

Performance of a Road Vehicle with Hydraulic Brake Systems Using Slip Control Strategy

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Abstract The capability of a road vehicle equipped with an anti-lock braking system (ABS) comes to a safe stop depends on factors such as dynamic force between tire and road surface adhesion coefficient, and the vertical profile of the road. When in panic, a driver's reaction is to step hard on the brakes to make the vehicle stop as soon as possible. Although the use of modern technologies such as ABS has reduced the number of accidents significantly, any further improvement in stopping distance would only complement these technologies. Mathematical simulation of an ABS has been implemented in Matlab, which employs a quarter car vehicle's model undergoing a straight line braking maneuver. The model also incorporates a hydraulic brake valve dynamics and road-tire interaction. The road-tire interaction model is given in the form of an empirical function (Magic formula) describing the nonlinear relation between adhesion (rolling) coefficient and wheel slip. A Bang-Bang controller has been implemented with the above model for controlling wheel slip at given desired reference value. The braking performances in both assisted ABS mode and non-ABS mode have been evaluated by simulation. Simulated results of stopping distances were confirmed using a road test setup. The results indicate that the braking performance of automotive assisted ABS was improved significantly, the braking time advanced, and the stopping distance shortened consequently, the braking safety of vehicle can be improved.

Keywords: Bang-Bang controller, Anti-locked Braking System, straight line braking maneuver, hydraulic brake actuator dynamics, road-tire friction, wheel slip

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1. Introduction

Antilock braking system (ABS) exhibits strongly nonlinear and uncertain characteristics. A grey sliding mode controller was proposed to track the reference wheel slip had been employed to overcome these difficulties. The concept of grey system theory, which has a certain prediction capability, offers an alternative approach to conventional control methods. The proposed controller anticipates the upcoming values of wheel slip, and takes the necessary action to keep wheel slip at the desired value. The control algorithm was applied to a quarter vehicle model, and it was verified through simulations indicating fast convergence and good performance of the designed controller. Simulated results were validated on real time applications using a laboratory experimental setup [1].

The braking performances in both ABS mode & non-ABS mode have been evaluated by simulation. A model based nonlinear observer had been implemented in Simulink. The mathematical model of an observer based emergency braking controller had been also implemented in Simulink. The nonlinear observer successfully estimate states like vehicle absolute velocity, road surface varying parameters and underestimation of longitudinal slip, with only wheel angular velocity information. The emergency

controller utilizes those estimated states for a feedback control law, achieves a near maximum deceleration. A Sliding mode controlled ABS had been implemented. Using Lyapunov theorem, the present worker had derived that the stability requirement for sliding mode controller as given in the reference was valid. Performance of the implemented sliding mode controller was compared to the Bang-Bang controller as far as regulation of wheel slip & vehicle stopping distance was concerned. The Bang-Bang controlled and the sliding mode controlled ABS were again evaluated with Lumped Lugre friction model and Burckhardt friction model [2].

In modern cars, braking systems are used to prevent the wheels from locking after brakes are applied. The dynamics of the controller needed for antilock braking system depends on various factors. The vehicle model often is in nonlinear form. Controller needs to provide a controlled torque necessary to maintain optimum value of the wheel slip ratio. The slip ratio was represented in terms of vehicle speed and wheel rotation. All system dynamic equations were explained and a slip ratio was expressed in terms of system variables namely vehicle linear velocity and angular velocity of the wheel. By applying a bias braking force system, response was obtained using Simulink models. Using the linear control strategies like PI-type, the effectiveness of maintaining desired slip ratio was tested. It was always observed that a

steady state error of 10% occurring in all the control system models [3].

The preliminary research and implementation of an experimental test bench set up for an electric vehicle Antilock Braking System (ABS)/Traction Control System (TCS) representing the dry, wet and icy road surfaces was described. A Fuzzy logic based controller to control the wheel slip for electric vehicle antilock braking system is presented. The test facility comprised of an induction machine load operating in the generating region. The test facility was used to simulate a variety of tire/road driving conditions, eliminating the initial requirement for skid-pan trials when developing algorithms. Simulation studies and results were provided [4].

An active safety device maximizes the braking force between the vehicle tyre and the road irrespective of the road conditions. This device is called anti-lock braking system (ABS). This is accomplished by regulating the wheel slip around its optimum value. Due to the high non-linearity of the tyre and road interaction, and uncertainties from vehicle dynamics, a standard PID controller will not suffice. A proposed nonlinear control model was designed to use input-output feedback linearization approach. To enhance the robustness of the non-linear controller, an integral feedback method was employed. The stability of the controller was analyzed in the Lyapunov sense. To demonstrate the robustness of the proposed controller, simulations were conducted on two different road conditions. The results from the proposed method exhibited a more superior performance and reduced the chattering effect on the braking torque compared to the performance of the standard feedback linearization method [5].

Many different control methods for ABS systems had been developed. These methods differ in their theoretical basis and performance under the changes of road conditions. The review was a part of research project entitled "Intelligent Antilock Brake System Design for Road-Surfaces of Saudi Arabia". In this work, the methods used in the design of ABS systems were reviewed. The main difficulties and summary of the more recent developments in their control techniques were highlighted. Intelligent control systems like fuzzy control can be used in ABS control to emulate the qualitative aspects of human knowledge with several advantages such

as robustness, universal approximation theorem and rule-based algorithms [6].

Several problems are existed in the control of brake systems including the development of control logic for antilock braking systems (ABS) and "base-braking." The base-braking control problem was studied where a controller was developed that can ensure the braking torque commanded by the driver will be achieved. In particular, a "fuzzy model reference learning controller," a "genetic model reference adaptive controller," and a "general genetic adaptive controller," were developed and their ability to reduce the effects of variations in the process due to temperature was investigated [7].

The above review has shown that almost the efforts which had been done in this subject was directed towards the studying of the road vehicle brake performance and its control. Their contributions were limited due to the consideration of the theoretical simulations are the only concept used and ignoring the experimental simulations, particularly that concerned with the field tests. In some cases the experimental simulation has carried out within the laboratories which have also given little attention. However, the objective of this study is to establish mathematical simulation of vehicle brake with ABS assisted has been implemented in Matlab, which employs a quarter car's model undergoing a straight line braking maneuver. The simulation also incorporates a hydraulic brake valve dynamics and road/tire friction. The road/tire friction model is given in the form of an empirical function (Magic formula) describing the nonlinear relation between adhesion coefficient and wheel slip. A Bang-Bang controller has been implemented with the above model for controlling wheel slip at given desired reference value.

2. Principles of Antilock-Brake System

Typical ABS components include: vehicle's physical brakes, wheel speed sensors (up to 4), an electronic control unit (ECU), brake master cylinder, a hydraulic modulator unit with pump and valves as shown in Figure 1. The reason for the development of antilock brakes is in essence very simple. Under braking, if one or more of a vehicle's wheels lock (begins to skid) then this has a number of consequences:

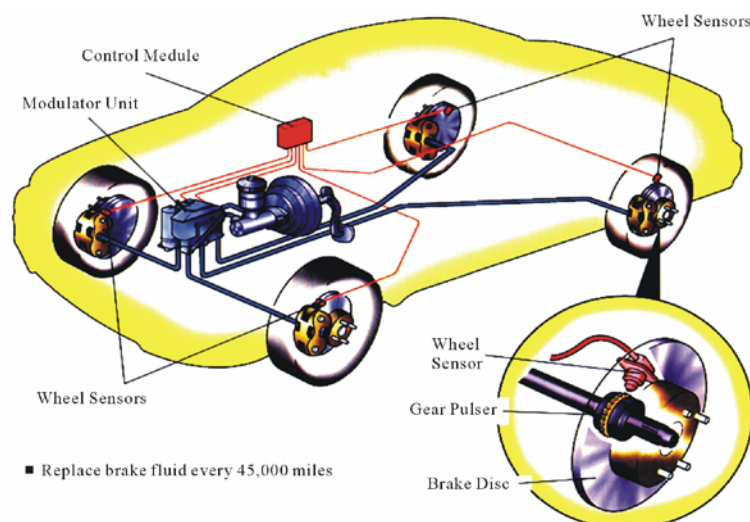


Figure 1. Typical ABS components [8]

- a) Braking distance increases, and
- b) Steering control is lost, and c) tire wear will be abnormal. The obvious consequence is that an accident is far more likely to occur.

The application of brakes generates a force that impedes a vehicles motion by applying a force in the opposite direction [8].

During severe braking scenarios, a point is obtained in which the tangential velocity of the tire surface and the velocity on road surface are not the same such that an optimal slip which corresponds to the maximum friction is obtained. The ABS controller must deal with the brake dynamics and the wheel dynamics as a whole plant. The wheel slip, S is defined as:

$$S = \frac{V - \omega R}{V} \quad (1)$$

where ω , R , and V denote the wheel angular velocity, the wheel rolling radius, and the vehicle forward velocity, respectively. In normal driving conditions, $V = \omega R$, therefore $S = 0$. In severe braking, it is common to have $\omega = 0$ while $S = 1$, which is called wheel lockup. Wheel

lockup is undesirable since it prolongs the stopping distance and causes the loss of direction control.

Figure 2 shows the relationship between braking coefficient and wheel slip. It is shown that the slide values for stopping/traction force are proportionately higher than the slide values for cornering/steering force. A locked-up wheel provides low road handling force and minimal steering force. Consequently the main benefit from ABS operation is to maintain directional control of the vehicle during heavy braking. In rare circumstances the stopping distance may be increased however, the directional control of the vehicle is substantially greater than if the wheels are locked up.

The main difficulty in the design of ABS assisted control arises from the strong nonlinearity and uncertainty of the problem. It is difficult and in many cases impossible to solve this problem by using classical linear, frequency domain methods [9]. ABS systems are designed around system hydraulics, sensors and control electronics. These systems are dependent on each other and the different system components are interchangeable with minor changes in the controller software.

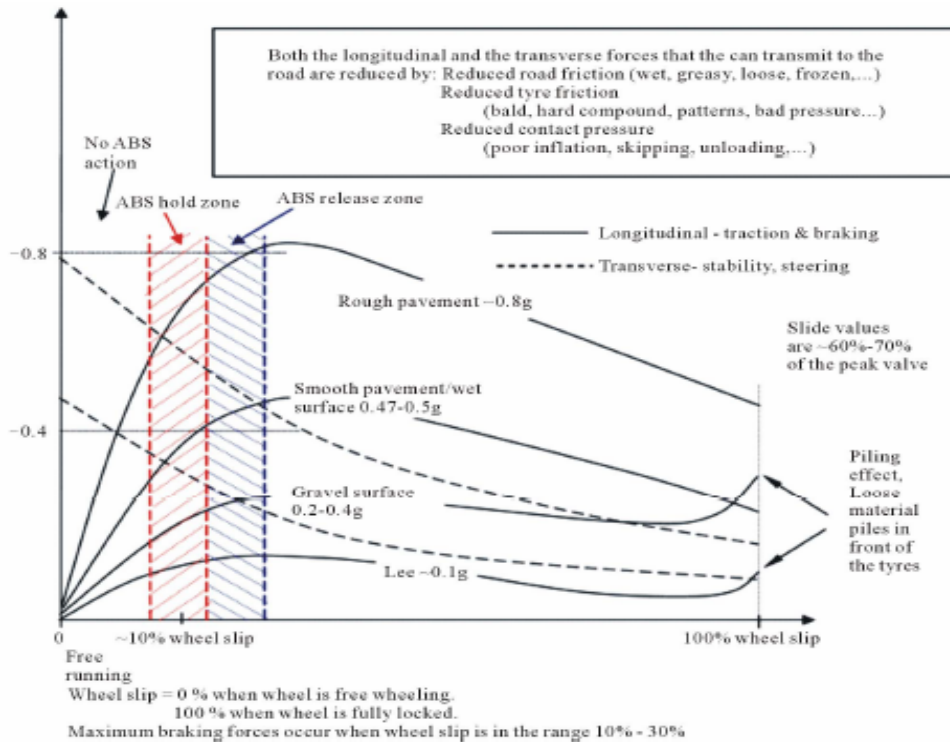


Figure 2. Illustration of the relationship between braking coefficient and wheel slip [10]

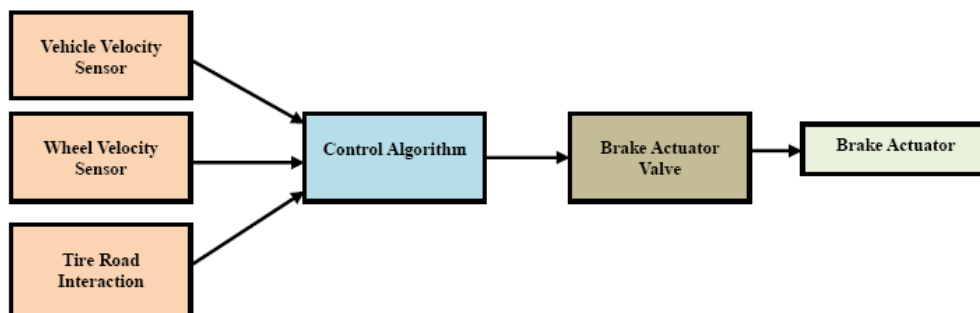


Figure 3. Block representation of an ABS

The wheel sensor feeds the wheel spin velocity to the electronic control unit, which based on some underlying control approach would give an output signal to the brake actuator control unit. The brake actuator control unit then controls the brake actuator based on the output from the electronic control unit. The control logic is based on the objective to keep the wheels from getting locked up and to maintain the traction between the tire and road surface at an optimal maximum. The task of keeping the wheels operating at maximum traction is complicated given that the friction-slip curve changes with vehicle, tire and road changes. The block diagram in Figure 3 shows the block representation of an antilock brake system. It shows the basic functionality of the various components in ABS systems and also shows the data/information flow. The ABS (shown in Figure 4) consists of a conventional hydraulic brake system plus antilock components [10].

The conventional brake system includes a vacuum booster, master cylinder, front disc brakes, rear drum brakes, interconnecting hydraulic brake pipes and hoses, brake fluid level sensor and the brake indicator. The ABS components include a hydraulic unit, an electronic brake control module (EBCM), two system fuses, four wheel speed sensors (one at each wheel), interconnecting wiring, the ABS indicator, and the rear drum brake. Most ABS systems employ hydraulic valve control to regulate the brake pressure during the anti-lock operation. Brake pressure is increased, decreased or held. The amount of time required to open, close or hold the hydraulic valve is the key point affecting the brake efficiency and steering controllability.

3. ABS Control

ABS brake controllers pose unique challenges to the designer:

- For optimal performance, the controller must operate at an unstable equilibrium point
- Depending on road conditions, the maximum braking torque may vary over a wide range
- The tire slippage measurement signal, crucial for controller performance, is both highly uncertain and noisy
- On rough roads, the tire slip ratio varies widely and rapidly due to tire bouncing
- Brake pad coefficient of friction changes, and
- The braking system contains transportation delays which limit the control system bandwidth [10].

As stated in the previous section of this paper, the ABS consists of a conventional hydraulic brake system plus antilock components which affect the control characteristics of the ABS. ABS control is a highly a nonlinear control problem due to the complicated relationship between friction and slip. Another impediment in this control problem is that the linear velocity of the wheel is not directly measurable and it has to be estimated. Friction between the road and tire is also not readily measurable or might need complicated sensors. Researchers have employed various control approaches to tackle this problem. A sampling of the research done for different control approaches is shown in Figure 4. One of the technologies that have been applied in the various aspects of ABS control is soft computing. Brief review of ideas of soft computing and how they are employed in ABS control are given below.

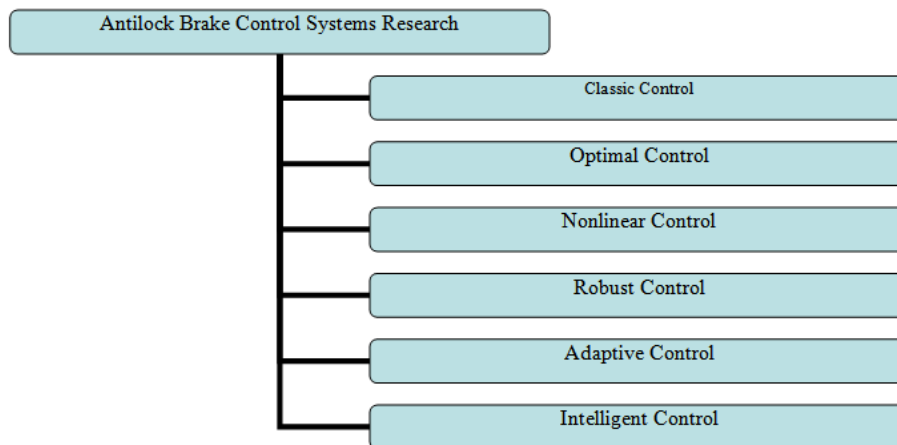


Figure 4. Sampling of ABS control

4. Wheel Slip Control of ABS with Bang-Bang Controller

4.1. Introduction

Mathematical modeling is the first and most crucial task in developing a control algorithm for the antilock braking system. However, modeling an antilock braking system is really a difficult task, considering the ABS dynamics being highly nonlinear and time varying. However, in this study, a simplified model for controller design and computer simulation is used. Towards the above goal, the

mathematical model of an ABS has been implemented in Matlab, which employs a quarter car vehicle's model undergoing a straight line braking maneuver. The model also incorporates a hydraulic brake actuator dynamics and road-tire friction. The road-tire friction model is given in the form of the Magic formula describing the nonlinear relation between adhesion coefficient and wheel slip. A Bang-Bang controller has been implemented with the above model for controlling wheel slip at given desired reference value. The braking performances in both in ABS assisted mode & non-ABS mode have been evaluated by simulation [11].

4.2. Vehicle Dynamics

In this study a simplified quarter car vehicle model undergoing perfectly straight line braking maneuver has been considered. Thus there is no lateral tire force and also yaw do not exist. Furthermore, the following assumptions are considered in the modeling process

- There is no steering input.
- Only longitudinal vehicle motion has been considered.
- The sprung mass is assumed to be connected to unsprung mass with a rigid body (no damping effect)
- Approximating the vertical forces as a static value.

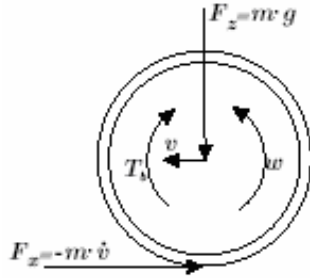


Figure 5. Quarter car or Single wheel model [12]

The equations of motion are given by:

$$V = -F_x / m \quad (2)$$

$$\dot{\omega} = (rF_x - T_b) / J \quad (3)$$

The aerodynamic drag force:

$$F_d = C_d \cdot A_f \cdot V^2 / 2 \quad (4)$$

The tire rolling resistance force:

$$F_r = f_r \cdot F_z$$

For radial-ply passenger car tires under rated loads and inflation pressures, the relationship between rolling resistance coefficient f_r , and vehicle speed V (up to 150 Km/h) the rolling resistance force may be expressed by

$$F_r = (0.0136 + 0.4 \cdot 10^{-7} \cdot V^2) F_z \quad (5)$$

V is in km/hr

The tire friction force is given by the magic formula:

Longitudinal force for pure slip, F_x , consists of coefficients B, C, D, E and S_v . The subscript x represents condition along x -axis. Slip ratio, κ , is the input of F_x as given [13]

$$F_x = D_x \sin \left[C_x \arctan \left\{ \begin{array}{l} B_x \kappa_x \\ -E_x \left(\begin{array}{l} B_x \kappa_x \\ -\arctan(B_x \kappa_x) \end{array} \right) \end{array} \right\} \right] + S_{V_y} \quad (6)$$

More details about Magic formula and their coefficient can be found Appendix.

4.3. Hydraulic Dynamics

For a conventional Anti-lock braking hydraulic system, a proportional directional control valve is used here as a reference system to control the ABS hydraulic unit, In the ABS the hydraulic system pressure is increased or decreased depending on the control valve driving, this valve is modeled as a first order differential equation as:

$$P + \frac{1}{t_d} = \frac{1}{t_d} \cdot P_d \quad (7)$$

Neglecting the effect of system nonlinearities and temperature, the brake torque T_b can be formulated as linear function of the brake pressure equation.

The ABS control function is designed based on the slip control strategy, in this control strategy, the brake pressure is triggered to maintain the longitudinal wheel slip at the desired values that gives a peak braking force on a wide variety of road surface. The controller uses input data stemming from the existing sensors of the ABS. Such input variables deceleration and speed of the car, deceleration and speed of the wheels, and hydraulic pressure of the brake fluid. These variables indirectly indicate the current operation point of the braking and its behavior over time, the slip can easily be evaluated by ration of vehicle and wheel speeds, the calculated slip is then evaluated and is then compared with the value of the desired slip to produce an error signal, the controller (Bang-Bang) returns (+1) if the value is greater than zero and hence the pressure is increased, returns (0) if the error equal zero and hence the pressure remains unchanged, and returns (-1) if it is less than zero and hence the pressure is reduced.

4.4. Bang-Bang Control Law

In this model, an ideal anti-lock braking controller was used that uses 'bang-bang' control based upon the error between actual slip and desired slip as presented in Matlab. The desired slip was set to the value of slip at which the μ -slip curve reaches a peak value, this being the optimum value for minimum braking distance. The control input, namely the brake torque T_b , is switched between the maximum value, T_{bmax} , and the minimum value, 0, so as to keep the slip operating in the desired region. From the figure below taking $\lambda_d = 0.2$. Also $k=100$; $T=0.01$ [2].

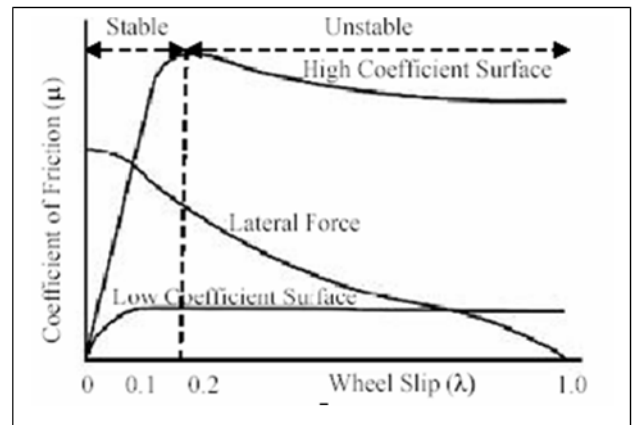


Figure 6. μ - λ pseudo static curve [11]

4.5. Braking Control Procedure

Consider the Vehicle is in braking, the slip ration (%) and the error signal can be given by:

$$S = \frac{V - V_w}{V} = \frac{V - \omega R}{V}; Error(e) = S_d - S \quad (8)$$

Braking torque from the brake aligning (T_b):

$$T_b = A_o.n.R_d.P \quad (9)$$

Desired braking torque depending on error signal:

$$T_b = A_o.n.R_d.Z_k \quad (10)$$

Z_k : the desired pressure at time (t), depending on the error signal by the control, which can be calculated based on the following set of equation:

$$Z_k(t+t_s) = e^{-\frac{t_s}{t_d}} Z_k(t) - (e^{-\frac{t_s}{t_d}} - 1) P_f(t) \quad (11)$$

P_f is obtained as a solution to:

$$P_f(t+t_s) = P_f(t) + S_f.Z_k(t) \quad (12)$$

In this case the S_f is a function which simulates the Bang-Bang controller [sign (e)] which returns (-1) for negative error, (0) for zero error, (+1) for positive error.

Transfer Function and Hydraulic Valve time delay: The transfer function produces the rate of change of pressure, which means the pressure demand of the system in a certain time, according to the signal produced by the bang-bang controller {i.e. sign () }, considering the valve time delay, until the valve is fully opened. Table 1 tabulates the parameters used in modeling and in experiments.

Table 1. System parameters and numerical values used in modeling and in experiments

No.	Parameter	Symbol	Value	Unit
1	Quarter vehicle mass	m	447.5	Kg
2	Acceleration due to gravity	g	9.81	m/s ²
3	Radius of wheel	r	0.308	m
4	Moment of inertia	J	1.7	Kg.m ²
5	Initial vehicle speed	V	11, 14, 17	m/s
6	Number of friction surfaces	n	2	-
7	Effective brake radius	R _d	0.112	m
8	Tyre radius	R _w	0.308	m
9	piston radius	r	0.0185	m
10	Drag coefficient	C _d	0.539	-
11	Projected Area	A _f	2.04	m ²
12	Max pressure	P	20.10 ⁶	N/m ²
13	Disk brake coefficient of friction	μ	0.35	%
14	Effective Disc Radius	R _d	0.112	m
15	Valve time delay	v _{td}	0.1	s

5. Experimental Methodology

5.1. Introduction



Figure 7. Vehicle used for testing



Figure 8. Road surface condition

Braking road tests were performed on Peugeot 406 vehicle is shown in Figure 7, while Figure 8 shows the of road surfaces used. The condition of road surface is dry asphalt. The tests were carried out to measure the stopping braking distance (SD) when the vehicle was braked at vehicle speed of 40, 50 and 60 km/hr to reach 0.0 km/hr, the vehicle was equipped by either assisted ABS or non-ABS system.

4.2. Instrumentation and Data Logging

The instrumentation used on the test vehicle is illustrated in Figure 9 and Figure 10. The G-tech Pro RR is a small device (smaller than a packet of cigarettes) that attaches to the windscreen using rubber cups and plugs into the cigarette lighter. With no other connections this little gadget can give 0-60, 0.4 km time and speed, 60-0 braking in feet, horsepower at the wheels and G-force measurements. The G-tech uses an accelerometer, a timer and a computer to calculate the above states. The accelerometer measures the G-force and the computer samples this 400 times a second and calculates the distance covered and the traveling speed. There is one multi-function switch/button on the device that is used to select the desired operating mode. All tests have to start from a stationary position as it is the only reference point available to the unit.



Figure 9. The G-tech Pro RR Performance Meter



Figure 10. The G-tech Pro RR with its computer Meter

4.3. Testing Procedure

In each test, the driver took the vehicle out of gear a short time before the vehicle brakes were engaged, the driver initiated the data-logger, so that a short period of coasting could be recorded, and then the braking solenoid was activated. A dry asphalt road conditions with rolling resistance coefficient (f_r) of 0.018 Three repeat tests were completed at nominal set brake pressures of 3 bar, 5 bar, and 8 bar on each of these surfaces. It was at first desired that the tests would be conducted from an initial speed of 11 m/s (40 km/h). This was so that the vehicle brakes would dissipate approximately the same amount of energy as they would under normal operation, with all the brakes operating, when stopping from 17 m/s (60 km/hr).

6. Results and Discussions

6.1. Theoretical Results

In order to evaluate the performance of the proposed controller, three performance indices are adopted. These are:

- the integral squared error [ISE] of the slip
- the integral squared control input

- the stopping distance

The desired performance will therefore be: small variations from the desired slip and less effective braking torque, to achieve a shorter stopping distance. Simulations are conducted on a straight-line braking operation, braking commenced at an initial longitudinal velocity of 11 (40 km/hr), 14 (50 km/hr) and 17 m/s (60 km/hr), and the braking torque was limited to 1200 Nm. The system parameters and numerical values used in simulation and in experiments are presented in Table 1. On each axle a brake-valve controller controls the brake pressure to the minimum of either the brake demand from the driver or the pressure set by the wheel-slip controller. Each wheel-slip controller is in turn controlled by the global controller. The multiple-input, multiple-output control problem is further complicated by the saturation of the brake pressure actuation at 0 and 2 bar and by the non-linear tyre characteristics. In order to speed up the simulation runs when assessing the improvement of wheel-slip control, the pressure demand was assumed to be a step input. This does not directly affect the braking algorithm. Figure 11 and Figure 12 depict samples from the results computed for the vehicle and wheel speeds, slip %, brake pressure and brake force with respect of time respectively.

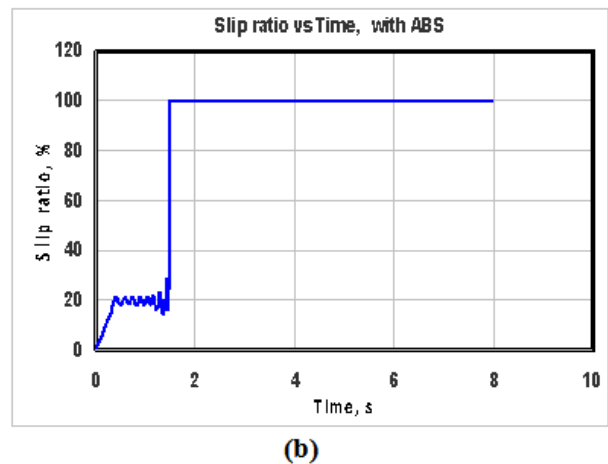
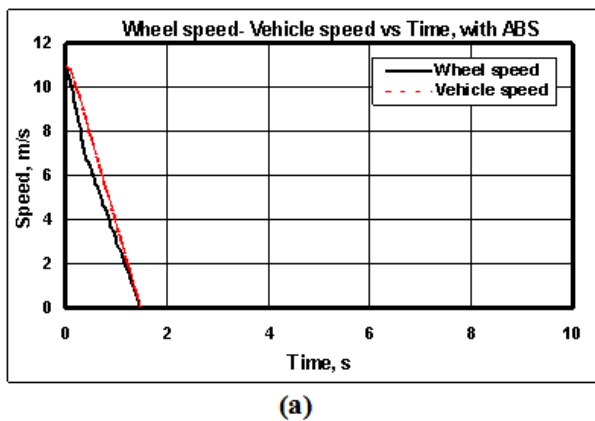


Figure 11. (a): Simulation results of a vehicle and wheel speeds for ABS at dry asphalt road condition and $V = 40$ km/hr (11 m/s); (b): Simulation results of slip ratio - VS, time for ABS at dry asphalt road condition and $V = 40$ km/hr (11 m/s)

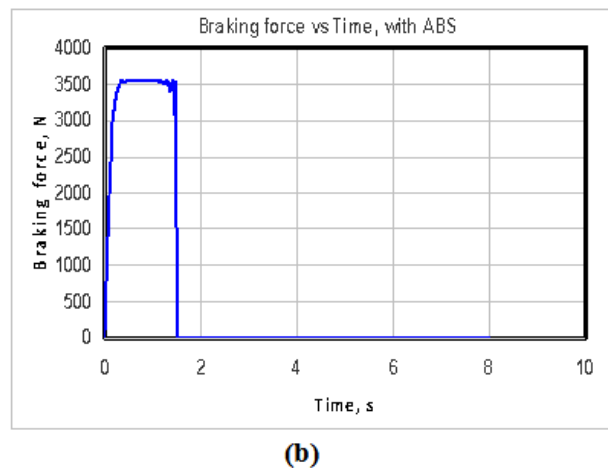
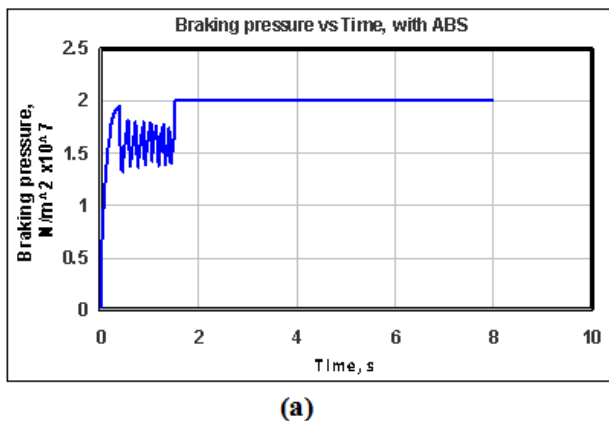


Figure 12. (a): Simulation results of brake pressure VS, time for ABS at dry asphalt road condition and $V = 40$ km/hr (11 m/s); (b): Simulation results of brake force VS, time for ABS at dry asphalt road condition and $V = 40$ km/hr (11 m/s)

Samples from the simulation results for ABS assisted, non-ABS, wheel and vehicle are shown in Figure 13 and

Figure 14 respectively at vehicle speed of 50 km/hr (14.5 m/s), where the relationship between both vehicle and

wheel speeds; and stopping distance are existed. The values of used parameters are given in Table 1. Based on Figure 13(a) and Figure 13(b), the data were collected and presented in Table 2 which indicate that the Bang-Bang controller parameter in ABS assisted is feasible and improved the vehicle stopping distance, consequently, can enhance vehicle braking performance. Based on Figure 14(a) and Figure 14(b), the influence of ABS and non-ABS on the wheel and vehicle stopping distance is shown respectively. Figure 13(a) demonstrates that an increase in slip, s , can increase the tractive force between the tire and road surface by virtue of an increase in μ . However, once the peak (μ_{max}) of the characteristic is encountered, any further increase in slip will reduce traction, and consequently induce an unstable acceleration of the wheel until the drive torque is reduced. The objective of an antilock braking system (ABS) is to manipulate the tractive force applied to the driven wheels in order to limit the slip, s , between the road surface and the tire, and consequently only operate within the stable control region of the μ -slip characteristic. This can be evaluated by letting ABS-assisted stopping distance to be compared to

those simulated with the non-ABS. To facilitate this comparison, the following equation was used:

$$ABS - (SDI) = \frac{SD_{non-ABS} - SD_{ABS}}{SD_{non-ABS}} \times 100\% \quad (13)$$

$SD_{non-ABS}$ = non-ABS stopping distance

SD_{ABS} = ABS assisted stopping distance.

Table 2. Simulated values of stopping distance when the vehicle equipped by ABS and non-ABS

a) Vehicle

No.	Vehicle speed, km/hr	Stopping distance, m		ABS Stopping distance improvement, %
		ABS	Non-ABS	
1	40	2.7	3.4	21
2	50	4.2	5.4	22
3	60	5.9	7.7	23

b) Wheel

No.	Vehicle speed, km/hr	Stopping distance, m		ABS Stopping distance improvement, %
		ABS	Non-ABS	
1	40	2.7	1.8	-50.0
2	50	4.2	2.7	-55.5
3	60	5.9	3.8	-58.0

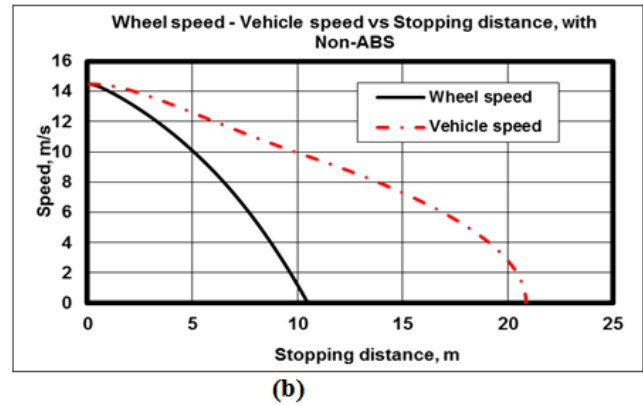
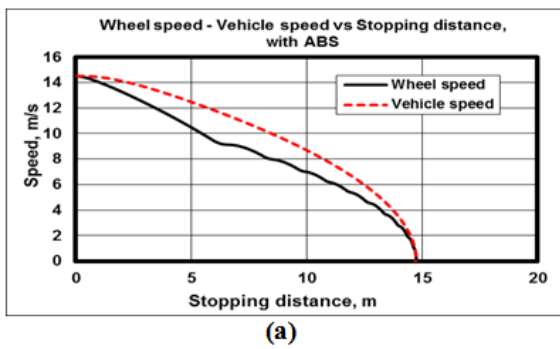


Figure 13. (a): Simulation results of wheel and vehicle speeds VS, stopping distance for ABS assisted, at wheel speed of $V=14.5$ m/s and dry asphalt; **(b):** Simulation results of wheel and vehicle speeds VS, stopping distance for non-ABS at wheel speed of $V=14.5$ m/s and dry asphalt

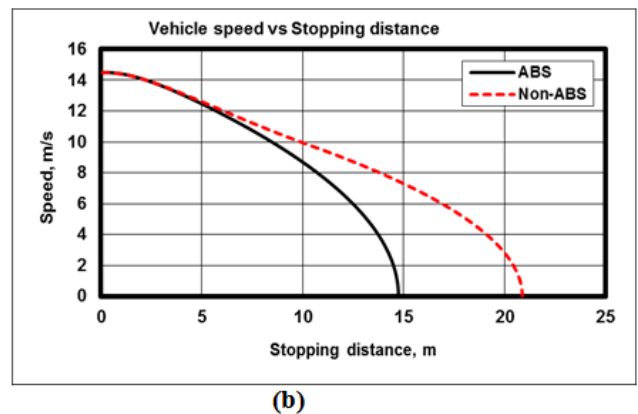
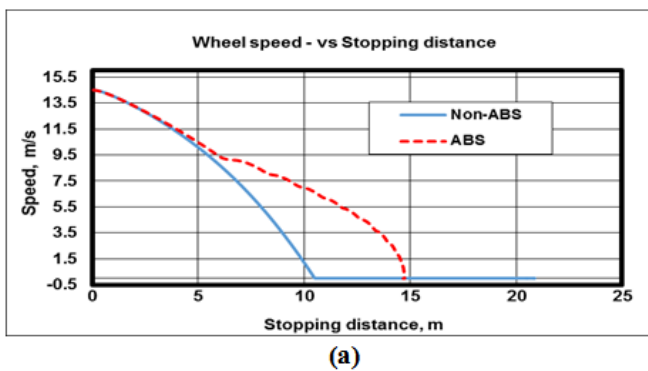


Figure 14. (a): Simulation results of wheel speeds VS, stopping distance at vehicle speed of $V=14.5$ m/s and dry asphalt; **(b):** Simulation results of vehicle speeds VS, stopping distance at vehicle speed of $V=14.5$ m/s and dry asphalt

Based on equation (13), all stopping distances reported in this simulation reflect the shortest distance observed as depicted in Figure 15 and tabulated in Table 2. The calculated results indicate that the vehicle in ABS assisted improve the stopping distance for either vehicle or wheel. This makes braking process safe and more controllable.

Simulations are conducted for the types of different road condition are tabulated in Table 3. The simulations

are terminated at speeds of about 4 km/h . This is because as the speed of the wheel approaches zero, the slip becomes unstable, therefore the ABS should disengage at low speeds to allow the vehicle to come to a stop. Figure 16 shows a sample of the relationship between the wheel slip and braking force for different frictional road surface conditions of dry asphalt, wet asphalt, compact snow (ice) and concrete of ABS technology to a preliminary wheel-

slip controller. The wheel-slip controller was tuned using a trial and error technique for choosing the gains for the controllers. The results indicate that wheel-slip control has the potential to reduce the braking. Moreover, if the road condition is getting harder the slip % is increased. Moreover, the simulation results also show that the developed control algorithm works good for all the road conditions except compact snow (ice) type surface. Figure 17 shows a sample of the simulation results for the relationship between both vehicle and wheel speeds; and stopping distance at dry asphalt road condition and vehicle speed of 40 km/hr.

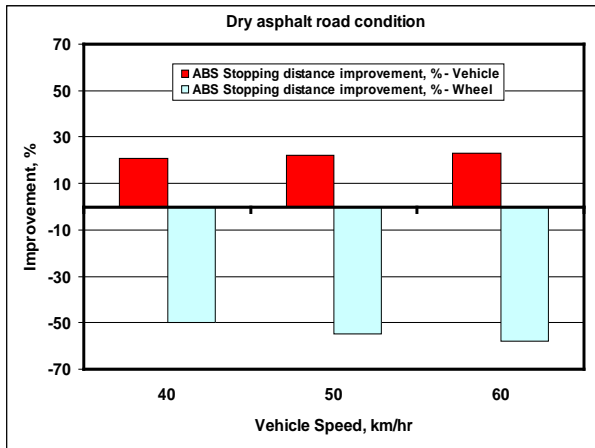


Figure 15. Improvement results of wheel and vehicle speeds VS, vehicle speed for ABS assisted and non-ABS at $V=14.5$ m/s and dry asphalt

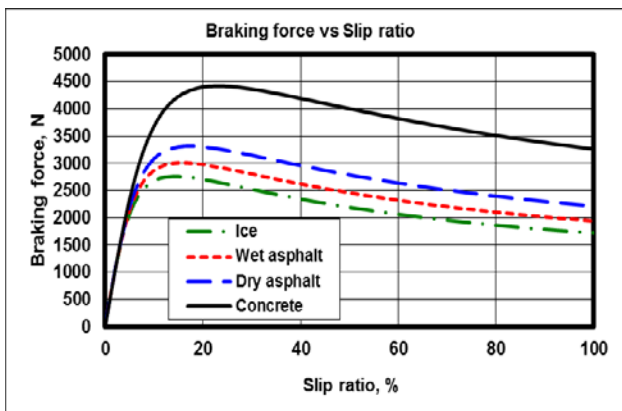


Figure 16. Simulation of braking force VS, slip at different road surface

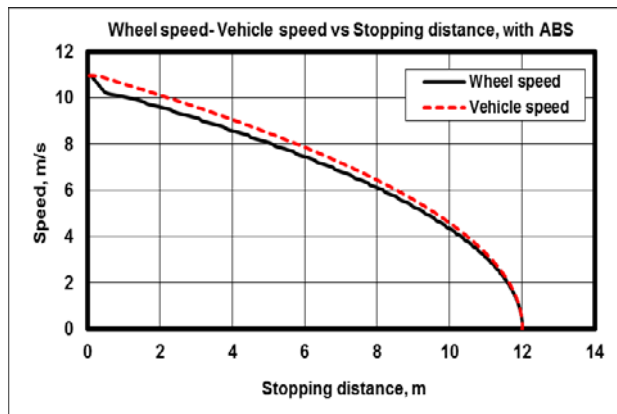


Figure 17. Simulation results of wheel and vehicle speeds VS, stopping distance for ABS at dry asphalt road condition, $V=40$ km/hr, $VTD=0.015$ s, $S_d=0.05$

Table 3. Simulated values of the types of different frictional road conditions and their properties

No.	Type of the road condition	Rolling resistance coefficient f_r (mean value)	Stopping distance, m
1	Dry asphalt	0.018	59
2	Concrete	0.020	61
3	Wet asphalt	0.050	67
4	Compact snow (ice)	0.100	74

Sample of the simulated results for determining the hydraulic valve time delays is depicted in Figure 18. Results are given for only apply brake signal. As the pressure increases so the time delays decrease, but only slightly due to choking of the hydraulic pipes. The hydraulic signal cannot travel any quicker than the speed of sound in air. Since the values of the time delays is not significantly dependent on the set brake pressures applied or if the brake is being applied or released, these values were averaged to calculate a single value. This considerably simplifies the simulation of the braking process. When simulating the performance of the ABS, the time delays are most important. These determine the delay of the ABS logic commands to the brake chambers and strongly influence the response speed of the system, where are some oscillations of the wheel speed due to the valve dynamics, however both the optimal slip and wheel speed are well tracked by the control strategy used. Table 4 gives values of stopping distance (SD) computed at different of valve time delays. It is noticed from the simulation results that the performances decay only at low speed when the wheel locks-up. As well known, the controllers based on acceleration thresholds induce oscillations on the braking pressure. Moreover, when the valve time delay increases the stopping distance increases too.

Table 4. Simulated values of the valve time delay

No.	Valve time delay (VTD), s	Stopping distance (SD), m
1	0.01	13
2	0.015	14.9
3	0.05	16

Sample from the simulation results for the vehicle and wheel braking for Bang-Bang controller are shown in Figure 19 for dry friction road conditions (asphalt), while Table 5 tabulates such results for different values for the desired slip. It can be observed that the desired slip increases the stopping distance decreases

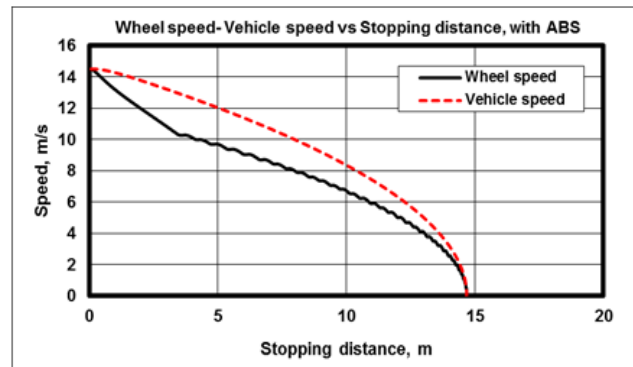


Figure 18. Simulation results of wheel and vehicle speeds VS, stopping distance for ABS for road condition (dry asphalt) at $V=50$ km/hr and valve time delay = 0.015

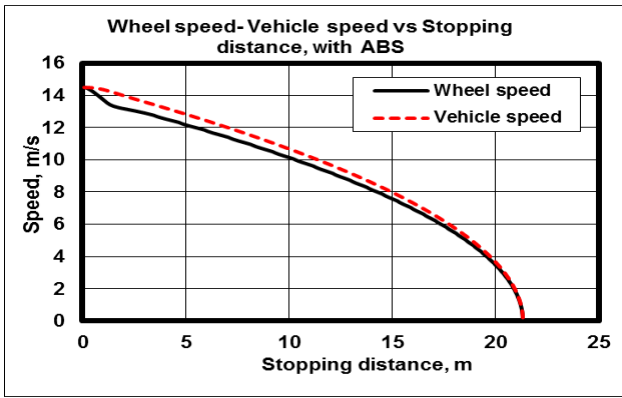


Figure 19. Simulation results of wheel and vehicle speeds VS, stopping distance for ABS for road condition (dry asphalt) at V= 50 km/hr and desired slip (S_d) = 0.05

Table 5. Simulated values of the desired slip

No.	Desired slip (S_d)	Stopping Distance (SD), m
1	0.20	37
2	0.15	28
3	0.10	23
4	0.05	21.5

6.2. Experimental Results

The simulation and experimental results for braking on dry asphalt ($S_d = 0.2$, brake demand = 2 bar and V=60 km/hr) are shown in Figure 20 to Figure 23 and Table 6. The results show good agreement between the stopping distance measured experimentally based on the procedure existed in section 5 and those simulated either for ABS assisted or non-ABS (Figure 20 and Figure 21). Moreover, the difference between stopping distance for ABS assisted and non-ABS either in simulation or experimental is clearly seen (Figure 22 and Figure 23). Further improvements could be made by modifying the tyre model to account for

the variation of maximum friction with speed. This is clearly evident from the experimental braking. Nevertheless the simulation is thought to be accurate enough for investigating the main behaviour of ABS systems.

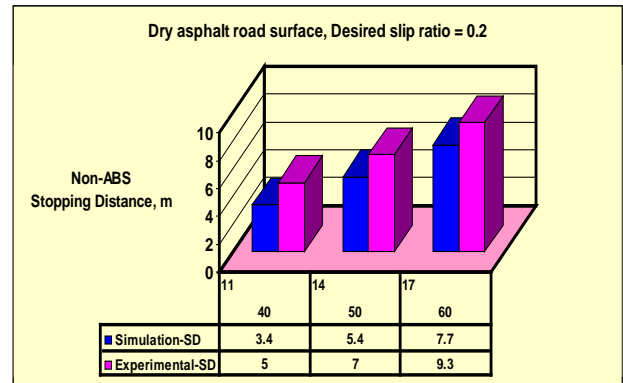


Figure 20. Simulation and experimental stopping distance results for ABS for road condition (dry asphalt) at desired slip of 0.2

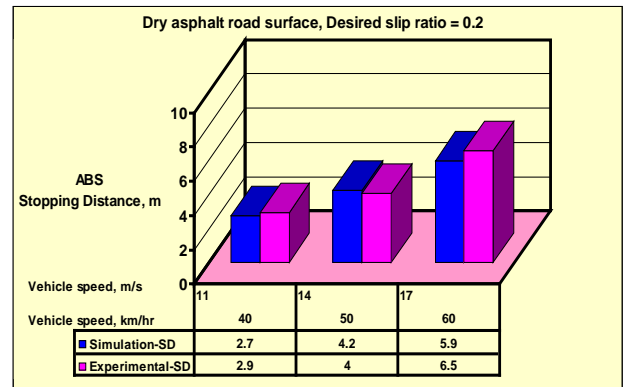


Figure 21. Simulation and experimental stopping distance results for non-ABS for road condition (dry asphalt) at desired slip of 0.2

Table 6. Experimental and simulated values of the stopping distance

No.	Vehicle speed, km/hr	Vehicle speed, m/s	Experimental stopping distance (SD), m		Theoretical stopping distance (SD), m	
			ABS	Non-ABS	ABS	Non-ABS
1	40	11.00	2.9	5	2.7	3.4
2	50	14.50	4	7	4.2	5.4
3	60	17.00	6.5	9.3	5.9	7.7

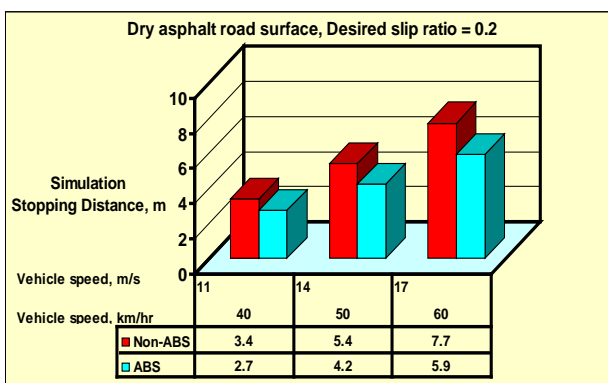


Figure 22. Simulation stopping distance results for road condition (dry asphalt) at desired slip of 0.2

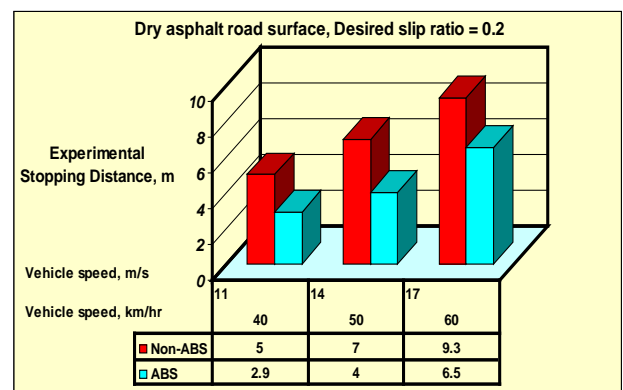


Figure 23. Experimental stopping distance results for road condition (dry asphalt) at desired slip of 0.2

7. Conclusions

The mathematical simulation of the braking performance of a vehicle has been developed. The simulation includes the details of vehicle dynamics, the ABS control algorithm as well as the mechanics of the foundation brakes and the hydraulic systems. The simulation of ABS was carried out, it can be drawn from the simulation results that the braking performance of automotive assisted ABS was improved significantly, the braking time advanced, and the stopping distance shorten consequently, the braking safety of vehicle can be improved. It can provide theoretical support which Bang-Bang algorithm is applied in the development of vehicle brake, and has a high application value.

The Bang-Bang control strategy for antilock braking systems has been established. The study has shown that it is possible to track the optimal slip value by measuring both the vehicle and wheel speeds. It is indicated that the control strategy is robust with respect to frictional road surface condition variations, takes into account the dynamics of the hydraulic systems and is almost hardware independent. The effectiveness of the control strategy has been tested by the experiments.

The vehicle braking simulation results are validated by comparing its performance with field tests (experimentally) on the vehicle. A significant agreement between simulation and experimental results has been gained. Furthermore, the difference between stopping distance for assisted ABS and non-ABS either in simulation or experimental is clearly estimated. Further improvements could be made by modifying the tyre model to account for the variation of maximum friction with speed. This is clearly evident from the experimental braking. Nevertheless the simulation is thought to be accurate enough for investigating the main behaviour of the assisted ABS systems.

The results of this study showed that there is potential to increase the braking performance of the vehicle using wheel-slip control, but improvement of the brake hydraulic system is required to maximize the performance. Moreover, It is well known that the dynamics of the hydraulic valves plays an important role for the system performances, where the amplitude of the wheel speed and slip oscillations increases with the valves slowness. However these oscillations are considered to be the optimal slip. Consequently the braking force is still around its maximum. The amplitude of the vehicle and wheel speeds increases with the valves slowness.

Nomenclature

P	Actual system pressure
P_d	Demand system pressure
T_d	Valve time delay

T_b	Braking torque applied on the wheel.
F_z	Normal load acting on the tire in z direction.
f_r	Rolling resistance coefficient.
B	Stiffness Factor
C	Shape Factor
D	Peak Factor
E	Curvature Factor
S	Actual slip
S_d	Desired Slip
V	Vehicle speed
V_ω	Wheel speed
ω	Wheel angular speed
R	Wheel radius
μ	Disk brake coefficient of friction
A_s	Brake wheel cylinder area
n	Number of friction surfaces
R_d	Effective brake radius.

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Appendix

A1. Magic formula

The magic tire model is an imperial formula which relates the longitudinal slip and the normal Force (F_z) to the longitudinal force (F_x) in a compact relationship. The following demands where put on the description of the tire behavior during braking. The basic form of the tire model is represented by the following equation:

$$F_x = D_x \sin[C_x \arctan\{B_{xK_x} - E_x(B_{xK_x} - \arctan(B_{xK_x}))\}] + S_{Vy} \tag{A1}$$

Where, B: Stiffness Factor = $\frac{a_3 \cdot f_z^2 + a_4 \cdot F_z}{C \cdot D \cdot e^{a_5 \cdot F_z}}$

C: Shape Factor (for skid) = 1.65

D: Peak Factor = $a_1 \cdot F_z^2 + a_2 \cdot F_z$

E: Curvature Factor = $a_6 \cdot F_z^2 + a_7 \cdot F_z \cdot a_2$

S: The percentage of tire longitudinal slip %
 a_1 to a_8 Are empirical coefficients

Coefficient					
1	a_1	-21.3	5	a_5	0.069
2	a_2	1144	6	a_6	-0.006
3	a_3	49.6	7	a_7	0.056
4	a_4	226	8	a_8	0.486

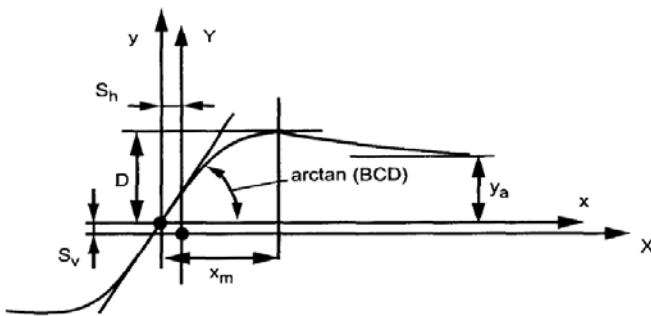


Figure A1. Characteristics of the Magic Formula for fitting tire test data