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Modelling and Simulation of Braking System in Automobile

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ABSTRACT

In recent years vehicles are fitted with antilock brake systems to enhance braking system in automobiles and prevent wheel lock during braking. To achieve this controller are needed controlled torque that will maintain the optimum wheel slip ratio and coefficient of friction, which is the vehicle speed in relation to wheel rotation. A model of a quarter vehicle was developed and differential equation of motion was formulated and was applied to linear and angular velocity. Control strategies of Proportional Feedback Control, Proportional Integral Feedback Control, Proportional Integral Derivative Feedback Control, were applied to determine the effectiveness of maintaining desired slip ratio and coefficient of friction. Mathlab/Simulink software were used to simulate the response of the different braking strategies. Results shows that by maintaining the slip ratio and coefficient of friction values, the braking time was reduce 3.6 seconds to 2.3 seconds for a distance of 45m.

KEYWORDS: *ABS, control feedback, slip ratio, coefficient of friction*

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1. INTRODUCTION

Since the development of the first motor driven vehicle in 1769 and the occurrence of first driving accident in 1770, engineers were determined to reduce driving accidents and improve the safety of vehicles [1]. By the 1960s, high end automobiles were fitted with rear-only ABS, and with the rapid progress of microcomputers and electronics technology, the trend exploded in the 1980s. Today, all wheel ABS can be found on the majority of our model vehicles and even on select motorcycles [2-6]. It important to note that wheels will slip and lockup during severe braking or when braking on a wet or icy road surface [7-8]. Such breaking results in a long stopping distance and sometimes the vehicle will lose steering stability [9-11].

However, Anti-lock brake systems (ABS) prevent brakes from locking during braking. Under normal braking conditions the driver controls the brakes. However, during severe braking or on slippery roadways, when the driver causes the wheels to approach lockup, the antilock system takes over. ABS modulate the brake line pressure independent of the pedal force, to bring the wheel speed back to the slip level range that is necessary for optimal braking performance. An antilock system consists of wheel speed sensors, a hydraulic modulator, and an electronic control unit. The ABS have a feedback control system that modulates the brake pressure in response to wheel deceleration and wheel angular velocity to prevent the controlled wheel from locking. The system shuts down when the vehicle speed is below a pre-set threshold.

The main difficulty in the design of ABS 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 [12]. 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 [13].

Antilock systems have effectively improved the stopping distances, improve stability, and also improve steerability during braking. Meanwhile the technology of ABS is also applied in traction control system (TCS) and vehicle dynamic stability control (VDSC) [14].

2. VEHICLE DYNAMICS

The ABS controller must deal with the brake dynamics and the wheel dynamics as a whole plant [15]. Wheel lockup is undesirable since it prolongs the stopping distance and causes the loss of direction control [16,17].

A model capturing the essential features of the vehicle system has to be employed for the controller design. The design considered here belongs to a quarter vehicle model as shown in Fig 2.1. This model has been already used to design the controller for ABS.

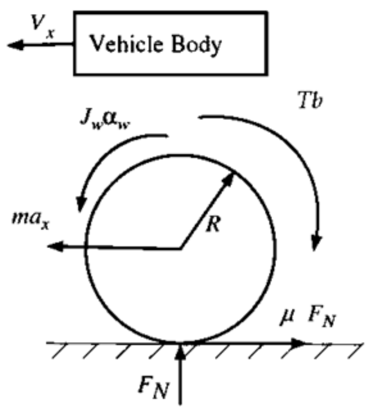


Fig. 1 Vehicle Model

The longitudinal velocity of the vehicle and the rotational speed of the wheel constitute the degrees of freedom for this model. The governing two equations for the motions of the vehicle model are as follows:

For braking force balance in longitudinal direction (vehicle)

$$ma_x = \mu F_n \rightarrow m\frac{dV_x}{dt} = \mu F_n \qquad (1)$$

Summing torque at wheel center

$$j_\omega \alpha_\omega - \mu r F_n - T_b \qquad (2)$$

For convenience a slip ratio is defined according to:

$$\lambda = \frac{V_x - \omega r}{V_x} \qquad (3)$$

Differentiating on both sides with respect to time (t), we get

$$\dot{\lambda} = \frac{\dot{V}_x(1-\lambda) - \dot{\omega} r}{V_x} \qquad (4)$$

Where; V_x= linear velocity of vehicle a_x = linear acceleration of vehicle ω = rotational speed of wheel α_ω = angular acceleration of wheel T_b = braking torque, λ = slip ratio μ = friction coefficient r = radius of tire m = mass of the model.

The slip ratio is dependent on the input torque u and the vehicle velocity V_x during breaking.

$$x_1 = s_x \qquad (5)$$

$$x_2 = V_x \qquad (7)$$

$$x_3 = \lambda, \qquad (8)$$

where s_x is the stopping distance

The state space equations are

$$\dot{x}_1 = x_2 \qquad (9)$$

$$\dot{x}_2 = \frac{-\mu F_n}{m} \qquad (10)$$

$$\dot{x}_3 = \frac{-\mu F_n}{x_2}\left(\frac{1-x_3}{m} + \frac{r^2}{j\omega}\right) + \frac{r}{j_\omega x_2} T_b \qquad (11)$$

The slip model in equation (5-8) gives value of coefficient of friction as a function of linear velocity and hence slip ratio.

$$\mu(\lambda, V_x) = \left[C_1\left(1-e^{-C_2\lambda}\right) - C_3\lambda\right]e^{-C_4 V_x} \qquad (12a)$$

$$\lambda = \frac{\left[C_1\left(1-e^{-C_2\lambda}\right) - C_3\lambda\right]e^{-C_4 V_x}}{\mu V_x} \qquad (12b)$$

Where

C_1 is the maximum value of friction curve; C_2 the friction curve shapes;

C_3 the friction curve difference between the maximum value and the value at = 1; and

C_4 is the wetness characteristic value. It lies in the range 0.02–0.04s/m. Where for dry asphalt as the surface condition, above parameters are

From studies it is shown that the effective coefficient of friction between the tire and the road has an optimum value at particular value of wheel slip ratio. Road type determine the value. Hence, frictional coefficient value is optimum for most roads, when the wheel slip ratio is approximately 0.2 and worst when the wheel slip ratio is 1. That is at the point when wheel is locked. Hence, what ABS does is that it tends to control or regulate the wheel slip ratio from 1 to 0.2 to maximize the frictional coefficient (μ) for any given road surface.

3. Feedback Control System

In order to regulate the brake system, there is need for a sensor that will monitor the slip ratio and sends signal to the controller which regulates the brake pressure modulator to maintain the correct slip ratio with reference to slip ratio value. Fig 2 shows a diagram of feedback control system.

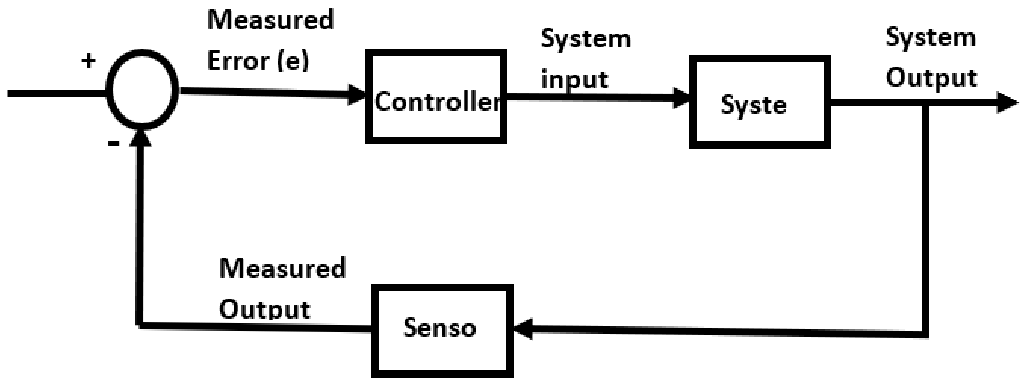


Fig 2 Diagram of Feedback Control System

The feedback control was performed under different conditions. These includes;

3.1. Proportional Feedback Control.

Where input to the system which is in proportion to measured error (e) between the output and the set-point.

Here control torque is

$$\mu = k_p e \qquad (13)$$

Where k_p is known as the proportional gain of the controller.

$$e = \lambda_d - \lambda \qquad (14)$$

where λ_d is desired output and is actual output measured by sensor Proportional Derivative Feedback Control (PD-type)

This controller feeds both the error with constant gain (k_p) and the differentiation of error with constant gain (k_d) to the system in order to maintain the output of system at the set point.

$$\mu = k_p e + k_d \frac{de}{dt} \qquad (15)$$

Where k_d is differential gain of the controller

3.2. Proportional Integral Feedback Control.

Here input to the system is the error with constant gain (k_p) plus the integral of error with constant gain (k_p) to control the system output.

$$\mu = k_p e + k_i \int edt \qquad (16)$$

Where K_i is integral gain of the controller.

3.3 Proportional Integral Derivative Feedback Control.

Here the input is the addition of error with constant gain (k_p), integral of error with constant gain (k_i), and differential of error with constant gain (k_d).

$$\mu = k_p e + k_i \int edt + k_d \frac{de}{dt} \qquad (17)$$

Simulink software was used to determine the ABS braking system with control system having the following parameters; k_p, k_i *and* k_d.

4. SIMULUATION MODELS

Simulink model of quarter vehicle

The quarter vehicle of ABS is modelled in Simulink environment incorporating the dynamic equations. Fig 4 shows the diagram of the Simulink model representing vehicle dynamics during straight line braking.

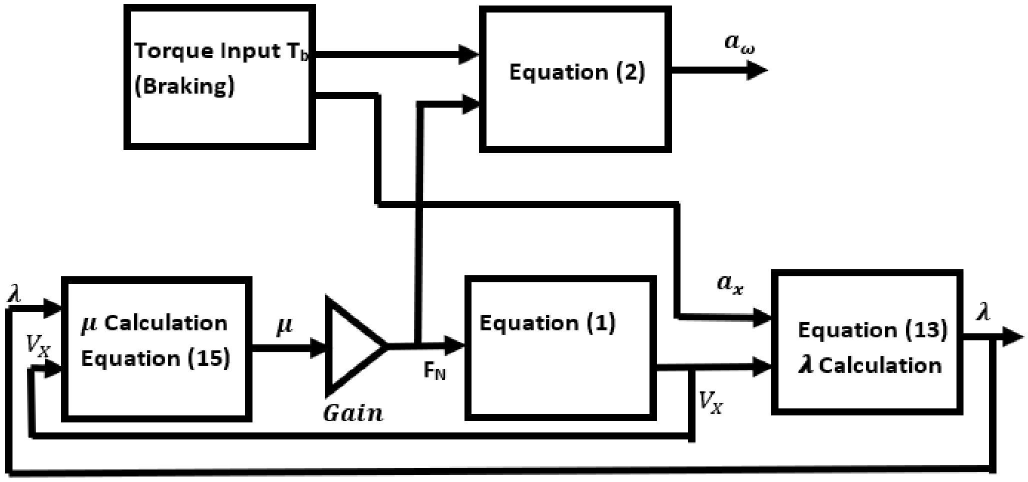


Fig 3 Diagram Representing Dynamics of Equations

Simulink model of quarter vehicle during straight line braking without feedback control was obtained by the combination of the slip ratio and coefficient of friction.

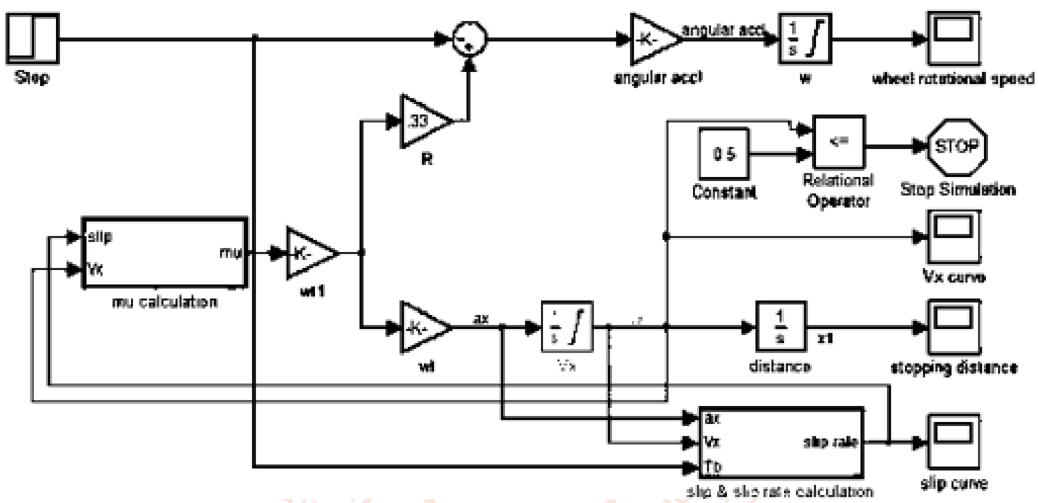


Fig 4 Vehicle Model Without Feedback Control

The above feedback is modified to include SUM box between input terminal, which is control torque u and brake torque T_b.

Hence total torque input T to wheel is $T = \mu + T_b$ This results in figure 5

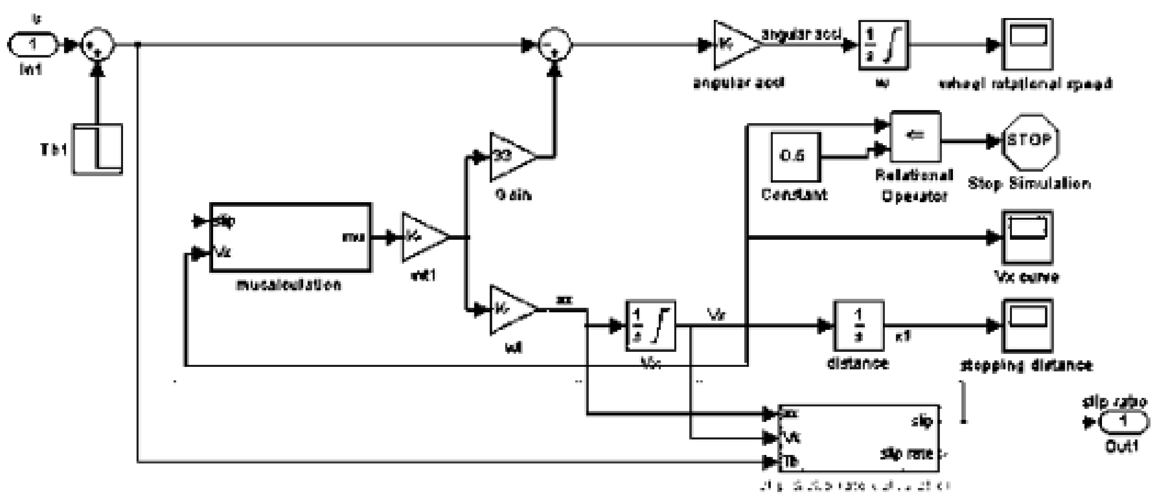


Fig 5 Modified Vehicle Model Without Feedback Control

With figure 4 and 5 we can have a simplified feedback control system with input and output as shown in figure 6.

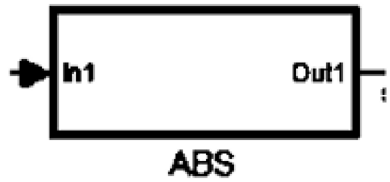


Fig 6. A simplified feedback control system

Integrating proportional feedback control with proportional gain K_p gives Fig 7.

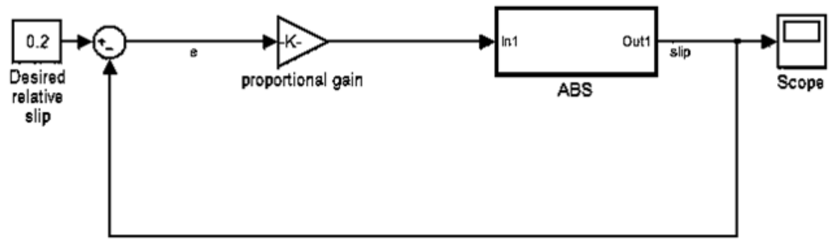


Fig 7. Integrating proportional feedback control

Also, the system is fed with proportional deferential feedback control. Where K_p is proportional gain and K_d is differential gain. This system is shown in Fig 8.

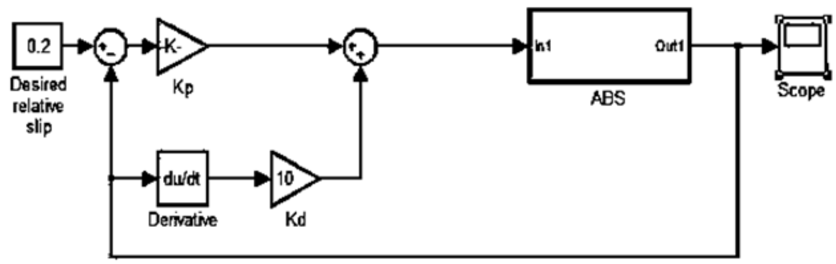


Fig 8 Proportional deferential feedback control

ABS System is fed with proportional integral feedback control where K_p is proportional gain and K_i is integral gain. This system is shown in Fig 9.

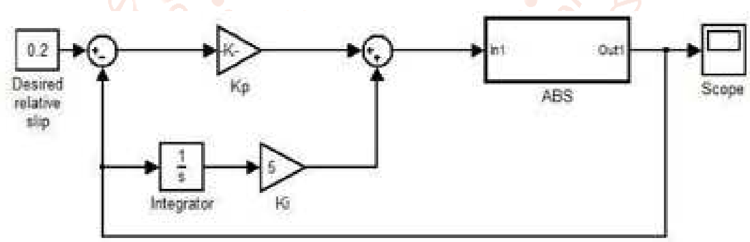


Fig 9 proportional integral feedback control

Simulink model of ABS using proportional integral deferential feedback control system where $k_p,$ is proportional gain k_d is differential gain and k_i . is integral gain are used. This system is shown in Fig 10.

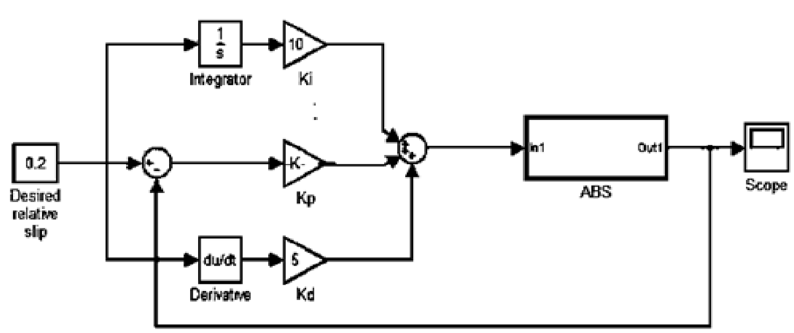


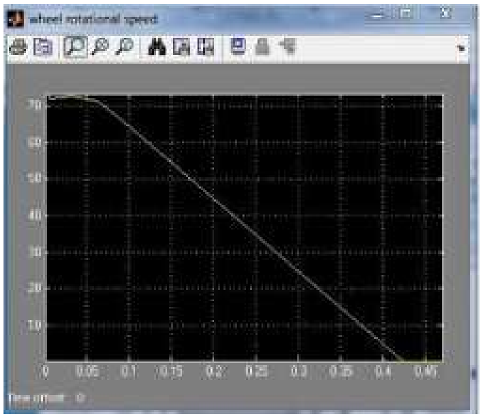
Fig 10 proportional integral deferential feedback control

5. RESULTS

Various vehicle performance in straight line braking conditions were simulated under different parameters, i.e with or without feedback.

The input parameters used are; r = 5.0 m, m =568 kg, J_w =1.13 kgm^2, g =9.81 m/s^2, Max braking torque = 2000Nm, Initial linear velocity = 33.33m/s = 120 km/h, Initial rotational speed = 33.33/5.0 = 6.67 rad/s, λ_d = 0.2, k_p= 250, k_a. = 5, K_i = 10.

The behavior of vehicle parameters during straight line braking without any controller. Figure 11 a and b shows plot of vehicle angular velocity, stopping distance.



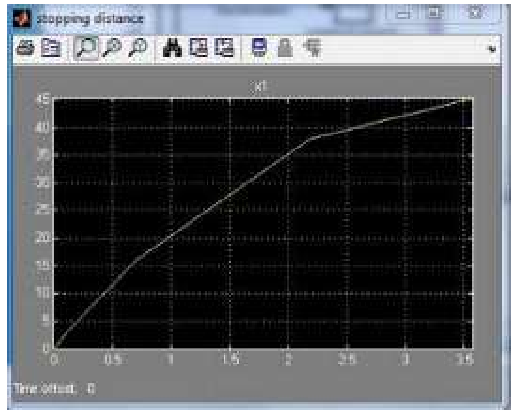
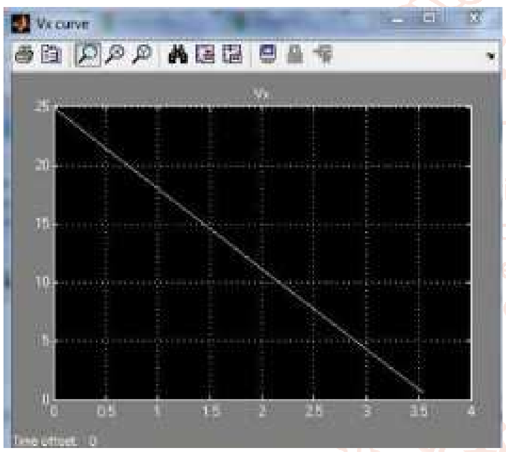


Fig. 11 a) angular speed against time; b) stopping distance against time

Figure 12 a and b shows vehicle linear velocity and slip ratio respectively versus time.



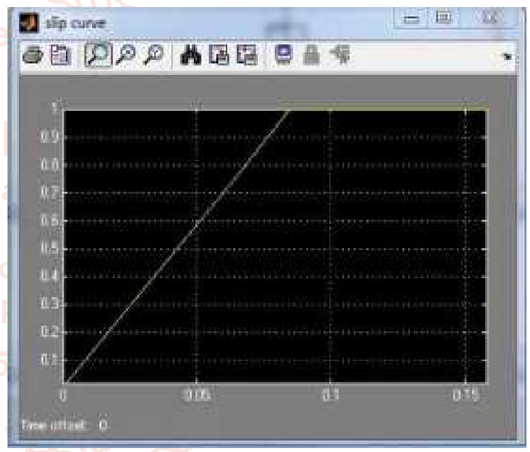
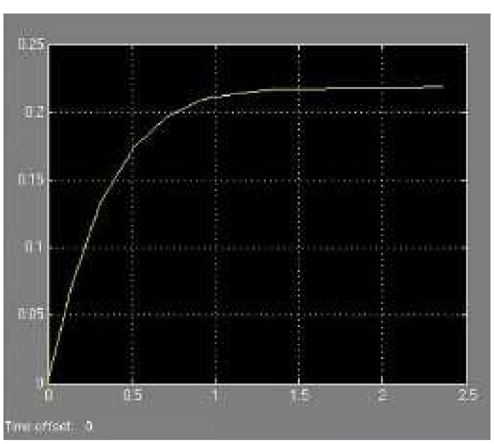


Fig 12 a) linear velocity against time; 12 b) slip ratio against time

From the plots it is observed that vehicle with its slip ratio and coefficient of friction not monitored with ABS, moving at a speed 70m/s stopped in 3.6 secs, at a stopping distance of 45m indicating wheel lock before vehicle comes to stop. The slip ratio has been varied between 0 to 1 on application of brakes to the wheels. It is also observed that braking steerability is lost at 0.42 seconds due to locking of wheel.

To maintain constant slip ratio a feedback control is incorporated in the system. simple linear model called Proportional Feedback Control with a constant gain k_p is first incorporated.



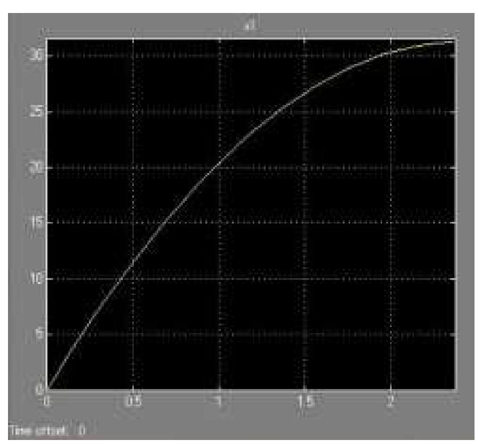


Fig 13 a) slip ratio v/s time (k_p = 250) 13b) stopping distance v/s time

From figure 13, the stopping distance was reduced 33 m because a P – controller supplies a control force and maintain slip ratio with 0.01.

When proportional derivative (PD type) feedback control incorporated, the stopping distance was further reduced to 32 m at stopping y=time of 24seconds as shown figure 14.

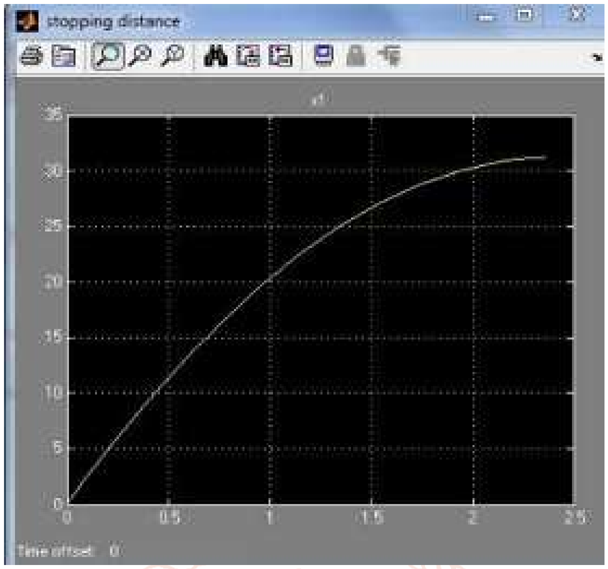


Fig 14 stopping distance v/s time

Finally, when proportional integral derivative control was incorporated, the plots shows that both stopping time and stopping distance are slightly reduced. Stopping time is 2.3 seconds and stopping distance is 31 m.

DISCUSSION

From figure 12-14, it is observed that ABS improve efficiency of the braking system. The graph of slip ratio against time for different control feedback indicate that a Proportional Feedback Control will reduce the braking time. Proportional Integral Feedback Control will eliminate the steady state error, but it may make the transient response worse. Proportional Integral Derivative Feedback Control will increase the stability of the system, reduce the overshoot, and improve the transient response. The results of the three-feedback control system on a closed-loop system are summarized in the table 1.

Table 3.1 Braking Performance Results

ABS Controller	Stopping distance (meters)	Stopping time (meters)
Without controller	45	3.6
Proportional Feedback Control	33	3
Proportional Integral Feedback Control	32	2.4
Proportional Integral Derivative Feedback Control	31.2	2.3

6. CONCLUSION

The work has shown the results of different application of various type of feedback controller used for antilock braking systems. The modeled quarter vehicle dynamics and differential equation of motion was formulated were applied using feedback control system. The slip ratio evaluation was the major aspect of the work as frictional force and normal reaction are function of slip ratio.

Three state space equations were used to obtain the block diagram representing dynamics of equations that was applied in the Simulink software that generated the result of the stopping distance, stopping time and slip factor variation. To achieve this, different feedback control like - proportional feedback control, proportional integral feedback control, proportional integral derivative feedback control was implemented and the result obtained shows that feedback control increases the stability of the system, reduce the overshoot, and improve the transient response.

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