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Modelling and PID Value Search for Antilock Braking System (ABS) of a Passenger Vehicle

M. M. Abdul Majid[1], S. A. Abu Bakar[*2], S. Mansor[2], M. K. Abdul Hamid[2] and N. H. Ismail[1]

[1]Department of Computer Aided Engineering (CAE), Vehicle Development and Engineering (VDE), Proton Holding Berhad, 40918 Shah Alam, Selangor
[2]Department of Aeronautics, Automotive and Ocean Engineering, Faculty of Mechanical Engineering, Universiti Teknologi Malaysia, 81310 Skudai, Johor

[*]*Corresponding author: saiful@mail.fkm.utm.my*

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Abstract – *This paper presents the methodologies use in determining the PID value of an Antilock Brake System (ABS) of a Malaysian made passenger vehicle. The research work involves experimental work for data acquisitions, development of braking model, parameter tuning for both simulation model parameter and PID values search. A Malaysian made car is equipped with instrumentation used to collect vehicle behaviour during normal and hard braking manoeuvres. The data collected are the vehicle's stopping distance and longitudinal speed. The data during the normal braking are used to validate a two degree of freedom (2 DOF) of vehicle's braking model, while the data collected during the hard braking are used to search for the PID value used to control the operation of the ABS system. The developed simulation model of a braking system correlates well with the experimental data and the tuning done on the PID algorithm indicates that the ABS is controlled by the PI system.*

Keywords: 2 DOF braking model, Antilock Braking System (ABS), PID

1.0 INTRODUCTION

The concept of Antilock Brake System (ABS) was introduced in early 1900s by introducing modulated hydraulic braking pressure with main objective in preventing tire fully locked and wheel slippage.

Nowadays, the ABS has become a mandatory safety feature in most automotive market regulations due to the proven testament in preventing tire locked-up during emergency braking events as encountered on slippery road condition or during panic response by driver (Bosch, 1994). In ensuring the steer-ability of vehicle in any emergency conditions, it is important to prevent tire locked-up so that the vehicle is remain responsive to steering input intervention to avoid any catastrophic accident. It is also known that the ability of ABS in enhancing stopping distance especially on low road friction surface such as wet and slippery conditions (Garrick et al., 1998; Brosnan et al., 2012).

Railways and aircraft industries are the pioneer application of preliminary ABS concept in 1900s. The application of ABS is crucial in aircraft landing valve system to achieve shortest stopping distance on any runways surface condition and length. Boeing Corporation was the first company that applied ABS for its commercial aircraft in 1947 (Altrock & Krause, 1994). However due to uneconomical ABS technology cost for medium margin industry, it is taken a decade for its first application in automotive industry. Only later in 1954, the first introduction of ABS, supplied by French aircraft supplier was seen in Lincoln's model but with limited number of production units (Petersen, 2003). More automotive companies later introduced the ABS feature in their exclusive models in late 60's such as Chrysler, Cadillac and Ford. The early production stage of ABS technology utilized vacuum based actuated modulators and conventional analogue computers (Peterson, 2003). It was reported to be commercially unsuccessful by Wellstead and Pettit (1997).

The quantum change in ABS technology development was only seen in late 70's through the introduction of electronically-controlled base system by BMW and Mercedes. Further development of ABS technology by automotive vendors took place in 1985 when Audi, BMW and Mercedes used Bosch's ABS system in their models whilst, Teves ABS system by Ford and General Motors (Petersen, 2003). In late 80's, ABS application can be normally found in luxury and sports car segments as exclusive feature. However due to technology evolution and awareness on vehicle safety system, it was then be a common and mandatory feature in current automotive markets with more additional advance features introduced to the base ABS foundation such as traction control, Electronic Stability Control (ESC), hill assist, etc.

In general, there are two types of algorithms that are commonly used in controlling ABS system operation: (a) acceleration-control based, and (b) slip-control based. In the acceleration-control based, it is implemented by controlling the wheel slip by using wheel's acceleration/deceleration which is computed by deriving the wheel angular velocity data. However, a major disadvantage of this method is significant vibrations which existed during braking (Pedro et. al., 2003). As for the slip-based control algorithm, it involves by keeping the actual slip rate at an optimal target slip using continuously computed slip data computed form the vehicle's velocity and wheel's speed.

Various researchers have suggested different approaches in controlling the operation of the ABS system. Most of the current mass production of ABS controllers are based on linear or non-linear algorithms. Solyom (2002) used PI and PID control with gain scheduling based on the vehicle speed while Jiang and Gao (2001) showed that nonlinear PID controller achieved better braking performance than conventional PID controller. Buckholtz (2002) considered sliding mode-type approaches to the wheel slip.

This paper explains on the methodologies used in determining the parameters value for the ABS algorithm used to control the operation of an ABS system in a mass-produced passenger vehicle.

2.0 MODELLING OF 2-DOF BRAKING MODEL

A 2 DOF of vehicle's wheel model is used in this study where it consists of a vehicle's linear dynamics and wheel's rotational dynamics. Figure 1 shows the 2 DOF of vehicle's braking model.

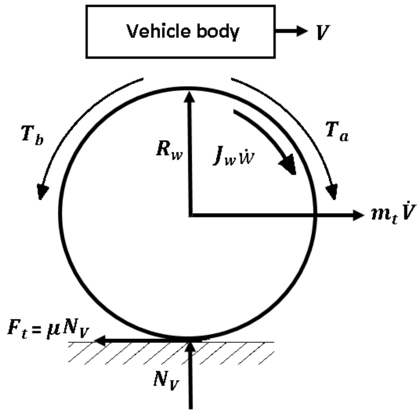

Figure 1: 2 DOF braking model

The linear dynamics of a vehicle during braking condition can be formulated as follows:

$$-V = \frac{N_w F_t}{M_v} \tag{1}$$

where $\dot{V}$ is vehicle linear deceleration, F_t is traction force and M_v is mass of vehicle. The vehicle linear acceleration is equal to the difference between total traction force acting at tire contact patch, divided by vehicle mass. Meanwhile, traction force is the average friction force of driving wheels for acceleration and the average friction force of all wheels for deceleration. The total traction force is equal to the product of average friction force, F_t and the number of wheels, N_w. The traction force model is crucial (Drakunov et al., 1995) and has been used in many studies. The assumption of nonlinear function of relative slip (λ) on traction force of each wheel during braking condition has been considered in this study. Traction force calculation is given by:

$$F_t = N_v A \mu \tag{2}$$

where N_v is wheel normal load, A is correctional gain for coefficient of friction and μ is coefficient of friction for road surface. Figure 2 shows the relationship between coefficient of friction and wheel slip ratio, S.

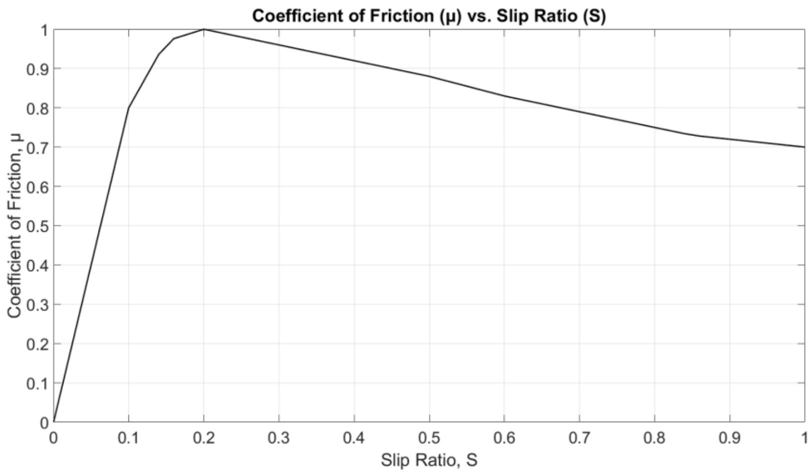

Figure 2: Typical friction curve characteristic (Petersen, 2003)

The normal load is described by standard Newtonian equation of motion given by:

$$N_v = m_t g + F_L \tag{3}$$

where, m_t is quarter vehicle mass. The longitudinal force due to weight transfer during braking, F_L is formulated by:

$$F_L = \frac{M_v h_{cg}}{2L} \dot{V} \tag{4}$$

where, M_v is vehicle mass, h_{cg} is height of vehicle's centre of gravity, L is wheelbase length. The rotational dynamics of the wheel is given by:

$$\dot{\omega} = \frac{T_a + R_w F_t - T_b}{J_w} \tag{5}$$

where, $\dot{\omega}$ is wheel angular acceleration, J_w is moment of inertia of wheel about the axis of rotation, T_b is the brake torque applied to the wheel, F_t is wheel's traction force.

The total torque acting on the wheel is divided by the moment of inertia of the wheel giving the wheel angular acceleration (deceleration). The total torque consists of shaft torque from the engine, which is opposed by the brake torque and the torque components due to the tire traction force. The engine torque is assumed to be zero (0) during braking. The longitudinal slip ratio, S of the tire during braking condition is determined by:

$$\lambda = \frac{\omega R_w - V}{\omega R_w} \tag{6}$$

Figure 3 shows the braking model developed in the Matlab/Simulink environment.

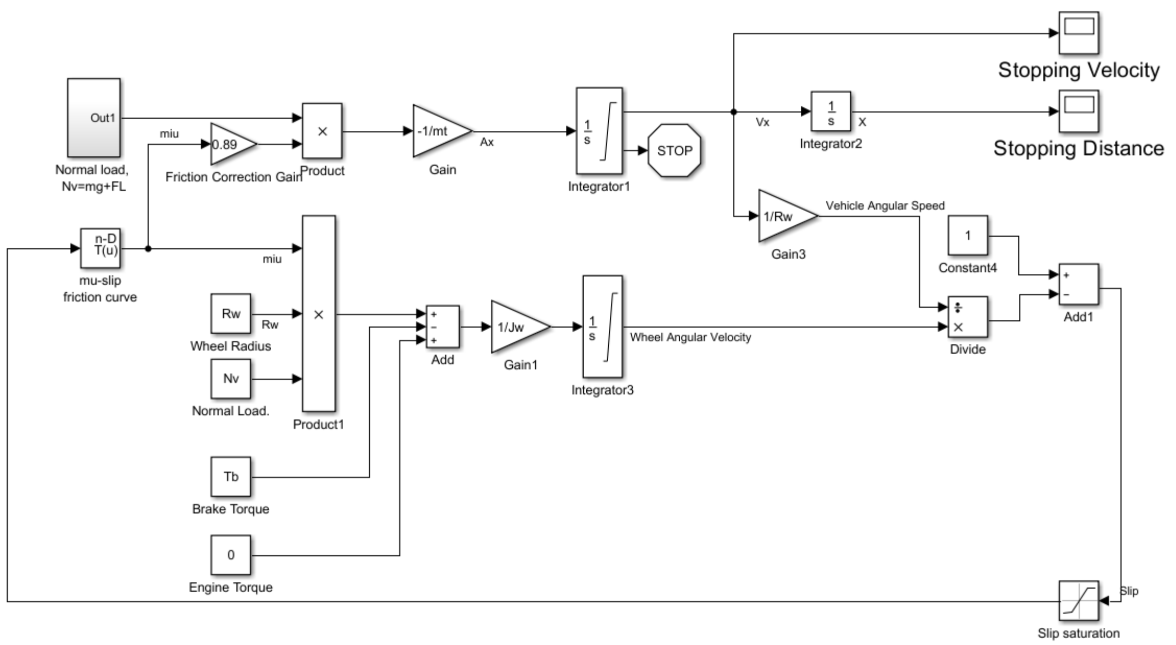

Figure 3: 2 DOF braking model developed in Matlab/Simulink environment

3.0 EXPERIMENTAL SETUP AND MODEL VALIDATIONS

For the purpose of validation and determining the parameter values for the braking model and the PID values used in the PID algorithm, the results from the simulation model will be compared with experimental data. For these purposes, the vehicle's longitudinal speed and stopping distance will be used as the main responses and to be collected during the experiment measurements. The vehicle's longitudinal speed and stopping distance were measured using a GPS system, connected to a Racelogic VBOX 3i which acted as a data logger system for the data collection.

There were two types of tests conducted in validating the braking model of the studied passenger vehicle.

- The first test involves a normal braking test which requires a normal braking effort by the driver until the vehicle stop, while
- The second test requires a hard braking (sudden braking) by the driving which will operates the vehicle's ABS.

Both tests were conducted on a 575m straight track with the speed before braking effort is 105km/h.

4.0 PID CONTROLLER AS THE ABS ALGORITHM

The PID algorithm is assumed to be used to control the operation of the ABS system of the studied vehicle due to the facts of algorithm's simplicity and only requires a small processing capability (Fu et al., 2012). The PID is the most common control algorithm (Bera et al., 2011) which is being widely used in the process control due to its advantages of requiring small amount of processing capability, good in real time response, easy to implement etc.

The PID controller, functions by estimating the corrections that need to be made base on the errors obtained from the actual model with the desired input. The PID algorithm is given by:

$$u(t) = K_P . e(t) + K_I \int_0^t e(\tau) d\tau + K_D \frac{de(t)}{dt} \tag{7}$$

where, $u(t)$ is the control output, K_p is the proportional gain, K_I is the integral gain, K_D is the derivative gain; e_t is the error between model and reference value; t is the time or instantaneous time; τ is the variable of integration.

Figure 4 shows the layout of the PID-controlled ABS implemented in Matlab/Simulink environment. Where the PID function is to estimate sufficient braking torque that need to be applied to the wheel so that the slip ratio, S will be no more than 0.2 (by referring to Figure 2, the coefficient of frictions will drop if the slip ratio is more than 0.2. The estimated value from the PID algorithm is considered as an ideal case, where the estimation of the braking torque will be applied directly into the braking model.

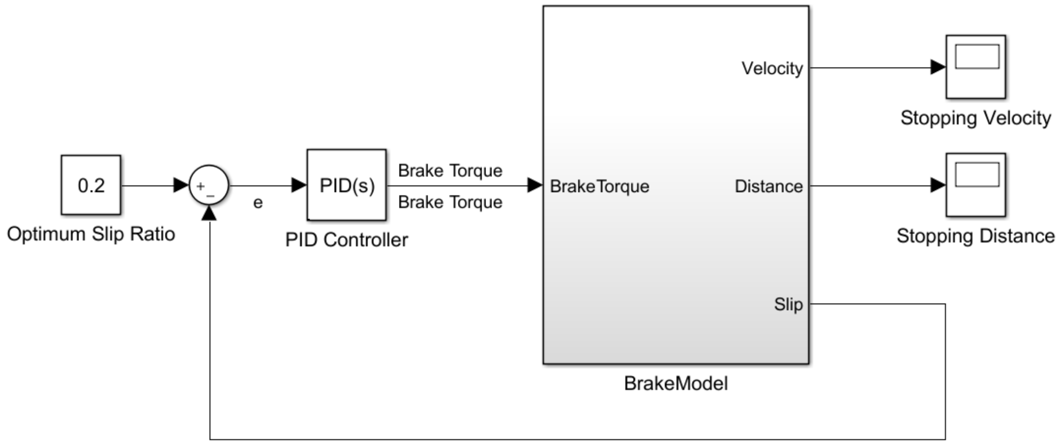

Figure 4: PID-controlled ABS

4.1 Parameter Tuning

Two level of parameter tuning were done in this study. The first level involves parameter tuning in search for the parameter values for the braking model, while the second level involves parameter tuning for the PID algorithm. The first level involves the search for two parameter values which are the values for wheel inertia value, J_w and the correction gain for the friction coefficient, A respectively. The second level of tuning involves the tuning for P, I and D values in the PID algorithm.

The tuning for both levels were done manually for in depth understanding on how each parameter value affects model's behaviours. The tuning was done by minimizing the root-mean-square (RMS) error between the simulation and experimental results of both stopping distance and stopping velocity. The RMS value is given by:

$$RMS = \sqrt{\frac{1}{T}\int \left\| u(t) \right\|^2 dt} \tag{8}$$

where, T is the total simulation time and $u(t)$ is the response signal data. The selection of parameter value is based on a minimum error of RMS values, between the simulation and experimental data which is given by:

$$RMS_{emin} = \left| RMS_{simulation} - RMS_{experiment} \right| \tag{9}$$

where, T is the total simulation time and $u(t)$ is the response signal data. The selection of parameter value is based on a minimum error of RMS values, between the simulation and experimental.

5.0 RESULTS AND DISCUSSION

Figures 5 to 8 show the validation results between the simulation and experimental results, for both stopping distance and stopping velocity during the normal braking an ABS braking conditions. It can be seen in general that the operation of the installed ABS in general are capable of improving the stopping distance and stopping time by 22.2 percent and 18 percent, respectively.

The manual tuning, which were done based on the method in Section 4, for searching the values for wheel inertial value, J_w and *t* correctional gain for friction coefficient, *A* gives a good correlation between the simulation results and the experimental results (refer to Figure 5 and Figure 6). The manual tuning done during the first level gives the value of 2.054 kg/m^2 for the wheel inertial parameter, while for the correctional gain for the friction coefficient, A the value obtained is 0.87.

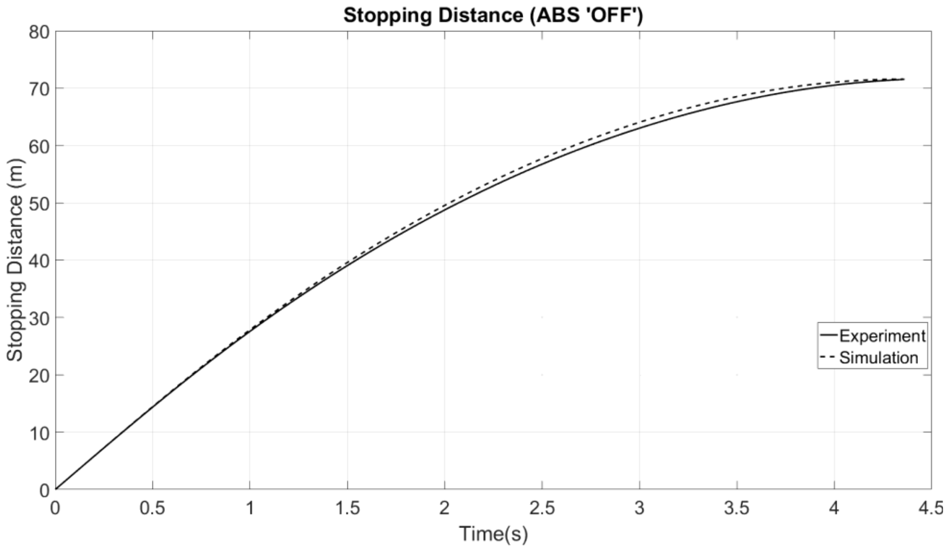

Figure 5: Stopping distance response validation (ABS 'OFF')

The values obtained during the first level of tuning were used in the second level of tuning, which involves the tuning for the PID values. The tuning for the PID values were done manually where its starts with the initial guess for P, while the value for I and D are set to 0. Initial value of P was set to 500 with an increment of 50 for each iterations. The tuning is stopped and the last value for P is selected when the RMS error (Equation 9) between the simulation and experimental data is the lowest. The tuning of P value affects the magnitudes of the stopping distance and stopping velocity.

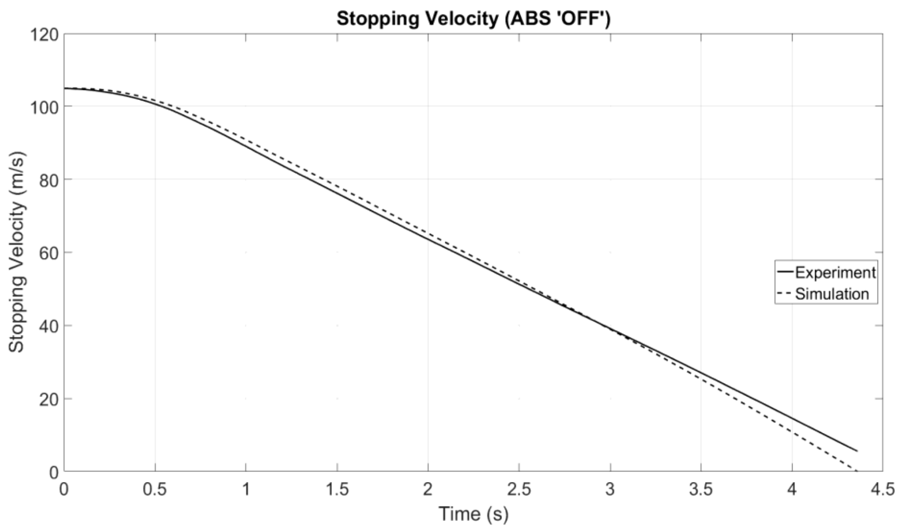

Figure 6: Stopping velocity response validation (ABS 'OFF')

The tuning for the I value, starts by fixing the P value which was previously selected. The initial value for I is set to 10 with an increment of 10 for each repetition done. As for the D value, the tuning done revealed that it does not give any changes in the RMS error between the simulation and experimental data, indicating that the D value does not effecting the ABS system operations. Figure 7 and Figure 8 show good correlation between the simulation and the experimental results. The value of P and I used in producing the correlations shown in Figures 7 and 8 are 5950 and 500, respectively.

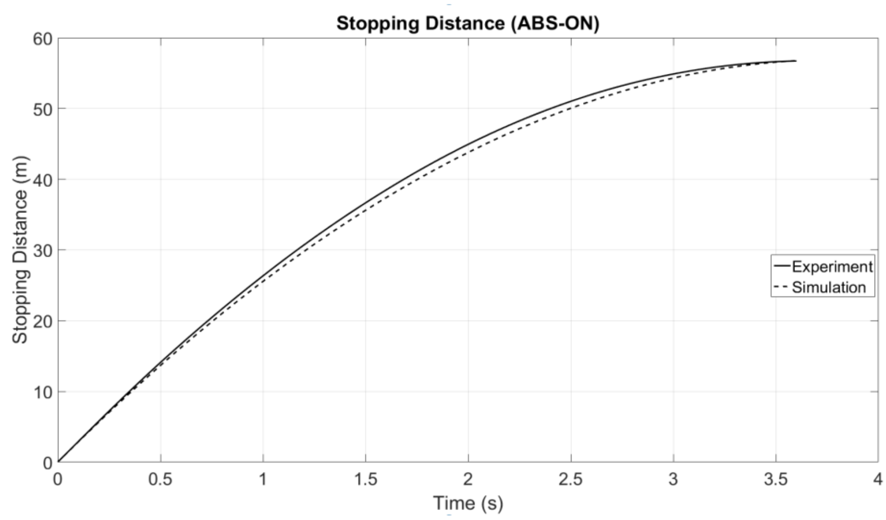

Figure 7: Stopping distance response validation (ABS 'ON')

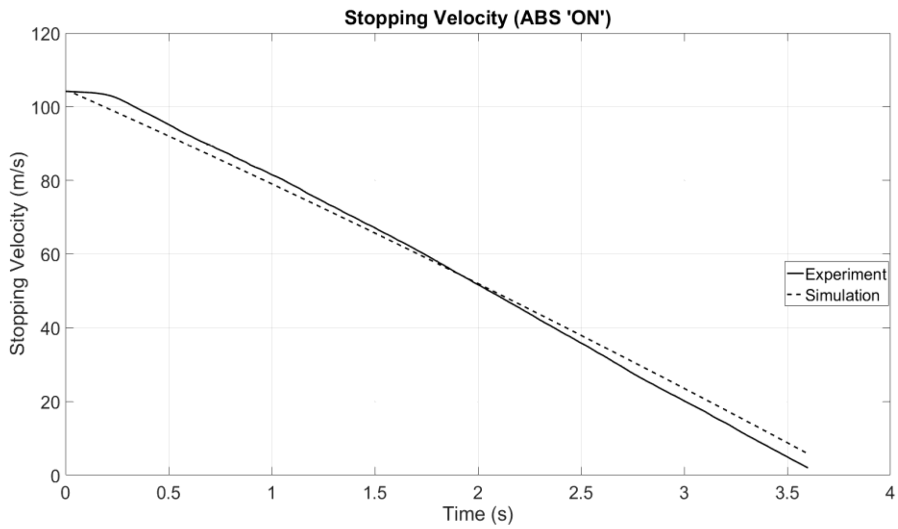

Figure 8: Stopping velocity response validation (ABS 'ON')

6.0 CONCLUSIONS

A 2 DOF of vehicle's braking model has been developed where some of its parameters were tuned in order to obtain a similar response with the experimental data (stopping distance and stopping velocity). Two levels of tuning were done, where the first level involves the tuning for the wheel's inertial value and correctional gain for the coefficient of friction, *A*. As for the second level of tuning which involves the PID value search for the PID algorithm used to control the operation of the ABS system, it were found only the tuning of P and D values affect the braking's model behaviour. The value of P and I, obtained from the tuning done were 5950 and 500, respectively, while the value for D is 0. As a recommendation for potential future works, the validated braking model could be used to study other algorithms that can control the operation of the ABS as well as considering an actual case of ABS which is by considering the modelling and integration of a hydraulic system in the braking model.

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