Novelty of Anti-Lock Braking System in Proposed Sliding Mode Controller

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Abstract: **Anti-lock Braking system (ABS) is a safety and control tool implemented in vehicles that prevents the wheel lock-up during severe or panic braking. The existing ABS controls have the ability to control and regulate the level of pressure to optimally maintain the wheel slip within the vehicle stability range. However, the ABS shows strong nonlinear characteristics for which the vehicle equipped with existing controllers have a tendency to over steer and become unstable. In this paper, the non-linear and time varying mathematical model of anti-lock braking system (ABS) is established. The proposed sliding mode controller is designed based on the slip ratio control technique, which makes the system state to lie on the sliding surface and improve the robustness of the sliding mode control system. The simulation results on a practical road surface prove that this strategy is appropriate and impressive, as it cuts-down the braking distance and braking time in comparison with other existing braking maneuver discussed in this paper. An extensive simulation study is carried out taking into account of the on/off control strategy, the conventional SMC strategy and the proposed SMC strategy. Further a comparative study is performed to judge the performance of the controllers.**

Keywords: Transport System, Anti-lock Braking System, SAR, SMC.

INTRODUCTION

In the present scenario there is a cut-throat competition among different vehicle manufacturing industries for adopting advanced technologies to improve the dynamicity of the vehicles. Due to this the security of the driving personnel and the fast growing traffic is at stake. For the security of the above the proper operation of the braking unit is badly essential in an automobile system. Brakes being energy conversion devices, convert kinetic energy of vehicle into thermal energy. This process of conversion has made many researchers around the globe to adopt many advanced techniques to improve the breaking process. Out of these, Anti-lock Braking system (ABS) is one of the most widely adopted technologies.

ABS is an electro-mechanical system which prevents the wheels from locking and hence skidding. ABS was introduced in the year 1929 for the use in aircraft to avoid the burned and burst tire problem. Later on it was introduced in four wheelers in 1970 and two wheelers in 1988. As reported by an Australian study in Monash University Accident Research Centre in 2014, the risk of multiple vehicle crashes is reduced by 24% and the risk of runoff road crashes is reduced by 41% [1].

Although, Anti-lock Braking System has been implemented in modern cars, still the ABS technology is having importance and providing significant improvement to the modern safety system. The ABS dynamics as well as the road-tire friction model being highly nonlinear, uncertain and time varying, modelling & development of ABS is really a meticulous effort. As far as automotive industry is concerned, the ABS technology is the most recent development in enhancing passenger safety or to be precise, accident avoidance. In particular an important aspect of ABS is the insight of friction model. Therefore the study of friction force characteristic at the road tire interface is of utmost importance for the designing of ABS. Moreover, the friction models are crucial for accurately reproducing friction force for simulation purpose.

A mathematical model of a vehicle that employs ABS control with on/off valve control has been implemented. The main objective being to optimize the friction force, so that vehicle stopping distance can be ceased to certain extent by regulating the wheel slip. Wheel slip dynamics being highly nonlinear & uncertain [3], a robust control strategy based on sliding mode is a standard approach to tackle the parametric and modelling uncertainties of a nonlinear system for the regulation of wheel slip [2].

In this context, a mathematical model of ABS with conventional SMC algorithm and proposed SMC have been implemented in Simulink and simulation studies were performed to verify the effectiveness of the controller and the results shows that proposed controller is able to maintain the wheel slip at the desired reference value & also reducing the vehicle stopping distance as compared to the existing controllers.

ROAD-TIRE FRICTION MODEL

The relation of the frictional coefficient (μ) and the wheel slip ratio (λ), provides the elucidation of the ability of the ABS to maintain vehicle stability, steerability and cropped 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.

However, there are many friction models but we consider the Burckhardt Friction model (1993) due to its simpler form. It gives value of coefficient of friction as a function of linear velocity and slip ratio.

$$\mu(\lambda, V_x) = [C_1(1 - e^{-C_2\lambda}) - C_3\lambda]e^{-C_4 V_x} \quad (1)$$

where,
C_1 is the maximum value of friction curve,

C_2 is the friction curve shapes,
C_3 is 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.

Table 1: Surface conditions for different surface.

SURFACES TYPES	C_1	C_2	C_3	C_4
Dry asphalt	1.2801	23.99	0.52	0.03
Wet asphalt	0.857	33.822	0.347	0.03
Dry concrete	1.1973	25.168	0.5373	0.03
Snow	0.1946	94.129	0.0646	0.03
Ice	0.05	306.39	0.00	0.03

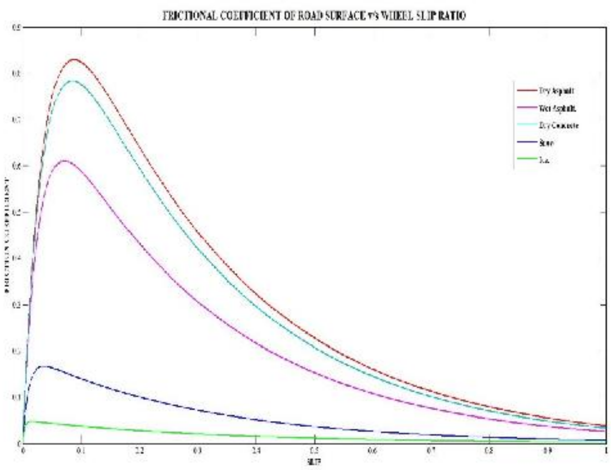

Fig. 1: Frictional coefficient of road surface v/s wheel slip ratio.

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 surfaces. From Fig. 1, it is resolved that for almost all road surfaces the frictional coefficient value is optimum when the wheel slip ratio is approximately 0.1 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 a desired value of 0.1 to maximize the frictional coefficient (μ) for any given road surface and thereby maximizing the braking force.

The curves shown in Fig. 1 are deduced empirically, based merely on steady-state experimental data using specially designed test vehicles in a highly controlled laboratory environment, i.e., constant linear and angular velocity are used to evaluate the steady state values. Under such steady-state conditions, experimental data seems to support the friction force vs. slip curves of Fig. 1. In reality, the linear and angular velocities can never be controlled independently and hence, such idealized steady-state conditions are not reached except during the rather uninteresting case of cruising with constant speed. The development of the friction force at the tire & road interface is very much a dynamic concept. Experiments performed in commercial vehicles, have shown that the tire & road forces do not necessarily vary along the curves shown Fig. 1, but instead they "jump" from one value to another when these forces are displayed in the ($\mu - \lambda$) plane. Also, in practical situations, these variations are more likely to deliver hysteresis loops, clearly displaying the dynamic nature of friction.

VEHICLE DYNAMICS

In this study a simplified quarter car vehicle model undergoing perfectly straight line braking manoeuvre has been considered. Thus there is no lateral tire force & also yaw movement do not exist.
Furthermore, the following points are considered in the modelling process.
a) There is no steering input,
b) Only longitudinal vehicle motion has been considered,
c) There is no damping effect, and
d) Approximating the vertical forces as a static value.

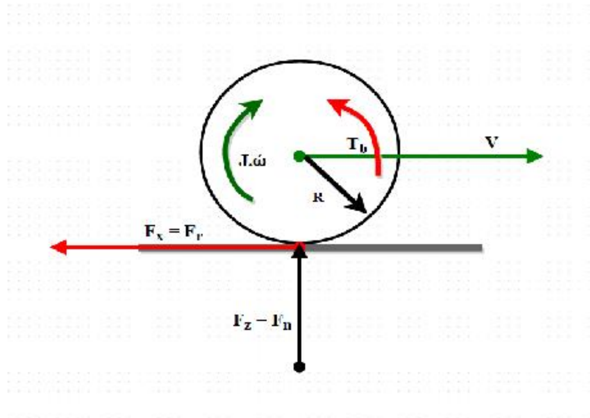

Fig. 2: Quarter car or single wheel model.

The equation of motion is given as,

$$\frac{dv}{dt} = -F_x/m \quad (2)$$

$$\frac{d\mu}{dt} = \frac{RF_x - T_b}{J} \quad (3)$$

The tire friction force is given by the Coulomb's law,

$$F_x = \mu(\lambda)F_z \quad (4)$$

The longitudinal wheel slip is given as,

$$\lambda = \frac{v - \mu R}{v} \quad (5)$$

And the friction coefficient can be written as,

$$\mu(\lambda) = 2\mu_p \lambda_p \left(\frac{\lambda}{\lambda_p^2 + \lambda^2} \right) \quad (6)$$

where, F_x = Tractive force,
F_z = Vertical load,
T_b = Braking torque,
v = Vehicle absolute velocity,
m = Quarter vehicle mass,
μ = Coefficient of friction,
μ_p = Peak friction coefficient,
λ_p = Wheel slip corresponding to μ_p,
ω = Wheel angular speed,
R = Rolling radius of the wheel, and
J = Moment of inertia of the wheel.

VEHICLE MODEL WITH ON/OFF VALVE CONTROL STRATEGY

It should be noted that the rate of change of brake pressure (or brake torque) is proportional to fluid flow rate and flow rate is proportional to control valve opening. Henceforth, the brake pressure rate is proportional to the valve command.

Here, to control the change rate of brake pressure, the model takes the difference of the actual slip and the desired slip and feeds this signal into an "On/Off valve controller" which assign the output as +1 or -1, depending on the sign of the error. This on/off rate passes through a first-order lag transfer function that stands for the delay associated with the hydraulic brake lines of the system. The model then integrates the filtered rate to obtain the actual brake pressure. The resulting signal, multiplied by a gain which represents the radius and piston area with respect to the wheel (K_f), is the brake torque applied to the wheel.

(A) MATHEMATICAL MODELLING

Let, $x_1 = v/R$ and $x_2 = \omega$ be the two state variables and $y = \lambda$ be the output variable.

Hence, from the above equations (1), (2), (3), and (6) we get,

$$\frac{dx_1}{dt} = -\left(\frac{F_z}{mR}\right) 2\mu_p \lambda_p \left(\frac{\lambda}{\lambda_p^2 + \lambda^2}\right) \tag{7}$$

$$\frac{dx_2}{dt} = \left(\frac{RF_z}{J}\right) 2\mu_p \lambda_p \left(\frac{\lambda}{\lambda_p^2 + \lambda^2}\right) - \left(\frac{T_b}{J}\right) \tag{8}$$

and

$$y = \lambda \tag{9}$$

(B) ON/OFF VALVE CONTROL LAW

In this model, an ideal anti-lock braking controller is used that incorporates On/Off valve control strategy based upon the error or difference between actual slip and desired slip. We set the desired slip to the value of slip at which the (μ-λ) curve reaches a peak value which is 0.1 (obtained from Fig. 1), this being the optimum value for the minimum braking distance as the braking force will be at its peak. The control input, namely the brake torque (T_b), is switched between the maximum value (T_{max}) and the minimum value (T_{max}) so as to keep the slip operating in the desired slip region.

(C) SIMULATION

The simulation parameters used in the simulation is tabulated below:

Table 2: Simulation parameters used for development of on/off controller.

Symbol	Quantity	Value
m	Quarter vehicle mass	375 kg
g	Acceleration due to gravity	9.81 m/s^2
R	Radius of wheel	0.326 m
J	Moment of inertia of wheel	1.7 kg-m^2
V_0	Initial vehicle velocity	60 km/hr
K	Gain	20

Running the simulation in ABS mode:

Setting the model variable Ctrl = 1,

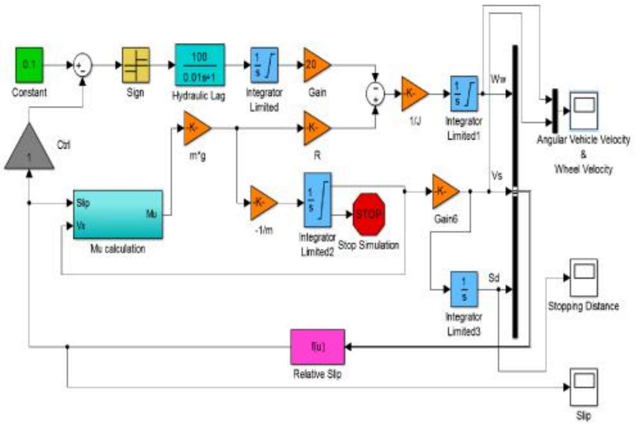

Fig. 3: Simulink model of vehicle with ABS with On/Off valve control strategy.

Running the simulation without ABS:

Setting the model variable Ctrl = 0,

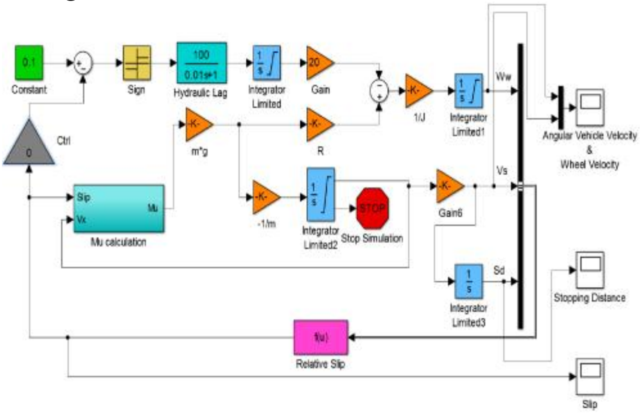

Fig. 4: Simulink model of vehicle without ABS control.

(D) SIMULATION RESULTS AND DISUSSION:

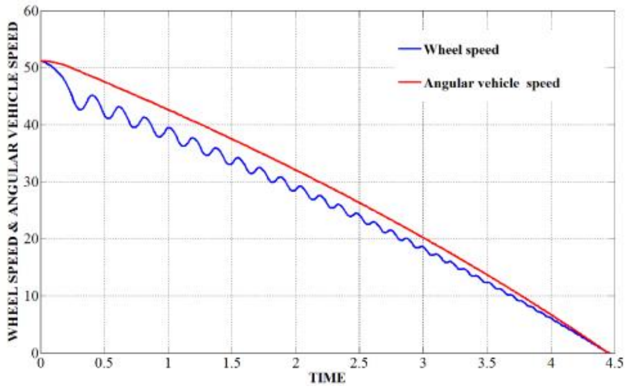

Fig. 5: Simulation result of wheel speed and vehicle speed with ABS employing On/Off controller.

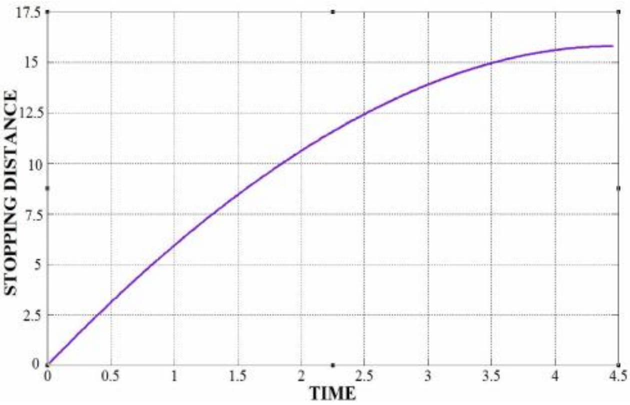

Fig. 6: Simulation result of vehicle stopping distance with ABS employing On/Off controller.

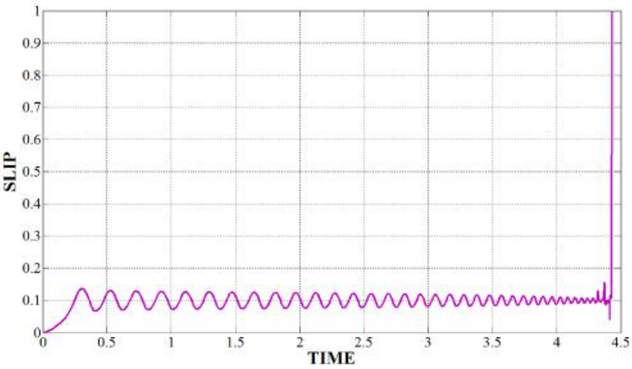

Fig. 7: Simulation result of relative wheel slip with ABS employing On/Off controller.

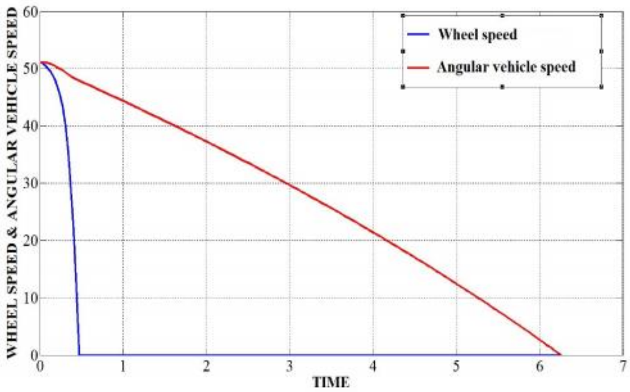

Fig. 8: Simulation result of wheel speed and vehicle speed without ABS.

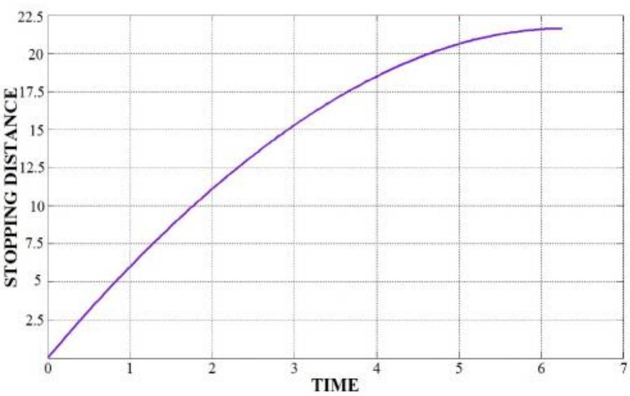

Fig. 9: Simulation result of vehicle stopping distance without ABS.

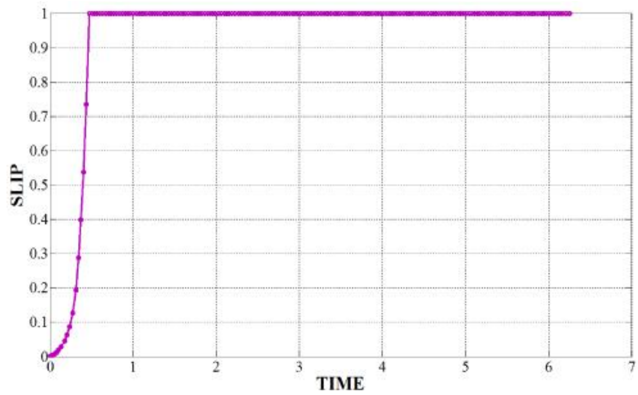

Fig. 10: Simulation result of relative wheel slip without ABS.

From the simulation results it can be observed that, in the non-ABS mode, the wheel locks up in about 0.5 seconds and it skids for next 6 seconds till the vehicle comes to rest. Whereas in the ABS mode, wheel never gets locked up. Also comparing the simulation results of ABS mode and non-ABS mode, we can say that, without ABS, the vehicle skids about an extra 5 meter, taking about 2 seconds longer to come to a stop.

Table 3: Comparison of parameters with ABS and without ABS.

	Stopping Distance*	**Stopping Time***
With ABS	16.5 m	4.5 sec
Without ABS	21.5 m	6.5 sec

* approximately

All the data are calculated taking initial velocity of 60 km/hr in a straight line manoeuvre taken in a practical road surface and the maximum braking torque for the vehicle was taken to be 1200 N-m.

So here we can conclude that with ABS we can obtain a shorter stopping distance.

VEHICLE MODEL WITH SMC STRATEGY

Generally ABS prevents wheel lockup and maintains steerability during a hard brake manoeuvre. Ideally, the ABS maintains the wheel slip at the peak of μ-slip curve. If we can regulate the wheel slip at which friction coefficient reaches its maximum value, then maximum friction force can be generated, consequently vehicle stopping distance is reduced compared to other operating points on the μ-slip curve. However the peak of μ-slip curve generally varies with varying surface and with varying vehicle longitudinal velocity. So it is very difficult to maintain the wheel slip at the peak of the (μ - λ) curve, due to the uncertainty of the problem. Moreover ABS dynamics is highly nonlinear (main nonlinearity does come from road tire friction). That's why, sliding mode control which is widely regarded as a robust control approach can be applied in ABS to regulate the wheel slip, to overcome the nonlinearity & uncertainty of the problem.

In this study, with reference to work of Wijaya [9] a sliding mode controlled ABS has been implemented in Simulink. Here a cascade form of controller is realized. A schematic of the cascade control is shown in the Fig. 11.

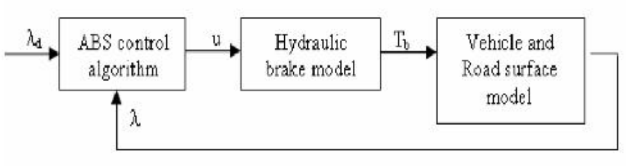

Fig. 11: Schematic of the ABS control design. [9]

(A) HYDRAULIC BRAKE DYNAMICS

The braking torque depends on the hydraulic pressure of the wheel cylinder. The pressure is controlled by the solenoid valves through the combination of build, hold & reduction mode. Hence if u(t) = Pressure that generates input signal for solenoid, and p(t) = Wheel cylinder pressure, which is taken as output, then the brake system including the solenoid valves can be approximated as a second order transfer function and is given as,

$$H(s) = \left(\mu_n^2 \Big/ \left(s^2 + 2¶\mu_n s + \mu_n^2 \right) \right) \tag{10}$$

where, ω_n = Undamped natural frequency, and

ξ_n = Damping ratio of the braking system.

And the braking torque is given by,

$$T_b = K_b p(t) \tag{11}$$

where, K_b = Torque gain in passenger car.

(B) MATHEMATICAL MODELLING

Let, x_1 = v/R and x_2 = ω, are the two state variables and y = λ is the output variable.

Hence from the above equations (1), (2), (3) and (6), we get,

$$\frac{dx_1}{dt} = -\left(\frac{F_z}{mR}\right) 2«_p \text{a}_p \left(\frac{\text{a}}{\text{a}_p^2 + \text{a}^2} \right) \tag{12}$$

$$\frac{dx_2}{dt} = \left(\frac{RF_z}{J}\right) 2«_p \text{a}_p \left(\frac{\text{a}}{\text{a}_p^2 + \text{a}^2} \right) - \left(\frac{T_b}{J}\right) \tag{13}$$

and,

$$y = \text{a} \tag{14}$$

Now, differentiating the output equation (14), we have,

$$\frac{dy}{dt} = \frac{\left(\frac{dx_1}{dt}(1 - \text{a}) - \frac{dx_2}{dt} \right)}{x_1} \tag{15}$$

Now substituting the values of dx_1/dt and dx_2/dt from above equations, we have,

$$\frac{dy}{dt} = -\left(\frac{F_z}{mR}\right) 2«_p \text{a}_p \left(\frac{\text{a}}{\text{a}_p^2 + \text{a}^2} \right)(1 - \text{a}) - \left(\frac{RF_z}{J}\right) 2«_p \text{a}_p \left(\frac{\text{a}}{\text{a}_p^2 + \text{a}^2} \right) + \frac{T_b}{J} \tag{16}$$

$$= \frac{1}{x_1}\left[2«_p \text{a}_p \left(\frac{\text{a}}{\text{a}_p^2 + \text{a}^2} \right)\left(\left(\frac{-F_z}{mR}\right)(1 - \text{a}) - \frac{RF_z}{J} \right) + \frac{T_b}{J} \right]$$

- ### *(C) SLIDING MODE CONTROL*

Here the control objective is to maintain the wheel slip at which frictional coefficient reaches its maximum value. Hence, we have to drive the wheel slip ratio λ to the desired constant value λ_d, which is the reference input to the SMC. Primarily SMC employs a high speed switching control law to drive the nonlinear plants' state trajectory onto a specified & user chosen surface in the state space, and to maintain the plants state trajectory on this surface for all subsequent time. This surface is called the switching surface because if the state trajectory of the plant is "above" the surface, a control path has one gain & a different gain if the trajectory drops "below" the surface. Hence, the first step for the design of SMC is the selection of sliding surface that models the desired close loop performance is chosen and then the control law such that the system state trajectories are forced towards the sliding surface is derived. Once the sliding surface is reached, the system state trajectories should stay on it.

Now SMC being used to track the reference wheel slip, hence sliding surface can be defined as,

$$s = \frac{d}{dt}\left((+¥)^{(r-1)} (\text{a} - \text{a}_d) \right) \tag{17}$$

where, γ is strictly a positive constant taken as the bandwidth of the system.

Since the relative order 'r' of the output for the system is one,

Hence,

$$s = (\text{a} - \text{a}_d) \tag{18}$$

The controller objective is to drive the system state to the sliding surface and maintain it on the surface for all subsequent time. In constructing the controller, the generalised Lyapunov approach can be used. Select the distance, v = $0.5s^2$ from the sliding surface, as the Lyapunov candidate function. Then, select the controller gain so that the time derivative of the chosen Lyapunov function evaluated on the solution of the controlled system is negative definite with respect to the sliding surface, thus ensuring the motion of the state trajectory to the surface. Therefore the goal is to find the control u such that,

$$\frac{d(0.5s^2)}{dt} = s\frac{ds}{dt} \langle 0 \tag{19}$$

To perform these task, the control *u*, consist of two terms. The first tem is called the equivalent control u_{eq}, which maintain the condition *s = 0* during the reduced order sliding motion. And to drive the system to the sliding surface, an additional control term called the hitting control or nonlinear control u_s has to be appended to the equivalent control u_{eq}.

Thus, u = u_{eq} + u_s

The dynamics along the sliding surface is given by,

$$\frac{ds}{dt} = 0 \tag{20}$$

Differentiating (18) on both sides, we have, ds/dt=dλ/dt,... $d\lambda_d/dt$ being a constant value.

Combining equation (15), (16) and (20), we have,

$$\frac{1}{x_1}\left[2«_p \text{a}_p \left(\frac{\text{a}}{\text{a}_p^2 + \text{a}^2} \right)\left(\left(\frac{-F_z}{mR}\right)(1 - \text{a}) - \frac{RF_z}{J} \right) + \frac{T_b}{J} \right] = 0 \tag{21}$$

Hence, the equivalent control is given by,

$$T_b = u_{eq}(t) = Jx_1\left[\left(\frac{2«_p \text{a}_p}{x_1} \right)\left(\frac{\text{a}}{\text{a}_p^2 + \text{a}^2} \right)\left(\frac{F_z}{mR}(1 - \text{a}) + \frac{RF_z}{J} \right) \right] \tag{22}$$

If the system state trajectory is not on the switching surface, an additional control term called hitting control should be added to the equivalent control. Thus,

$$u(t) = u_{eq}(t) + u_s(t) \tag{23}$$

Such that the reaching condition is satisfied, i.e.,

$$\left(s(t)\frac{ds}{dt}\right)<0 \tag{24}$$

By fulfilling the above condition, the system state trajectory is always directed towards the sliding surface, thus can guarantee the stability under parametric variations.

Conventional SMC:

Switching control or hitting control input used in this study is given by,

$$u_s(t) = -(Jx_1)\rho\left(\frac{s(t)}{|s(t)|+\delta}\right) \tag{25}$$

where, δ is the boundary layer thickness which is selected to reduce the chattering problem and ρ is a designer parameter.

Now, to evaluate the reaching condition, we have,

$$s(t).\frac{d\dot{s}}{dt} = s(t)\left[\frac{1}{x_1}\left[2\mu_p\lambda_p\left(\frac{\lambda}{\lambda_p^2+\lambda^2}\right)\left(-a(1-\lambda)-\;\right)+\frac{T_b}{J}\right]\right] \tag{26}$$

where, $\alpha = F_z/mR$ and $\beta = RF_z/J$ are two constants.

$$s(t).\frac{d\dot{s}}{dt} = s(t)\left[\frac{1}{x_1}\left[2\mu_p\lambda_p\left(\frac{\lambda}{\lambda_p^2+\lambda^2}\right)\left(-a(1-\lambda)-\;\right)+\frac{u(t)}{J}\right]\right]$$

where, $T_b = u(t)$, therefore,

$$s(t).\frac{d\dot{s}}{dt} \tag{27}$$

$$= s(t)\left[\frac{1}{x_1}\left[2\mu_p\lambda_p\left(\frac{\lambda}{\lambda_p^2+\lambda^2}\right)\left(-a(1-\lambda)-\;\right)\right]+\frac{1}{x_1}\left[2\mu_p\lambda_p\left(\frac{\lambda}{\lambda_p^2+\lambda^2}\right)\left(a(1-\lambda)+\;\right)\right]-\rho\left(\frac{s(t)}{|s(t)|+\delta}\right)\right]$$

$$= s(t)\left[-\rho\left(s(t)/(|s(t)|+\delta)\right)\right]$$

Hence, the sliding condition is satisfied for ρ > 0 and the control law $u(t) = u_{eq}(t) + u_s(t)$ drives the system state trajectory onto the sliding surface and remains on it thereafter.

D) SIMULATION

The vehicle parameters used in the simulation are tabulated in table below.

A typical road profile is assumed to be fixed for the whole test process. The curve generated and used in the simulation to represent the coefficient of friction and slip relationship is as in Fig. 1.

Table 4: Simulation parameters used for conventional SMC.

Symbol	Quantity	Value
m	Quarter vehicle mass	375 kg
g	Acceleration due to gravity	9.81 m/s^2
R	Radius of wheel	0.326 m
J	Moment of inertia of wheel	1.7 $kg\text{-}m^2$
V_0	Initial vehicle velocity	60 km/hr
ξ_n	Damping ratio	354
ω_n	Natural frequency of brake system	7.07 s^{-1}
K_b	Torque gain	100
ρ	Sliding gain	500
δ	Boundary layer thickness.	0.001

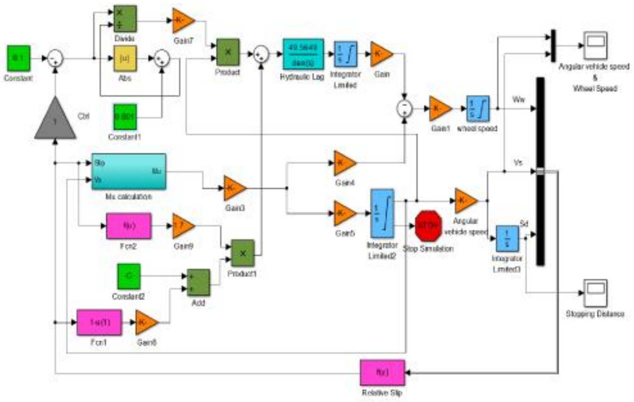

Fig. 12: Simulation model of ABS with conventional SMC.

(E) SIMULATION RESULTS:

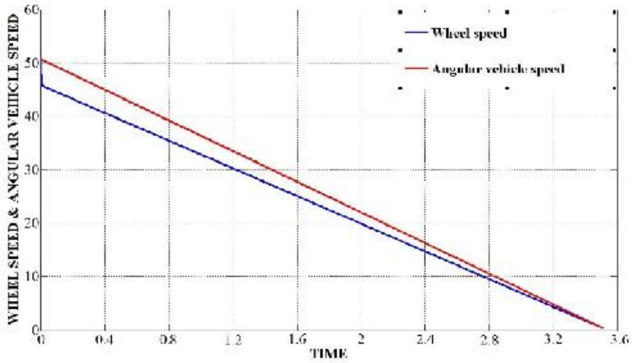

Fig. 13: Simulation results of wheel speed and vehicle speed.

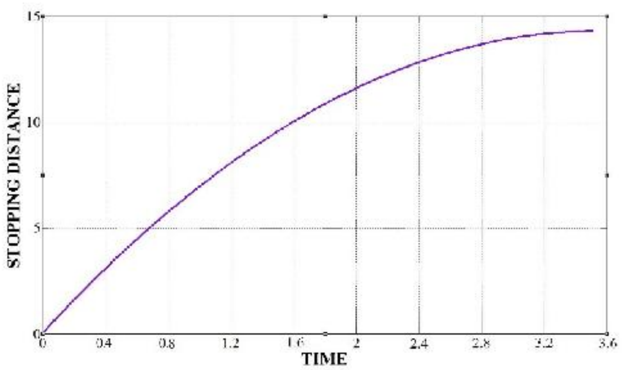

Fig. 14: Simulation result of vehicle stopping distance.

Proposed SMC:

The new switching control or hitting control input used in this study is given by,

$$u_s(t) = -(Jx_1)\rho\left[\frac{s(t)}{\tanh(s(t))+\delta}\right] \tag{28}$$

where, δ is the boundary layer thickness which is selected to reduce the chattering problem and ρ is a designer parameter.

Now, to evaluate the reaching condition, we have,

where, $\alpha = F_z/mR$ and $\beta = RF_z/J$ are two constants.

$$s(t).\frac{d\dot{s}}{dt} = s(t)\left[\frac{1}{x_1}\left[2\mu_p\lambda_p\left(\frac{\lambda}{\lambda_p^2+\lambda^2}\right)\left(-a(1-\lambda)-\;\right)+\frac{T_b}{J}\right]\right] \tag{29}$$

$$= s(t)\left[\frac{1}{x_1}\left[2\mu_p\lambda_p\left(\frac{\lambda}{\lambda_p^2+\lambda^2}\right)\left(-a(1-\lambda)-\;\right)+\frac{u(t)}{J}\right]\right]$$

where, $T_b = u(t)$,

$$s(t).\frac{\dot{ds}}{dt} \tag{30}$$
$$= s(t)\left[\frac{1}{x_1}\left[2«_p{}^{ª}{}_p\left(\frac{ª}{ª_p^2+ª^2}\right)\left(-a\left(1-{}^{ª}\right)-\ \right)\right]+\frac{1}{x_1}\left[2«_p{}^{ª}{}_p\left(\frac{ª}{ª_p^2+ª^2}\right)\left(a\left(1-{}^{ª}\right)+\ \right)\right]-°\left(\frac{s(t)}{\left|\tanh(s(t))+¢\right|}\right)\right]$$
$$= s(t)[-°(s(t)/(\tanh(s(t))+¢))]$$

Hence, the sliding condition is satisfied for ρ > 0 and the control law $u(t) = u_{eq}(t) + u_s(t)$ drives the system state trajectory onto the sliding surface and remains on it thereafter.

(F) SIMULATION

The vehicle parameters used in the simulation are tabulated in table below.

A typical road profile is assumed to be fixed for the whole test process. The curve generated and used in the simulation to represent the coefficient of friction and slip relationship is as in Fig. 1.

Table 5: Simulation parameters used for proposed SMC.

Symbol	Quantity	Value
m	Quarter vehicle mass	375 kg
g	Acceleration due to gravity	9.81 m/s^2
R	Radius of wheel	0.326 m
J	Moment of inertia of wheel	1.7 kg-m^2
V_0	Initial vehicle velocity	60 km/hr
ξ_n	Damping ratio	354
ω_n	Natural frequency of brake system	7.07 s^{-1}
K_b	Torque gain	100
ρ	Sliding gain	500
δ	Boundary layer thickness.	0.75

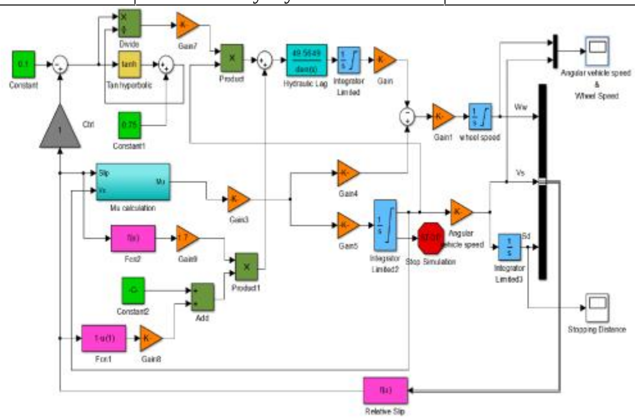

Fig. 15: Simulation model of ABS with proposed SMC.

(G) SIMULATION RESULTS AND DISCUSSION:

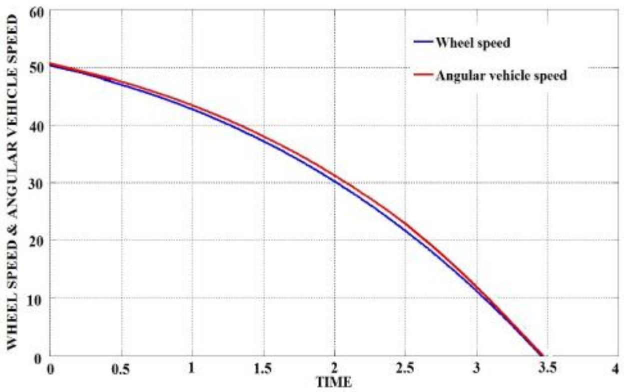

Fig. 16: Simulation results of wheel speed and vehicle speed.

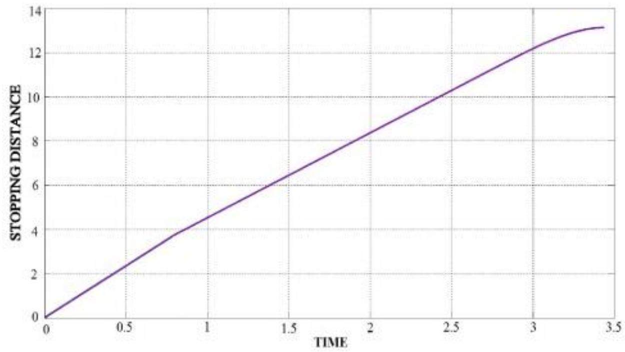

Fig. 17: Simulation result of vehicle stopping distance.

In the previous section we saw that there was no perfect rolling during braking which could possibly led to skidding of the vehicle. Therefore, a new control law or to be specific a new hitting control law was introduced and we can see better performance comparatively.

Analytically the stopping time and stopping distance were reduced, along with the system dynamics was improved.

Table 7: Comparison Parameters between on/off controller, conventional SMC and proposed SMC.

	Stopping Distance*	Stopping Time*
With On/Off Valve Control	16.5 m	4.5 sec
With Conventional SMC	15 m	3.5 sec
With Proposed SMC	13 m	3.4 sec

* approximately

CONCLUSIONS

Three different strategies were modelled for the development of anti-lock braking system. These three strategies employ regulation of wheel slip ratio. The first controller was driven by an on/off control valve, the brake toque being dependent on the flow rate of the fluid it was controlled by the valve opening, thereby regulating the slip ratio so as to obtain maximum braking force. The second and third controller implements sliding mode control because of its robustness to withstand the nonlinearity and uncertainty of anti-lock braking system. By comparing the parameters obtained from the modelling and simulation, it is justified that the proposed SMC has better performance over the other two controllers as it reduced the stopping distance and stopping time relatively higher.

Considering the initial speed of the vehicle to be 60 kmph and practical road surface, with a maximum braking torque of 1200N-m the stopping distance obtained in case of on/off controller is found out to be 16.5 m in a braking period of 4.5 seconds. However when we implement conventional SMC this stopping distance comes down to 15 m and stops 1 second faster than in on/off controller taking 3.5 seconds. But when the proposed SMC was implemented stopping distance is further reduced to 13 m, however the braking time is almost same as in case of conventional SMC that is 3.4 seconds. From this set of results it can be concluded that the proposed sliding mode controller can be used for the development of ABS as it incorporates rolling during the braking thereby avoiding skidding improving the stability of the vehicle.

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