Chapter 5

INTRODUCTION TO LONGITUDINAL CONTROL

5.1 INTRODUCTION

The term "longitudinal controller" is typically used in referring to any control system that controls the longitudinal motion of the vehicle, for example, its longitudinal velocity, acceleration or its longitudinal distance from another preceding vehicle in the same lane on the highway. The throttle and brakes are the actuators used to implement longitudinal control.

A very familiar example of longitudinal control is the standard cruise control system available on most vehicles today. With a standard cruise control system, the driver sets a constant desired speed at which he/she would like the vehicle to travel. The cruise control system then automatically controls the throttle to maintain the desired speed. It is the driver's responsibility to ensure that the vehicle can indeed safely travel at that speed on the highway. If there happens to appear a preceding vehicle on the highway that is traveling at a slower speed or is too close to the ego vehicle, the driver must take action and if necessary apply brakes. Application of the brakes automatically disengages the cruise control system and returns control of the throttle to the driver.

The following examples describe other types of advanced longitudinal control systems.

5.1.1 Adaptive cruise control

An adaptive cruise control (ACC) system is an extension of the standard cruise control system. An ACC equipped vehicle has a radar or other sensor that measures the distance to other preceding vehicles (downstream vehicles) on the highway. In the absence of preceding vehicles, the ACC vehicle travels at a user-set speed, much like a standard cruise controlled vehicle. However, if a preceding vehicle is detected on the highway by the vehicle's radar, the ACC system determines whether or not the vehicle can continue to travel safely at the desired speed. If the preceding vehicle is too close or traveling too slowly, then the ACC system switches from speed control to spacing control (see Figure 5-1). In spacing control, the ACC vehicle controls the throttle and/ or brakes so as to maintain a desired spacing from the preceding vehicle.

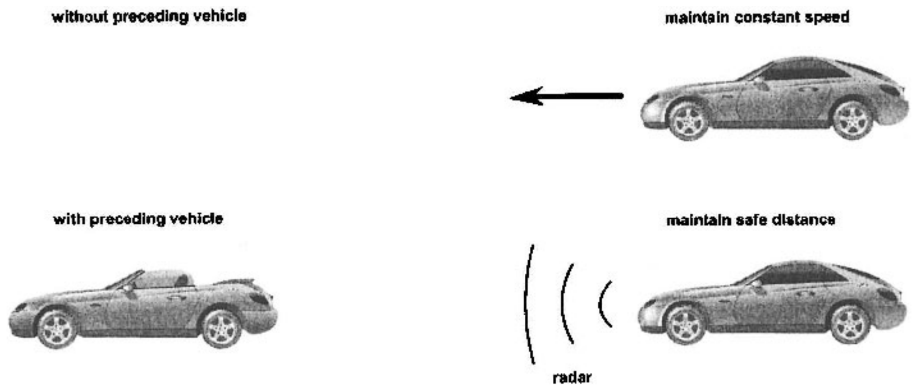

Figure 5-1. Adaptive cruise control

An ACC systems is "autonomous" - it only uses on-board sensors such as radar to accomplish the task of maintaining the desired spacing. It does not depend on wireless communication or on cooperation from other vehicles on the highway. ACC systems were first introduced in Japan (Watanabe, et. al., 1997) and Europe and are now available in the North American market (Fancher, et. al., 1997, Reichart, et. al., 1996 and Woll, 1997). The 2003 Mercedes S-class and E-class passenger sedans come with the option of a radar based Distronic adaptive cruise control system. The 2003 Lexus LS340 comes with an optional laser based adaptive cruise control system.

The design of ACC systems is discussed in detail in Chapter 7.

5.1.2 Collision avoidance

Instead of an ACC system, some vehicles come equipped with a "collision avoidance" (CA) system. A collision avoidance system also operates like a standard cruise control system in the absence of preceding vehicles and maintains a constant desired speed. If a preceding vehicle appears and the CA system determines that the desired speed can no longer be safely maintained, then the CA system reduces the throttle and/or applies brakes so as to slow the vehicle down. In addition, a warning is provided to the driver indicating the presence of other vehicles which necessitate that he or she should take over longitudinal control.

5.1.3 Automated highway systems

A completely different paradigm of longitudinal control is the control of vehicles to travel together in a tightly spaced platoon in automated highway systems (AHS). Automated highway systems have been the subject of intense research and development by several research groups, most notably by the California PATH program at the University of California, Berkeley. In an AHS, the objective is to dramatically improve the traffic flow capacity on a highway by enabling vehicles to travel together in tightly spaced platoons. The system requires that only adequately instrumented fully automated vehicles be allowed on this special highway. Manually driven vehicles cannot be allowed to operate on such a highway. Figure 5-2 below shows a photograph of eight fully automated cars traveling together in a tightly spaced platoon during a demonstration conducted by California PATH in August 1997. More details on this experimental demonstration are described in section 7.9. Automated highway systems are the focus of detailed discussion in chapter 7.

Figure 5-2. Platoon of Buicks used in the NAHSC Demonstration

5.2 BENEFITS OF LONGITUDINAL AUTOMATION

The development of the longitudinal vehicle control systems described in the previous section has been fueled by a number of motivations, including the desire to enhance driver comfort and convenience, the desire to improve highway safety and the desire to develop solutions to alleviate the traffic congestion on highways.

An ACC system provides enhanced driver comfort and convenience by allowing extended operation of the cruise control option even in the presence of other traffic. ACC systems and other automated systems in general are also expected to contribute towards increased safety on the highways. This is because statistics of highway accidents show that over 90% of accidents are caused by human error (United States DOT Report, 1992). Only a very small percentage of accidents are the result of vehicle equipment failure or even due to environmental conditions (like, for example, slippery roads). Since automated systems reduce driver burden and provide driver assistance, it is expected that the use of well-designed automated systems will certainly lead to reduced accidents.

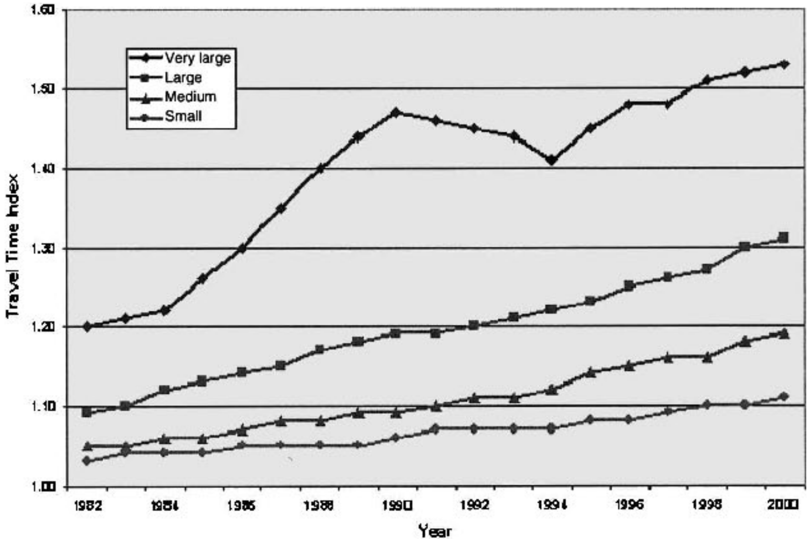

Figure 5-3. Growth in peak period travel time, 1982 to 2000
(Source: Texas Transportation Institute Report, 2001)

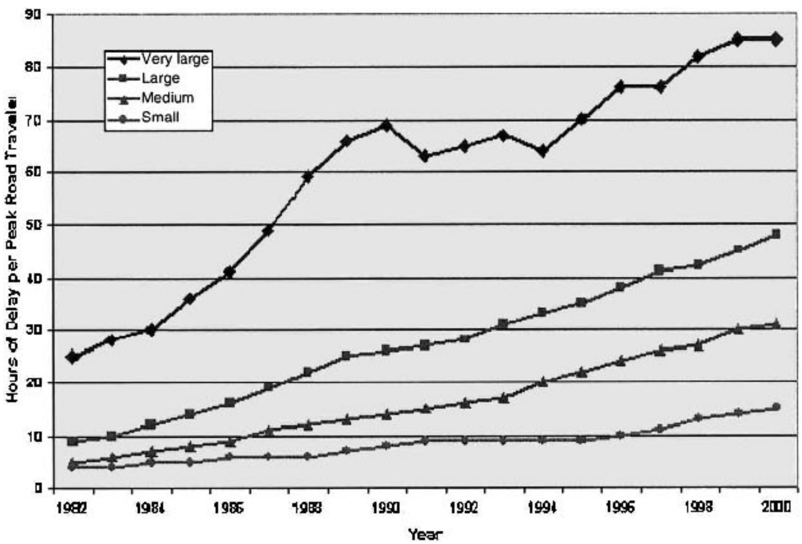

Figure 5-4. Growth in annual delay per peak road traveler, 1982 to 2000
(Source: Texas Transportation Institute Report, 2001)

The development of automated highway systems has been the direct result of the motivation to address traffic congestion on highways. Congestion has been increasing steadily in the country's major metropolitan areas to an extent where two-thirds of all highway travel today is congested travel. Using both the Travel Time Index (Figure 5-3) and annual delay per peak traveler (Figure 5-4), congestion appears to be increasing in cities of all sizes (Texas Transportation Institute Report, 2001). It appears unlikely that the congestion problem will be solved in the foreseeable future by highway expansion. The increase in traffic every year outpaces the increase in capacity due to additional highway construction (Texas Transportation Institute Report, 2001). Thus highway congestion is only expected to worsen every year. The development of AHS is an attempt to use technology to address the traffic congestion issue. An AHS in which vehicles travel in closely packed platoons can provide a highway capacity that is three times the capacity of a typical highway (Varaiya, 1993).

Having introduced the types of longitudinal control systems under development by various automotive researchers, we next move on to studying the technical details of designing longitudinal control systems.

5.3 CRUISE CONTROL

In a standard cruise control system, the speed of the vehicle is controlled to a desired value using the throttle control input. The longitudinal control system architecture for the cruise control vehicle will be designed to be hierarchical, with an upper level controller and a lower level controller as shown in Figure 5-5.

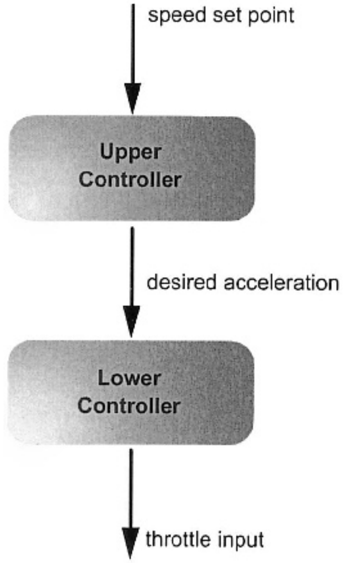

Figure 5-5. Structure of cruise control system

The upper level controller determines the desired acceleration for the vehicle. The lower level controller determines the throttle input required to track the desired acceleration. Vehicle dynamic models, engine maps and nonlinear control synthesis techniques (Choi and Devlin, 1995a and 1995b, Hedrick et al, 1991, Hedrick, et. al., 1993) are used by the lower controller in calculating the real-time throttle input required to track the desired acceleration.

In performance specifications for the design of the upper controller, it is necessary to specify that the steady state tracking error of the controller should be zero. In other words, the speed of the vehicle should converge to the desired speed set by the driver. Other desirable performance specifications might include zero overshoot and adequately fast rise time.

As far as the upper level controller is concerned, the plant model used for control design is

$$\ddot{x} = \frac{1}{\tau s + 1}\ddot{x}_{des} \tag{5.1}$$

or

$$\tau \dddot{x} + \ddot{x} = \ddot{x}_{des} \tag{5.2}$$

where x is the longitudinal position of the vehicle measured from an inertial reference. This means that the upper controller uses desired acceleration as the control input. The actual acceleration of the vehicle is assumed to track the desired acceleration with a time constant τ.

As far as the lower level controller is concerned, the driveline dynamics discussed in chapter 4 and the engine dynamics discussed in chapter 9 constitute the actual longitudinal vehicle model that must be utilized in control design. The lower level controller must ensure that the vehicle acceleration tracks the desired acceleration determined by the upper controller.

Due to the finite bandwidth associated with the lower controller, the vehicle is expected to track its desired acceleration imperfectly. Thus there is a first order lag in the lower level controller performance and hence the use of the model equation (5.1) for the upper controller which incorporates a lag in tracking desired acceleration.

This chapter assumes a lag of $\tau = 0.5$ for analysis and simulation.

5.4 UPPER LEVEL CONTROLLER FOR CRUISE CONTROL

A typical algorithm used for the upper controller is PI control using error in speed as the feedback signal:

$$\ddot{x}_{des}(t) = -k_p\left(V_x - V_{ref}\right) - k_I \int_0^t \left(V_x - V_{ref}\right) dt \tag{5.3}$$

where V_{ref} is the desired vehicle speed set by the user.

Define the following reference position

$$x_{des} = \int_0^t V_{ref}\, d\tau \tag{5.4}$$

Here $x_{des}(t)$ is the position of an imagined reference vehicle that is traveling at the reference or desired speed. Then the upper controller can be rewritten as

$$\ddot{x}_{des} = -k_p(\dot{x} - \dot{x}_{des}) - k_I(x - x_{des}) \tag{5.5}$$

This is equivalent to inter-vehicle spacing control with $x - x_{des}$ being the spacing from a fictitious vehicle traveling at the desired reference speed.

The unity feedback loop denoting this closed-loop system is shown below in Figure 5-6.

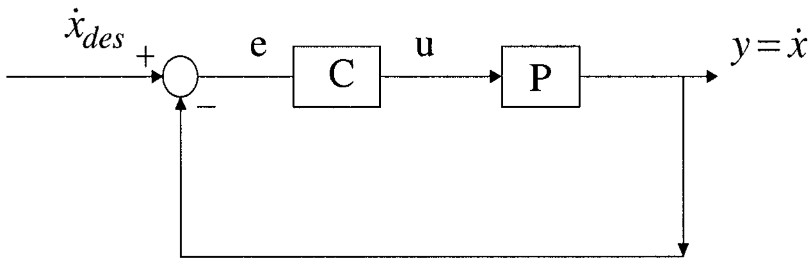

Figure 5-6. Unity feedback loop for upper controller for cruise control

As discussed previously, the plant model for the upper controller is the transfer function between desired acceleration and actual vehicle speed and is given by

$$P(s) = \frac{1}{s(\tau s + 1)} \tag{5.6}$$

The PI controller is

$$C(s) = k_p + \frac{k_i}{s} \tag{5.7}$$

Hence the closed-loop transfer function is

$$\frac{V_x}{V_{ref}} = \frac{PC}{1 + PC} = \frac{k_p s + k_i}{\tau s^3 + s^2 + k_p s + k_i} \tag{5.8}$$

A root locus of the feedback system is shown in Figure 5-7 for varying k_p with the ratio $\frac{k_p}{k_i}$ fixed at 4. A value of $\tau = 0.5$ was assumed for the system lag. Values of k_p varying from 0 to 0.75 were used. It can be seen from Figure 5-7 that the closed system is stable for all non-zero k_p. There is one closed-loop real pole and a pair of complex conjugate poles. For a value of $k_p = 0.75$, the complex poles have a damping ratio of 0.87. If the value of k_p is increased further beyond 0.75, the damping ratio of the complex poles decreases and the system becomes less damped.

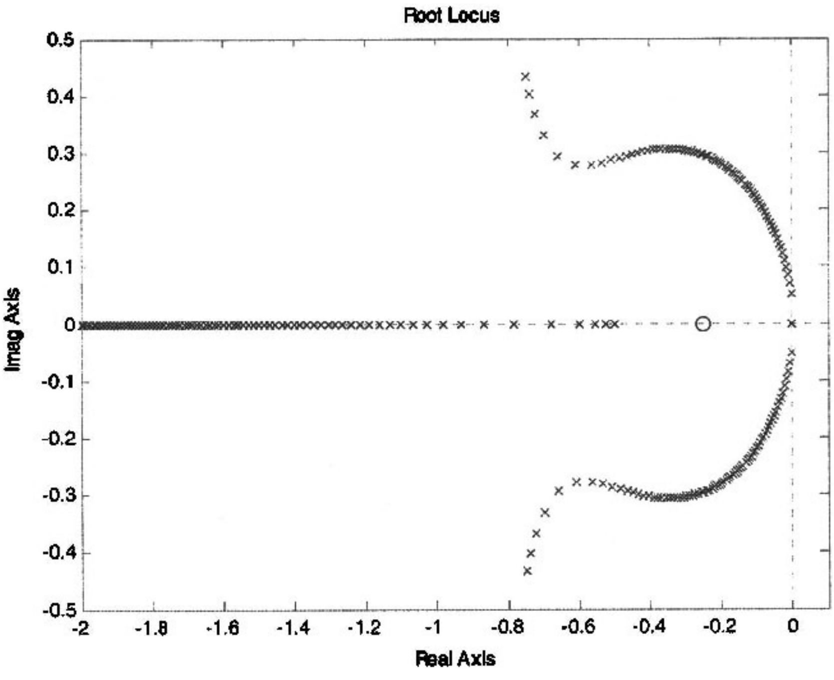

Figure 5-7. Closed-loop transfer function with PI controller

The Bode magnitude plot of the closed-loop transfer function is shown in Figure 5-8 for a value of $k_p = 0.75$. As seen in the figure, the resulting bandwidth of the closed-loop system is 0.2 Hz.

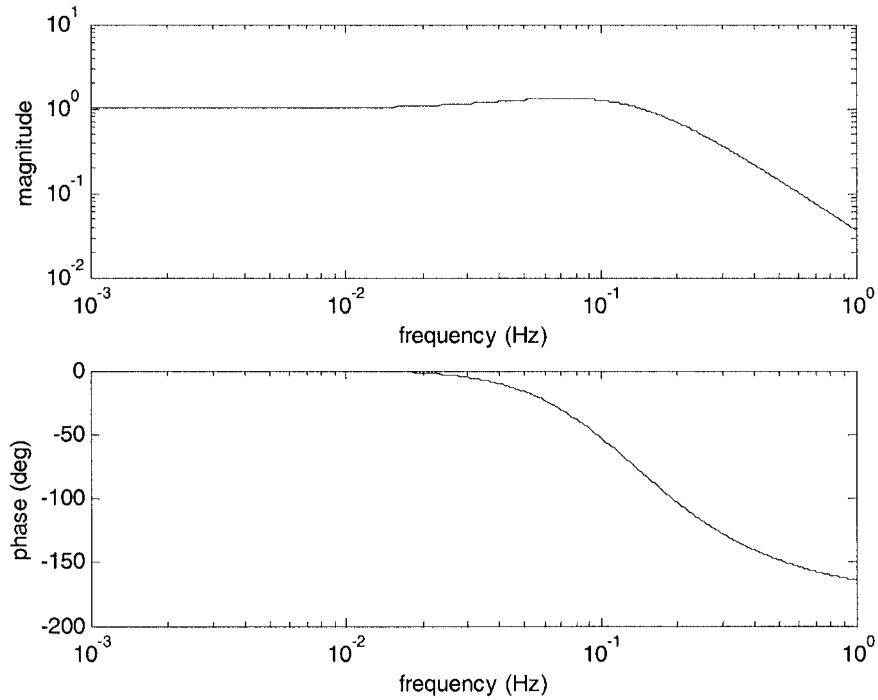

Figure 5-8. Closed-loop transfer function with PI controller

5.5 LOWER LEVEL CONTROLLER FOR CRUISE CONTROL

In the lower controller, the throttle input is calculated so as to track the desired acceleration determined by the upper controller. A simplified model of longitudinal vehicle dynamics can be used in the design of the lower level controller. This simplified model is typically based on the assumptions that the torque converter in the vehicle is locked and that there is zero-slip between the tires and the road (Hedrick, et. al., 1991). These are very reasonable assumptions during cruise control because

a) The cruise control system is typically engaged in gears 3 and higher where the torque converter is indeed locked.

b) The tire slip is small since the longitudinal maneuvers involved in cruise control are very gentle.

Using the above assumptions, the engine torque required to track the desired acceleration command is first calculated. This calculation is described in section 5.5.1. Once the required engine torque has been obtained, engine maps and nonlinear control techniques are used to calculate the throttle input that will provide this required torque.

5.5.1 Engine Torque Calculation for Desired Acceleration

A model of the driveline dynamics was discussed in section 4.2 of this book and should be reviewed by the reader. Consider the case where the torque converter is locked ($T_t = T_p$), the transmission is in steady state (it is not undergoing a gear shift) and the longitudinal tire slip is negligible. In this case, the wheel speed ω_w is proportional to the engine speed ω_e and related through the gear ratio R as follows

$$\omega_w = R\omega_e \tag{5.9}$$

and the transmission shaft speed is equal to the engine speed

$$\omega_t = \omega_e \tag{5.10}$$

The longitudinal vehicle velocity is approximated by $\dot{x} = r_{eff}\,\omega_w$ where r_{eff} is the effective tire radius and hence the longitudinal acceleration is

$$\ddot{x} = r_{eff} R\dot{\omega}_e \tag{5.11}$$

The longitudinal vehicle equation is

$$m\ddot{x} = F_x - R_x - F_{aero}$$

where F_x is the total longitudinal tire force from all tires, R_x is the rolling resistance force and F_{aero} is the aerodynamic drag force. Using equation (5.11), this can be rewritten as

$$mRr_{eff}\dot{\omega}_e = F_x - R_x - F_{aero} \tag{5.12}$$

Hence

$$F_x = mRr_{eff}\dot{\omega}_e + R_x + F_{aero} \tag{5.13}$$

Substituting from equation (5.13) into the equation for the wheel rotational dynamics (4.38)

$$I_w\dot{\omega}_w = T_{wheel} - r_{eff}(F_x) = T_{wheel} - mRr_{eff}^{\ 2}\dot{\omega}_e - r_{eff}R_x - r_{eff}F_{aero} \tag{5.14}$$

Hence, the torque at the wheels required to produce the desired acceleration is

$$T_{wheel} = I_wR\dot{\omega}_e + mRr_{eff}^{\ 2}\dot{\omega}_e + r_{eff}F_{aero} + r_{eff}R_x \tag{5.15}$$

Substituting from equation (5.15) into the equation for the transmission dynamics

$$I_t\dot{\omega}_t = T_t - RT_{wheel} = T_t - I_wR^2\dot{\omega}_e - mR^2r_{eff}^{\ 2}\dot{\omega}_e - Rr_{eff}F_{aero} - Rr_{eff}R_x$$

Since $\omega_t = \omega_e$ and $T_t = T_p$, we have

$$I_t\dot{\omega}_e = T_p - I_wR^2\dot{\omega}_e - mR^2r_{eff}^{\ 2}\dot{\omega}_e - Rr_{eff}F_{aero} - Rr_{eff}R_x$$

Hence the pump torque load on the engine is

$$T_p = (I_t + I_wR^2 + mR^2r_{eff}^{\ 2})\dot{\omega}_e + Rr_{eff}F_{aero} + Rr_{eff}R_x \tag{5.16}$$

Substituting from equation (5.16) into the engine rotational dynamics equation (4.35)

$$I_e\dot{\omega}_e = T_{net} - T_p$$
$$= T_{net} - (I_t + I_w R^2 + mR^2 r_{eff}{}^2)\dot{\omega}_e - Rr_{eff}F_{aero} - Rr_{eff}R_x$$

Hence

$$I_e\dot{\omega}_e = T_{net} - (I_t + I_w R^2 + mR^2 r_{eff}{}^2)\dot{\omega}_e - Rr_{eff}F_{aero} - Rr_{eff}R_x$$

or

$$J_e\dot{\omega}_e = T_{net} - Rr_{eff}F_{aero} - Rr_{eff}R_x \tag{5.17}$$

where

$$J_e = I_e + I_t + R^2 I_w + mR^2 r_{eff}{}^2 \tag{5.18}$$

Since F_{aero} is a quadratic function of vehicle velocity and can also be expressed in terms of a quadratic in ω_e, equation (5.16) represents a single first order o.d.e. that describes the vehicle dynamics in the case where the torque converter is locked and the slip is assumed to be negligible.

Substituting for F_{aero} as $F_{aero} = c_a\left(r_{eff}R\omega_e\right)^2$, the dynamics relating engine speed ω_e to the pseudo-input "net combustion torque" T_{net} can be modeled by the single first-order ode

$$\dot{\omega}_e = \frac{T_{net} - c_a R^3 r_{eff}{}^3 \omega_e{}^2 - R(r_{eff}R_x)}{J_e} \tag{5.19}$$

where $J_e = I_e + I_t + (mr_{eff}{}^2 + I_\omega)R^2$ is the effective inertia reflected on the engine side.

From equation (5.19), it is clear that if the net combustion torque is chosen as

$$(T_{net}) = \frac{J_e}{R\, r_{eff}} \ddot{x}_{des} + [c_a R^3 r_{eff}{}^3 \omega_e{}^2 + R(r_{eff} R_x)] \tag{5.20}$$

then the acceleration of the car is equal to the desired acceleration defined by the upper level controller i.e. $\ddot{x} = \ddot{x}_{des}$.

5.5.2 Engine Control

Once the required combustion torque is obtained from (5.20), the control law to calculate the throttle angle to provide this torque can be obtained by using engine dynamic models and applying nonlinear control synthesis techniques. Engine dynamic models for both SI and diesel engines and nonlinear control design to provide a desired engine torque are discussed in Chapter 9 of this book.

5.6 ANTI-LOCK BRAKE SYSTEMS

5.6.1 Motivation

Anti-lock brake systems (ABS) were originally developed to prevent wheels from locking up during hard braking. Modern ABS systems not only try to prevent wheels from locking but also try to maximize the braking forces generated by the tires by preventing the longitudinal slip ratio from exceeding an optimum value.

First, note that locking of the wheels reduces the braking forces generated by the tires and results in the vehicle taking a longer time to come to a stop. Further, locking of the front wheels prevents the driver from being able to steer the vehicle while it is coming to a stop.

To understand the influence of longitudinal slip ratio on braking forces, consider the tire force characteristics shown in Figure 5-9. As seen in Figure 5-9, the magnitude of the tire longitudinal force typically increases linearly with slip ratio for small slip ratios. It reaches a maximum (peak) value typically at a slip ratio value between 0.1 and 0.15. At slip ratios beyond this value, the magnitude of tire force decreases and levels out to a constant value.

If the driver presses hard on the brakes, the wheels will slow down considerably faster than the vehicle slows down, resulting in a big slip ratio value. However, as described above, slip ratios higher than an optimum value actually result in reduced braking forces. The vehicle would take longer to come to a stop if the slip ratio exceeded the optimum value. The ABS solution then is to prevent excessive brake torque from being applied on the wheels, so that the slip ratio doesn't exceed the optimum value. This would also prevent or delay the wheels from locking up and increase steerability of the vehicle during braking.

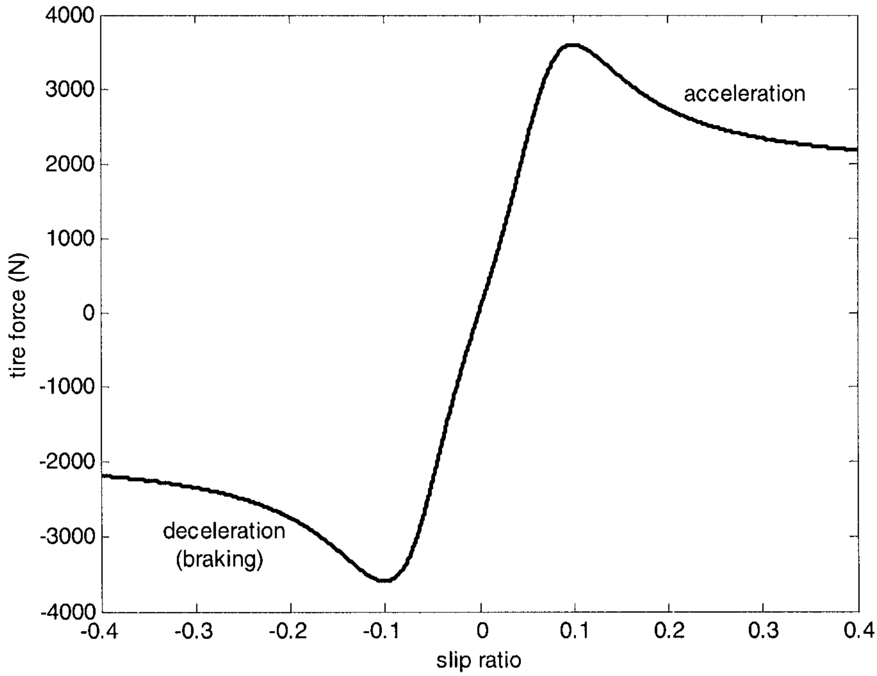

Figure 5-9. Tire longitudinal force as a function of longitudinal slip ratio

The following simulation plots demonstrate the negative consequences of very hard braking. Figures 5-10 and 5-11 show vehicle speed and slip ratio respectively during hard braking. As seen in Figure 5-11, the wheels lock during braking and result in a slip value of –1 within 1 second of the initiation of braking. As seen in Figure 5-10, while the wheels come to a stop in 1 second, the vehicle itself does not come to a stop and only reduces in speed from 30 m/s to 13 m/s in 12 seconds.

Figures 5-12 and 5-13 show slip ratio and vehicle speed during *reduced braking* designed to just prevent the wheels from locking up. As seen in Figure 5-12, the slip ratio is maintained at 0.09 which is close to the optimum value of 0.1. The wheels don't lock, as seen in Figure 5-13, thus allowing the vehicle to be steered. Further, the speed of the vehicle is reduced from 30 m/s to 2 m/s in 12 seconds. Thus a significantly greater reduction in vehicle speed is obtained by limiting the amount of braking torque applied to the wheels.

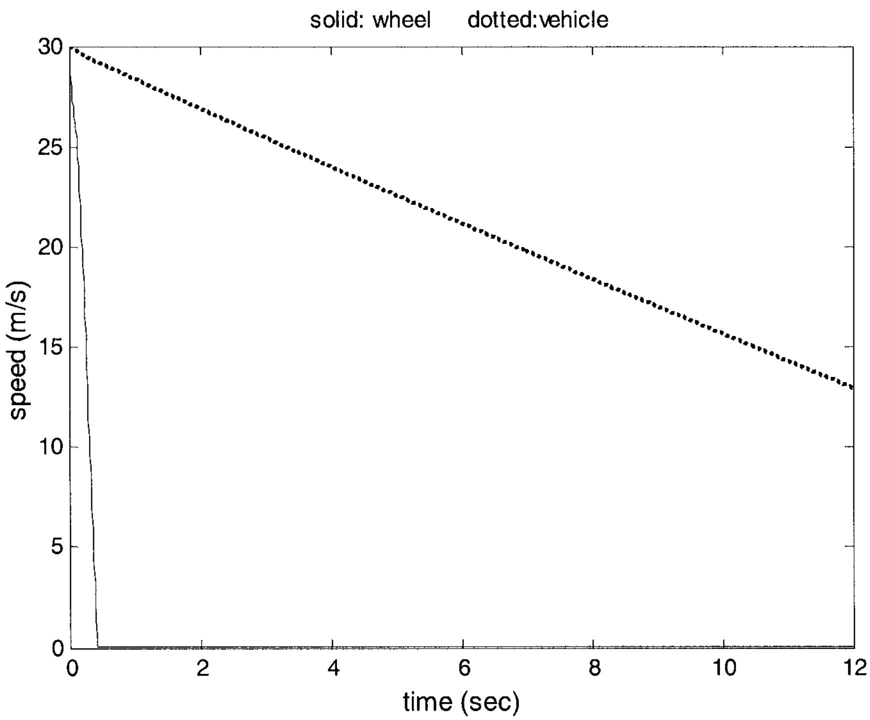

Figure 5-10. Vehicle speed during hard braking (No ABS)

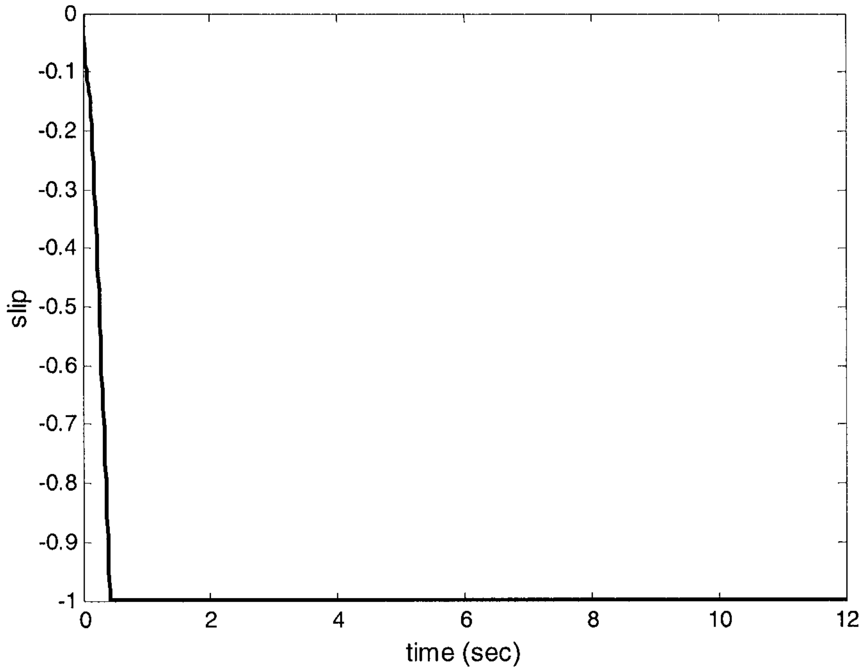

Figure 5-11. Slip Ratio during hard braking (No ABS)

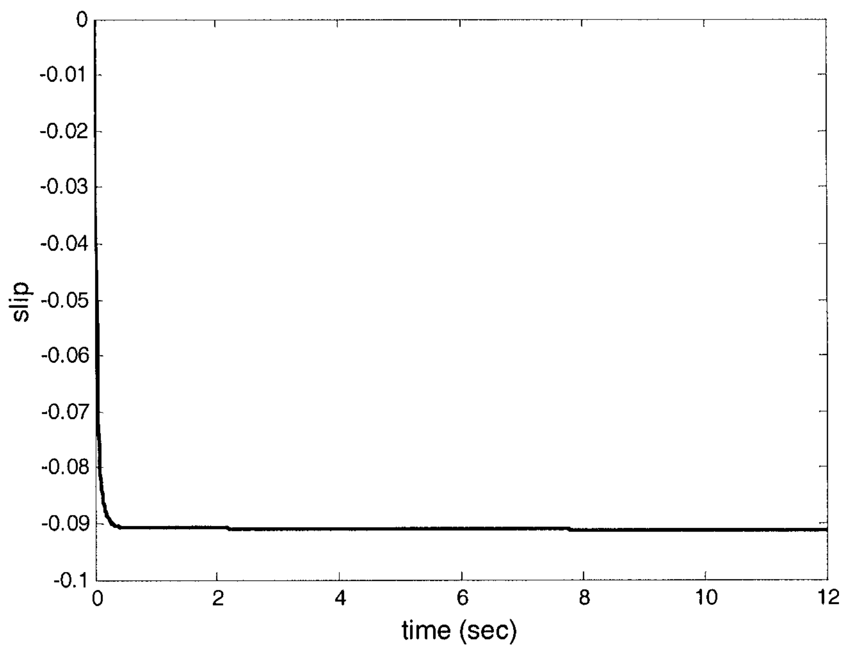

Figure 5-12. Slip Ratio with reduced braking (ABS)

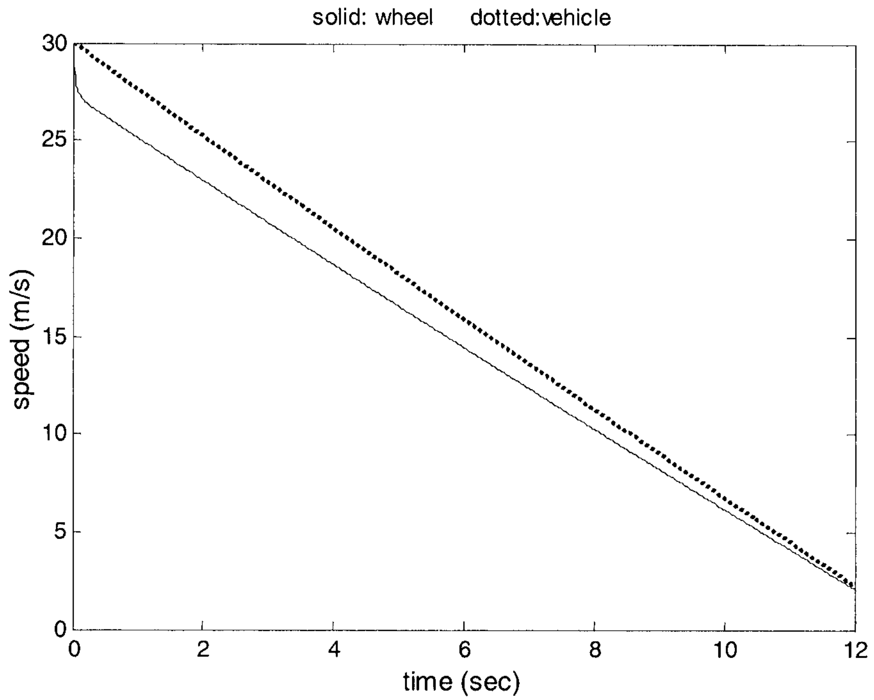

Figure 5-13. Vehicle speed with reduced braking (ABS)

5.6.2 ABS Functions

The basic objective of the ABS is to either hold or release the braking pressure on the wheels if there is a danger of the wheels locking. At the same time, the ABS needs to re-permit application of the brakes again once the danger of locking has been averted. The ABS system could also hold or release the braking pressure in order to keep the slip ratio at the wheel from exceeding an optimum value.

Depending on the number of wheels the ABS controls, ABS can be four channel four sensor, three channel three sensor or one channel one sensor. Each channel controlled by the ABS has a valve. Depending on the position of the valve, brake pressure on the wheel is held, released or controlled by the driver:

When the valve is open, pressure from the master cylinder is passed right through to the brake. This allows the brake to be controlled by the driver, allowing the amount of brake pressure desired by the driver to be applied to the brake.

When the valve is closed or blocked, that brake is isolated from the master cylinder. This holds the brake pressure and prevents it from increasing even if the driver pushes the brake pedal harder.

When the valve is in the release position, the pressure from the brake is released. In this position, not only is the brake isolated from any further braking actions of the driver, but the amount of braking pressure on the wheel is actively reduced.

A major practical problem in ABS systems is that wheel slip cannot be measured with inexpensive sensors on a passenger vehicle. Often the only measurements available to the ABS system are measurements of the individual wheel speeds at the four wheels. Algorithms that utilize these wheel speed measurements to predict if the wheels will lock and to predict if the danger of locking has been averted have to be used.

The process of determining whether or not the wheel is going to lock is called *prediction*. Prediction point slip is defined as the wheel slip at the instant the control unit predicts for the first time in a brake cycle that the wheel is going to lock.

The process of determining whether or not the danger of locking has been averted is called *reselection*. Reselection point slip is defined as the wheel slip at the instant it is predicted for the first time in a brake cycle that the danger of locking is averted.

5.6.3 Deceleration Threshold Based Algorithms

One of the most common ABS algorithms is the deceleration threshold based algorithm (Bosch Automotive Handbook, 2000). The wheel deceleration signal is used to predict if the wheel is about to lock. Here wheel deceleration is defined as angular deceleration multiplied by effective tire radius.

A common version of the deceleration threshold algorithm is summarized in Figures 5-14, 5-15, 5-16 and 5-17 (Kiencke and Nielsen, 2000 and Bosch Automotive Handbook, 2000).

Let $\dot{V}_R$ be the wheel deceleration defined as

$$\dot{V}_R = r_{eff}\,\dot{\omega}_w \tag{5.21}$$

where r_{eff} is the effective tire radius and ω_w is the angular wheel speed. Let a_1, a_2, a_3 and a_4 be acceleration threshold values, all defined to be positive with $a_2 > a_1$ and $a_4 > a_3$.

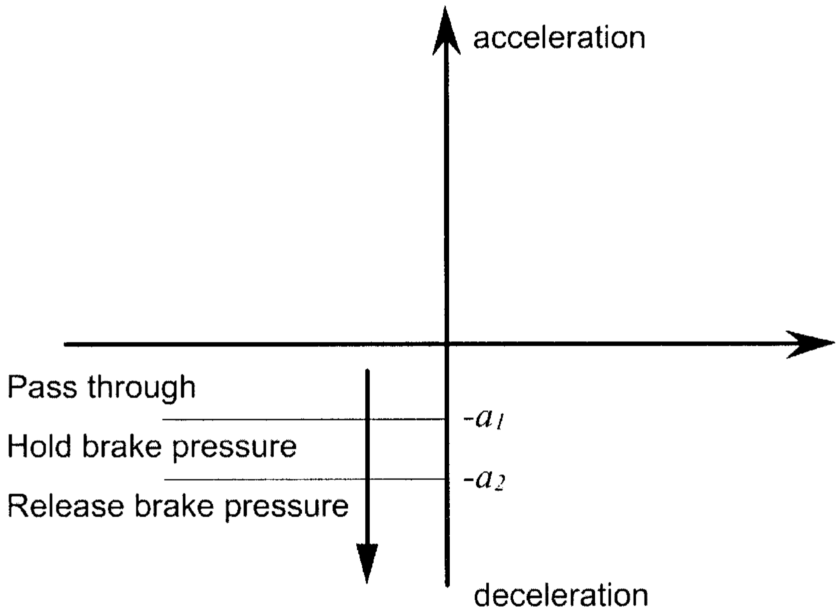

Figure 5-14. Deceleration in the first cycle

When the driver presses on the brake pedal, if the deceleration is less than a_1 (i.e. if $\dot{v}_R > -a_1$), then the driver's braking action is directly passed through to the brakes. When the deceleration exceeds a_1 for the first time (i.e $\dot{v}_R < -a_1$), the driver's braking action is no longer directly passed through to the brakes. Instead the braking pressure is held constant at the pressure value achieved when the deceleration first exceeded a_1. If the wheel deceleration continues to increase further and exceeds the value a_2 (i.e. $\dot{v}_R < -a_2$), then the braking pressure at the wheel is decreased. This will prevent the wheel from decelerating any further and could eventually result in the wheel gaining speed or accelerating. If the wheel deceleration reduces to the value a_2 (i.e. $\dot{v}_R > -a_2$), then the pressure drop is stopped. If the wheel deceleration drops below the value a_1 (i.e. $\dot{v}_R > -a_1$), then the driver's braking action is once again directly passed through to the brakes. If the wheel actually starts accelerating, and the acceleration exceeds the relatively high threshold a_4, then the braking pressure is actually increased beyond that dictated by the driver's actions, so as to prevent the wheel from over acceleration. In this case, when the wheel's acceleration drops to the value below a_3 (i.e. $\dot{v}_R < a_3$), the driver's braking action are again passed through to the brakes. When the wheel deceleration goes below a_1 ($\dot{v}_R < -a_1$) again the second cycle starts. Running through such cycles, the wheels are prevented from locking and the wheel rotational speed is kept in an area where wheel slip is close to that of the maximum friction coefficient. Note that a_4 is a relatively high deceleration level. (much larger than a_3).

During the second braking cycle, the braking pressure is reduced right away when the deceleration first exceeds a_1 (i.e. the phase of holding brake pressure constant between a_1 and a_2 is no longer done during the second braking cycle). In the first cycle, the short pressure holding phase is used for the filtering of disturbances.

Figure 5-14 and Figure 5-15 summarize the deceleration threshold based algorithm during wheel deceleration. Figure 5-16 and Figure 5-17 summarize the algorithm during wheel acceleration.

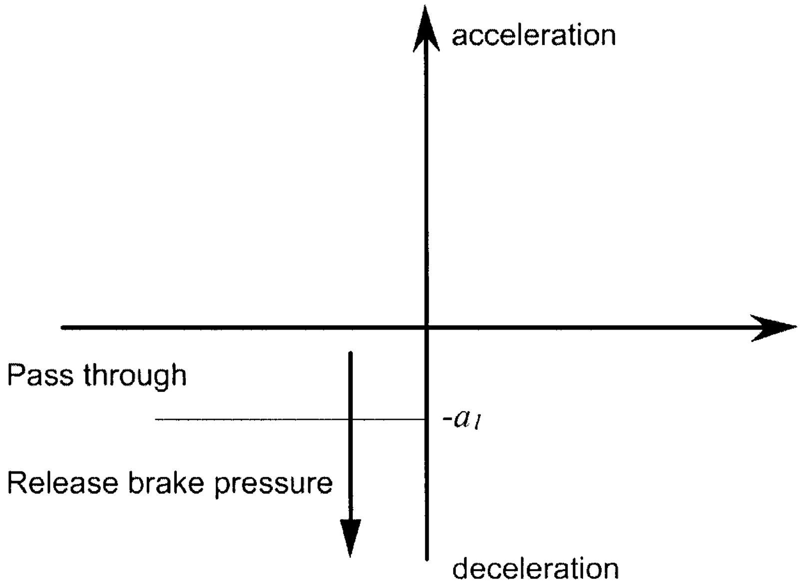

Figure 5-15. Deceleration in the second and subsequent cycles

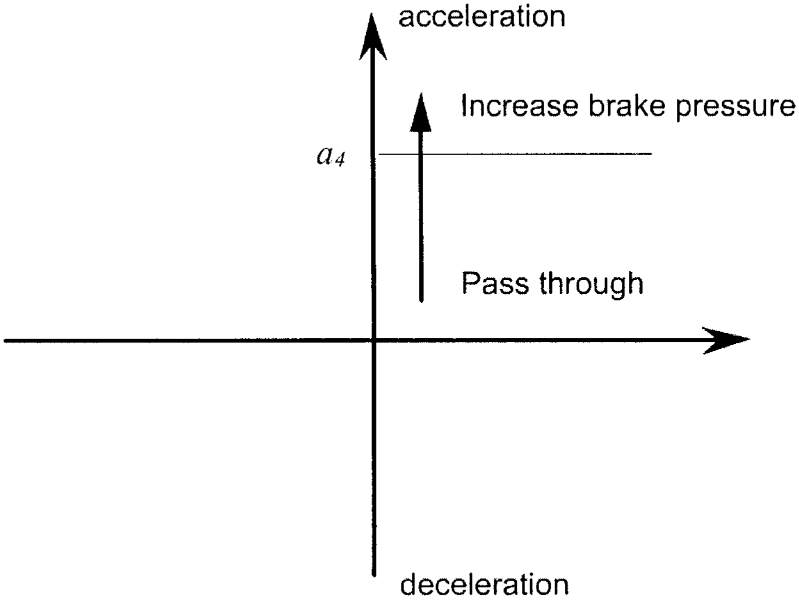

Figure 5-16. Increasing acceleration

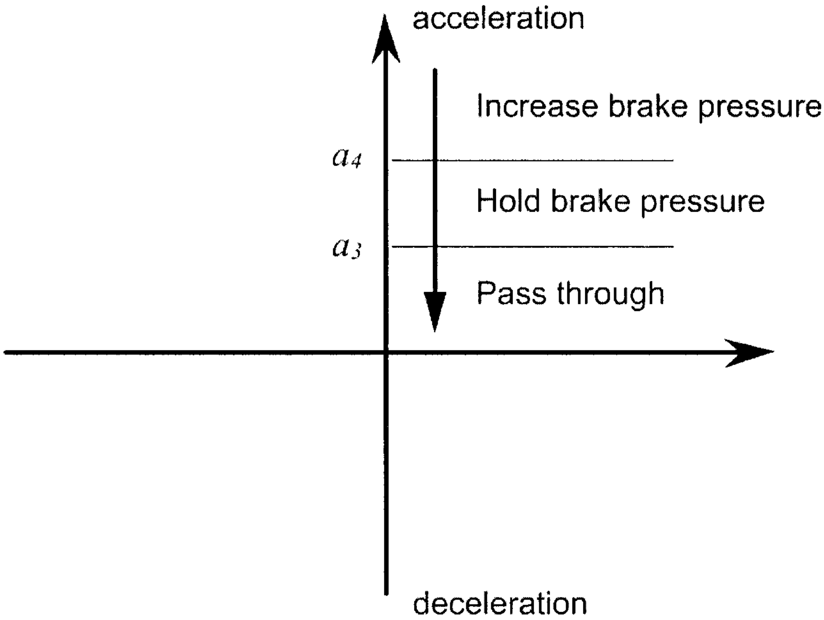

Figure 5-17. Decreasing acceleration

In a modified version of this algorithm, during the first cycle, if the deceleration exceeds a_1 and the wheel speed falls below a slip-switching threshold (determined based on the initial speed when braking first started), then the braking pressure is reduced. Thus the deceleration threshold a_2 is not used in this modified algorithm. From the second braking cycle onwards, pressure is reduced right away when the deceleration first exceeds a_1 (Bosch Automotive Handbook, 2000).

5.6.4 Other Logic Based ABS Control Systems

A number of factors influence the working of the ABS system. These include

The value of the tire-road friction coefficient, since it influences the range within which the wheel slip ratio should be maintained.

The rate of application of the brake torque (brake dynamics). During the first cycle, this depends on how the driver of the vehicle presses the brake pedal. In the subsequent cycles, it depends on the pressure build characteristics of the modulator.

Initial longitudinal velocity of the vehicle is also important, since it determines how quickly the vehicle can come to a stop.

The brake effort distribution from front to rear is also important

The performance of the ABS system for variations in the above parameters is an important consideration in ABS system design. Many logic based ABS control systems have been developed and reported in literature to address performance in the presence of the above variations.

The work by Guntur and Ouwerkerk, 1972 contains a good discussion of logic based ABS system design. It compares different logic controllers by evaluating their performance in simulations based on a mathematical vehicle model. In the simulations the authors vary three important parameters: rate of application of the brake, tire-road friction coefficients (i.e. different road conditions) and initial velocity of the vehicle. Different logic controllers are compared on the basis that, for variations in these parameters, the control unit should

1. Not fail to indicate locking of the wheel
2. Not make false predictions about locking of the wheel
3. Maintain the wheel slip within the desired range

Four different algorithms are evaluated in terms of their prediction of wheel lock. Based on their simulations results, the authors conclude that a compound condition consisting of two algorithms A_p and B_p results in the best performance (Guntur and Ouwerkerk, 1972). Method A_p sets a maximum threshold deceleration on the wheel speed, while method B_p sets another maximum threshold on the ratio of the deceleration of the wheel speed to the angular wheel speed. In the proposed compound condition, provision is made for an adaptive feature that changes the threshold values for initial velocities exceeding 35 m/s. For initial velocities lower than 35 m/s, a static threshold algorithm is found to be adequate. In considering the suitability of methods for the prediction point, the authors allow locking of the rear wheels as long as it does not cause instability of the vehicle.

Eight different algorithms are evaluated in the same paper in terms of their identification of the reselection point (Guntur and Ouwerkerk, 1972). The authors found that a compound condition consisting of methods A_r, D_r and F_r gives a good estimation of the reselection point. Method A_r is a fixed time delay condition which ensures the reapplication of the brake after a certain fixed time lapse after each time the brake is released. Method D_r is a variable condition on the desired angular velocity. The angular velocity of the wheel at the point of initial braking in the first cycle, or the corresponding signal at the point of reapplication in a subsequent cycle, is stored and the desired angular velocity is assumed to be proportional to this value. This method is used to ensure that the driver of the vehicle can conveniently influence the performance of the anti-skid system by interrupting a given braking maneuver. Method F_r reapplies the brakes whenever a threshold on the ratio of the deceleration of the wheel speed to the angular wheel speed is exceeded. It is added to improve the braking effectiveness at low vehicle speed, and also render the anti-skid system inoperative at very low speed. The compound reselection condition devised by the authors does not incorporate an adaptive feature like the one used for the prediction point condition.

5.6.5 Recent Research Publications on ABS

The development of ABS algorithms continues be an active area of research. Many research papers have concentrated on the development of algorithms that can ensure that a desired wheel slip ratio is tracked at the wheels. Detailed dynamic models of the wheel, tire, vehicle and the hydraulic system are used and the resulting system model is nonlinear. Nonlinear control system techniques are often used to ensure tracking of a desired wheel slip ratio. The measurable states of the system are the hydraulic pressure and the wheel speed. The fact that the vehicle absolute velocity cannot be measured means that the slip ratio itself cannot be measured. It must be estimated from a observer and this constitutes a very challenging problem. Accounting for changes in road surface conditions in the dynamic tire model (e.g. low friction coefficient on a slippery road) is an additional difficulty. Interesting research papers in this area include Unsal and Kachroo (1999) and Drakunov, et. al. (1995).

5.7 CHAPTER SUMMARY

This chapter provided an introduction to several longitudinal control systems, including standard cruise control, adaptive cruise control, collision avoidance, longitudinal control for operation of vehicles in platoons and anti lock brake systems. Control system design for standard cruise control and anti lock brake systems were discussed in detail. Chapter 6 will next provide a detailed discussion of adaptive cruise control while Chapter 7 will discuss longitudinal control for operation of vehicles in platoons.

NOMENCLATURE

x	longitudinal position of the vehicle from an inertial reference
$\dot{x}$ or V_x	longitudinal velocity of the vehicle
x_{des}	imaginary longitudinal position of a vehicle traveling with the reference speed
$\dot{x}_{ref}$ or V_{ref}	desired vehicle speed set by the driver
k_p, k_i	gains used in PI controller for cruise control
τ	time constant for lag in tracking desired acceleration
T_{net}	net combustion torque of the engine
T_{br}	brake torque
T_{wheel}	torque to the drive wheels
T_p	pump torque
ω_e	engine angular speed
ω_w	wheel angular speed
ω_t	turbine angular speed
c_a	aerodynamic drag coefficient
R	gear ratio
r_{eff}	effective tire radius
R_x	rolling resistance of the tires
F_x	total longitudinal tire force
F_{aero}	aerodynamic drag force
I_e	engine moment of inertia
I_t	transmission shaft moment of inertia
I_w	wheel moment of inertia
I_e	engine moment of inertia
J_e	effective inertia reflected on the engine side

m vehicle mass

V_R equivalent linear velocity of rotating wheel

a_1, a_2, a_3, a_4 acceleration thresholds used in ABS algorithm

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