DEVELOPMENT OF ANTILOCK BRAKING SYSTEM USING ELECTRONIC WEDGE BRAKE MODEL

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ABSTRACT

The development of an Antilock Braking System (ABS) using a quarter vehicle brake model and electronic wedge brake (EWB) actuator is presented. A quarter-vehicle model is derived and simulated in the longitudinal direction. The quarter vehicle brake model is then used to develop an outer loop control structure. Three types of controller are proposed for the outer loop controller. These are conventional PID, adaptive PID and fuzzy logic controller. The adaptive PID controller is developed based on model reference adaptive control (MRAC) scheme. Meanwhile, fuzzy logic controller is developed based on Takagi-Sugeno technique. A brake actuator model based on Gaussian cumulative distribution technique, known as Bell-Shaped curve is used to represent the real actuator. The inner loop controls the EWB model within the ABS control system. The performance of the ABS system is evaluated on stopping distance and longitudinal slip of vehicle. Fuzzy Logic controller shows good performance for ABS model by reducing the stopping distance up to 17.4% compared to the conventional PID and Adaptive PID control which are only 7.38% and 12.08%.

KEYWORDS*: Anti-lock braking system; Electronic wedge brake; MRAC; Adaptive PID control; Fuzzy logic*

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1.0 INTRODUCTION

Currently, vehicles have become major assets for every human in their daily life usage. Owning personal vehicles will tend to reduce time and save energy for humans to travel from one place to another, but this has significantly increased the risks to the human's life. According to the Malaysian Institute of Road Safety Research (MIROS), there were 3.55 deaths per 10,000 vehicles and 28.8 deaths for 100,000 populations in Malaysia every year (MROADS, 2009; Ghani & Musa, 2011). Based on statistical analysis by MIROS, the main cause of vehicle accidents is instability condition in which drivers lost control of the vehicle by steering, throttle or braking input (Ghani & Musa, 2011). The major effects of driver losing control of the vehicle are rollover and skidding effect during cornering and braking input.

Furthermore, vehicle skidding may cause oversteer where the rear end of vehicle slides out of the road or understeer, a condition where the front vehicle turns over outside the road without following the curve of the road. Other major skid effect is called four wheels skid where all the four wheels lock up and the vehicles starts to slip in the direction of forward momentum of the vehicle. During this condition occurring, there is a high possibility for the wheels to stop from rotating before the vehicle approaches to a halt. This process is known as 'locking up' where the braking force on the wheel is not being transferred efficiently to stop the vehicle because of the wheels skid on the road surface (Li & Yang, 2012). This leads to increase in the stopping distance of a vehicle since the grip between the wheel and the road has reduced drastically. Hence, it leads to high risk for a driver to lose control of the vehicle and unable to avoid the disturbance (Cabrera et al., 2005).

Henceforth, a new type of braking technology has been designed and developed by automotive designers by considering after effects of vehicle skidding. A wide range of controllable braking technology has been invented by including the conventional braking system with electrical and electronics components (Aparow et al., 2013a). This type of braking technology is commonly known as active braking system such as electronic brake distribution (Laine & Andreasson, 2007), electronic stability control (Zhao et al., 2006; Piyabongkarn et al., 2007), automatic braking system (Littlejohn et al., 2004; Avery & Weekes, 2009) and also antilock braking system (Oudghiri et al., 2007; Aparow et al., 2013a). All this controllable braking system has been enhanced by automotive designers as part of safety system in vehicle. However, automotive designers have focused more on antilock braking system (ABS) since this active braking system has high potential to avoid wheel lock up and reduce stopping distance once emergency brake is applied.

Basically, active braking known as antilock braking system was first introduced in aircraft technology to reduce the stopping distance of an aircraft. However, automotive researchers have adopted the knowledge of this active braking to ground vehicle design as vehicle safety system. Automotive researchers realized the capability of this system to reduce stopping distance in a safer method (Oniz et al., 2009; Aparow et al., 2013a). This active braking system was developed in order to control locking of the wheel during sudden braking (Oudghiri et al., 2007). In general, this type of active braking can be classified into three categories. Firstly, ABS via hydraulic system which is developed using hydraulic fluid used as a medium to transfer brake force from brake pedal to brake pad and rotors (Junxia & Ziming, 2010). Secondly, ABS via pneumatic system is designed by using air as a medium to transfer brake pressure which has been implemented in early 60's. Unfortunately, this type of active braking system is applied mainly in commercial heavy vehicle such as truck, lorry and bus (Zhang et al., 2009; Zhang et al., 2010). Thirdly, ABS via combination of electronic and mechanical system is invented which used combination of electronic and mechanical systems (Yang et al., 2006). This type of braking system is the latest technology which is also invented based on aircraft and this technology is known as brake-by-wire.

Many types of control strategies have been developed as the results of extensive research on antilock braking system (ABS). Control strategies used in order to control the ABS using electro hydro brake (EHB) and recently introduced for electronic wedge brake (EWB). Intelligent control strategies that have been applied for ABS using EHB are adaptive fuzzy-PID controller (Li et al., 2010), fuzzy logic control used by Aly et al., (2011), fuzzy sliding mode controller proposed by Oudghiri et al., (2007) and PID-particle swarm optimization (PSO) control technique proposed by Li and Yang (2012). Since ABS via EWB is still a new technology, many researchers such as Nantais and Minaker (2008) and Kim et al. (2010) have put their effort in designing ABS using simulation and applied in passenger vehicle. Moreover, Han et al. (2012) has initiated the controller research using sliding mode control using EWB actuator to estimate clamping force for braking. Most of the researchers used basic control algorithm such as PID to control ABS via EWB. However, the researchers do not identify a suitable controller by studying different controllers and make evaluation of their performance.

Based on previous research works such as by Oudghiri et al. (2007), (Li et al., 2010) and Aly et al., (2011) for ABS using electro hydro brake and Nantaiz and Minaker (2008), Kim et al. (2010), Han et al. (2012)

for ABS using electronic wedge brake, an antilock braking system is proposed in this study. The antilock braking system was developed using electronic wedge brake model using Bell-Shaped curve. The proposed ABS-EWB model is a hydraulic free active braking system where this system tends to help humans to control vehicle steering and stability under heavy braking by preventing the wheels from locking up. Moreover, the proposed ABS-EWB model represents an active braking system based on brake-by-wire technology. Meanwhile, three types of control strategy are proposed and compared in this study for ABS-EWB model. The controllers are conventional PID, adaptive PID and fuzzy logic controller. The main purpose of proposing these types of controller is to identify a suitable control scheme for ABS-EWB by evaluating the performance of the model in terms of wheel longitudinal slip and also stopping distance of a vehicle. Adaptive PID control is developed using model reference adaptive control (MRAC) technique based on MIT rule while fuzzy logic is developed using Takagi-Sugeno configuration method.

The paper is organized as follows: The second section presents the dynamic model of the vehicle in longitudinal direction. The following section presents the proposed control structure for the ABS inner loop control using EWB model and outer loop control using validated quarter vehicle model. The next section describes the integration of both outer loop and inner loop control structures to develop an ABS control scheme using PID, adaptive PID, fuzzy logic as the closed loop controller. The sixth section presents the performance evaluation of the proposed ABS control systems. The last section contains the conclusions of this study.

2.0 DEVELOPMENT OF A QUARTER VEHICLE BRAKE MODELING

Referring to works done by Oniz et al. (2009) and Li and Yang (2012), the brake modeling for a single wheel with quarter vehicle mass can be derived as mathematical equations. There are few types of responses of the vehicle due to brake input from driver and one of the response analyze in this study is skidding due to excessive braking. Four types of output response can be obtained using quarter vehicle brake model which are vehicle velocity, wheel velocity, longitudinal slip and stopping distance of a vehicle. The following section describes in detail on the development of the quarter vehicle model.

2.1 Quarter Vehicle Brake Mathematical Modeling

The free body diagram of a quarter vehicle model is as shown in Figure 1 describes the longitudinal motion of the vehicle and angular motion of the wheel during braking. Even though the model is quite general but it can sufficiently represent the actual vehicle system's fundamental characteristics (Oniz et al., 2009). There are several assumptions made in deriving the dynamic equations of the system. First, the vehicle is considered in a longitudinal direction. The lateral and vertical motion of the vehicle is neglected. Secondly, the rolling resistance force is ignored since it is very small during braking. Thirdly, it is assumed that there is no interaction between four wheels of the vehicle (Oniz et al., 2009; Li & Yang, 2012). Figure 1 shows the non-braking condition and braking condition of a quarter vehicle model.

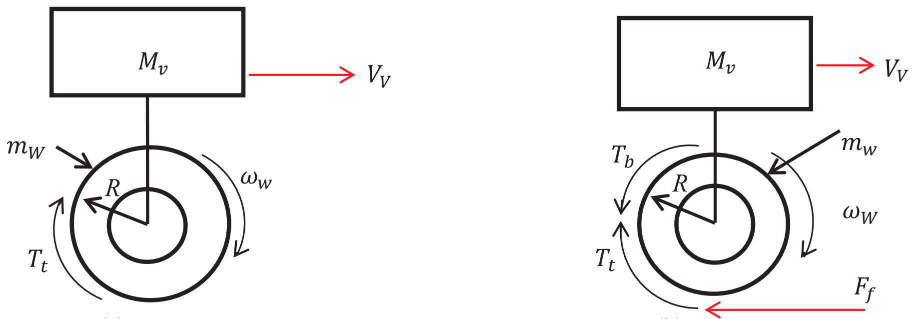

Figure 1(a). Non-braking condition and (b) braking condition (Ahmad et al., 2009)

Based on Newton's second law, the equation of motion for the simplified vehicle model can be expressed as:

$$F_f = -((M_v + m_w) \times a_v) \tag{1}$$

where M_v represents the total mass of a vehicle body, m_w represents the wheel mass and a_v represents the acceleration of the vehicle while F_f represents the tire friction force and based on the law of Coulomb is represented by:

$$F_f \;=\; \mu_s F_N \tag{2}$$

Thus

$$-(M_v + m_w) \times a_v = \mu_s F_N \tag{3}$$

In Equations (2) and (3), F_N represents the total normal load while the road adhesion coefficient, μ_s is a function of vehicle velocity and wheel slip.

During deceleration, braking torque is applied to the wheels to reduce wheel speed and vehicle speed. The rolling resistance force occurred at the wheel is neglected since it is very much smaller than friction force between the wheel and road (Oniz et al., 2009). According to Newton's second law, the equation of motion of the wheel can be written as:

$$\sum T_w = T_t - T_b \tag{4}$$

where T_w is wheel torque, while T_b and T_t represent the brake torque and tire torque respectively, Based on Equation (4), the following equations are derived as:

where the wheel torque, T_w, is

$$\sum T_w = I_\omega \times \alpha_\omega \tag{5}$$

where I_ω is the wheel inertia, angular acceleration is represented by α_ω, while the tire torque, T_t and tire friction force, F_f, are

$$T_t = F_f \times R_r \tag{6}$$

and

$$F_f = \mu_s F_N \tag{7}$$

where R_r is the radius of the wheel. By substituting Equations (5), (6) and (7) into (4), Equation (8) can be obtained as

$$\mu_s F_N \times R_r - T_b = I_\omega \times \alpha_\omega \tag{8}$$

where the total normal load, F_N, is

$$F_N = (M_v + m_w)g \tag{9}$$

substitute Equation (9) into Equation (8) and rearrange to obtain angular acceleration, α_ω,

$$\alpha_{\omega} = \frac{[\mu_s(M_v+m_w)g\times R_r]- T_b}{I_{\omega}} \tag{10}$$

The longitudinal wheel speed, V_w, and vehicle speed, V_v, is represented in Equations (11) and (12):

$$V_v = \int a_v\, dt = \int -\frac{F_f}{(M_v+m_w)}dt \text{ (vehicle speed)} \tag{11}$$

and

$$V_w = \int a_{\omega}\, dt = \int \frac{[\mu_s(M_v+m_w)g\times R_r]- T_b}{I_{\omega}}dt \text{ (wheel speed)} \tag{12}$$

The rotational velocity of the wheel would be similar with the forward velocity of the vehicle under normal operating conditions. Once the brakes are applied, braking forces are generated at the interface between the wheel and road surface. The braking force will cause the wheel to reduce the speed. Increasing the forces at the wheel will increase slippage between the tire and road surface and this will cause the wheel speed to be lower than the vehicle speed (Oniz et al., 2009). The parameter used to obtain the slippage occurring at the wheel using both vehicle and wheel velocity is known as wheel slip, λ,

$$\lambda = \frac{V_v - V_{\omega}}{max(V_v,V_{\omega})} \tag{13}$$

The forward velocity and tire rolling speed are equal when the slip ratios of the vehicle approach zero. Zero slip ratio implies absence of either engine torque or brake torque. A *positive* slip ratio implies the vehicle approaches a greater finite forward velocity with the tire having a positive finite rolling velocity. While a negative slip ratio implies the tire has a greater equivalent positive rolling velocity meanwhile the vehicle has a finite forward velocity (Short et al., 2004). Two conditions occurred when the longitudinal slip approaches either +1 or -1. The conditions are the wheel is 'spinning' with the vehicle at zero speed or the wheel is 'locked' at zero speed. The longitudinal slip is mathematically undefined when both tires and vehicle velocities are equal to zero. Figure 2 shows the graph relating friction coefficient, μ_s with wheel slip, λ (Aparow et al., 2013a).

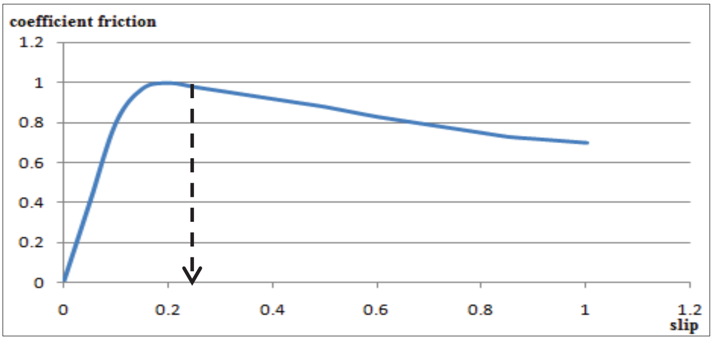

Figure 2. Experiment data for μ_s vs slip (Aparow et al., 2013a)

The value of friction coefficient, μ_s, is obtained using a lockup table based on the experiment on normal road surface. These values were gathered from tire experiment conducted on a normal surface road. The response from the variation of friction coefficient with slip ratio is used as the input for quarter vehicle brake model.

2.2 Description of Simulation Model

The vehicle dynamics model is developed based on the mathematical equations from the previous quarter vehicle model by using MATLAB SIMULINK. Figure 3 shows the simulation model of the quarter vehicle brake model with a step function as the brake input. The step response represents the brake torque input from brake pedal during braking. This signal is used in the quarter vehicle model for dynamic performance analysis in longitudinal direction only. Meanwhile, the parameters for the quarter vehicle model are listed in the Table 1.

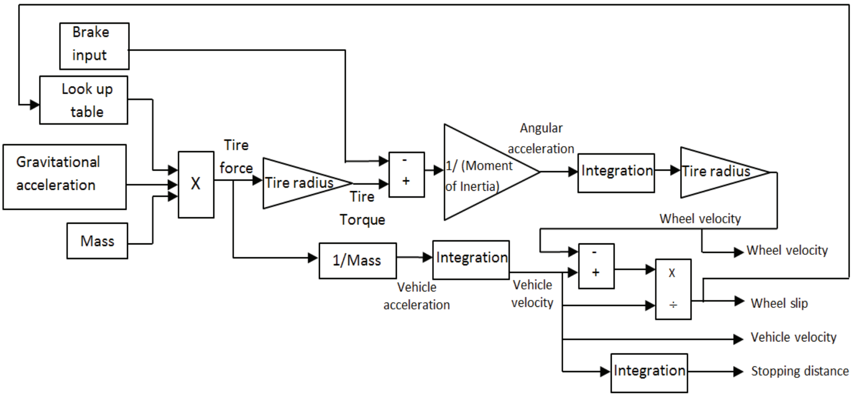

Figure 3. Quarter vehicle model in Matlab SIMULINK

Table 1. Simulation model parameters for the vehicle model (Ahmad et al., 2009)

Description	Symbol	Value
Wheel Inertia	I_ω	5 kgm^2
Brake coefficient	K_b	0.8
Vehicle mass	M_v	690 kg
Wheel mass	m_w	50 kg
Tire radius	R_t	285 mm
Gravitational acceleration	g	9.81 m/s^2
Coefficient of friction	μ	0.4
Vehicle speed	v0	25 m/s

3.0 EWB MODELING USING BELL-SHAPED CURVE

By refering to Aparow et al. (2013b), a mathematical equation using Bell-Shaped curve has been used to developed electronic wedge brake model. The mathematical equation can represent the actual force and brake torque produced by a real EWB actuator. The derived mathematical equation using trigonometric function such as tangential function (Aparow et al., 2013b). In this study, Bell-Shaped curve, a non-linear equation is used described by

$$F_M = F_{cs}[1/(1 + \left|\frac{x-c}{a}\right|^{2b}) \tag{14}$$

where
F_M is the maximum clamping force for EWB system (output)
F_{cs} is the real time clamping force of EWB system
x is the piston displacement (input)
a and b are the width of the curve
c is the center of the curve

The parameters used in the Equation (14) such as a, b and c are estimated to sufficiently the behaviour of EWB simulation model to replace an EWB actuator. The parameters of a, b and c are estimated through trial and error technique and the data has been analyzed using root mean square error (RMSE) technique. The parameters for Equation (14) is obtained based on the least error results from the RMSE analysis. The parameters is listed in Table 2 based on the simulation and experiment work of EWB. According to Equation (14), the input of the proposed equation is obtained from the worm pinion displacement while the output of this model represents the clamping force of EWB model. Hence, brake torque of the EWB model using Bell-Shaped curve can be obtained using Equation (15) which was obtained from Rahman et al., (2012)

$$T_b = F_{fc} r_{eff}$$

where

$F_{fc} = 2\mu_f F_N$

Hence it can be conclude that

$$T_b = 2\mu_f \, F_N \, r_{eff} \tag{15}$$

where

T_b is the brake torque produced
F_{fc} is the friction force generated at the contact interface
r_{eff} is the effective pad radius
F_N is the normal force
μ_f is the friction coefficient of on the surface

It is also necessary to define the coefficient of friction between disc and pad contact interface to calculate the brake torque. The coefficient of friction is dependent on the braking conditions such as disc speed, temperature of the disc, brake line pressure and other factors (Rahman et al., 2012; Aparow et al., 2013b).

Table 2. Simulation parameters for bell-shaped curve and brake torque

Description	Value
F_{cs}	3500 N
μ_f	0.45
r_{eff}	0.15 m
a	6.295×10^{-4} m
b	2.820×10^{-4} m
c	9.570×10^{-4} m

4.0 CONTROL STRCTURE DESIGN

In the previous section, the development of a quarter vehicle model and electronic wedge brake model using Bell-Shaped curve has been discussed in detail. In this section, an active braking system, Anti-lock Braking System (ABS) is developed using the quarter vehicle brake model and EWB model. Figure 4 shows the proposed outer and inner loop control for ABS model. Two control loops are needed to be designed in developing an ABS model control scheme using both quarter vehicle model and EWB model. These control loops are known as an inner loop control and outer loop control. The inner loop control is used to control the actuator of the proposed system. In this study, EWB model is also used as the actuator for the ABS control system.

The actuator model is the focus of on electronic wedge brake mechanism (EWB) (Rahman et al., 2012) system since the brake actuator model is developed based on brake-by-wire concept. Meanwhile, the outer loop control for ABS system is to control the quarter vehicle model. Initially, the longitudinal slip for quarter vehicle model is controlled without involving brake actuator model or inner loop control. The controller for this model will control the quarter vehicle model's wheel slip directly without any interference until the actual wheel longitudinal slip is obtained that closely follow with the desired wheel longitudinal slip (Aparow et al., 2013a). Similar procedure is applied for inner loop control model in order to control brake torque.

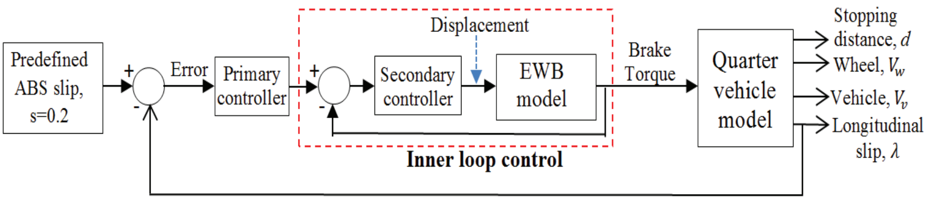

Figure 4. Proposed outer and inner loop controls for ABS model

Then, both loop controls are integrated to become a closed loop model known as ABS-EWB model to control the wheel longitudinal slip and stopping distance of quarter vehicle model. The proposed controller for inner loop control is PID controller while three types of controller, namely PID, adaptive PID and fuzzy logic controllers are proposed for the overall ABS-EWB control structure. The results using all three controllers are compared and most optimum controller is proposed for ABS-EWB model.

4.1 PID controller

PID algorithm stil is remain as one of the most widely used controller in industrial process control. It is not only because PID algorithm has a simple structure whereby easy to implement but also provides adequate performance in most applications. PID controller may also be considered as a phase lead-lag compensator with one pole formation at two points which is at the origin and other at infinity. A standard PID controller is known as " three-term" controller where the transfer function of PID is generally written in the "parallel form" or "ideal form"

$$G_{PID}(s) = \frac{U(s)}{E(s)} = K_p + K_i/s + K_d s \quad (16)$$

$$G_{PID}(s) = K_p(1 + \frac{1}{T_i s} + T_d s) \quad (17)$$

where $U(s)$ is the control signal acting on the error signal E(s), K_p is the proportional gain, K_i is the intergal gain, K_d the derivative gain, T_i the intergal time constant and T_d the derivative time constant. In this study, PID controller is used in the inner loop system to regulate the brake torque from the electronic wedge brake (EWB). 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 the wheel longitudinal slip to avoid the vehicle from skidding during braking. The PID parameter are tuned using Knowledge Based Tuning (KBT) method (Calvo-Rolle et al., 2011) until actual brake torque produced by EWB model is same as desired brake torque in order to control the wheel longitudinal slip of quarter vehicle model.

4.2 Outer loop Control Design using Adaptive PID controller

An adaptive mechansim is needed to tune the PID parameters to make overall system behaves as the reference model. Therefore, a model reference adaptive control (MRAC) using MIT rule is proposed as an adaptive mechanism for PID controller (Adrian et al., 2008). The adaptive PID controller which is applied in the outer loop model is shown in Figure 5 while mathematical derivations is described by the following equations.

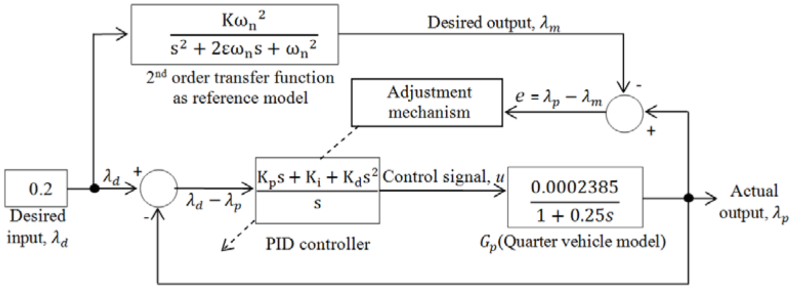

Figure 5. Adaptive PID using MIT Rule for outer loop model

The adaptive PID controller parameters are adjusted to give desired closed-loop performance for ABS model. The controller parameters are estimated to cause the desired change in plant transfer function so that the actual output can be made similar to the reference model (Swarnkar et al., 2011).

4.2.1 Reference model

Normally, the reference model will be the desired overall system's performance by the designer. The adaptive mechanism is developed based on the error between the output response of the plant with the reference model. The adaptive controller is designed to force the plant model to act similar to reference model. Model output, λ_m, is compared to the actual output, λ_p, and the difference is used to adjust feedback PID controller parameters. The reference model used for the adjustment mechanism is

$$\frac{\lambda_m(s)}{\lambda_{p(s)}} = \frac{K\omega_n{}^2}{s^2+2\varepsilon\omega_n s+\omega_n{}^2} \tag{18}$$

where, ω_n is the undamped natural frequency, ε, is the damping ratio while K is the gain of the system. The value of ω_n = 15 rad/s, ε = 0.65, K = 1. These values are chosen for the closed loop system to have maximum overshoot ≤ 5% at rise time 0.1 s and peak time at 0.25 s when excited with a step function.

4.2.2 Transfer Function for Quarter Vehicle Brake Model

The inner loop model is ignored in this stage in order to develop outer loop ABS model. The purpose of ignoring the inner loop model is to avoid interference during the development of the adaptive control. The Laplace transfer function of the quarter vehicle model, G_p, is needed to derive the adjustment mechanism. This transfer function model is developed using single input single output (SISO) concept. The step response is used as the brake input for both quarter vehicle brake model and first order transfer function. The comparison results of the quarter vehicle brake model and the approximation of the response as a first order system are shown in Figure 6.

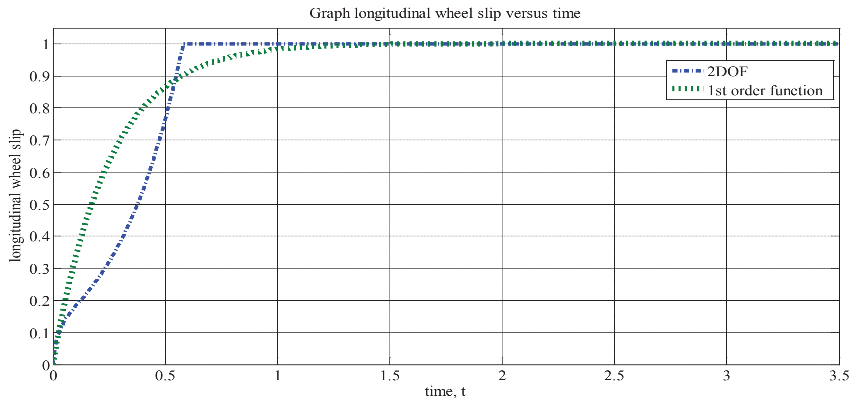

Figure 6. Comparison of 1st order transfer function and quarter vehicle model

Transfer function of plant model, $G_{p,}$ using quarter vehicle model is shown in Equation (19)

$$G_p = \left(\frac{b}{a_1+a_0 s}\right) = \left(\frac{0.002385}{1+0.25s}\right) \tag{19}$$

Meanwhile the output from plant model, λ_p, is

$$\lambda_p = \left(\frac{0.002385}{1+0.25s}\right)\left(\frac{K_p s+K_i+K_d s^2}{s}\right)\left(\lambda_d - \lambda_p\right) \tag{20}$$

where λ_d-λ_p represents the difference between desired input and actual response for the outer loop model. Rearrange Equation (20) to obtain the closed function of outer loop model

$$\frac{\lambda_p}{\lambda_d} = \left(\frac{0.002385(K_p s+K_i+K_d s^2)}{(0.25+0.002385K_d)s^2+(1+0.002385K_p)s+0.002385K_i}\right) \tag{21}$$

By applying the MIT gradient rule, the adaptation value for PID controller parameters K_p,K_i and K_d are obtained. The parameters are derived based

$$\frac{dK_x}{dt} = -\, \gamma_p \left(\frac{\partial J}{\partial K_x}\right) = -\, \gamma_x \left[\left(\frac{\partial J}{\partial \varepsilon}\right)\left(\frac{\partial e}{\partial y}\right)\left(\frac{\partial y}{\partial K_x}\right)\right] \tag{22}$$

where K_x = Proportional (P) or Intergral (I) or Derivative (D), $\partial J/\partial e$= e and $\partial e/\partial y$ = 1. Equation (22) has been derived mathematically in order to develop an adjustment mechanism for PID controller.

$$\frac{\partial Y_p}{\partial K_p} = \left(\frac{0.002385s}{(0.25+0.002385K_d)s^2+(1+0.002385K_p)s+0.002385K_i}\right)(\lambda_d - \lambda_p) \quad (23)$$

$$\frac{\partial Y_p}{\partial K_i} = \left(\frac{0.002385}{(0.25+0.002385\mathrm{K_d})\mathrm{s}^2+(1+0.002385\mathrm{K_p})\mathrm{s}+0.002385\mathrm{K_i}}\right)(\lambda_d - \lambda_p) \quad (24)$$

$$\frac{\partial Y_p}{\partial K_d} = \left(\frac{0.002385\mathrm{s}^2}{(0.25+0.002385\mathrm{K_d})\mathrm{s}^2+(1+0.002385\mathrm{K_p})\mathrm{s}+0.002385\mathrm{K_i}}\right)(\lambda_d - \lambda_p) \quad (25)$$

By refering to Equations (23) to (25), the adaptation value for PID parameters are obtained as Equations (26) to (28) using 1st order transfer function. This equations will be used as adjustment mechanism for the PID controller of outer loop control to develop an adaptive controller using PID.

$$\frac{dK_p}{dt} = -\gamma_p \left(\frac{\partial J}{\partial K_p}\right) = -\gamma_p\, e\left[\left(\frac{0.002385s}{(0.25+0.002385K_d)s^2+(1+0.002385K_p)s+0.002385K_i}\right)(\lambda_d - \lambda_p)\right] \quad (26)$$

$$\frac{dK_i}{dt} = -\gamma_\mathrm{i} \left(\frac{\partial \mathrm{J}}{\partial \mathrm{K_i}}\right) = -\gamma_\mathrm{i}\, \mathrm{e}\left[\left(\frac{0.002385}{(0.25+0.002385K_d)s^2+(1+0.002385K_p)s+0.002385K_i}\right)(\lambda_\mathrm{d} - \lambda_\mathrm{p})\right] \quad (27)$$

$$\frac{dK_d}{dt} = -\gamma_d \left(\frac{\partial J}{\partial K_d}\right) = -\gamma_d\, e\left[\left(\frac{0.002385s^2}{(0.25+0.002385K_d)s^2+(1+0.002385K_p)s+0.002385K_i}\right)(\lambda_d - \lambda_p)\right] \quad (28)$$

where γ represents the learning rate of the adaptation mechanism and ε are the defined error based on the difference between actual output and desired output. The parameters for the adjustment mechanism in the Equations (18) to (28) are obtained from linearization of the controller gain with the desired slip value of antilock braking, λ. Before linearization is made, it is necessary to identify the PID controller parameter for the braking model using sudden braking test input. The identified PID controller is used as a bench mark in developing adaptation mechanism for MRAC model. The initial values of the PID controller parameters and learning rate used in this model are shown in Table 3.

Table 3. Controller and learning rate parameters

Description	Symbol	Value
Proportional gain, P	K_p	550
Intergal gain, I	K_i	2950
Derivative gain, D	K_d	10
Learning rate of P	γ_p	0.001
Learning rate of I	γ_i	0.05
Learning rate of D	γ_d	0.05

4.3 Outer loop Control Design using Fuzzy Logic controller

Another advance controller, fuzzy logic, was developed in this study based on *Takagi-Sugeno* method for the outer loop control system besides adaptive PID controller. In this network model, the controller consists of three stages which are known as fuzzification, rule base and defuzzification (Ping et al., 2010). Figure 7 shows the overall control structure of fuzzy logic control using Gaussian membership function for the outer loop. The inputs to the fuzzy controller are error and error rate. These fuzzy or linguistic variables are obtained through fuzzification process. The output of the fuzzy controller is gap control of brake piston in EWB actuator. Membership functions of the input and outpu fuzzy variables are defined. The output-input variables are linked by rules. The membership functions and rules are defined based on expert experinece. Finally, defuzzification method converts the fuzzy output into a crisp signal and become the command input for the plant model.

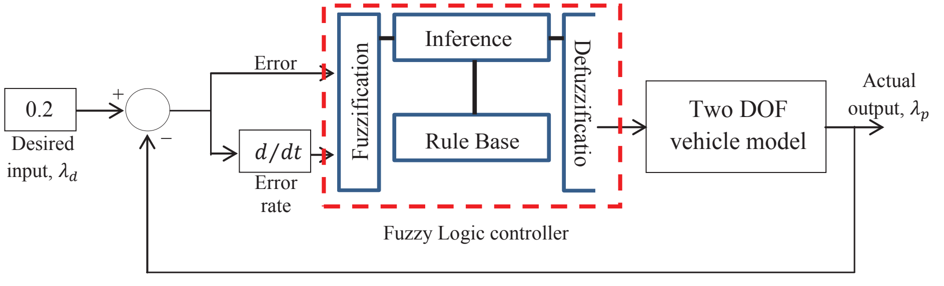

Figure 7. Fuzzy logic controller using Gaussian membership function for outer loop model

In the Figure 7, the inputs of the fuzzy controller i.e. slip error and slip error rate are defined during the initialization stage and all fuzzy membership functions are defined in order to form the complete rules. A Gaussian membership function is used and defined by

$$\mu = e^{-0.5(\frac{x-c}{\sigma})} \tag{29}$$

where
x is the Gaussian input from error and error rate
c is the center of the curve
σ is the spread of the curve

The fuzzy logic network model used a singleton fuzzification, Gaussian membership input using two variables such as centers, c_j^i, , and spreads, σ_j^i, output membership function uses center b_i, product for the implication and also center average defuzzification. The final equation representing the network model is

$$f(x|\theta) = \frac{\sum_{i=1}^{R} b_i \prod_{j=1}^{n} exp\,[-0.5\frac{x_j^i - c_j^i}{\sigma_j^i}]}{\sum_{i=1}^{R} \prod_{j=1}^{n} exp\,[-0.5\frac{x_j^i - c_j^i}{\sigma_i^i}]} \tag{30}$$

where x_j^i is input to the fuzzy network, j = 1, …, n (n is the number of inputs) and i = 1, …, R (R is the number of rules and it will be determined during the initialization stage and fixed during learning stage). The parameter b_i is the center of i th consequent fuzzy set and c_j^i and σ_j^i are center and spread of Gaussian antecedent membership functions at rule i and input j, respectively. In order to develop the Fuzzy Logic controller, slip error and slip rate error are chosen as the input variables. These variables reflect the fuzzy input sets for the closed loop model which is shown in Figures 8 and 9. The parameter of center, c and spread, σ are determined based on the range of maximum and minimum of the error and error rate values. The range is measured from the outer loop control structure using PID controller since this control structure is used as the benchmark to develop advance controllers. Meanwhile, the parameter b is determined based on the maximum and minimum brake torque required during braking a vehicle in order to avoid skidding disturbance.

In order to improve fuzzification speed, Gaussian function is proposed as the membership function (Ping et al., 2010). Figures 8 and 9 show the membership functions of input fuzzy variables and Figure 10 for the output fuzzy variable where the range of slip error is (-0.8, 0.01) and slip rate error is (-0.154, 0). Meanwhile, Figure 11 shows the range of output fuzzy variable is a singleton within -700 to 700 and the relationship between slip error, slip error rate and brake torque output in 3D surface.

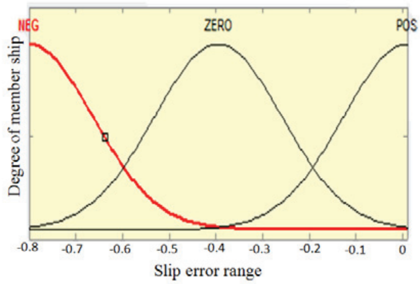

Figure 8. Slip error input variable

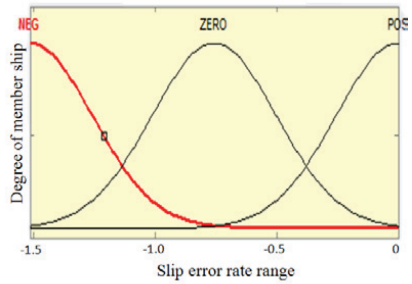

Figure 9. Slip error rate input variable

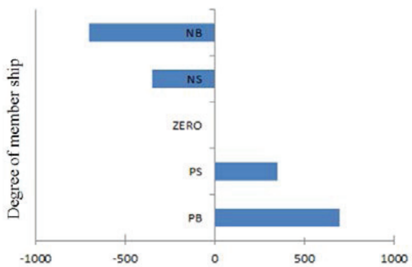

Figure 10. Brake torque output fuzzy variables

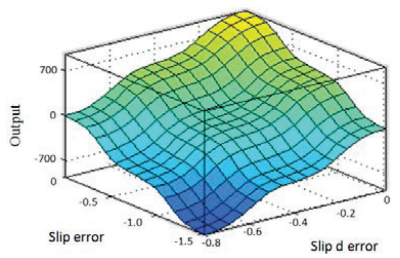

Figure 11. Output-input relation

The fuzzy sets for the input fuzzy variables are defined as positive (POS), zero (ZERO) and negative (NEG). The variables are maintained to initial condition since this inputs itself is sufficient to control the ABS longitudinal slip. Since there are 3 elements of each fuzzy set, the probability of 9 types of fuzzy rules has been used as base rule for this controller. The output fuzzy set representing brake torque is defined as positive big (PB), positive small (PS), zero (ZERO), negative small (NS) and negative big (NB).

The rules in Table 4 are derived basically using a combination of experience, trial and error approach and the knowledge of the response of the ABS system. In this study, ABS control system using PID controller is used as a benchmark to develop fuzzy logic rules. The error and error rate from ABS control system using PID controller is used to derive the combinations rules for ABS using fuzzy logic controller. Normally, the rules are determined using "if and then rules" condition. For an example, if slip error is positive and slip error rate is positive then the output is positive big. This means that the output signal is increasing and responding as overshoot response.

Table 4. Fuzzy rule base

Input variable		Slip error rate		
		POS	ZERO	NEG
Slip error	POS	PB (rule 9)	PS (rule 8)	ZERO (rule 7)
	ZERO	PS (rule 6)	ZERO (rule 5)	NS (rule 4)
	NEG	ZERO (rule 3)	NS (rule 2)	NB (rule 1)

4.4 Inner loop control design using EWB model

Inner loop control was developed for the electronic wedge brake (EWB) mechanism. From the previous section, the Bell-Shaped curve has been used as the model for the EWB actuator. Since the behaviour of the Bell-Shaped curve closely follows the actual behaviour of the EWB actuator, this mathematical equation is used as the plant model of the inner loop control. The desired trajectory of the inner loop control has been set up to 420 Nm based on brake torque of a passenger vehicle. Figure 12 shows the proposed control structure using EWB model.

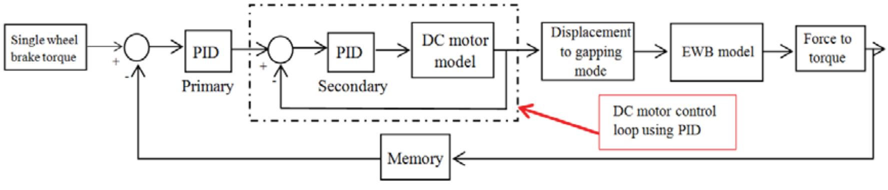

Figure 12. Proposed control structure using EWB model

Figure 12 shows the proposed control structure using EWB model to develop an inner loop controller. The reference input to the EWB model is the wheel brake torque. Meanwhile, PID controller is used to control the error of the inner loop control. Refering to Figure 12, there are two PID controllers used in this control structure. The primary PID controller is used to control brake torque obtained from the EWB model as described in Equation (14) and (15). The secondary PID controller is used specifically to command the DC motor to actuate the desired value from the primary PID controller (Aparow et al., 2013c).

The rotational displacement from the DC motor is converted to linear displacement (gapping mode). This will be the input to the Bell-Shaped curve which represents the EWB actuator. The EWB force will be converted to wheel brake torque and feedback to the comparator. The actual brake torque is compared with the desired brake torque by measuring the difference between actual and desired values in order to obtain minimum error. The performance of EWB model has been discussed in Aparow et al., (2013c).

4.5 Proposed ABS-EWB closed loop control structure

For an ABS-EWB closed loop scheme, the disturbance is measured in the form of wheel slip occurred after brake being applied and a corrective action is taken to maintain the wheel slip to 0.2. The outer loop model is developed using the quarter vehicle model as the plant to be controlled. Meanwhile, EWB model is used as the plant model for inner loop model. Since the outer closed loop is developed using a quarter vehicle brake model, only a single wheel brake torque is needed to be controlled by the controller. Besides, validated electronic wedge brake model is used as an inner loop control. The input for inner loop control is obtained from the controller of the outer loop control. Based on the proposed inner and outer loop controls, an ABS model is developed by integrating both inner loop and outer loop into a system. The desired trajectory used a constant slip value of 0.2. The response from inner loop control is used as brake torque for the plant model and the wheel slip is feedback and compared with the desired trajectory.

4.5.1 Closed Loop Control Design using Quarter Vehicle and EWB Model

The ABS control system using three type of controllers which are conventional PID, adaptive PID and fuzzy logic control are studied. For comparison, the adaptive PID and fuzzy logic controller are compared with both conventional PID controller and passive system. The performance of the controllers are evaluated by examining the vehicle braking performance namely vehicle and wheel velocity, stopping distance and vehicle longitudinal slip. Figures 13 and 14 show the closed loop control structure of the proposed ABS model using adaptive PID and fuzzy logic control.

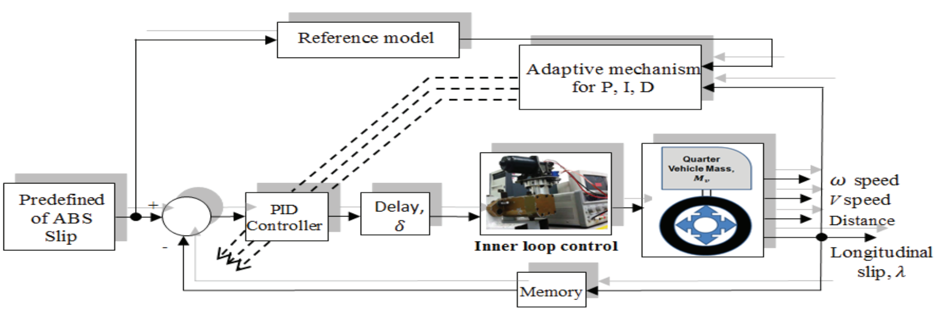

Figure 13. Control structure of ABS-EWB closed loop control using adaptive PID

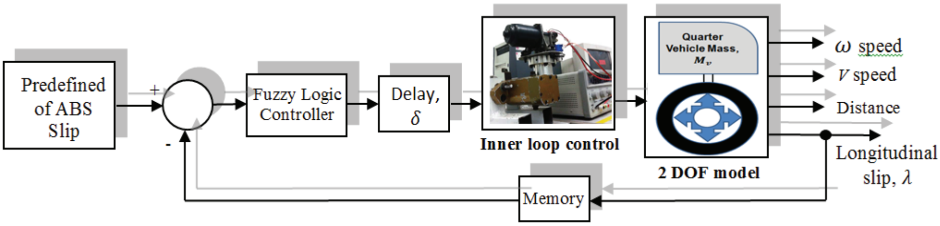

Figure 14. Control structure of ABS-EWB closed loop control using fuzzy logic

The actual response of longitudinal slip from the quarter vehicle model will be feedback to the comparator. The error is obtained from the difference between the predefined trajectory of ABS slip and actual longitudinal slip from the plant model. The error will be used as the input to the controller to optimize and reduce the error until the actual value will closely follow the reference value. According to the ABS concept, the wheel of the quarter vehicle should produce longitudinal slip in the range of 0.15 to 0.25. Wheel skidding during sudden braking can be reduced if the longitudinal wheel slip is maintained between the range of 0.15 to 0.25. Further, wheel speed and stopping distance of the vehicle are also determined.

5.0 SIMULATION RESULTS

The simulation results of the ABS closed loop control structure using PID, adaptive PID using model reference adaptive control (MRAC) technique, and fuzzy logic control using Gaussian membership function compared with conventional braking. This control system was developed based on the outer loop control using quarter vehicle model and inner loop control using electronic wedge brake (EWB) actuator model as shown in Figures 13 and 14. The control structure is analyzed using sudden braking test where the brake is applied when the velocity of the vehicle is 25 m/s and stopping distance, longitudinal slip and velocities of vehicle and wheel are observed. ABS is developed by controlling the brake torque obtained from the EWB actuator.

Brake torque from EWB model is used as input to the quarter vehicle model. Conventional PID controller is used as the benchmark for the adaptive PID and fuzzy logic to control the brake torque of a single wheel. The brake torque will control the longitudinal wheel slip to be in between 0.15 to 0.25. This will reduce the stopping distance of the vehicle and managed to halt wheel velocity without skidding. The results of the study are shown in Figures 15 to 18 which compare the vehicle and wheel velocities, longitudinal slip and the vehicle stopping distance. Figures 15 and 16 show the comparison of vehicle and wheel velocity of the quarter vehicle model using conventional PID, adaptive PID, fuzzy logic and conventional braking.

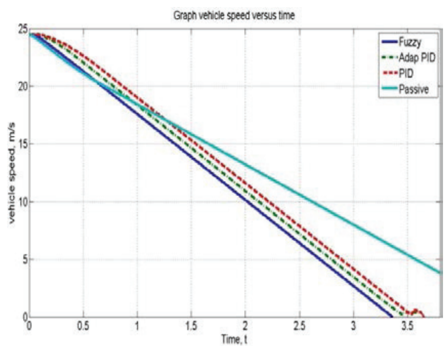

Figure 15. Comparison of vehicle velocity

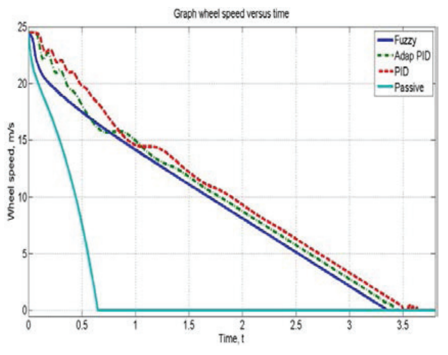

Figure 16. Comparison of wheel velocity

Conventional braking causes the wheel to lock up causing skidding after 0.6 seconds brake is applied. Meanwhile, ABS model using conventional PID, adaptive PID and fuzzy logic control the brake torque in order to avoid the wheel from locking during braking. ABS using conventional PID shows the wheel stops decelerate at 3.65 second and ABS using adaptive PID stops at 3.5 second. However, for ABS-EWB using fuzzy logic control, the vehicle is able to halt at 3.3 second. ABS-EWB using fuzzy logic control has improve the performance of the wheel velocity by reducing jerking motion occurred in a wheel during braking compared to the other two controllers. Skidding of the wheel has been reduced and thus improving the stopping distance of the vehicle.

Figure 17 shows the comparison of the response of longitudinal wheel slip during sudden braking. It can be seen that conventional braking results approaches +1 after 0.6 second which represents wheel locking during braking and the vehicle starts to skid until stops at 3.9 second. Three type of controllers are used to reduce address issue occurring in the wheel by controlling wheel longitudinal slip to 0.2. Conventional PID and adaptive PID controllers are used in this study to control longitudinal slip of wheel to be equal to 0.2 once brake is applied. The performance of both controllers is investigated and the responses show that ABS using PID controller reached settling time condition for longitudinal wheel slip within 2.0 second with minor jerk before halt. Meanwhile, ABS using adaptive PID controller is also able to reach settling time condition within 1.5 second with jerking slighly better than previous controller.

Jerking condition as shown in Figure 17 is occurred because of overshoot response in wheel longitudinal slip before approaching steady state conditions.Generally, adaptive PID works better than conventional PID because of the parameter of PID will change by the adjustment mechanism until they converge to optimal values. However, the parameter adjustment for adaptive PID will take some time and this has been effected the simulation to slow down the longitudinal slip response to reach 0.2 in shortest time. Even tough the result shows overshoot response using both conventional PID and adaptive PID controllers, the system still unable to produce fast response to reach steady-state condition. Hence, by using fuzzy logic as ABS controller the wheel stopping capability has improved drastically. This system is able to reach steady-state condition in 0.58 second even tough the wheel longitudinal slip shows overdamped behaviour. Additionally, this system is also able to avoid jerking which occurred for adaptive PID at 0.6 second to 1.5 second and 0.72 second to 1.8 second for conventional PID.

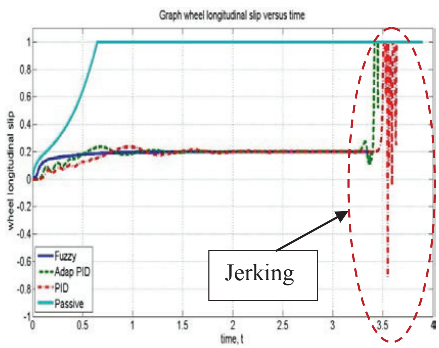

Figure 17. Comparison of longitudinal slip

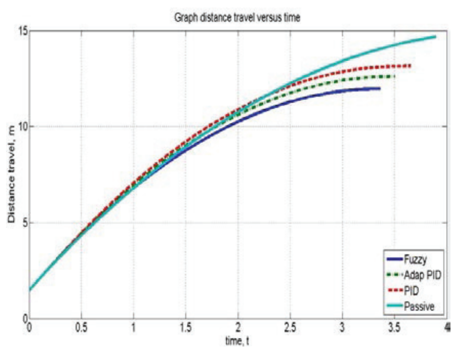

Figure 18. Comparison of vehicle distance travel

Figure 18 shows the distance travelled by the vehicle after the brake input is applied. The figure shows fuzzy logic controller is able to reduce the distance travelled by the vehicle compared to other controllers and conventional braking system. The distance travel by the vehicle controlled by fuzzy logic control is about 12.3 meter and the vehicle stops 3.3 second after braking. Meanwhile, vehicle with adaptive PID and conventional PID controllers are able to stop 3.5 seconds with the distance travelled of 13.8 meter and 3.65 seconds with the distance travelled of 14.9 meter respectively. By controling the wheel longitudinal slip, the distance travel bu the vehicle has been improved significantly compared to conventional braking system. At the same time, fuzzy logic controller is shown to halt the vehicle with the shortest distance compared to other controllers.

Figure 19 shows the comparison of vehicle velocity and wheel velocity of a vehicle using fuzzy logic controller. The ABS-EWB model using fuzzy logic controller shows a good performance by stopping the vehicle and wheel at the same time without any jerking problems in the wheel. Besides, the stopping distance has been reduced compared to ABS model using conventional PID and adaptive PID. The proposed controller shows better performance in controlling the vehicle and EWB brake model to produce a good ABS model.

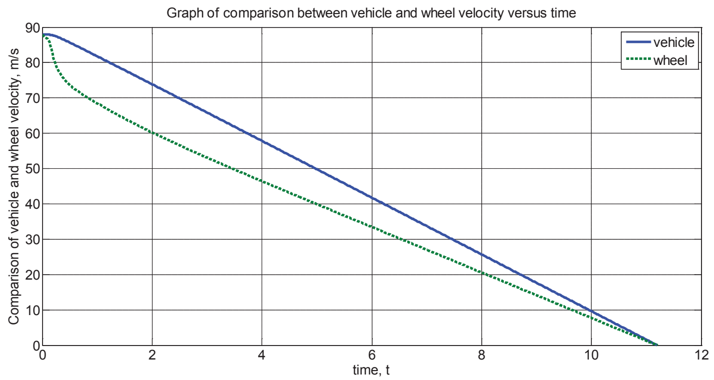

Figure 19. Comparison between wheel and vehicle velocity for ABS using fuzzy logic

6.0 CONCLUSION

A comprehensive study of the ABS model using Electronic wedge brake (EWB) mechanism and the design of the conventional PID, adaptive PID and fuzzy logic controller have been presented. A quarter vehicle model is modeled and validated using CarSim simulator. A mathematical model using Bell-Shaped is used to represent an EWB. This model is used to represent the actual EWB behaivour. An outer loop model has been proposed using validated quarter vehicle model and EWB model is used to develop an inner loop model. Both outer and inner loop are integrated in order to develop the overall ABS closed system. The simulation studies considered three types of controllers for ABS model. Comparison has been made for the proposed ABS model to investigate the performance of the proposed controllers. From the simulation results, the Fuzzy Logic controller shows good performance for ABS-EWB control structure by reducing the stopping distance up to 17.4 % compared to the conventional PID and Adaptive PID control is 7.38% and 12.08 %. The proposed control scheme is able to improve the stopping distance and wheel longitudinal slip of vehicle during braking. Hence, it can be concluded that fuzzy logic controller could provide better active brake performance than the adaptive PID and conventional PID control.

ACKNOWLEDGEMENT

This work is supported by the University Teknikal Malaysia Melaka (UTeM) through short term grant project entitled "Safety and Stability Enhancement of Automotive Vehicle using ABS with Electronic Wedge Brake Mechanism" led by Fauzi Bin Ahmad at the Universiti Teknikal Malaysia Melaka. This financial support is gratefully acknowledged.

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