MODELLING & DEVELOPMENT OF ANTILOCK BRAKING SYSTEM

Thesis submitted in partial fulfillment of the requirements for the degree of

Bachelor of Technology (B. Tech)

In

Mechanical Engineering

By

PARTH BHARAT BHIVATE

Roll No: - 107ME051

NATIONAL INSTITUTE OF TECHNOLOGY
ROURKELA 769008, INDIA

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Certificate of Approval

This is to certify that the thesis entitled **MODELLING & DEVELOPMENT OF ANTILOCK BRAKING SYSTEM** submitted by **Sri Parth Bharat Bhivate** (Roll No. 107ME051) has been carried out under my supervision in partial fulfillment of the requirements for the Degree of Bachelor of Technology (B. Tech.) in Mechanical Engineering at National Institute of Technology, NIT Rourkela, and this work has not been submitted elsewhere before for any other academic degree/diploma.

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CONTENTS

ABSTARCT

Antilock braking systems are used in modern cars 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 is represented in terms of vehicle speed and wheel rotation.

In present work first of all system dynamic equations are explained and a slip ratio is expressed in terms of system variables namely vehicle linear velocity and angular velocity of the wheel. By applying a bias braking force system, response is obtained using Simulink models. Using the linear control strategies like P - type, PD - type, PI - type, PID - type the effectiveness of maintaining desired slip ratio is tested. It is always observed that a steady state error of 10% occurring in all the control system models.

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CHAPTER 1

INTRODUCTION

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 modulates 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 has 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.

1.1 IMPORTANCE OF ANTILOCK BRAKING SYSTEMS

The objectives of antilock systems are threefold:

1. to reduce stopping distances,
2. to improve stability, and
3. to improve steerability during braking.

These are explained below

Stopping Distance The distance to stop is a function of the mass of the vehicle, the initial velocity, and the braking force. By maximizing the braking force the stopping distance will be minimized if all other factors remain constant. However, on all types of surfaces, to a greater or lesser extent, there exists a peak in fiction coefficient. It follows that by keeping all of the wheels of a vehicle near the peak, an antilock system can attain maximum fictional force and, therefore, minimum stopping distance. This objective of antilock systems however, is tempered by the need for vehicle stability and steerability.

Stability Although decelerating and stopping vehicles constitutes a fundamental purpose of braking systems, maximum friction force may not be desirable in all cases, for example not if the vehicle is on a so-called p-split surface (asphalt and ice, for example), such that significantly more braking force is obtainable on one side of the vehicle than on the other side. Applying maximum braking force on both sides will result in a yaw moment that will tend to pull the vehicle to the high friction side and contribute to vehicle instability, and forces the operator to make excessive steering corrections to counteract the yaw moment. If an antilock system can maintain the slip of both rear wheels at the level where the lower of the two friction coefficients peaks, then lateral force is reasonably high, though not maximized. This contributes to stability and is an objective of antilock systems.

Steerability Good peak frictional force control is necessary in order to achieve satisfactory lateral forces and, therefore, satisfactory steerability. Steerability while braking is important not only for minor course corrections but also for the possibility of steering around an obstacle.

Tire characteristics play an important role in the braking and steering response of a vehicle. For ABS-equipped vehicles the tire performance is of critical significance. All braking and steering forces must be generated within the small tire contact patch between the vehicle and the road. Tire traction forces as well as side forces can only be produced when a difference exists between the speed of the tire circumference and the speed of the vehicle relative to the road surface. This difference is denoted as slip. It is common to relate the tire braking force to the tire braking slip. After the peak value has been reached, increased tire slip causes reduction of tire-road friction coefficient. ABS has to limit the slip to values below the peak value to prevent wheel from locking. Tires with a high peak friction point achieve maximum friction at 10 to 20% slip. The optimum slip value decreases as tire-road friction decreases.

The ABS system consists of the following major subsystems:

Wheel-Speed Sensors

Electro-magnetic or Hall-effect pulse pickups with toothed wheels mounted directly on the rotating components of the drivetrain or wheel hubs. As the wheel turns the toothed wheel (pulse ring) generates an AC voltage at the wheel-speed sensor. The voltage frequency is directly proportional to the wheel's rotational speed.

Electronic Control Unit (ECU)

The electronic control unit receives, amplifies and filters the sensor signals for calculating the wheel rotational speed and acceleration. This unit also uses the speeds of two diagonally opposed wheels to calculate an estimate for the speed of the vehicle. The slip at each wheel is derived by comparing this reference speed with the speeds of the individual wheels. The "wheel acceleration" and "wheel slip" signals serve to alert the ECU to any locking tendency. The microcomputers respond to such an alert by sending a signal to trigger the pressure control valve solenoids of the pressure modulator to modulate the brake pressure in the individual wheel-brake cylinders. The ECU also incorporates a number of features for error recognition for the entire ABS system (wheel-speed sensors, the ECU itself, pressure-control valves, wiring harness). The ECU reacts to a recognized defect or error by switching off the malfunctioning part of the system or shutting down the entire ABS.

Hydraulic Pressure Modulator

The hydraulic pressure modulator is an electro-hydraulic device for reducing, holding, and restoring the pressure of the wheel brakes by manipulating the solenoid valves in the hydraulic brake system. It forms the hydraulic link between the brake master cylinder and the wheel-brake cylinders. The hydraulic modulator is mounted in the engine compartment to minimize the length of the lines to the brake master cylinder and the wheel-brake cylinders. Depending on the design, this device may include a pump, motor assembly, accumulator and reservoir. Fig 1 shows relationship between modulator, dynamics and controller.

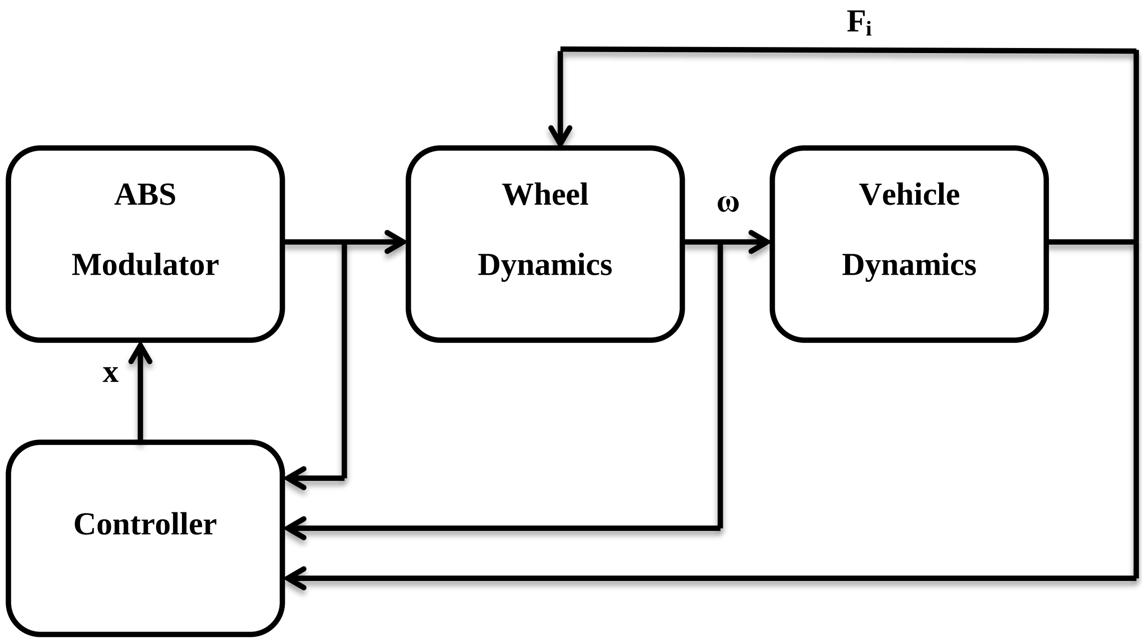

Fig 1.1 Scheme of ABS

Following brakes are generally used in automobiles.

In disk brake, a force is applied to both sides of a rotor and braking action is achieved through the frictional action of inboard and outboard brake pads against the rotor.

In drum brakes, a force is applied to a pair of brake shoes. A variety of configurations exists, including 1eading, trailing shoe (simplex), duo-duplex, and duo-servo. Drum brakes feature high gains compared to disk brakes, but some configurations tend to be more nonlinear and sensitive to fading.

1.2 LITERATURE REVIEW

Following literature is surveyed relating to ABS.

Mirzaeinejad and Mirzaei [1] have applied a predictive approach to design a non- linear model-based controller for the wheel slip. The integral feedback technique is also employed to increase the robustness of the designed controller. Therefore, the control law is developed by minimizing the difference between the predicted and desired responses of the wheel slip and it’s integral.

Baslamisli*et al.* [2] proposed a static-state feedback control algorithm for ABS control. The robustness of the controller against model uncertainties such as tire longitudinal force and road adhesion coefficient has been guaranteed through the satisfaction of a set of linear matrix inequalities. Robustness of the controller against actuator time delays along with a method for tuning controller gains has been addressed. Further tuning strategies have been given through a general robustness analysis, where especially the design conflict imposed by noise rejection and actuator time delay has been addressed.

Choi [3] has developed a new continuous wheel slip ABS algorithm. here ABS algorithm, rule-based control of wheel velocity is reduced to the minimum. Rear wheels cycles independently through pressure apply, hold, and dump modes, but the cycling is done by continuous feedback control. While cycling rear wheel speeds, the wheel peak slips that maximize tire-to-road friction are estimated. From the estimated peak slips, reference velocities of front wheels are calculated. The front wheels are controlled continuously to track the reference velocities. By the continuous tracking control of front wheels without cycling, braking performance is maximized.

Rangelov [4] described the model of a quarter-vehicle and an ABS in MATLAB-SIMULINK. In this report, to model the tire characteristics and the dynamic behavior on a flat as well as an uneven road, the SWIFT-tire model is employed.

Sharkawy [5] studied the performance of ABS with variation of weight, friction coefficient of road, road inclination etc. A self-tuning PID control scheme to overcome these effects via fuzzy GA is developed; with a control objective to minimize stopping distance while keeping slip ratio of the tires within the desired range.

Poursmad [6] has proposed an adaptive NN- based controller for ABS. The proposed controller is designed to tackle the drawbacks of feedback linearization controller for ABS.

Topalov*et al.* [7] proposed a neurofuzzy adaptive control approach for nonlinear system with model uncertainties, in antilock braking systems. The control scheme consists of PD controller and an inverse reference model of the response of controlled system. Its output is used as an error signal by an online algorithm to update the parameters of a neuro-fuzzy feedback controller.

Patil and Longoria[8] have used decoupling feature in frictional disk brake mechanism derived through kinematic analysis of ABS to specify reference braking torque is presented. Modelling of ABS actuator and control design are described.

Layne *et al.* [9] have illustrated the fuzzy model reference learning control (FMRLC). Braking effectiveness when there are transition between icy and wet road surfaces is studied.

Huang and Shih [10] have used the fuzzy controller to control the hydraulic modulator and hence the brake pressure. The performance of controller and hydraulic modulator are assessed by the hardware in loop (HIL) experiments.

Onit*et al.* [11] have proposed a novel strategy for the design of sliding mode controller (SMC). As velocity of the vehicle changes, the optimum value of the wheel slip will also alter. Gray predictor is employed to anticipate the future output of the system.

1.3 SCOPE & OBJECTIVE OF PRESENT WORK

During the design of ABS, nonlinear vehicle dynamics and unknown environment characters as well as parameters, change due to mechanical wear have to be considered. PID controller are very easy to understand and easy to implement. However PID loop require continuous monitoring and adjustments. In this line there is a scope to understand improved PID controllers with mathematical models.

The present work, it is planned to understand and obtain the dynamic solution of quarter car vehicle model to obtain the time varying vehicle velocity and wheel. After identification of system dynamics a slip factor defined at each instance of time will be modified to desired value by means of a control scheme. Various feedback control schemes can be used for this purpose. Simulation are carried out to achieve a desired slip factor with different control scheme such as

1) Proportional Feedback control
2) Proportional Derivative Feedback Control
3) Proportional Integral Feedback Control
4) Proportional Integral Derivative Feedback Control

Graphs of linear velocity, stopping distance and slip ratio for each system is plotted and compared with each other. At the end, possible alternate solutions are discussed.

The work is inspired from the demo model of ABS provided in Simulink software.

1.4 ORGANISATION OF THESIS

Chapter 2 describes the mathematical modelling of quarter vehicle and vehicle dynamic equations used to describe the system. Feedback control systems which are used for ABS are explained. Simulink Models of each control system are described.

Chapter 3 contains various graphs obtained from each of Simulink models. Comparison and discussion between different control schemes are shown.

Chapter 4 concludes the above work. It contains summary of work and throws light on future scope for further studies and development.

CHAPTER 2

MATHEMATICAL MODELLING

2.1 VEHICLE DYNAMICS

Basically, a complete vehicle model that includes all relevant characteristics of the vehicle is too complicated for use in the control system design. Therefore, for simplification 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.

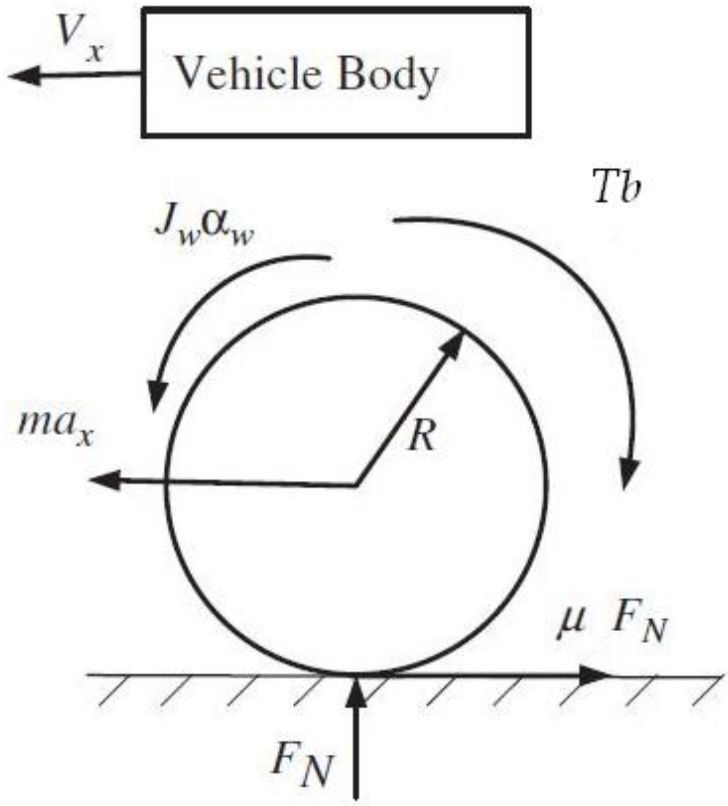

Fig 2.1 Quarter Vehicle Model

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)

$$m\, a_x = -\mu\, F_N \implies m \frac{d\,V_x}{d\,t} = -\mu\, F_N \qquad (1)$$

Summing torque at wheel centre (wheel)

$$J_\omega\, \alpha_\omega = \mu\, R\, F_N - T_b \implies J_\omega\, \dot{\omega} = \mu\, R\, F_N - T_b \qquad (2)$$

For convenience a slip ratio is defined according to:

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

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

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

The nomenclature in above equations is presented as follows

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

State space representation of above equation is presented below. During braking, the slip ratio is dependent on the input torque u and the vehicle velocity V_x. The system state variables are:

$$x_1 = S_x, \tag{5a}$$

$$x_2 = V_x, \tag{5b}$$

$$x_3 = \lambda, \tag{5c}$$

where S_x is the stopping distance. The state space equations are

$$\dot{x}_1 = x_2 \tag{6}$$

$$\dot{x}_2 = \frac{-\mu F_N}{m} \tag{7}$$

$$\dot{x}_3 = \frac{-\mu F_N}{x_2} \left(\frac{1-x_3}{m} + \frac{R^2}{J_w}\right) + \frac{R}{J_w x_2} T_b \tag{8}$$

By controlling the braking torque u in the simulation tests to evaluate the performance of ABS, using different control strategies.

2.2 PROBLEM FORMULATION

The relation of the frictional coefficient μ versus wheel slip ratio λ, provides the explanation of the ability of the ABS to maintain vehicle steerability and stability, and still produce shorter stopping distances than those of locked wheel stop. The friction coefficient can vary in a very wide range, depending on factors like:

(a) Road surface conditions (dry or wet),

(b) Tire side-slip angle,

(c) Tire brand (summer tire, winter tire),

(d) Vehicle speed, and

(e) The slip ratio between the tire and the road.

Friction model used in [5] is used here. It gives value of coefficient of friction as a function of linear velocity and slip ratio.

$$\mu\ (\lambda, V_x) = [c_1 (1- e^{-c_2 \lambda}) - c_3 \lambda]\ e^{-c_4 V_x} \qquad (9)$$

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 $\lambda = 1$; 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

C_1= 1.2801, C_2 = 23.99, C_3=0.52, C_4= .03 (assumed)

The effective coefficient of friction between the tire and the road has an optimum value at particular value of wheel slip ratio. This value differs according to the road type. From Fig 2.2 it is clear that, for almost all road surfaces the frictional coefficient value is optimum when the wheel slip ratio is approximately 0.2 and worst when the wheel slip ratio is 1 in other words when wheel is locked. So, objective of ABS controller is to regulate the wheel slip ratio (λ) to target value of 0.2 to maximize the frictional coefficient (μ) for any given road surface.

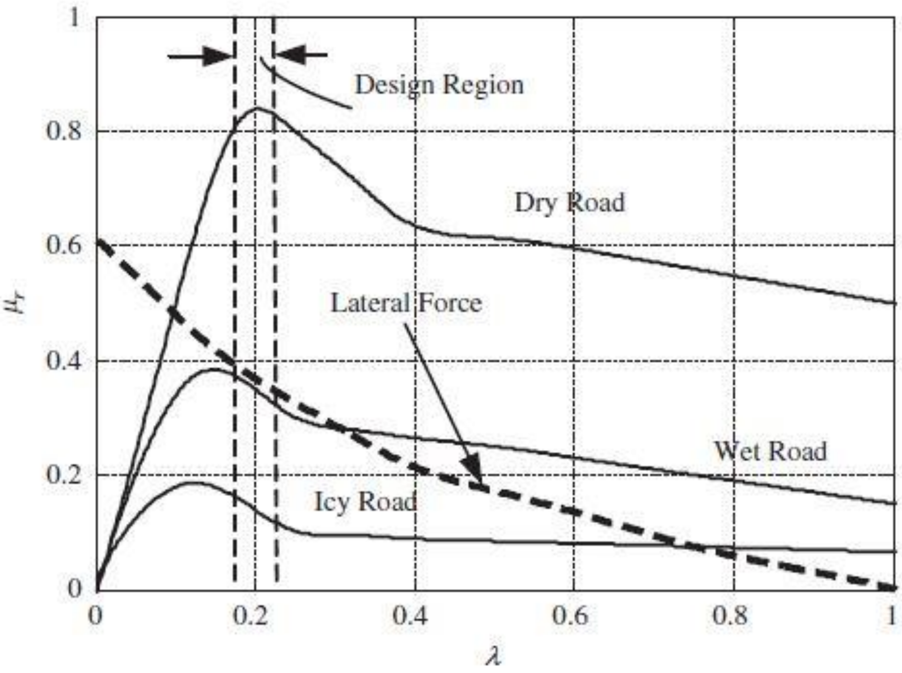

Fig 2.2 Frictional Coefficient of Road Surface v/s Wheel Slip Ratio

2.3 CONTROL SYSTEM

A feedback control system is a closed loop control system in which a sensor monitors the output (slip ratio) and feeds data to the controller which adjusts the control (brake pressure modulator) as necessary to maintain the desired system output (match the wheel slip ratio to the reference value of slip ratio).

Fig 2.3 shows the block diagram of feedback control system

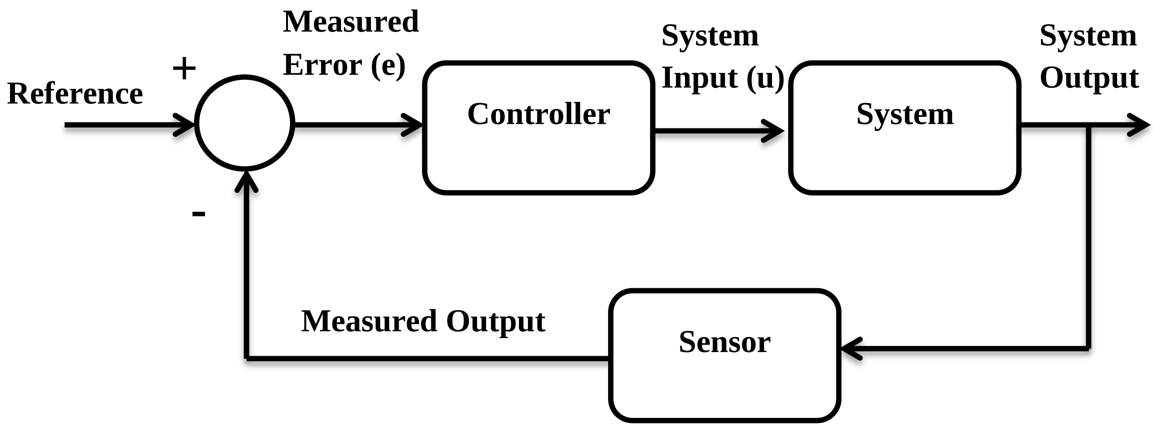

Fig 2.3 Block Diagram of Feedback Control System

This feedback controller can be any one of

1) Proportional Control
2) Proportional Derivative Control
3) Proportional Integral Control
4) Proportional Integral Derivative Control

Proportional Feedback Control (P-type)

A proportional controller attempts to control the output by applying input to the system which is in proportion to measured error (e) between the output and the set-point. Here control torque is

$$u = K_p e \tag{10}$$

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

$$e = \lambda_d - \lambda \tag{11}$$

where λ_d is desired output and λ 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.

$$u = K_p e + K_d \frac{de}{dt} \tag{12}$$

Where K_d is differential gain of the controller

Proportional Integral Feedback Control(PI-type)

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

$$u = K_p e + K_i \int e\, dt \tag{13}$$

Where K_i is integral gain of the controller.

Proportional Integral Derivative Feedback Control (PID-type)

In PID controller system 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).

$$u = K_p e + K_i \int e \, dt + K_d \frac{de}{dt} \qquad (14)$$

To design ABS with control systems mentioned above the values of K_p, K_i and K_d of the controller can be determined by

1) Trial and error,
2) Manual tuning, and
3) Simulation.

Appropriate values of K_p, K_i and K_d are calculated using trial and error method by observing trend of graph of slip ratio versus time obtained using Simulink software tool.

2.4 SIMULINK MODELS

Simulink model of quarter vehicle

In order to model the ABS with different controllers system incorporating the dynamic equations is modelled in Simulink environment. Fig 2.4 shows the block diagram of the Simulink model representing vehicle dynamics during straight line braking.

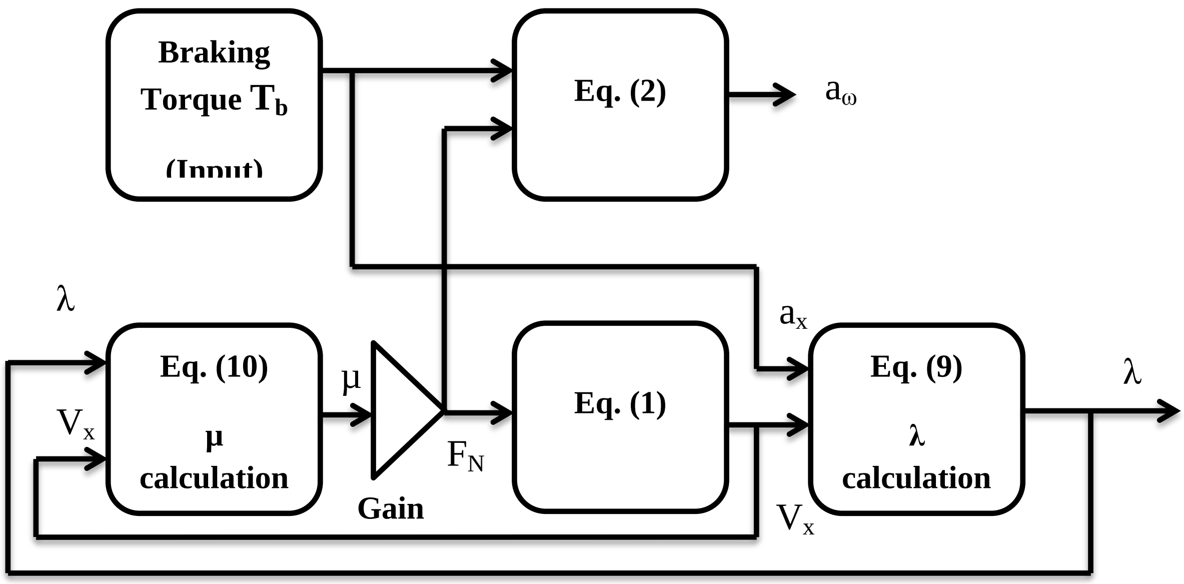

Fig 2.4 Block Diagram Representing Dynamics of Equations

To model this system in Simulink, several subgroups are used to avoid confusion.

Slip ratio λ calculation given in Eq. (9) can be formed as a subgroup shown in fig 2.5

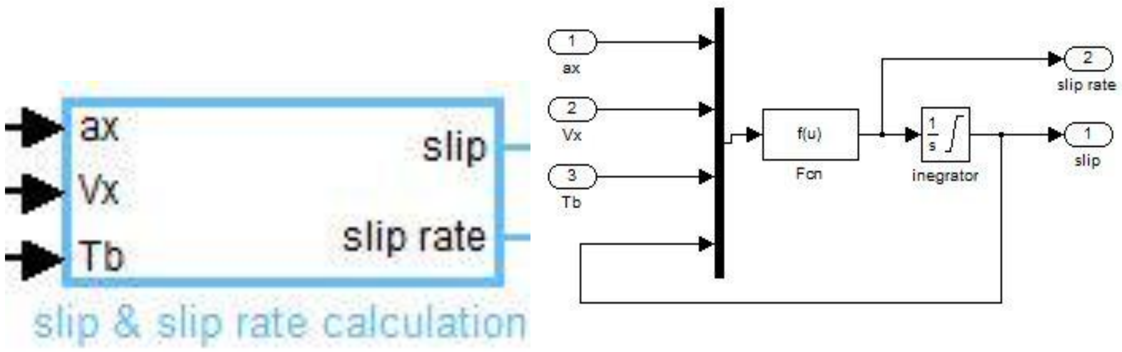

Fig 2.5 Subgroup of Slip Ratio Calculation

Similarly friction coefficient (µ) calculation can be formed in one subgroup

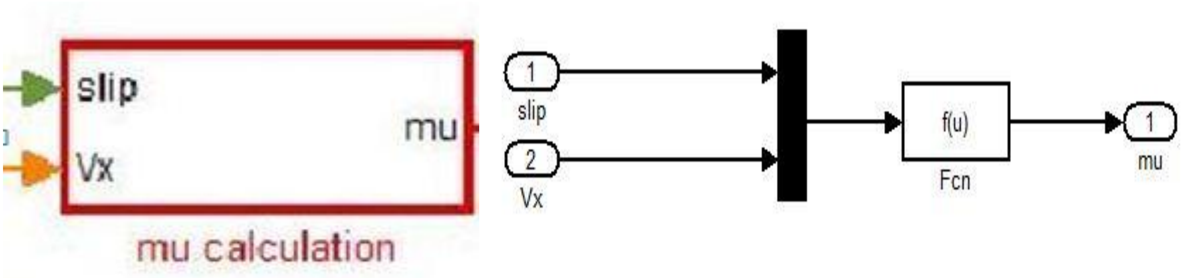

Fig 2.6 Subgroup of µ Calculation

Combining sub groups and modelling remaining equations into Simulink model, we get complete Simulink model of quarter vehicle during straight line braking without feedback control as shown in Fig 2.7

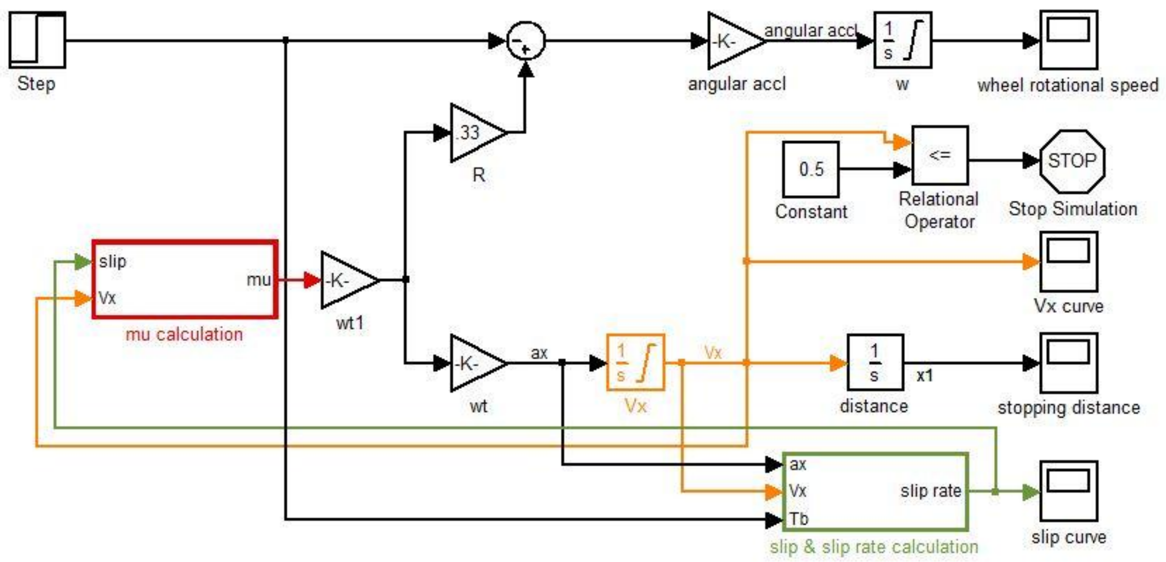

Fig 2.7 Vehicle Model Without Feedback Control

Simulink model of ABS using proportional feedback control

Simulink model shown in Fig 2.7 is modified to use it as a system subgroup in modelling of feedback control system. Fig 2.8 shows the modified version in which a SUM box is added between input terminal (which is control torque u) and brake torque T_b. So the total torque input T to wheel is

$$T = u + T_b \qquad (15)$$

This subgroup formed is shown in Fig 2.9

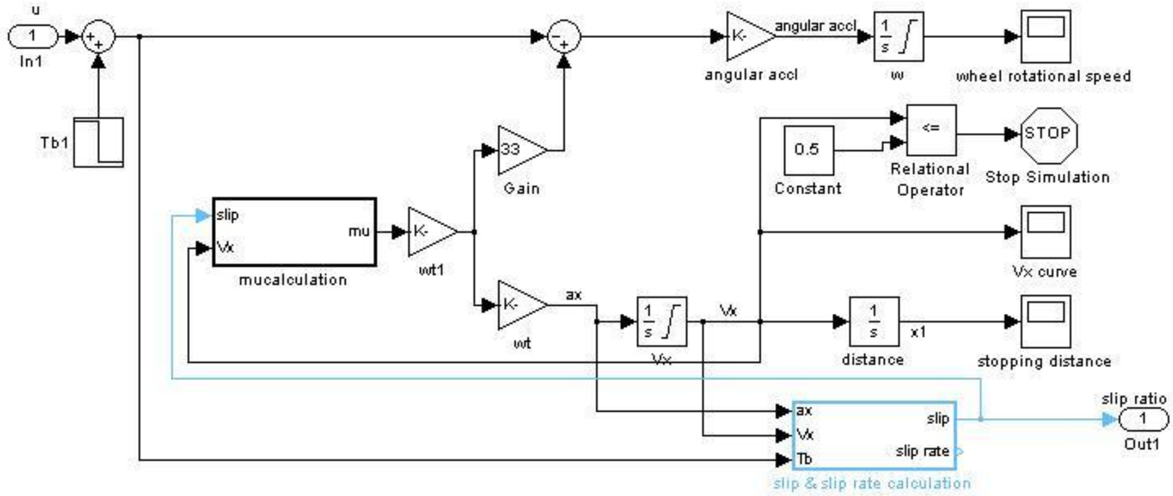

Fig 2.8 Modified Vehicle Model Without Feedback Control

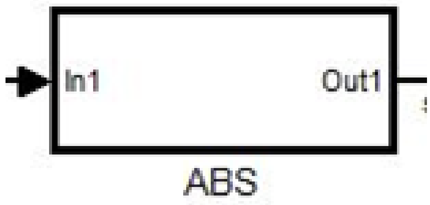

Fig 2.9 Subgroup of System

Newly formed subgroup shown in Fig 2.8 is integrated with proportional feedback control with proportional gain K_p as shown in Fig 2.10

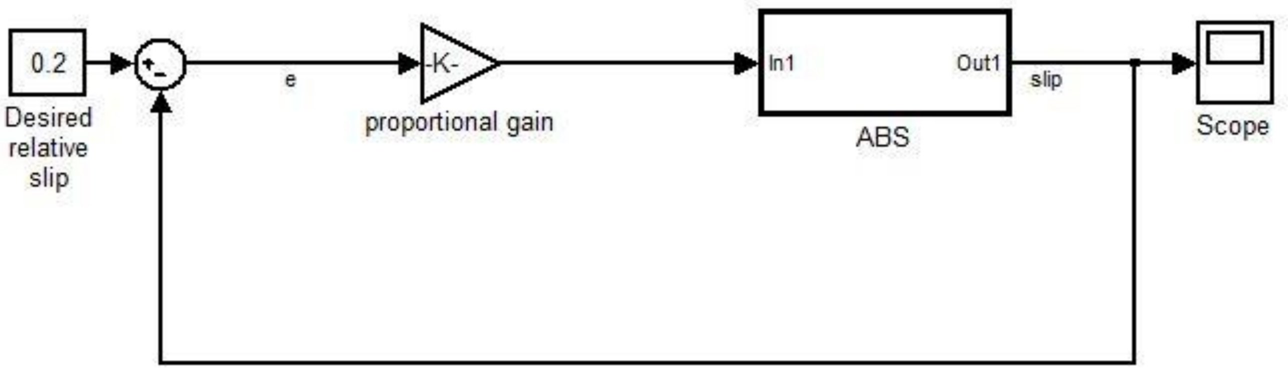

Fig 2.10 P-type Feedback control

Simulink model of ABS using proportional deferential feedback control

In this case 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 2.11

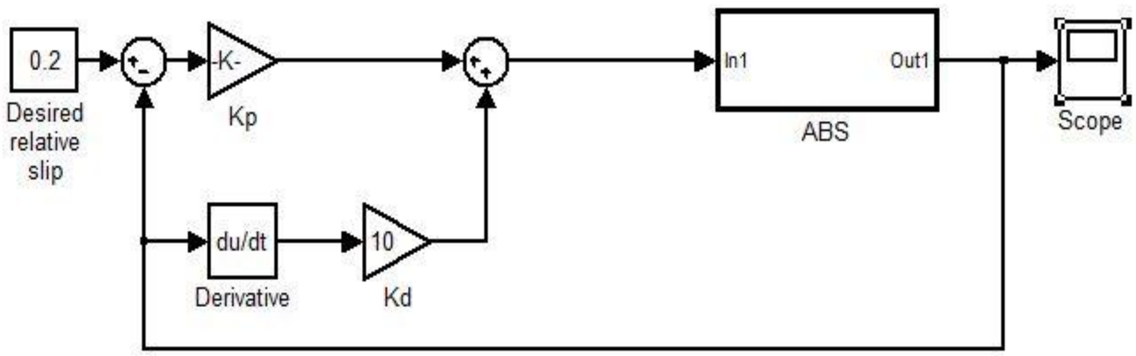

Fig 2.11 PD-type Feedback control

Simulink model of ABS using proportional integral feedback control

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 2.12

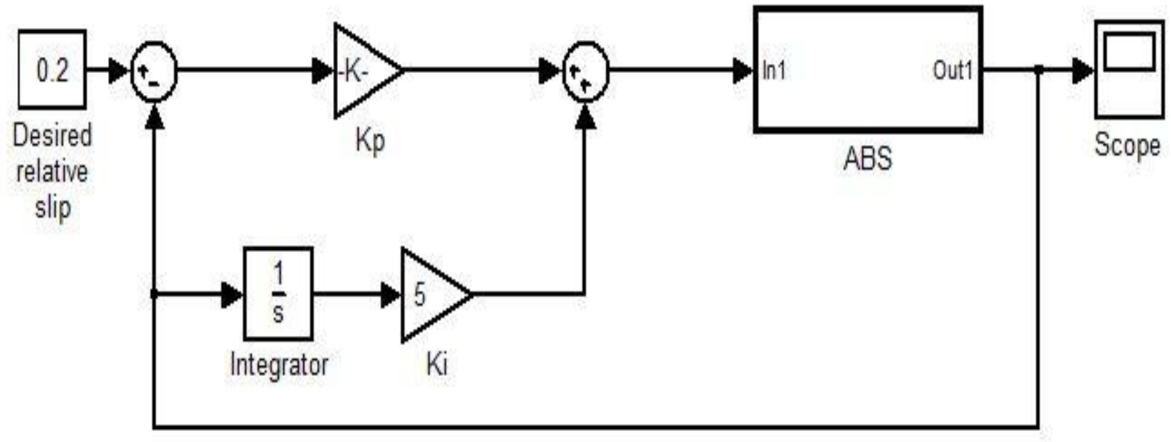

Fig 2.12 PI-type Feedback control

Simulink model of ABS using proportional integral deferential feedback control

By combination of above systems we get PID- type 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 2.13

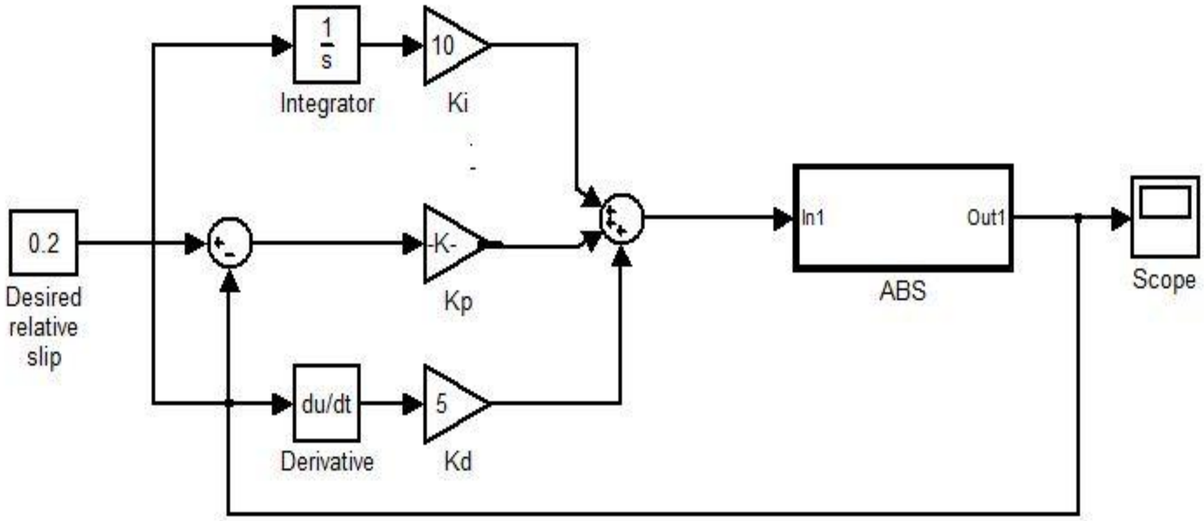

Fig 2.13 PID-type Feedback control

CHAPTER 3

RESULTS & DISCUSSION

This chapter describes the controlled slip response outputs using linear control models.

3.1 INPUT PARAMETERS USED

To simulate the performance of different vehicle parameters with and without any feedback control system under straight line braking following input parameters are considered [5].

R=0.33 m,

m =342 kg,

J_w =1.13 kgm^2,

g =9.81 m/s^2,

Max braking torque = 1200Nm

Initial linear velocity = 27.78m/s = 100 km/h

Initial rotational speed = $\frac{27.78}{.33}$ = 84.18 rad/s

$\lambda_d = 0.2$

$K_p = 250$

$K_d = 5$

$K_i = 10$

3.2 STRAIGHT LINE BRAKING OF VEHICLE WITHOUT FEEDBACK

Fig 3.1 and 3.2 shows the behaviour of vehicle parameters during straight line braking without any controller.

Fig 3.1 a, b and Fig 3.2 a, b are plot of vehicle angular velocity, stopping distance, vehicle linear velocity and slip ratio respectively versus time.

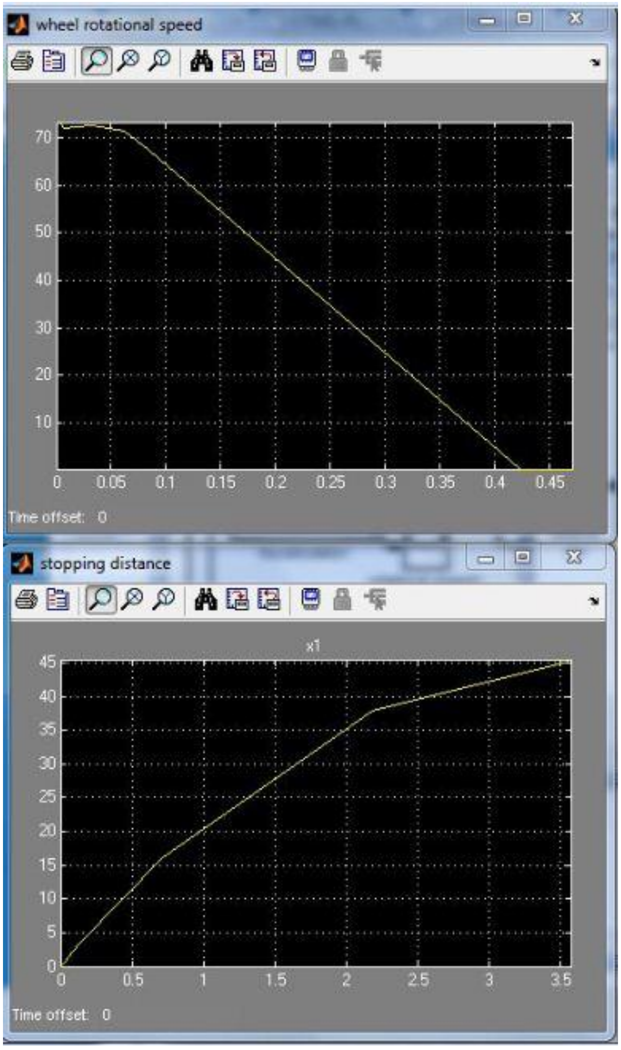

Fig 3.1: a) wheel angular speed v/s time; b) stopping distance v/s time

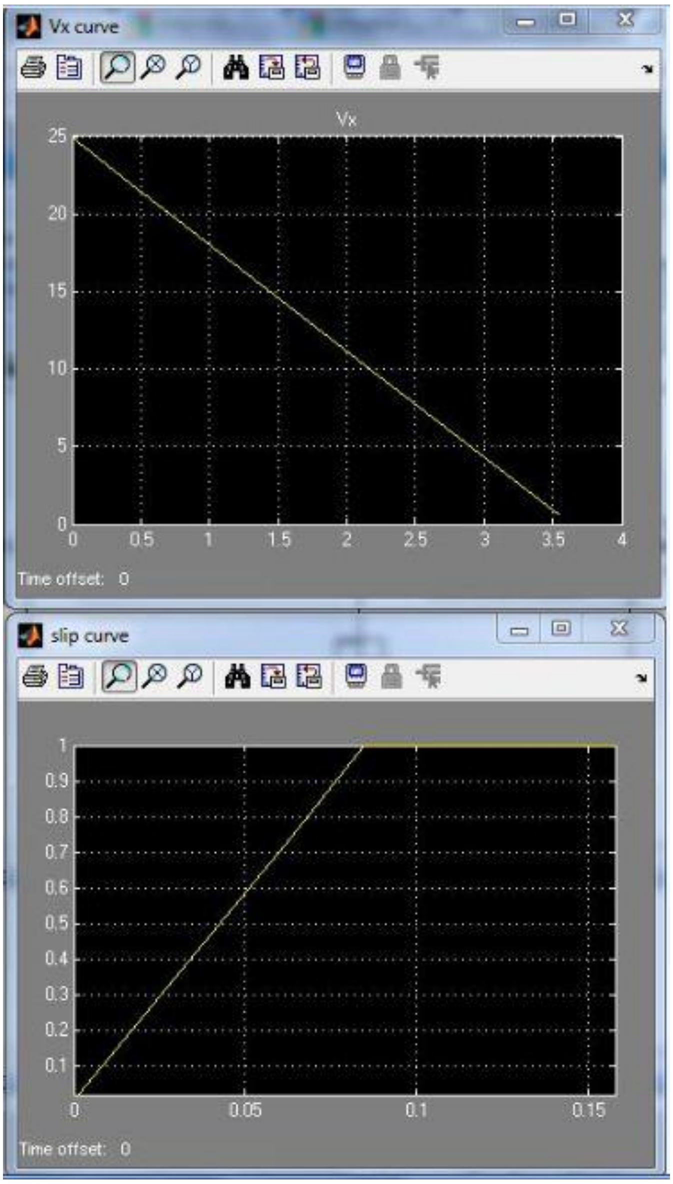

Fig 3.2: a)vehicle linear velocity v/s time; b) slip ratio v/s time

It is seen that slip ratio has been varying from 0 to 1 from application of brakes to the wheel stopping instant. Even the wheel speed is zero at .42 seconds, the stopping distance of 45 m occurs at 3.6 seconds. This indicates that wheel has been locked before vehicle comes to halt. That means during braking steerability is lost at .42 seconds due to locking of wheel

3.3 PROPORTIONAL CONTROL

When feedback control is incorporated in the system to maintain constant slip ratio value, simple linear model called P- control with a constant gain K_p comes first.

Fig 3.3 and Fig 3.4 shows plot of slip ratio versus time and stopping distance versus time respectively.

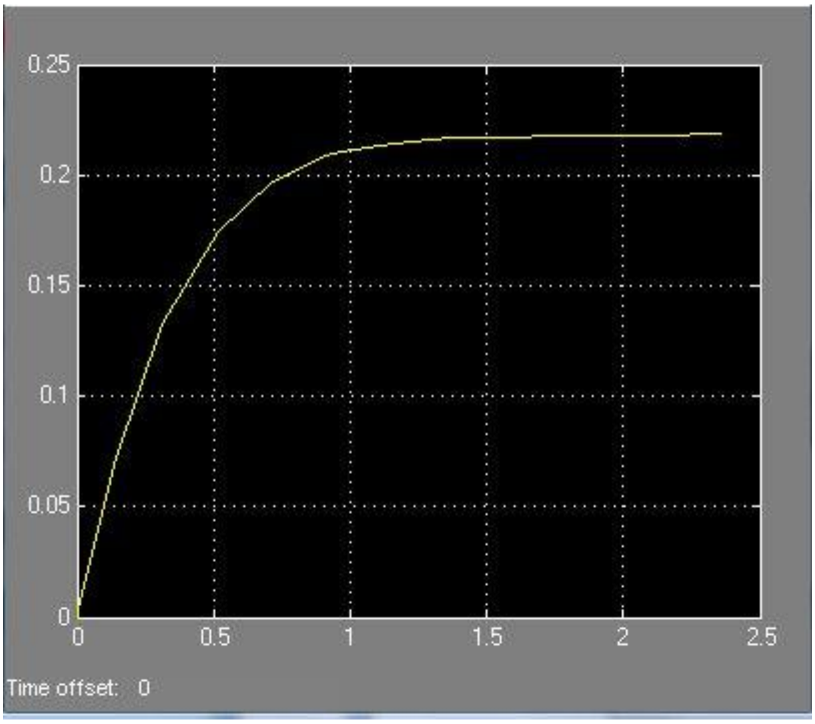

Fig 3.3: slip ratio v/s time (K_p = 250)

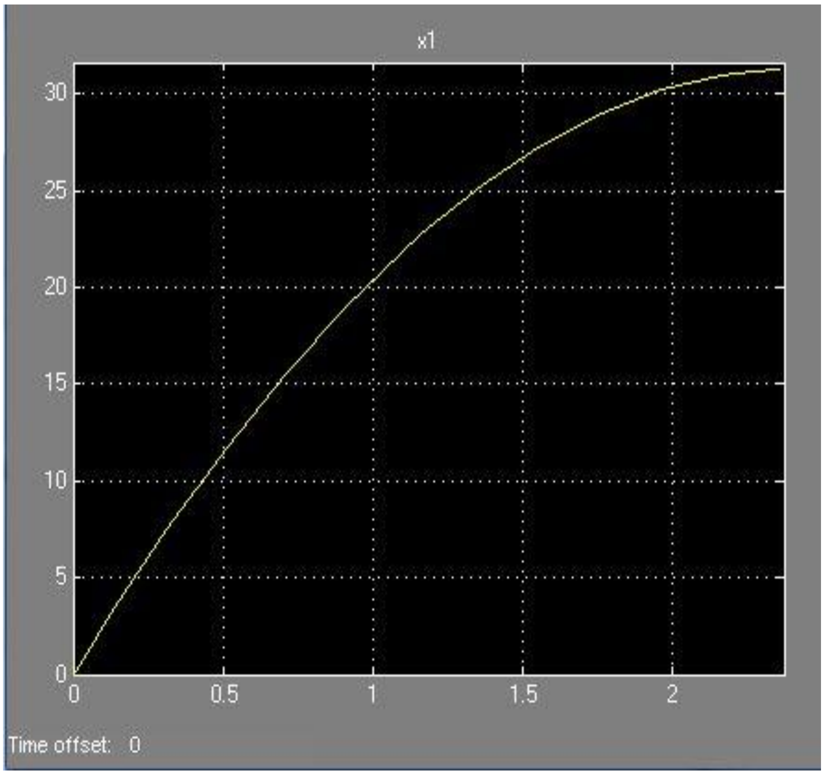

Fig 3.4: stopping distance v/s time

Compare to 45 m stopping distance and increasing slip ratio in open loop case, P – controller supplies a control force and maintain slip ratio with 0.01 steady state error and the stopping distance reduced to 33 m.

3.4 PROPORTIONAL DERIVATIVE CONTROL

For PD type feedback control, plots of slip ratio versus time and stopping distance versus time are obtained. These plots are shown in Fig 3.5 a and b.

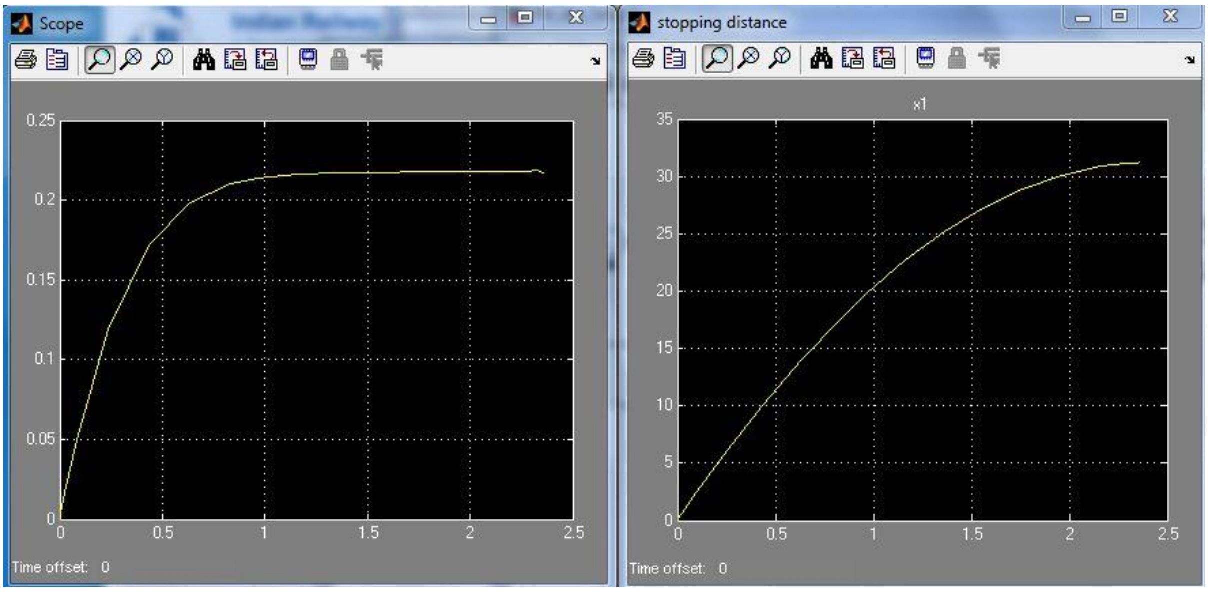

Fig 3.5: a) slip ratio v/s time; b) stopping distance v/s time

As seen it is similar to P type controller. In this case stopping time is 2.4 seconds and stopping distance is 32 m. Stopping time and stopping distance are decreased slightly.

3.5 PROPORTIONAL INTEGRAL CONTROL

In this case also the plots of slip ratio versus time and stopping distance versus time are obtained. These plots are shown in Fig 3.6 a and b

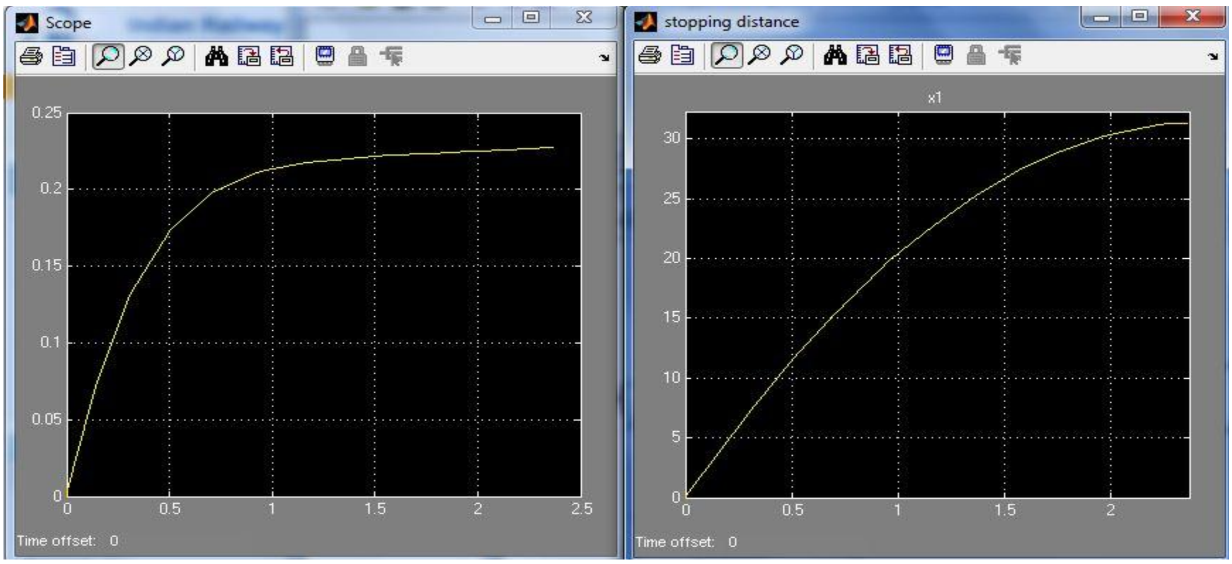

Fig 3.6: a) slip ratio v/s time; b) stopping distance v/s time

There is no much variation. This time stopping distance found to be 33 m and stopping time found to be 2.5 seconds.

3.6 PROPORTIONAL INTEGRAL DERIVATIVE CONTROL

Similarly in case of PID type feedback control plots of slip ratio versus time and stopping distance versus time are obtained.

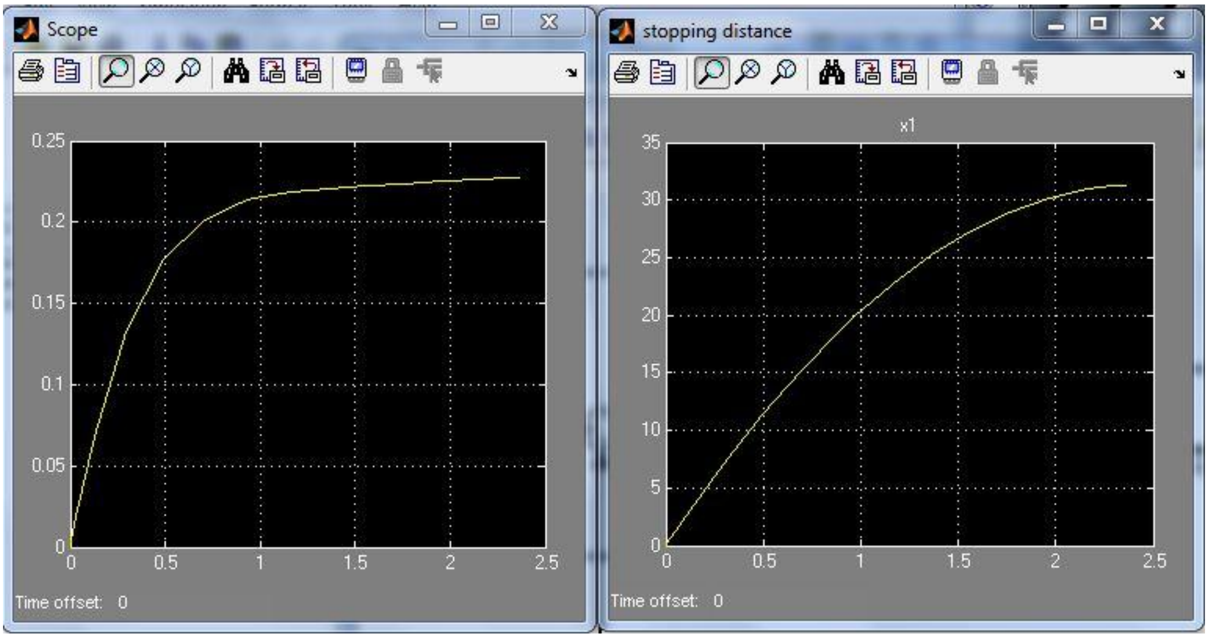

Fig 3.7: a) slip ratio v/s time; b) stopping distance v/s time

Here the stopping time and stopping distance are slightly reduced. Stopping time is 2.3 seconds and stopping distance is 31 m. Overall Comparisons are tabulated in Table 3.1.

Table 3.1 Braking Performance Results

ABS Controller	Stopping time (meters)	Stopping distance (seconds)
Braking Without controller	45	3.6
P-type	33	3
PD-type	32	2.4
PI-type	33	2.5
PID-type	31	2.3

3.7 DISCUSSION

From Table 3.1, it is clear that ABS improves braking performance of vehicle. Comparing slip ratio v/s time graphs of different control schemes suggests that a proportional controller (K_p) will have the effect of reducing the rise time and will reduce but never eliminate the steady-state error. An integral control (K_i) will have the effect of eliminating the steady-state error, but it may make the transient response worse. A derivative control (K_d) will have the effect of increasing the stability of the system, reducing the overshoot, and improving the transient response. Effects of each of controllers K_p, K_d, and K_i on a closed-loop system are summarized in the table shown below.

Table 3.2 General Effects

Gain Response	Rise time	Over shoot	Settling time
K_p	Decrease	Increase	Small Change
K_i	Decrease	Increase	Eliminate
K_d	Small Change	Decrease	Decrease

CHAPTER 4

CONCLUSION

In this thesis an attempt is made to understand the application of various type of linear controller used for antilock braking systems. The system was modeled with a quarter vehicle dynamics and differential equation of motion was formulated. The slip ratio is used control as a criterion for this control work. Friction force and normal reaction are function of slip ratio and in turn entire equations were nonlinear. The second order differential equations were written as three state space equations (1^{st} order equations) and solutions are obtained by time integration method and are directly achieved with MATLAB^{TM} Simulink block diagrams. The time histories of the wheel, stopping distance of the vehicle, and slip factor variation are obtained for benchmark problem available in literature. Various central strategies like P-type, PD-type, PI-type, and PID-type have been implemented to augment the constant braking torque so as to control the slip ratio.

4.1 FUTURE SCOPE

In this work system is nonlinear model and controller is a linear type hence the effectiveness of the controller may not be good. In this line, as a future scope of the work well known linear controllers like neural networks, neuro-fuzzy, and fuzzy PID systems may be employed. Also, real time implementation of the control logic is needed with a on board micro-controller mounted over a small scaled model of the vehicle.

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