

Linear Control Technique for Anti-Lock Braking System

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Abstract- 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 PI-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.

I. 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.

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 friction coefficient. It follows that by keeping all of the wheels of a vehicle near the peak, an antilock system can attain maximum frictional 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 drive train 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.

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.

II. LITERATURE REVIEW

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 its 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. In this ABS algorithm, rule-based control of wheel velocity is reduced to the minimum. Rear wheels cycle 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.

Poursamad [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 neuro-fuzzy 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. Modeling 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.

III. 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. PI controller are very easy to understand and easy to implement. However PI loop require continuous monitoring and adjustments. In

this line there is a scope to understand improved PI 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 control scheme such as pi controller 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.

IV. 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. This model has been already used to design the controller for ABS.

The controller, actuator and quarter car models are all in the feed forward path. The calculated wheel slip (which is to be controlled) is fed back and compared to a desired slip value, with the error fed into the controller. This is shown in Figure 1.

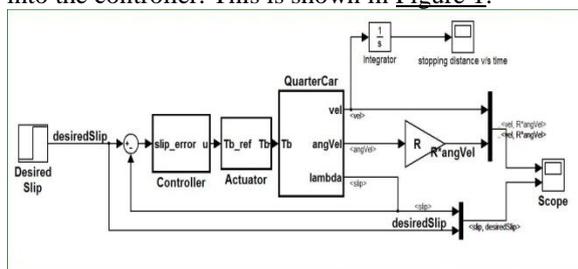


Figure 1: Quarter Car Slip Control Loop.

The model is only valid for vehicle speeds above 1–2 ms⁻¹. Hence it includes a *Stop Block* which stops the simulation when the speed drops below a small value – in this case 0.5 ms⁻¹.

A.THE TIRE MODEL

The tire model implemented in this example uses the standard Pacejk *magic formula*. In Pacejk’s model, the equation for the longitudinal friction coefficient μ_x is given by

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

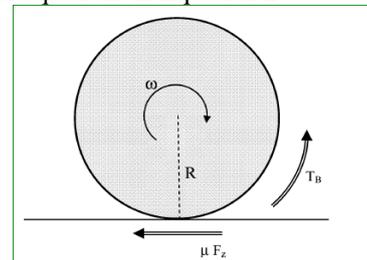
Equation 1: Pacejk’s Magic Formula Tire Model.

where λ (lambda) is the wheel slip, and the coefficients a , b and c change depending on the

current road surface. This example assumes that the vehicle is being driven on dry asphalt and hence the coefficients are $a=c1= 1.28$, $b= c2=23.99$ and $c=c3= 0.52$.

B.THE QUARTER CAR MODEL

This example uses a standard set of equations for the dynamics of a quarter car. It contains two continuous time states, and is described by the set of non-linear equations in Equation 2.



$$J\dot{\omega} = RF_x - \text{sign}(\omega)T_b$$

$$m\dot{v} = -F_x$$

$$\lambda = \frac{v - R\omega}{v}$$

$$F_x = F_z\mu_x$$

$$F_z = mg$$

Equation 2: Quarter Car Equations.

where $\omega > 0$, $v > 0$, and hence $-1 < \lambda < 1$. The following table lists the definition of the notation used in Equation 2.

| Name | Description | Value |
|-----------|---------------------------|-----------------------|
| ω | Angular Speed | Output Signal |
| v | Longitudinal Speed | Output Signal |
| J | Inertia | 1 Kg m ² |
| R | Wheel Radius | 0.32 m |
| T_b | Brake Torque | Input Signal |
| F_x | Longitudinal Force | Calculated |
| λ | Longitudinal Wheel Slip | Calculated |
| F_z | Vertical Force | Calculated |
| μ_x | Road Friction Coefficient | Calculated |
| m | Quarter Vehicle Mass | 450 Kg |
| g | Gravitational Force | 9.81 ms ⁻² |

Table 1: Notation for the Quarter Car Model.

C.THE ACTUATOR MODEL

Actuator dynamics, and particular time delays, are often critical to the design of a sufficiently accurate control algorithm. This example uses a simple first order lag in series with a time delay to

model the actuator. In practice a second order model is almost always required, and often actuators have different responses when they are opening and closing, and hence need to be modeled in considerable more detail than is done here. The model for the actuator are given by Equation 3,

$$T_b = e^{-\tau s} \frac{a}{s + a} T_{ref}$$

Equation 3: Actuator Equation.

subject to the constraint that $0 < T_b < T_{b_sat}$. The following table lists the definition of the notation used in Equation 3.

| Name | Description | Value |
|--------------|----------------------|--------|
| T | Time Delay | 0.05 s |
| A | Filter Pole Location | 70 |
| T_{b_sat} | Saturation | 4000 |

Table 2: Notation for the Actuator Model

Note that the first order lag (transfer function) has been implemented using an integrator, gain and summation/negation block rather than a Transfer Function Block (from the *Continuous* library). This has been done as the Transfer Function block does not allow vector signals as an input, but the current implementation does. Hence the model being developed can more easily be expanded to allow for 4-channels, i.e. one for each wheel on a four wheeled vehicle. A Transfer Function Block would preclude that from happening (without replacing the block).

D.THE CONTROLLER MODEL

There are many different potential implementations for the controller. Here a simple PI (proportional–integral) controller has been shown to be adequate. Note that the subsystem has been made *atomic*, and given a discrete sample rate of $T_s = 5ms$. Controller gains that have been determined to work reasonably well for the configuration chosen here are $K_p = 1200$ and $K_i = 100000$.

E.SIMULATING THE COMPLETE MODEL

Once the model components have been implemented and connected, and the parameters defined, then the model may be simulated. The results for the current configuration are shown in Figure 2.

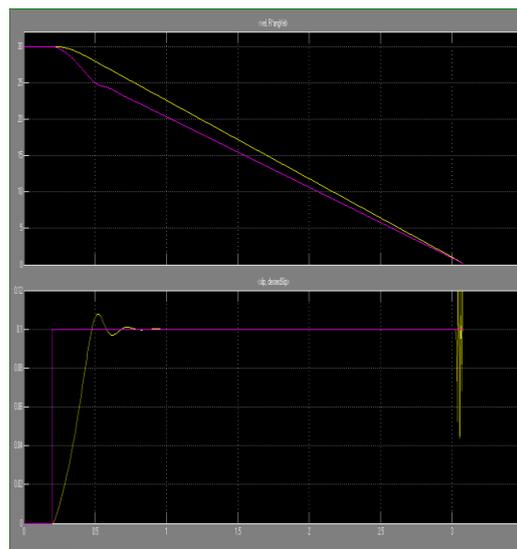


Figure 2: The Simulation Results.

The yellow line in Figure 6 shows the vehicle longitudinal velocity and the magenta line shows the wheel's linear velocity (calculated as the wheel radius multiplied by the angular velocity). When there is no wheel slip the two lines are the same. When the magenta line is below the yellow line there is wheel slip. Figure 2 shows that the vehicle is initially moving forward at $30ms^{-1}$ with no wheel slip. At $T=0.2s$ a slip of $\lambda = 10\%$ is demanded, which is achieved by applying the brake to the wheel. Consequently the vehicle decelerates reducing its speed down towards zero.

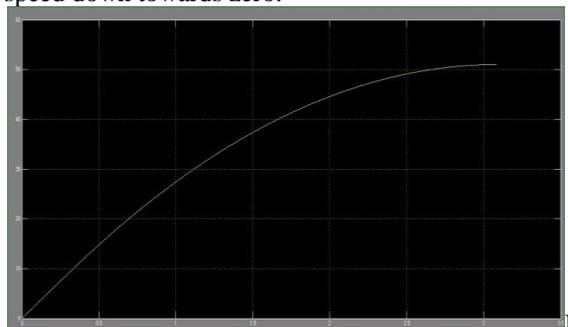


Figure 3: stopping distance vs. time for (lambda=0.2)

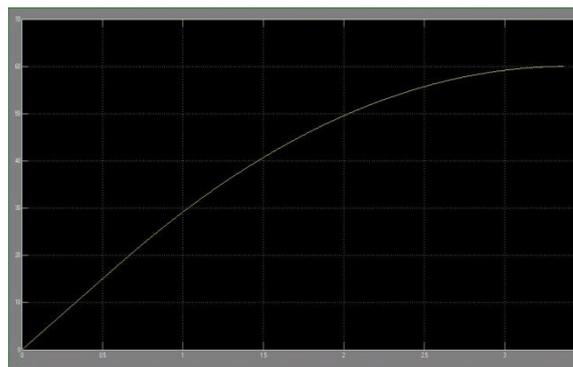


Figure 4: Stopping distance vs. time for (lambda=0.5)

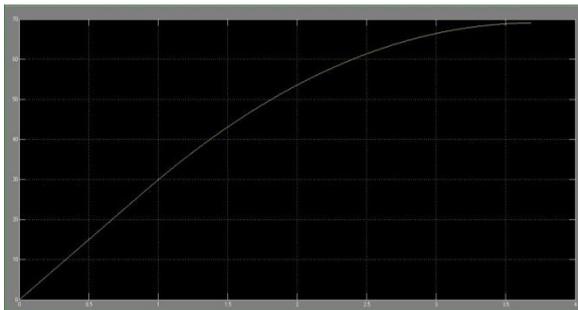


Figure 5: Stopping distance vs. time for ($\lambda=0.8$)

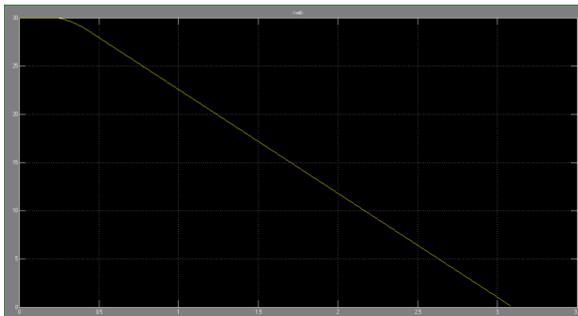


Figure 6: Linear velocity vs. time for ($\lambda=0.2$)

The controller has the desired effect which is to track the demanded slip at the required value of 10%. It takes approximately 0.2s to achieve this level of slip, which in some applications may be too slow. In that case the controller could be redesigned to try to achieve faster tracking. Figure 3,4 & 5 is comparison of stopping distance v/s time for different values of desired slip and figure 6 shows the graph between linear velocity wrt time.

V. CONCLUSION

In this paper 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 time histories of the wheel, stopping distance of the vehicle, and slip factor variation are obtained for benchmark problem available in literature. Various control 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.

VI. 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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