×10^{-3}
a_{14}	4.6139×10^{-3}
a_{15}	-11.6039
a_{16}	-5.2087×10^{-2}
a_{17}	4.7438×10^{-2}

4.0 DEVELOPMENT OF AN INNER LOOP MODEL FOR THE ACTIVE IDLE STOP SYSTEM

The inner loop model plays an important role in the development of the AIS system, improving the dynamic performance of the passenger vehicle. The throttle body is categorized as the actuator for the vehicle system to supply the required air fuel mixture to the combustion chamber based on the driver's throttle pedal input. The performance of the passenger vehicle can be enhanced by controlling the throttle valve plate electronically using a controller. Therefore, the throttle body is used as an actuator to develop the inner loop model for the AIS system.

4.1 PID Controller Design

The PID algorithm is mostly used in automatic process control applications in industry. PID can be applied in many industrial process variables due to its simple structure, which allows easy implementation and adequate performance in most applications. PID controllers have three control modes, namely proportional control, integral control and derivative control; each of the modes reacts differently to an error. A standard PID controller can generally be written as a 'parallel form' or an 'ideal form' as expressed in Equation (16) and (17):

$$G_{PID}\,(s) = \frac{U(s)}{E(s)} = K_p \; + \frac{K_i}{s} + K_d \; s \tag{16}$$

$$G_{PID}\,(s) = \; K_p \; (1 + \frac{1}{T_{is}} + T_d \; s) \tag{17}$$

where $U(s)$ is the control signal acting on the error signal $E(s)$, K_p is the proportional gain, K_i is the integral gain and K_d is the derivative gain, T_i is as the integral time constant and T_d is the derivative time constant. In this study, the PID controller is used as the inner loop system to control the opening of the throttle valve of the electronic throttle body model. At the same time, the output from the inner loop control is used as an input for the outer loop control. The output from the outer loop control is used to control the vehicle's dynamic performance to avoid the vehicle decelerating and rolling back when driving uphill (Wang, Sponck & Tracht, 2003).

4.2 Control Structure of the Inner Loop Model

The electronic throttle body model is developed based on the mathematical equations discussed in Section 3. The developed ETB model has been evaluated using four types of throttle inputs. The model is tested using various inputs to verify its performance before it can be implemented in the vehicle model. A conventional PID controller is used to control the throttle opening valve of the ETB model. The complete control structure of the electronic throttle body (ETB) model is shown in Figure 8. Meanwhile, the parameters used for the ETB model are listed in Table 3.

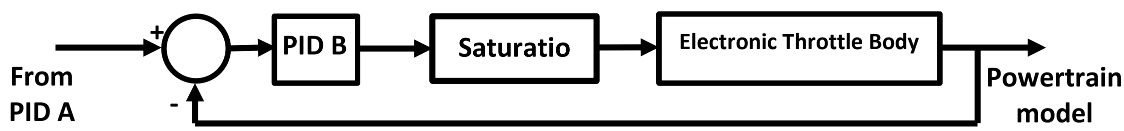
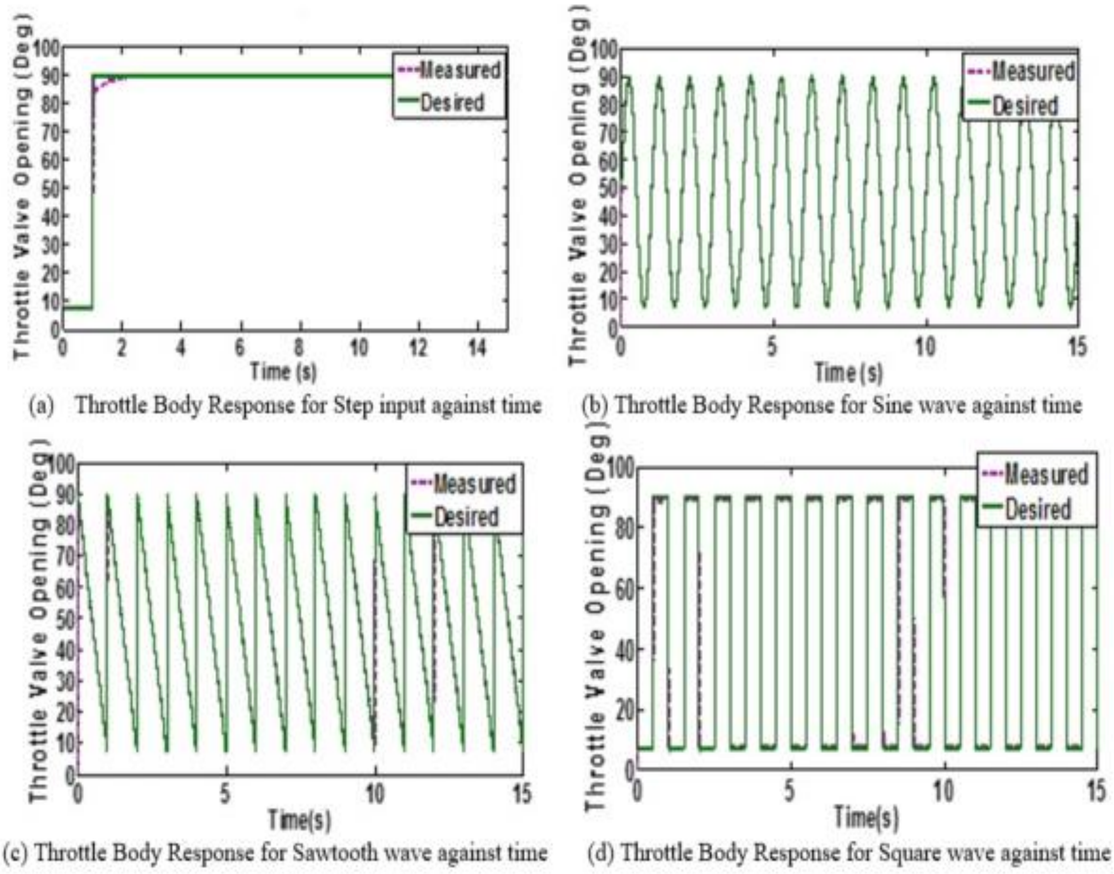


Figure 8. Control structure of the inner loop model

4.3 Position Tracking Control Results using the Inner Loop Model

Figures 9 (a) to (d) show the simulation results of the ETB with four types of input using maximum amplitude of 0.57 with frequency of 1 hertz. The inputs used in this simulation are square, sine, square and step inputs. The configuration parameter of the simulation model is based on Runga-Kutta solvers with fixed time of 0.001 seconds. Two types of result are illustrated in each result: desired and measured value. According to Figures 9 (a) to (d), the measured values from the simulation model are able to follow the desired inputs with and without delay time response. However, for the simulation results using square and step responses, the results show overdamped responses with a slight delay in approaching a settling time. The measured value required almost one second to approach the desired trajectory without any chattering during the settling period.



(a) Throttle Body Response for Step input against time
(b) Throttle Body Response for Sine wave against time
(c) Throttle Body Response for Sawtooth wave against time
(d) Throttle Body Response for Square wave against time

Figure 9. Position tracking response of the inner loop model using electronic throttle body

The performance results show that the control structure developed using the electronic throttle body model is able to follow desired trajectories with minimum errors. Thus, the model can be used as an inner loop model for a vehicle model to control the throttle of a vehicle system. In this study, the developed electronic throttle body is used as the inner loop model for AIS in a vehicle while traveling on a road surface with a gradient.

5.0 DEVELOPMENT OF THE ACTIVE IDLE STOP SYSTEM OPTIMIZED USING A GENETIC ALGORITHM

The AIS system can be developed using the vehicle longitudinal model in order to investigate the performance of the vehicle when driving uphill. In this case, the dynamic motion of the vehicle when on a level road (*at gradient*, $\theta_{vehicle}$ = 0°) is used as the reference point to minimize the rollback effect during uphill driving conditions, as shown in Figure 10. Therefore, the vehicle speed while traveling on a level road is measured and recorded as desired speed in order to initiate the AIS system. The AIS system will be activated only if the Conditional Desired Speed detects the road gradient ($\theta_{vehicle}$ > 0°) while in traveling mode.

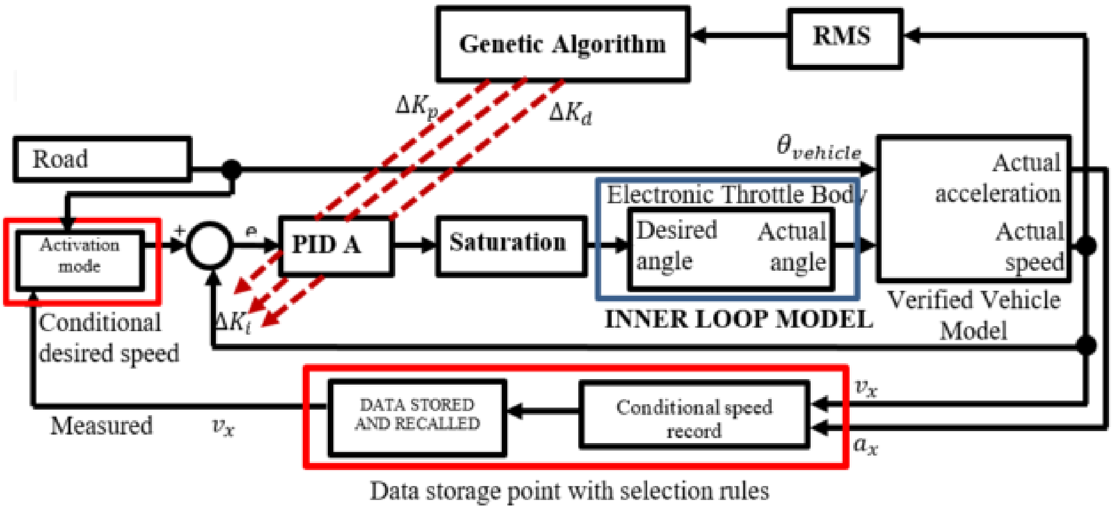


Figure 10. AIS system control structure

The AIS system can be activated based on two conditional rules which are used to define the operating condition of a vehicle either on a level road or one with an incline. The conditional rules are used to record the actual speed of the vehicle during level road traveling conditions and to indicate the desired speed in order to initiate the AIS system. The data storage point with selection rules can be defined as given in Equation (18):

$$\text{Vehicle data storing point} = \begin{cases} \text{If} \quad a_x \neq 0; \text{ then the vehicle speed is not stored} \\ \text{If} \quad a_x = 0; \text{ then the vehicle speed is stored} \end{cases} \tag{18}$$

The vehicle data storing point mainly depends on the vehicle's deceleration condition when traveling uphill. The data will be stored if there is no gradient while traveling in a longitudinal direction. This vehicle speed without uphill climbing conditions is used to define the conditional desired speed of the AIS system. Meanwhile, the conditional desired speed indication is defined based on the given rules in Equation (19):

$$\text{Conditional desired speed} = \begin{cases} \text{If } \theta_{vehicle} > 0°\text{; then desired speed and AIS is activated} \\ \text{If } \theta_{vehicle} = 0°\text{; then desired speed and AIS is deactivated} \end{cases} \quad (19)$$

The conditional desired speed is used for the AIS system based on the road gradient approach for the vehicle while traveling in a longitudinal direction. Once the control system of the AIS system detects an increment in the road gradient, the stored data on the vehicle speed while traveling on a level road gradient will be automatically used as the desired speed. Normally, the rollback condition occurs in most passenger vehicles when the vehicle stops on an inclined road gradient surface. Thus, this AIS system uses the measured desired speed during level road gradient as a reference to improve the deceleration and eventually reduce the rollback effect on the vehicle.

In order to optimize the control parameters used in the PID controller, a trial and error approach has been used as an initial stage of tuning the parameters. However, the parameters need to be optimized using a learning algorithm based on the required conditions of this AIS system. Therefore, a GA approach has been used as the optimization tool to obtain suitable control parameters for the AIS system. Genetic algorithms were first proposed by John Holland from the University of Michigan. This GA approach is a stochastic optimization method where the concepts are based on a natural selection and evolution process which is highly required for the AIS system. The procedure of optimizing using a genetic algorithm is shown as a flowchart in Figure 11 (Xiaofang, Yimin, Hui & Yaonan, 2012).

In the GA approach, each of the parameters should be encoded into a binary string known as a “Gene”. The genes are the first optimization layer and the results of the genes will be combined to develop “Chromosomes”. Thus, a non-negative real value known as “Fitness” is designed for each chromosome. The main target of GA is to maximize the fitness value in such a way the error in the AIS control structure can be reduced. Therefore, the fitness value needs to be defined accordingly where the results of the fitness value are able to minimize the objective function. The parameter values used to define the genetic algorithm are population size, selection method, fitness value, upper and lower bound, stopping criteria, mutation and crossover levels. The parameters of the genetic algorithm process set in the MATLAB Simulink are given in Table 4.

Table 4. Parameters of the genetic algorithm process

Parameters	Remarks
Population size	500
Selection method	Tournament
Stopping criteria	1000 generations
Fitness value	<0.00001
Mutation level	Default
Crossover level	Default
Upper bound	50
Lower bound	0

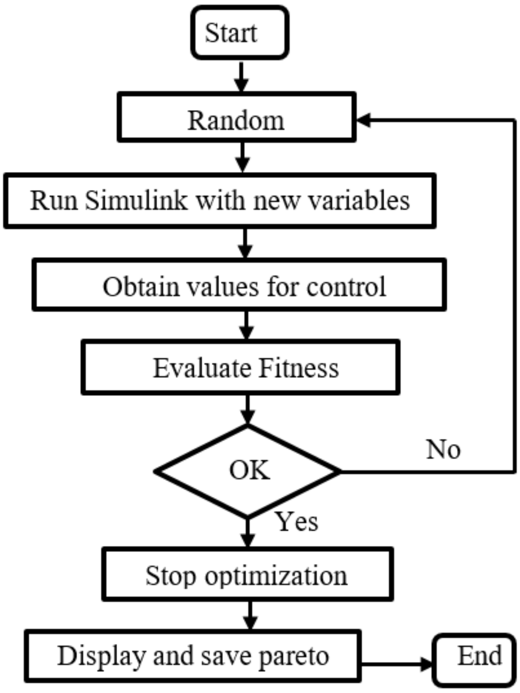


Figure 11. Flowchart of optimization process using a genetic algorithm

6.0 PERFORMANCE EVALUATION

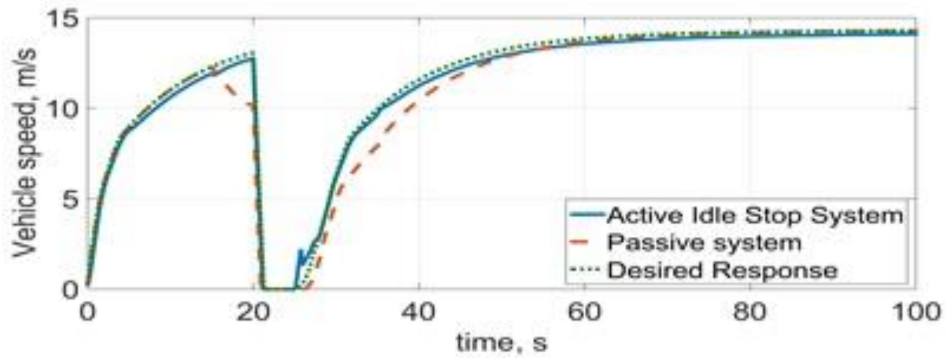
This section discusses the AIS system performance of a passenger vehicle while driving on an inclined road with various road gradients, $\theta_{vehicle}$. The developed control strategy of the AIS system is evaluated using three types of gradient values: 15°, 20° and 25°. The passenger vehicle is assumed to travel in a longitudinal direction from zero velocity until the vehicle travels from a level road surface to an inclined road surface and a sudden brake is applied. Then, the vehicle starts from zero velocity on an inclined road to observe the rollback effect due to the vehicle's deceleration. As discussed earlier, the AIS system will be activated once the vehicle detects a non-zero-gradient road surface. The simulation was performed using MATLAB Simulink software for a period of 100 seconds using Runge-Kutta solver with a fixed step size of 0.001 seconds. Three sets of PID control parameters as shown in Table 5 have been used to investigate the AIS performance when traveling uphill.

Table 5. PID controller parameters for the AIS system optimized by the GA tool

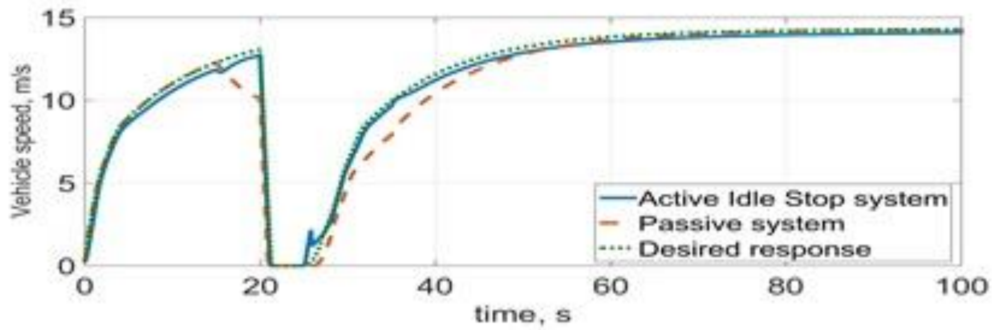
Angle (°)	Controller Parameters		
	K_p	K_i	K_d
15	8.012	0.113	3.003
20	8.031	0.121	3.012
25	8.023	0.123	3.013

6.1 Driving Conditions at Various Road Gradients

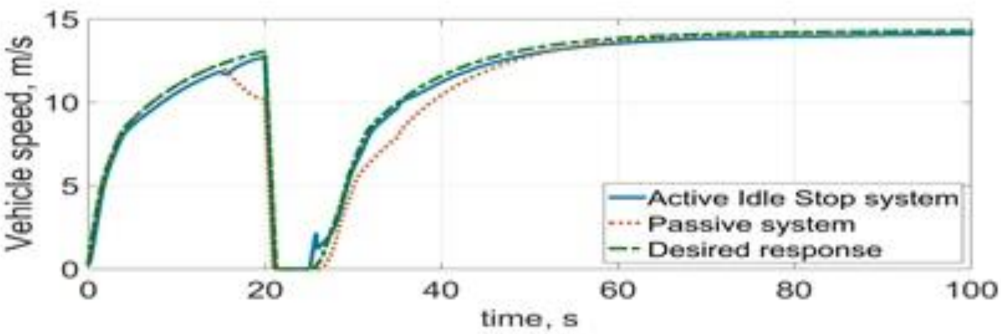
For the first condition, the vehicle travels in a straight direction by applying full throttle to the pedal. At 15 seconds, the vehicle approaches an inclined road surface, 15°, 20°, 25°, with full throttle input. Then, the vehicle is stopped by applying full brake input on the inclined road surface at 20 seconds. The vehicle is maintained on the incline for 5 seconds before being re-started from the halt condition. During the starting point, a full throttle is given to the vehicle until it approaches its maximum speed. The vehicle approaches the level road surface at 35 seconds and stays on the level road surface until the 100th second of testing, as shown in Figures 12, 13 and 14, which describe the vehicle speed performance, acceleration and distance travel response.



(a) Vehicle speed against time at 15 degree road gradient



(b) Vehicle speed against time at 20 degree road gradient



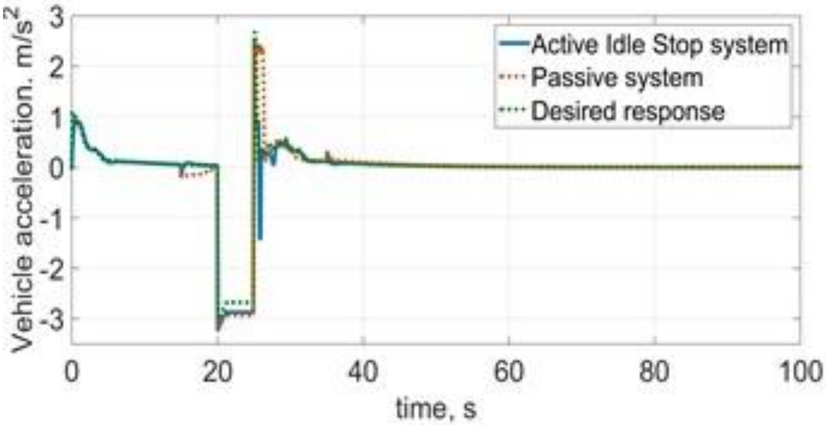
(c) Vehicle speed against time at 25 degree road gradient

Figures 12. Vehicle speed response at incline road of 15°, 20° and 25°

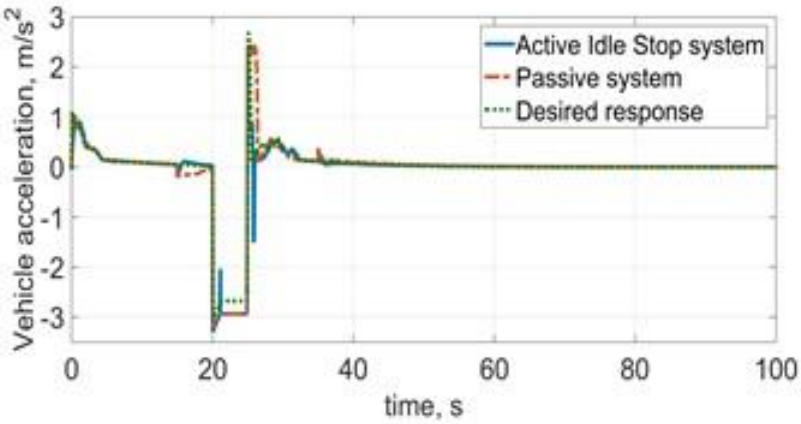
It can be observed that the vehicle shows the desired response when traveling on a road without any incline without any drop in vehicle speed or acceleration. However, during the uphill driving condition, a sudden drop in vehicle speed and acceleration for the passive vehicle can be observed, as shown in Figures 12 and 13. The sudden drop in vehicle speed and acceleration is observed at 15 seconds. By using the AIS system, the vehicle is able to improve the sudden drop in vehicle speed and acceleration that it experienced when climbing uphill and follow the desired response. For the road gradient of 15°, the vehicle speed drop at the 15th second is improved up to 88.6%, and for the 20° and 25° gradients, the vehicle speed drops have improved up to 83.4%. It can be assumed that the road gradient increment increases the vehicle speed drops due to the effect of the normal force and gravitational pull, *g*, acting in the opposite direction of travel to the vehicle.

The AIS system is able to improve the effect of vehicle dropping by more than 50% with minor deviation due to the gravitational force. Meanwhile, the main intention in using the AIS system is to reduce the deceleration which causes the vehicle to roll back during uphill driving from the halt condition. By implementing the AIS system, the vehicle speed while climbing at 15°, 20° and 25° is able to follow the desired response with percentage of similarity of 91.3 ± 5 %. Slight initial velocity spikes occurred in each case while starting the vehicle with the AIS system. This condition may occur due to the sudden torque generated by the AIS system to maximize the throttle opening valve plate. This condition gives an additional boost to the vehicle system so that it accelerates faster than the passive vehicle system. This spiking moment occurred 3 seconds before the AIS vehicle was able to follow the desired vehicle speed, as shown in Figures 12.

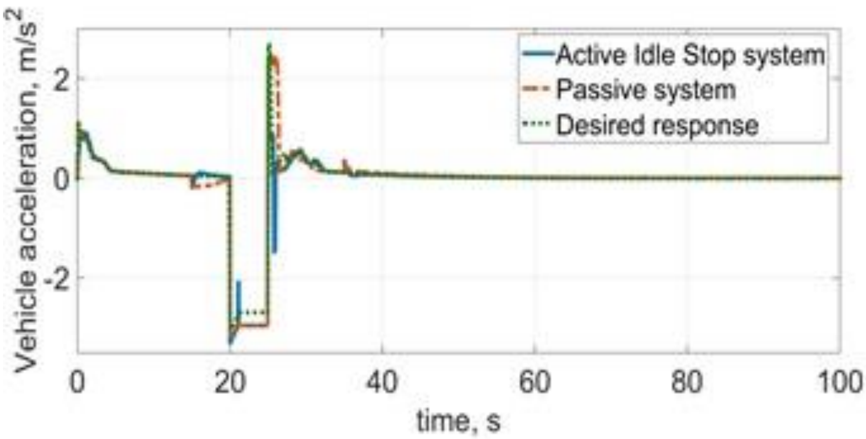
Meanwhile, the vehicle acceleration for the three cases is able to follow the desired response starting from the 15th to the 20th second with maximum overshoot of 6.5 % before the vehicle approaches the halt condition. Besides, the deceleration of the vehicle with the AIS system once it approaches the inclined road surface has been reduced compared with passive vehicle. Its shows that the AIS system is capable of improving dynamic performance of the vehicle while traveling uphill without reducing the vehicle's speed. However, there is a slight peak point in acceleration for the inclined roads with gradients of 20° and 25°, as shown in Figures 13(b) and 13(c). This is explaining by the effect of gravitational force which generates backward force opposing the vehicle while in a static position at the 22nd second. In terms of rollback effect, the vehicle deceleration has been significantly reduced, as can be seen in Figures 13(a), 13(b) and 13(c). The vehicle with the AIS system has been able to improve the rollback effects that occurred 25 seconds after the vehicle started from the static position in the uphill condition. The AIS system has improved the rollback of the vehicle by minimizing the deceleration effects. The AIS system is capable of reducing the effects by giving additional throttle angle input to the powertrain system at the 25th second. The additional opening of the throttle valve angle has increased the engine torque, which causes a sudden spike in vehicle acceleration speed response, as shown in Figures 12 and 13. However, the effect of additional opening of the throttle valve is reduced in less than 4.5 seconds.



(a) Vehicle acceleration against time at 15 degree road gradient



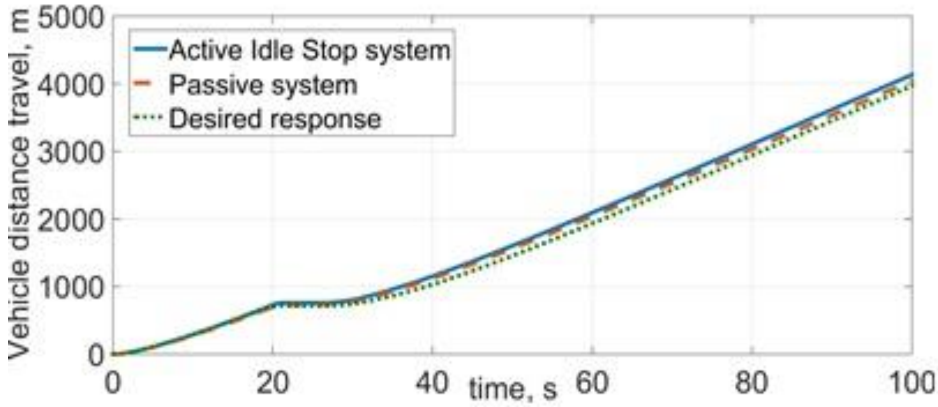
(b) Vehicle acceleration against time at 20 degree road gradient



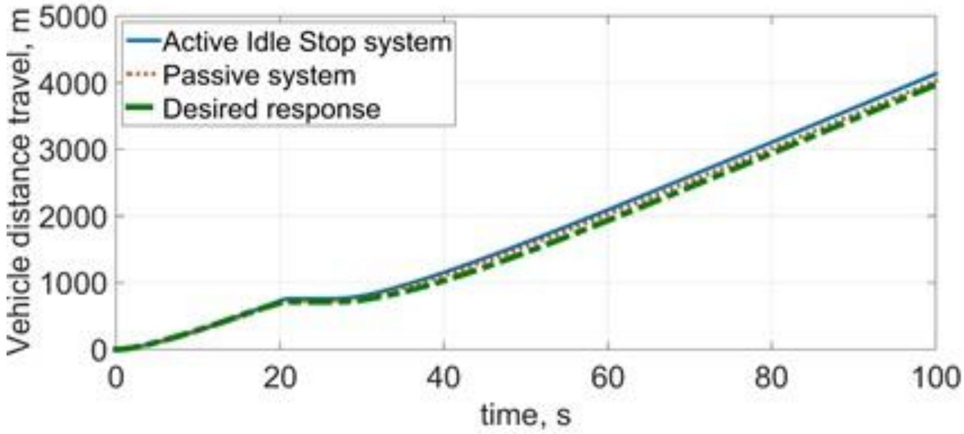
(c) Vehicle acceleration against time at 20 degree road gradient

Figure 13. Vehicle acceleration response at incline road of 15°, 20° and 25°

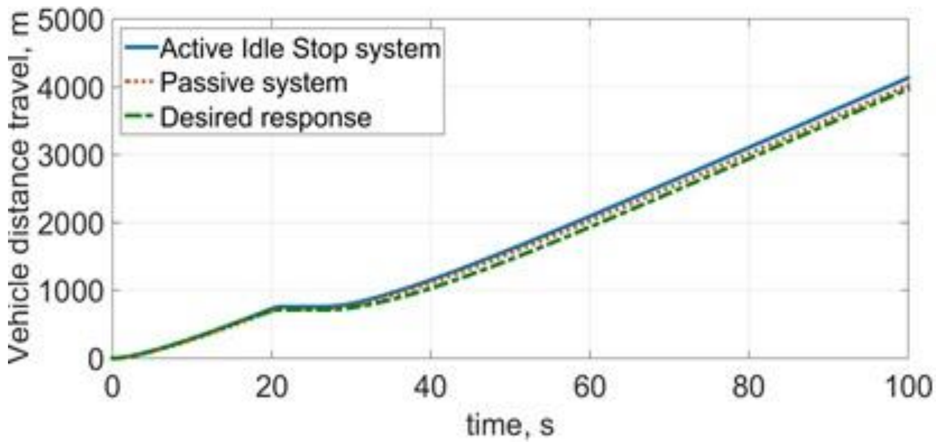
Figures 14(a) to 14(c) show the distance traveled by the vehicle with the AIS system, the passive vehicle and the desired vehicle travel response. The vehicle travels from the starting point of 0 m and approaches 1000 m at the 20th second. Then, the vehicle is in a static position (*on the uphill road condition*) until the 35th second, and then it starts to travel back from the current position until the 100th second, where the total distance traveled by the vehicle is 4800 m. Based on the results shown, it is clear that by introducing the AIS system the distance the vehicle travels is not disrupted compared with the passive vehicle system. The AIS system operates by improving the longitudinal dynamic motion of the vehicle and so maintaining the distance traveled.



(a) Vehicle distance travel against time at 15 degree road gradient



(b) Vehicle distance travel against time at 20 degree road gradient



(c) Vehicle distance travel against time at 25 degree road gradient

Figure 14. Vehicle distance travel response at incline road of 15°, 20° and 25°

7.0 CONCLUSION

Based on the simulation results, it can be concluded that the AIS system using the vehicle longitudinal model shows a significant improvement in reducing the rollback of the vehicle while climbing an uphill road surface. The vehicle model is able to follow the desired speed to reduce the deceleration effects which occurred in the vehicle without the AIS system. The developed AIS system is able to operate during uphill driving until 25 degrees and the vehicle is able to maintain the desired speed up to 85%. Thus, the AIS system has proven to be a very effective safety system to be implemented in passenger vehicles, which will be very useful when driving up an incline.

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