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Simulation Performance of Antilock Braking System under Different Drag Coefficients

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Abstract—*Simulation performance of antilock braking system under different drag coefficient has been presented. The research is aimed at finding the effect of air drag force on the performance of a controller for an ABS during hard braking. Simulink model of a single tyre is obtained. A proportional integral and derivative (PID) controller is developed and integrated with the single tyre model. Simulation was performed in Matlab/Simulink. The results obtained shows that as the drag coefficient increases the braking or stopping distance decreases. This shows that the controller performed effectively irrespective of the increase in the air drag force around a braking vehicle under severe road condition.*

Keywords— *ABS, Controller ,Hard braking, PID, Drag coefficient increases.*

I. Introduction

Vehicle traction control system (VTCS) is a technology introduced by automobile manufacturers that provides traction and tracking control during acceleration and deceleration. It is the technology that plays the essential role in electronic stability control (ESC) system. In order to ensure performance improvement of the vehicle traction on low friction road surface and maximize the forward friction coefficient, traction control system (TCS) is used to maintain optimal slip ratio while antilock braking system (ABS) is used during braking.

The essence of VTCS is to ensure safety during acceleration or deceleration when road condition is severe or hard braking. There are basically two requirements of any controller that will be used in VTCS. One the controller must be robust so as to be able to maintain wheel slip around the desired slip ratio under severe road surface conditions

Vehicle traction control system (VTCS) provides for traction and braking control. The basic function of both traction control system (TCS) and antilock braking system (ABS) is to control the wheel of a tyre whose speed shows significant variation from the average speed of the wheels of other tyres. In a situation where both TCS and ABS are integrated in a vehicle, these control functions are normally combined into hydraulic control unit, where they share a common electronic control unit (ECU) or controller. The objective in this case is for both to operate so as to provide improved vehicle tracking and stability when accelerating and braking under severe road surface conditions.

Since the function of VTCS is to improve vehicle performance on road with respect to the interaction force between the tyre and the road surface condition during hard braking or acceleration in unfavorable road surface condition; in this context, a controller is designed to improve the slip tracking performance of a vehicle during braking on poor road surface conditions. In order to minimize slip around a stable operating point, a slip controller should be able to track a set point slip which is desired for optimal performance.

II. Liteature Review

A slip tracking approach in which the design objective is for each wheel to track a reference trajectory for the longitudinal wheel slip is proposed in [1]. A nonlinear proportional integral and derivative (NPID) controller is proposed in [2]. The NPID is achieved by incorporating a nonlinear function to the linear PID. Sliding mode control (SMC) is a robust control approach. The class of systems in which it is applied, sliding mode controller design offers a systematic approach to the problem of keeping the system stable and ensuring consistent performance. However, the main disadvantage with the SMC is the chattering caused by the non-linearity in the dynamic equations of the wheel slip control model, which could affect the life span of the component parts of the antilock braking system. In the study carried out in [3], it was reported that some researchers have tried solve the problem of chattering by introducing a saturation function in place of switching sign function for different road conditions. A moving sliding surface is proposed in [4] for the slip control based on global sliding mode control technique. Another sliding model control (SMC) called grey sliding mode control (GSMC) is proposed in [5] for slip the control of wheel slip depending on the forward velocity. A self-learning fuzzy sliding mode control (SLFSMC) design method for slip control in antilock braking system is proposed in [6]. A hybrid system that combines feedback linearization (FBL) and PID controllers to realize the hybrid FBLPID controller for slip control in antilock braking system is proposed in [7]. Mathematical simulation and implementation of slip control in antilock braking system in Matlab using a Bang-Bang control technique is presented in [8]. Slip control model for purposes of performing slip tracking of target slip is formulated in [9]. In order to maintain a desire slip ratio, [10] formulated a linear slip control model.

In this paper, the objective is to study the linear control performance of a vehicle antilock braking system under different drag coefficients with respect to stopping distance.

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III. System Modelling

A. Dynamic Equations of a Vehicle

In this section, the dynamic equations of a vehicle are obtained using a quarter-car or single tyre model. Fig. 1 represents a single tyre model.

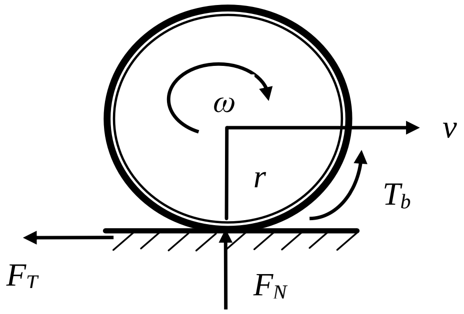

Fig. 1. Single tyre model.

A single tyre model is used to obtain the forward or longitudinal braking dynamics. It comprises a single tyre carrying a quarter mass, m, of the vehicle, such that the vehicle is moving with a longitudinal velocity $v(t)$ at any time, t. Before brakes are applied, the wheel moves with an angular velocity of $\omega(t)$, driven by the mass, m in the direction of the longitudinal motion. The modelling of the system is carried out using Simulink block.

The single tyre Simulink model:

The single tyre model shown in Fig. 2 is a combination of wheel and vehicle dynamics.

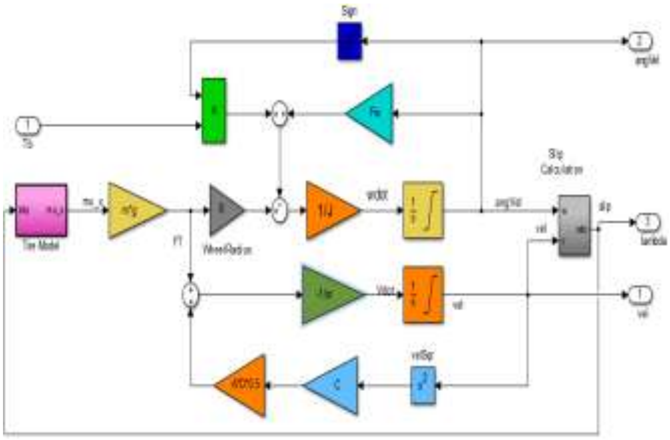

Fig. 2. Single tyre Simulink model.

The actuator Simulink model:

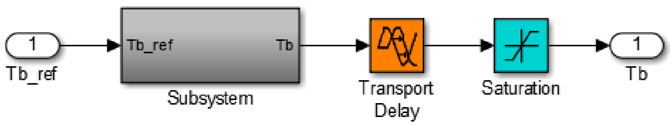

Fig. 3. Simulink model of actuator.

B. Controller Design

Fig. 4 shows the Simulink model of the PID controller in discrete form. The values of the gains k_p, k_i and k_d were chosen for the implementation of the controller by combining proportional control (P), integral control (I) and derivative control (D) simultaneously and watching the output for key response parameter of interest.

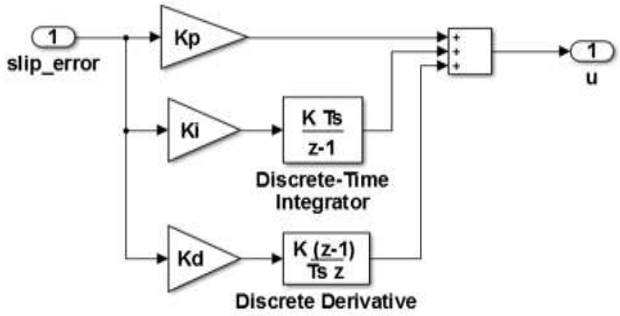

Fig. 4. Simulink model of PID controller.

The controller gains that have been evaluated and chosen to work well for the model used here are $k_p = 2000$, $k_i = 1200000$, $k_d = 1$. The discrete sample rate chosen is $T_S = 0.005s$.

C. System Configuration

The block diagram of the slip control system is shown in Fig. 5. It is a closed-loop control model which integrates a standard quarter car model, a brake actuator, and a proportional integral and derivative controller. The control loop follows a very standard form. The controller, actuator, and the quarter-car models are all in the forward path. The wheel slip, which is the output, is fed back and compared with a desired slip value, with error fed into the controller.

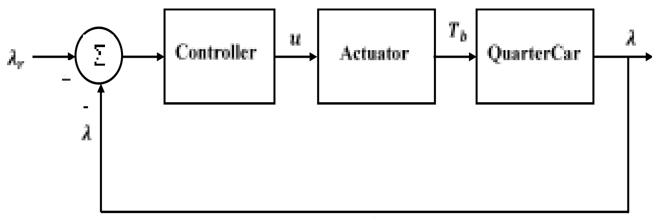

Fig. 5. Block diagram of slip control.

D. Optimal Performance Evaluation

The main objective of slip control in ABS is to maintain the maximum brake force to reduce the brake distance. The braking distance can be chosen as the cost function to define the optimality of the system. In this context, the braking distance is compared from the initial speed braking v_0 to final speed of 1.8km/h (or 0.5m/s), and then the cost function can be defined as below:

$$d = \int_{v0}^{1.8} v dt \quad (1)$$

The performance of the controller based on the error is evaluated using the integral squared error [ISE] of the slip. A system is considered an optimum control system when the system parameters are adjusted so that the index reaches an extremum value, commonly a minimum value. The model for the evaluation done using the Matlab/Simulink is shown in Fig. 6.

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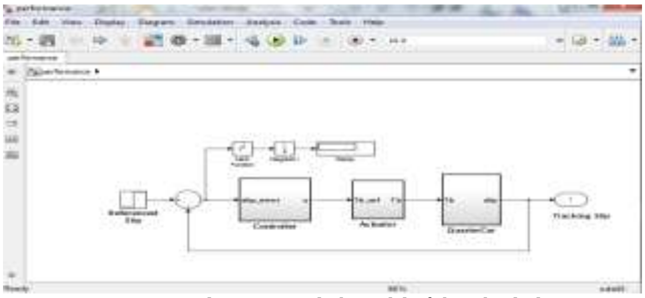

Fig. 6. ISE evaluation Simulink model of the wheel Slip.

E. Simulation Parameters and Numerical Values

In this paper, the definitions and the numerical values of the parameters used for this work are stated in Table 1. The numerical values used here were those of [7] and [10] are adopted for the present work.

The parameters in Table 1 are assumed the same for the simulations of four different cars with different drag coefficients. The reason is that aerodynamic drag is a significant factor in car's ability to accelerate or decelerate (braking) and also affects the fuel economy of a vehicle.

The Pacejka friction model is very detailed, and it is the tyre-road friction description most commonly used in commercial vehicle simulators such as, for example, CarSim, Adams/Tire, and Bikesim [11]. The Pacejka friction model is given in (2) and the parameters are defined in Table 2.

$$\mu_x = a(1 - e^{-b\lambda} - c\lambda) \qquad (2)$$

where a, b, c are constants

TABLE 1. System parameters and numerical values [8] and [10].

m	Single tyre mass	450	kg
J	Moment of inertia	1.6	Kgm^2
R	Wheel Radius	0.32	m
F_w	Wheel friction coefficient	0.08	Nms/rad
τ	Hydraulic time constant	0.0143	s
g	Gravitational acceleration	9.81	m/s^2
λ_r	Desired slip	0.1	Dimensionless
A_p	Actuator Pole	70	Dimensionless
K	Hydraulic gain	1.0	Constant

TABLE 2. Values of different road conditions [11].

Road condition	ϑ_{r1}	ϑ_{r2}	ϑ_{r3}
Dry asphalt	1.28	23.990	0.52
Wet asphalt	0.86	33.82	0.35
Cobblestone	1.37	6.46	0.67
Snow	0.19	94.13	0.06

The drag coefficient of some commercial vehicle are listed in Table 3.

TABLE 3. Drag coefficients of some commercial cars [12].

Car	Drag Coefficient
BMW i8	0.26
BMW i3	0.29
Jaguar XE	0.26
Mazda3	0.26
Mercedes-Benz B-Class	0.26
Mercedes-Benz C-Class/ S-Class	0.24
Nissan GT-R	0.26
Nissan Leaf	0.28
Toyota Prius	0.25
Tesla Model S	0.24

IV. Simulation Results, Performance Analysis and Discussion

A. Simulation Results

In this paper, the analysis and discussion of the results obtained from the modeling and simulation of the quarter car model for slip minimization in vehicle traction control is presented. To evaluate the performance of the PID controller on different road conditions, simulations were implemented in Matlab/Simulink environment.

Simulation for Dry Asphalt Road Condition:

Fig. 7 to 10 show the tracking performance of the control system on dry asphalt road condition under different drag coefficients, C.

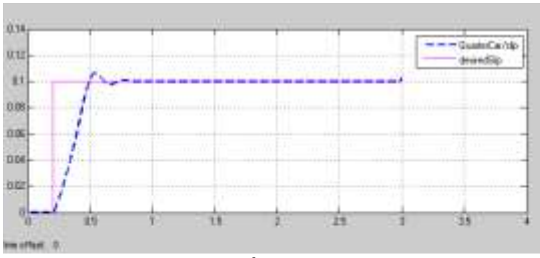

Fig. 7. for C = 0.24.

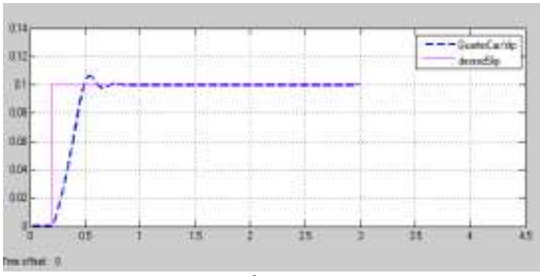

Fig. 8. for C = 0.26.

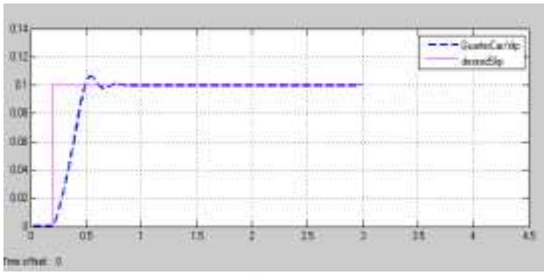

Fig. 9. for C = 0.28.

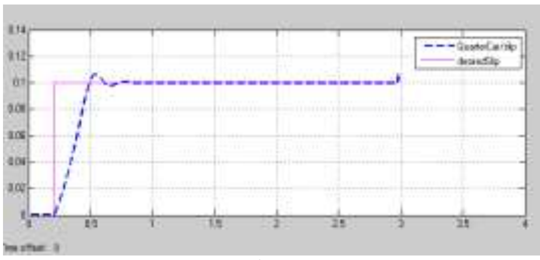

Fig. 10. for C = 0.29.

Fig. 11 to 14 show the stopping distance performance of the control system on dry asphalt road condition under different drag coefficients, C.

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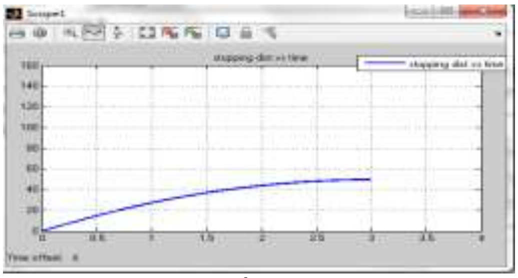

Fig. 11. for C = 0.24.

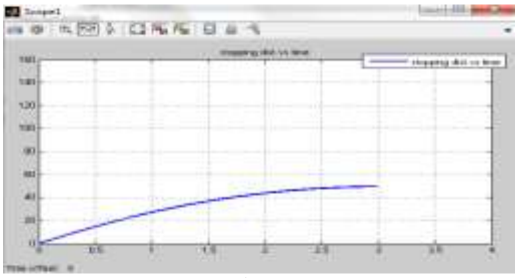

Fig. 12. for C = 0.26.

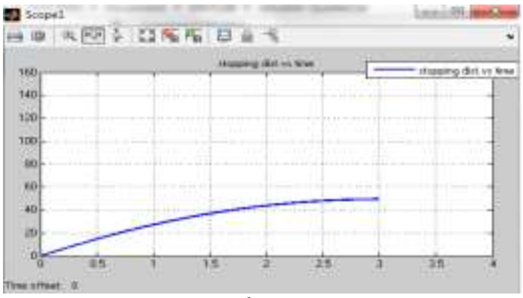

Fig. 13. for C = 0.28.

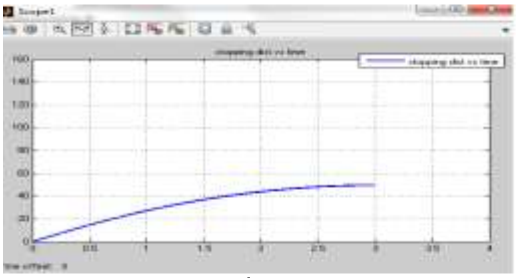

Fig. 14. for C = 0.29.

Simulation for Wet Asphalt Road Condition:

Fig. 15 to 18 show the tracking performance of the control system on wet asphalt road condition under different drag coefficients, C.

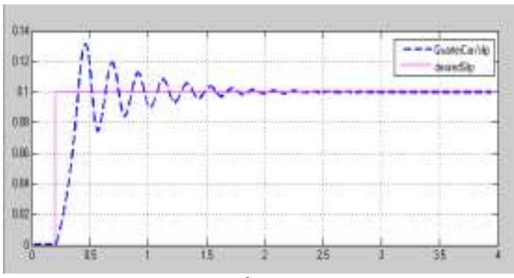

Fig. 15. for C= 0.24.

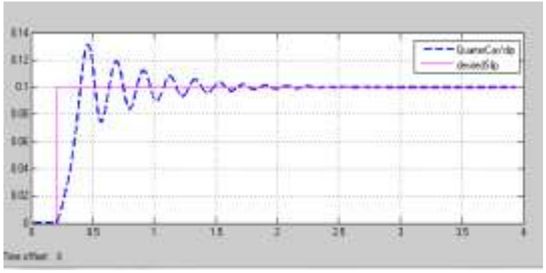

Fig. 16. for C = 0.26.

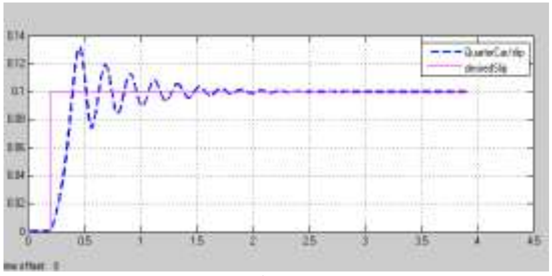

Fig. 17. for C = 0.28.

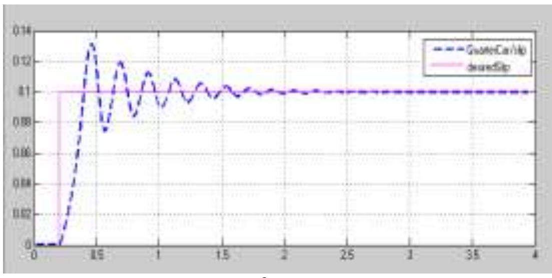

Fig. 18. for C = 0.29.

Fig. 19 to 22 show the stopping distance performance of the control system on dry asphalt road condition under different drag coefficients, C.

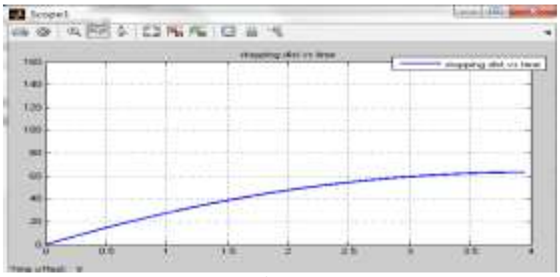

Fig. 19. for C = 0.24.

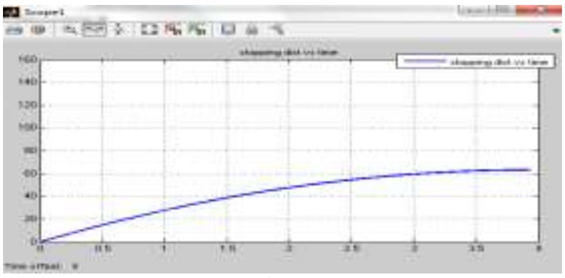

Fig. 20. for C = 0.26.

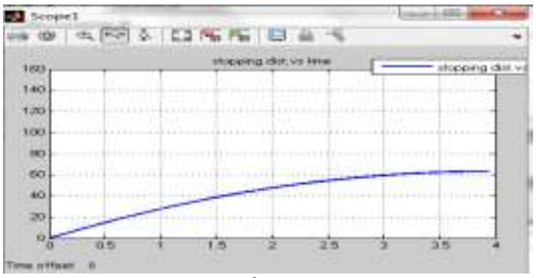

Fig. 21. for C = 0.28.

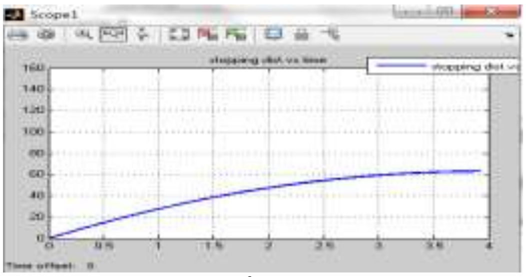

Fig. 22. for C = 0.29.

B. Performance Analysis

The objective of slip control in anti-lock braking system (ABS) is to track a predetermined set point. The performance of the system with and without controller can be evaluated by letting the ABS-assisted stopping distance to be compared to those simulated with non-ABS assisted. In order to make easy this comparison, the following equation was used [8]:

$$ABS-(SDI)=\frac{SD_{non-ABS}-SD_{ABS}}{SD_{non-ABS}}\times 100\% \tag{3}$$

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

SD_{ABS} = ABS assisted stopping distance.

TABLE 4. Simulated values of stopping distance when the vehicle is equipped with ABS and non-ABS for dry and wet asphalt road conditions respectively.

Drag Coeff.	DA SD_{ABS}	WA SD_{ABS}	DWA $SD_{non-ABS}$	Stopping Distance Improvement Dry/Wet (%)
0.24	49.63	63.43	143.9	65.51
0.26	49.37	63.01	142.8	65.43
0.28	49.53	63.26	143.5	65.48
0.29	49.42	63.09	143.0	65.44

DA = Dry Asphalt, WA = Wet Asphalt, DWA = Dry/Wet Asphalt

The bar charts shown in Fig. 23 and 24 are used to explain the performance of the controller on different road conditions and also the effect of the wind drag on the braking.

The bar charts in Fig. 23 and 24 show the comparisons of the performance of the PID controller on both dry asphalt road surface and wet asphalt road surface conditions. In Fig. 23 it can be seen that the controller achieves lower stopping distance in dry asphalt road than in wet asphalt road. In Figure 24, the percentage improvement of the controller in dry asphalt road surface is greater than that of the wet asphalt road surface for all drag coefficients.

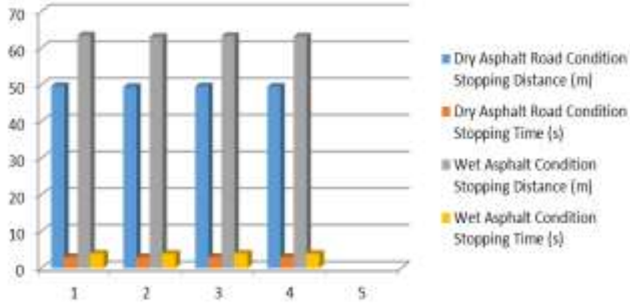

Fig. 23. Bar chart of stopping distance for dry/ wet asphalt road conditions and their stopping time.

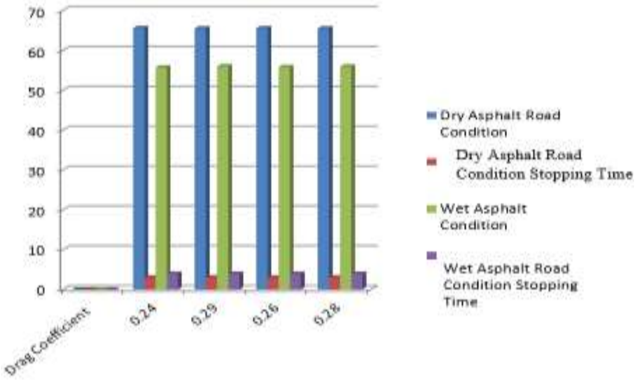

Fig. 24. Bar chart of percentage improvement of stopping distance for dry/ wet asphalt road conditions and their stopping time.

C. Discussion

In general, the proportional plus integral plus derivative (PID) controller is able to reduce the stopping distances in both dry asphalt and wet asphalt road conditions. The PID controller achieved stopping distance of lower value in dry asphalt road surface than the wet asphalt road surface because the dry asphalt road surface is a high friction road surface and the wet asphalt road surface is a low friction road surface.

It can be seen from the simulation results obtained for the stopping distance on dry and wet asphalt road conditions considering different drag coefficients that the stopping distance during severe braking was significantly improved. The simulation results presented in Table 4 show that when the controller is activated with slip occurring, it brings about improved stopping distance and reduced braking time.

Considering the drag coefficients, as the drag coefficient increases the stopping distance decreases. Hence the controller is able to reduce the stopping distance of hard braking vehicle irrespective of the air drag of the car.

Generally, simulation results of Fig. 7 to 10 and Fig. 15 to 18 show that the PID controller maintains an optimal wheel slip of 10% for both road surface conditions. Though a comparison of the slip plots for the dry and wet asphalt road conditions shows that the controller have a superior performance of regulating the longitudinal slip with respect to the desired slip value (λ_r) on dry asphalt road than wet asphalt road. For both road conditions, the controller achieved rise time of 0.2 s.

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V. Conclusion

The paper has presented Simulation performance of antilock braking system (ABS) under different drag coefficient. It shows that the performance of a braking system can be improved with respect to the stopping distance considering air drag force around the braking vehicle. Hence, the research is geared towards finding the effect of air drag force on the performance of a controller for an ABS during hard braking.

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