CHAPTER

Active Systems

8

We can do anything — we've been to the moon. The trick is in working out what we want.

Damian Harty, AVEC 10 Keynote Address

8.1 Introduction

Modern passenger vehicles are developed to a very high level in terms of their dynamic behaviour. Indeed, there are many who believe that little more can be done to improve the performance of the road car with passive means. While it may be a little premature to make such a statement, there have been many steps forward that have been made since the start of the vehicle engineering industry:

- addition of suspension damping
- adoption of independent suspension
- adoption of hydraulic suspension damping
- adoption of hydraulic brakes
- progressive stiffening of body structure
- addition of isolating elastomers in suspension
- adoption of radial tyre construction
- decoupling of lateral and longitudinal loadpaths
- optimisation of suspension geometry
- optimisation of elastokinematic behaviour
- adoption of low profile radial tyre construction.

It is probably true that there is less to do than has been done. However, some fundamental difficulties remain with pneumatically tyred vehicles. In particular, the fall-off in yaw damping with speed is problematic. Since the fundamental nature of the tyres is to generate side forces with respect to slip angle, this leads to increased lateral velocities at increased vehicle speeds in order to respond to a given level of disturbance (Figure 8.1). This behaviour leads directly to Fonda's 'tyre-as-damper' analogy (Fonda, 1956). Since lane width in general does not increase with speed, this means the vehicle's ability to reject a given disturbance within the lane is reduced with increasing speed and hence driver workload increases. As well as the

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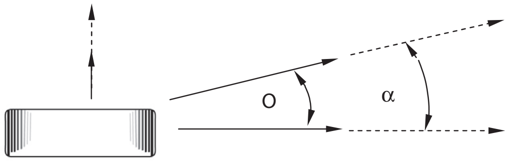

FIGURE 8.1

An increase in forward speed requires an increase in lateral speed to respond to the same disturbance.

straightforward increase in lateral velocity, consideration of the stability derivatives for the classical vehicle dynamics equations given in Milliken and Milliken (1995), and derived again in Chapter 7, shows that four important derivatives have vehicle speed as a denominator:

$$Y_r = \left(\frac{a \cdot C_F - b \cdot C_r}{V_x}\right) \tag{8.1}$$

$$N_r = \left(\frac{a^2 \cdot C_F - b^2 \cdot C_r}{V_x}\right) \tag{8.2}$$

$$Y_{VY} = \frac{C_F + C_R}{V_x} \tag{8.3}$$

$$N_{VY} = \frac{a \cdot C_F - b \cdot C_r}{V_x} \tag{8.4}$$

The dimensions a and b are the longitudinal distances of the vehicle mass centre from the front and rear axles as shown previously in Figure 7.22. The first two derivatives are the side force and yaw moment with respect to yaw velocity. The remaining two are side force and yaw moment with respect to body slip angle, which may be expressed as a function of lateral and forward velocities and hence implicitly contains forward velocity in the denominator. It is thus a fundamental consequence of the tyre behaviour that the vehicle loses restoring force with speed. This is in marked contrast to aircraft (in the subsonic region, at least) where restoring forces increase with speed since aerodynamic forces rise with the square of speed.

Note the contrast between these statements and those at the start of Chapter 7, where the control gains for ground vehicles can be seen to increase significantly with speed. This presents vehicle engineers with some particular problems since control authority is increasing while system stability is reducing — a ‘closing gap’ scenario in which the safe behaviour of vehicles at Autobahn speeds remains difficult to engineer.

8.2 Active systems

This 'closing gap' means the industry as a whole is considering the use of active systems to complement the passive behaviour of the vehicle. Traditionally, 'active' has meant a system in which energy is added in some substantial amount. In modern usage, this distinction is becoming lost and so the term is being used for any system that performs something other than a passive mechanical reaction. Previously, systems that modified their behaviour without adding energy were referred to as 'adaptive'. In general, systems that combine mechanical actuation with electronic control are referred to as 'mechatronic', although this term appears infrequently in industry literature.

To model any type of mechatronic system requires the introduction of sensors into the vehicle model and the implementation of the control law. Note that sensors in this context are different to software entities that may be also called 'sensors'. One convenient way is to add state variables in a manner similar to that described for driver modelling. One important difference, however, is that the driver model is 'continuous' – that is to say the resulting outputs may be differentiated without discontinuity.

An aspect of many active systems is that they use 'logic controller' principles (see Chapter 6) and thus may branch between conditions in a very short space of time. If carelessly modelled, this may give convergence problems inside the vehicle model and so the use of some smoothing method is required. Note that these difficulties are present when implementing real-life systems and not confined to simulation. In both MSC ADAMS and Simpack, the STEP function described in Chapter 3 allows the transition between one level and another using a half sinusoidal form that is continuously differentiable.

8.2.1 Full authority active suspension and variable damping

Lotus performed a great deal of work in the 1980s looking at 'active suspension', a fast acting system that varied the vertical load on each corner in accordance with control laws attempting to preserve ride height, minimise wheel load variation and minimise body acceleration. This system was undoubtedly very effective, particularly on the attitude-sensitive Formula 1 car. However, expensive components and significant power consumption mean that full authority active systems, which have the authority to substantially raise and lower the vehicle if so commanded, remain uncommon. In the 10 years since the first edition of the book, to the authors' knowledge only the Mercedes 'Active Body Control' system has remained available (Figure 8.2), despite the earnest attempts of marketing departments elsewhere to convince us that there is no difference between variable damping and full authority active suspension.

When compared to the original Lotus system, the Mercedes system is a down-specified version, working only with primary ride motions (up to around 5 Hz) rather than the higher frequency wheel-hop control achieved by the Lotus motorsport and SID research vehicles. In April 1990, Inifiniti, the luxury division of Nissan,

FIGURE 8.2

Mercedes Active Body Control (left) and a passively suspended vehicle (right) in action at high lateral accelerations – around 1° of body roll (a combination of tyre compliance and some programmed roll compliance for 'feel') with the ABC car compared to 4.5° of body roll for the passively suspended car.

(Courtesy of Auto Motor und Sport.)

introduced a similar hydraulic 5 Hz system for the 1991 model year Q45, badged as Q45a when carrying the 'Full Active Suspension' (FAS). This only remained on the market until the 1996 model year, when it was withdrawn the year before the face-lifted model was introduced. Nissan cited low take-up of the FAS option as the reason for its deletion, with internet rumours suggesting only 15% of US customers subscribing to it. Around the same time, Toyota showcased similar technology as part of a well-integrated *tour de force* on the UZZ32 Soarer (Sato et al., 1992) and subsequently on the 1992 Corolla. Neither vehicle sold in large volumes, suggesting the difficulty with full authority active suspension is not the technicalities but the commercial reality of persuading customers that the value is worth the cost.

Recently, Mercedes has introduced the system many engineers dreamed of in the early 1980s by adding a previewing feed-forward aspect to their active suspension, marketing it as 'Magic Body Control' in an attempt to distance it from damper control systems (Figure 8.3). Previewing is a good way to circumvent bandwidth limitations. It is easy to demonstrate conceptually in multibody systems (MBS) models if a deterministic road profile (e.g. a flat road with a cosine step at a certain distance) is employed, by using a location offset in looking up the road profile function. However, in a more general model with an arbitrary road surface – a scanned physical surface, for example – then the preview is more difficult to achieve. Visual terrain mapping of the kind used by the Mercedes system is more or less impossible to implement inside general purpose multibody software. Mercedes' own publicity material suggests the camera previews to about 15 m ahead of the vehicle but does not describe the spatial resolution. If such a preview were desired with an existing road profile then it could conceivably be implemented by conceptually extending the vehicle body component the requisite distance ahead of the vehicle and attaching a large number of small contact blocks to it. The contact properties for the blocks would be set to generate very little force so as not to influence the solution in any

FIGURE 8.3

Mercedes Magic Body Control (top) with road preview compared to an unspecified configuration (bottom) — the improved absolute pitch control is very apparent.

(Courtesy of Mercedes.)

meaningful way. Interrogating the contact elements for their penetration will make a preview of the road profile available to control algorithm.

Partial active systems, working only against body roll, have included Citroën's 'Activa' system fitted to the Xantia (not to be confused with the two concept cars of the same name) from 1995 until 2000, when it was replaced by the C5 model that reverted to the adaptive roll stiffness/damping system known as 'Hydractive II'. Land Rover's 'Active Cornering Enhancement' is a hydraulically actuated fast roll levelling system, working around 5 Hz and available on Land Rover Discoveries from the 1999 Model year to date, and has made its way upmarket onto Range Rover models in a reversal of the normal trend. BMW's Active Roll Stabilisation system is conceptually similar and available on X5, X6 and X7 series platforms.

Active suspension systems and their cousins, continuously variable damper systems, are easy in concept to implement in MBS models. Action/reaction force pairs are introduced into each vehicle suspension unit in addition to the normal spring, damper and anti-roll bar forces. The magnitude of these forces is controlled by control laws in a similar fashion to that in the real vehicle. For example, a system (Figure 8.4) might use an open-loop roll moment method by applying forces proportional to the output from a lateral accelerometer. A further modification might be the addition of a roll damping term based on relative velocities of left and right suspension units, with a lookup table (spline) for use at different vehicle speeds.

Table 8.1 shows a sample of an MSC ADAMS command file implementing such a system on the front suspension of a vehicle.

Active suspension in principle allows the control of individual tyre reaction forces in a manner decoupled from platform orientation. As well as allowing a level platform during braking and cornering events (preserving suspension travel for disturbance events and improving the angle at which the tyre is presented to the road), it allows the redistribution of roll moment reaction on a moment-by-moment basis. The mechanisms discussed in Chapter 7 for inducing over- and

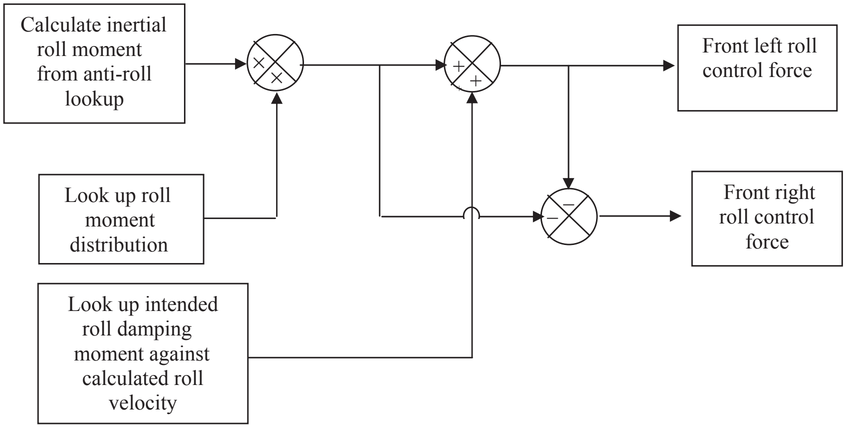

FIGURE 8.4

Simplified active suspension model.

understeer can thus be harnessed and used to improve turn-in behaviour, yaw damping and so on.

Note that in the example, no attempt has been made to filter the output down to any particular bandwidth. Filtering is a lengthy topic in its own right and is best not approached glibly. Many software environments allow the use of filtering techniques within their postprocessing tools. However, for simulating active systems, some kind of filtering is almost always necessary at run-time. Whether to simulate the limited bandwidth of an actuation device or to smooth noisy data, some form of filter must be implemented while the model is running.

Some MBS codes have access to a sophisticated run-time filtering library, such as those set up in the Matlab product environment. Others, like MSC ADAMS and Simpack, lack such run-time tools but do have the ability to enter a generic transfer function command. The filter transfer function itself must be arrived at using some external filter design tool. Particular caution is needed since some software environments use descending order for the terms in the transfer function while some use ascending order, as illustrated in Eqn (8.5), which represents a 1 Hz 2nd order Butterworth filter in the Laplace domain as used in Matlab and MSC ADAMS:

$$g(s) = \underbrace{\frac{39.884}{s^2 + 8.8869s + 39.884}}_{\boxed{\text{Matlab Description}}} = \underbrace{\frac{39.884}{39.884 + 8.8869s + s^2}}_{\boxed{\text{MSC.ADAMS Description}}} \tag{8.5}$$

More sophisticated filtering is possible using higher order transfer functions and using the more general state-space modelling methods, which are available in most MBS environments. The interested reader is referred to 'Digital Filtering in the Time and Frequency Domain' (Blinchikoff and Zverev, 2001) for detailed discussion of

Table 8.1 MSC ADAMS Command File Sample for Simplified Active Suspension

```
data_element create variable &
 variable_name = inertial_sprung_roll_moment &
 function = "(VARVAL(latacc)* (mass)*", &
        "(DZ(m_body_CG,base) - ", &
        " (AKISPL(varval(fl_suspension_position),0,front_anti_roll_spline) ", &
        " * (front_track)/2)* varval(fl_suspension_load)/varval(total_load)", &
        " + AKISPL(varval(fr_suspension_position),0,front_anti_roll_spline) ", &
        " * (front_track)/2)* varval(fr_suspension_load)/varval(total_load)", &
        " + AKISPL(varval(rl_suspension_position),0,rear_anti_roll_spline) ", &
        " * (rear_track)/2)* varval(rl_suspension_load)/varval(total_load)", &
        " + AKISPL(varval(rr_suspension_position),0,rear_anti_roll_spline) ", &
        " * (rear_track)/2)* varval(rr_suspension_load)/varval(total_load)", &
        ")))
data_element create variable &
 variable_name = roll_moment_load_front &
 function = "varval(inertial_sprung_roll_moment)*", &
            "AKISPL(varval(latacc),0,roll_moment_distribution_spline)", &
            "/(front_track)"
data_element create variable &
 variable_name = roll_damping_load &
 function = "( ", &
            " (VZ(fl_damper_top,fl_damper_bottom) + ", &
            "  VZ(rl_damper_top,rl_damper_bottom)) -", &
            " (VZ(fr_damper_top,fr_damper_bottom) + ", &
            "  VZ(rr_damper_top,rr_damper_bottom)) ", &
            ")* ((front_track)+(rear_track))/2* ", &
            "* AKISPL(varval(velocity),0,roll_damping_velocity_spline)
force create direct single_component_force &
      single_component_force_name = front_left_active_force &
      i_marker_name = .fl_damper_top &
      j_marker_name = .fl_damper_bottom &
      function = "(+varval(roll_moment_load_front) + varval(roll_damping_load)", &
                 "* STEP(TIME,0.0,0.0,1.0,1.0)"
force create direct single_component_force &
      single_component_force_name = front_right_active_force &
      i_marker_name = .fr_damper_top &
      j_marker_name = .fr_damper_bottom &
      function = "( - varval(roll_moment_load_front) - varval(roll_damping_load)", &
                 "* STEP(TIME,0.0,0.0,1.0,1.0)"
```

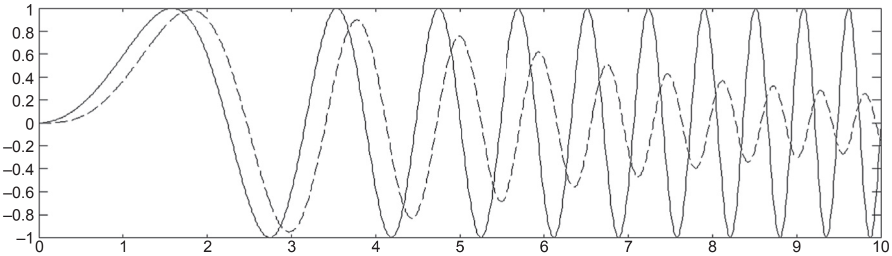

FIGURE 8.5

2-pole Butterworth filter described in Eqn (8.5) applied to signal in Eqn (8.6) (solid) — resulting signal in dashed line.

filtering methods and their repercussions. The most important thing to recognise is that almost any form of real-time/run-time filtering introduces some form of phase delay. Using the example transfer function in Eqn (8.5), an input chirp signal in Matlab defined by Eqn (8.6) produces an output as shown in Figure 8.5.

$$x = \sin(2\pi \cdot \frac{\text{time}^2}{10}) \tag{8.6}$$

Note that the amplitude attenuation is very gentle but that a phase delay has been introduced even at frequencies well below the notional cut-off frequency. Many software packages offer 'zero phase shift' Butterworth filtering by running the filter forwards and backwards on logged data, but a moment's thought should reveal this is unavailable in real-time applications in the absence of time travel technology.

The 2-pole Butterworth filter represents a linear second-order mechanical system and so its intelligent use can be very helpful in avoiding modelling complexity. The interested reader is encouraged to study its formulation and implementation in different forms — pole/zero and Laplace polynomial — as a launch-pad to a deeper understanding of real-time filtering issues and complexities.

Adaptive damping logic can be implemented in a similar fashion to active suspension inside an MBS model, scaling a damper spline or adding a scaled 'damper variation' spline to a 'minimum damping' spline according to a controller demand. The 1988 Lancia Thema 8.32 is believed by the authors to be the first production implementation of an adaptive damping system, with two-state dampers switching rapidly between 'soft' and 'hard' settings according to an algorithm that examined ride accelerations and driver inputs.

Active suspensions and continuously variable dampers seek to address the traditional three-way trade-off in suspension calibration (Figure 8.6). Given enough working space it is possible to improve both ride and handling simultaneously, as evidenced by desert race vehicles.

However, in road cars with suspension travel limited by prosaic packaging constraints, generally ride is achieved at the expense of handling or vice versa. Crolla comments (Crolla, 1995) that good control of the suspension dampers delivers

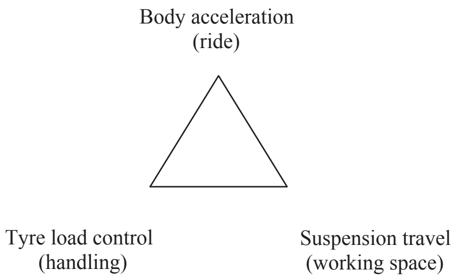

FIGURE 8.6

Suspension trade-offs.

many of the benefits of active control without the power consumption. Thus it seems to the authors to be the 'thinking man's' solution to improved suspension calibration. It also offers the prospect of software control for calibration of suspension dampers and making inroads into development times. The recent proliferation of adaptive damping systems suggests Crolla's observations were indeed correct.

Adaptive damping systems also offer an intriguing possibility for logistical convenience for companies making use of so-called 'platform engineering', when common hardware is used across a diverse range of products. The Volkswagen group of companies, — making vehicles badged Skoda, Seat, VW and Audi on the same hardware platforms — have a range of dampers that are physically interchangeable. When considered in conjunction with the engine and body style variants this might make something like 20 dampers with unique internals but an interchangeable vehicle interface — which is to say 19 opportunities to fit the wrong damper. A damper that communicates with the vehicle and is instructed by the vehicle to select a different set of parameters could in concept become a single part serving the entire platform family. This certainly brings logistical savings and, given the remorselessly downward trend in electronics costs, there must logically arrive some tipping point when the cost of the adaptive hardware is offset by the savings in logistics costs (which, in contrast, are not changing a great deal).

8.2.2 Brake-based systems

Other systems are less comprehensive but can have large benefits. In 1978, Mercedes introduced the Bosch 'Anti-Blockier System' as an option on the Mercedes S-Class. This system, universally abbreviated to ABS (albeit with some contortions to translate the German expression for 'anti-lock' into English), releases, holds and reapplies brake pressure in individual wheel brakes in order to retain directional stability and steering control with very little sacrifice in braking capability. Although technically it is possible for a skilled driver to outbrake anti-lock braking system (ABS), in truth the level of rehearsal this requires means it is unlikely in a road car in any realistic situation. Highly rehearsed circuit driving is an area where skilled drivers may improve over ABS performance and so circuit cars rarely use ABS.

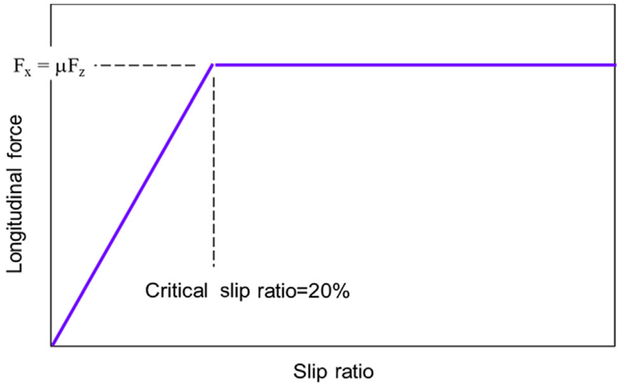

FIGURE 8.7

A simplified slip ratio versus force curve for the following ABS discussions.

Described in some detail in Limpert's work (1999) is the function of a vehicle ABS system. The main variable is the brake pressure. In the work by Ozdalyan (1998) a slip control model was initially developed as a precursor to the implementation of an ABS model. In order to discuss ABS operation, a simplified tyre slip characteristic can be used. Such a characteristic is shown in Figure 8.7. It can be seen to be made up of two linear portions and has a constant friction coefficient, μ.

ABS operation is essentially the interplay between two opposing torques, the frictional torque from the brake rotor and the spin-up torque resulting from friction between the tyre and the ground as illustrated in Figure 8.8. At any speed substantially above zero, the brake friction torque is broadly independent of speed but the tyre friction torque varies significantly with slip ratio (which is in essence a measure

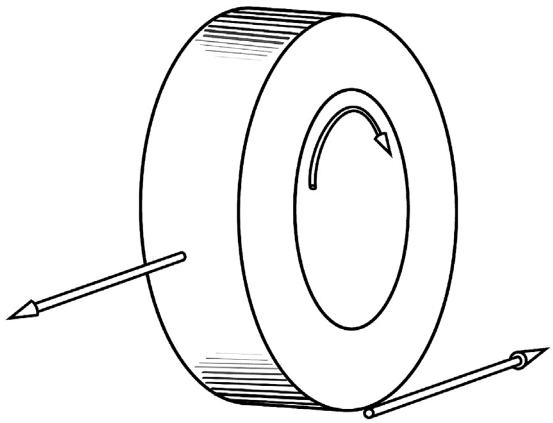

FIGURE 8.8

Anti-lock braking system operation is the balancing of two torques against each other.

of the sliding velocity of the contact patch, as discussed in Chapter 5). Spin-up torque also varies substantially with surface condition, varying by an order of magnitude between dry, smooth road and slightly wet ice.

The angular acceleration of a wheel, $\dot{\Omega}_{wheel}$, can be simply described with Newton's Second Law.

$$\dot{\Omega}_{wheel} = \frac{T_B - T_S}{I_{wheel}} \tag{8.7}$$

Where T_B is brake torque, T_S is spin-up torque and I_{wheel} is the mass moment of inertia in the rolling direction. If this wheel is connected to a single mass representing the vehicle then the linear deceleration of the wheel is given by:

$$A_{vehicle} = \frac{T_S}{R_1 m} \tag{8.8}$$

The above equations can be readily implemented in, say, MS-Excel to investigate their behaviour. Ramping on brake torque to a certain level, the wheel can be made to hold at 19.9% slip, as shown in Figure 8.9. A period of initial deceleration of the wheel can be seen, followed by a settling of the effective linear wheel velocity (its angular velocity multiplied by its radius) to its steady state value of approximately 80% of the vehicle linear velocity during the stop. A more aggressive brake torque ramp to a higher level causes the wheel to rapidly decelerate. The rapid nature of the deceleration can be explained by considering the net torque on the wheel during the non-locked stop in comparison to the locking stop — the saturation of the spin-up torque means the full additional brake torque is applied unopposed to the wheel, and it is very large in comparison to the net torque prior to locking.

Considering Figure 8.9, it can be seen that monitoring wheel angular deceleration gives a clear signature for a wheel entering a lock event. It could be imagined that some software might watch for a value below -400 rad/s and intervene when this is detected.

For real tyres, in Figure 8.10 it can be seen that on initial application of the brakes the brake force rises approximately linearly with slip ratio depending on the wheel load. If the braking is severe the slip ratio increases past the point where the optimum brake force is generated. To prevent the slip ratio increasing further to the point where the wheel is locked, an ABS system will then cycle the brake pressure on and off maintaining peak braking performance and a rolling wheel to assist manoeuvres during the braking event.

In the Ozdalyan (1998) model, the brake pressure is found by integrating the rate of change of brake pressure, this having set values for any initial brake application or subsequent application during the ABS cycle phase. The key ingredient of such a system is the ability to control brake pressure in one of three modes, often described as 'hold, dump and pump'. Hold is fairly self-explanatory, the wheel cylinder pressure is maintained regardless of further demanded increases in pressure from the driver's pedal. 'Dump' is a controlled reduction in pressure, usually at a predetermined rate and 'pump' is a controlled increase in pressure, again usually at a

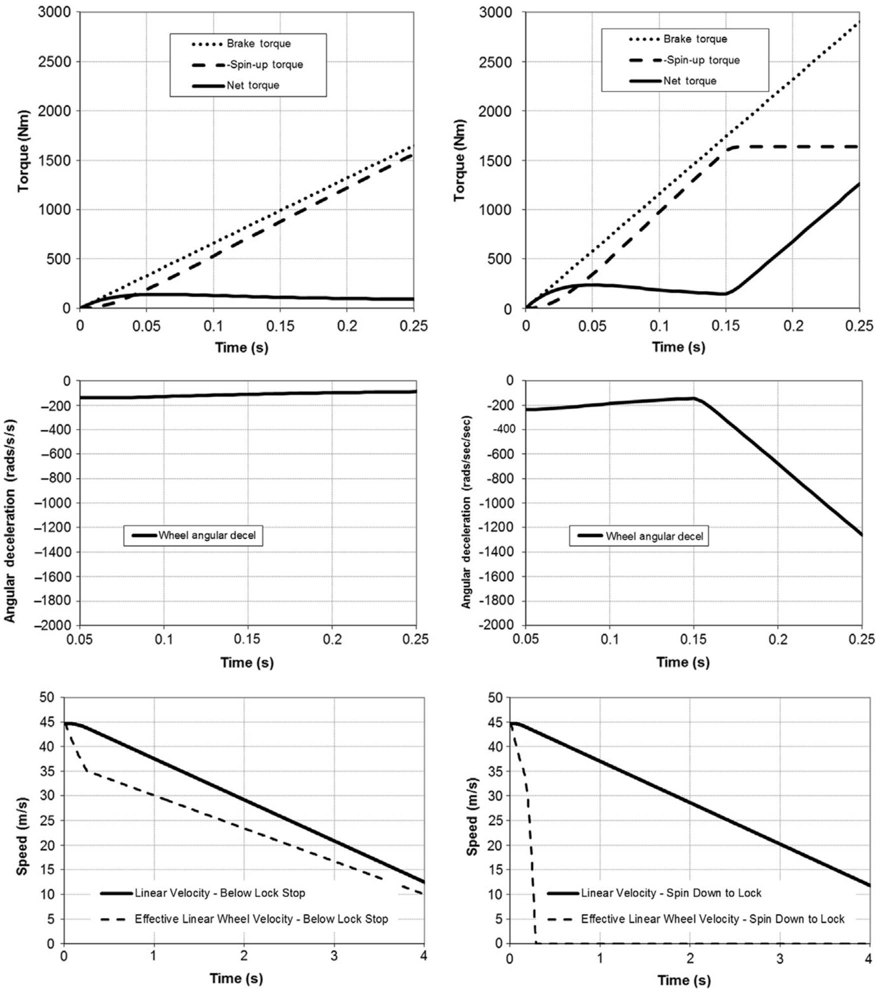

FIGURE 8.9

A 'below lock' stop (left) in contrast to a 'locked wheel' stop (right); note linear velocities are indistinguishable between the two (bottom graphs).

predetermined rate. Implementation of these changing dump, pump and hold states requires care to ensure no discontinuities in the brake pressure formulation.

To explore their influence in an ABS system, we can return to the simplified formulation used earlier and imagine four ascending levels of hardware complexity, as shown in Figure 8.11.

Using the hold valve alone is ineffective for controlling wheel speed, resulting in either a slight slowing of the wheel lock event or a reduction in vehicle deceleration, depending on whether it is triggered early or late (Figure 8.12).

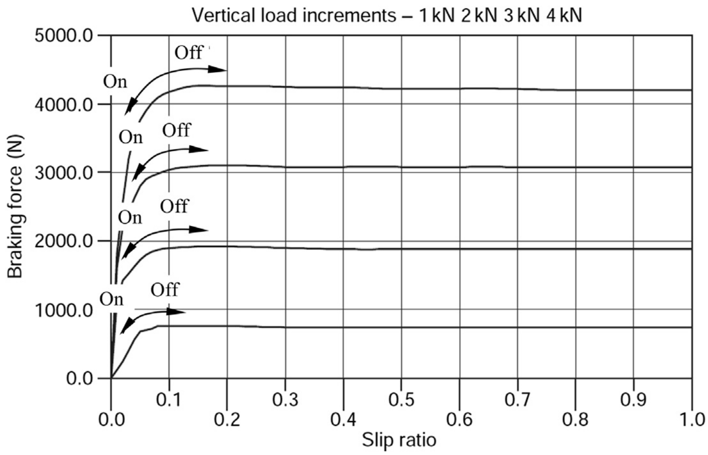

FIGURE 8.10

Principle of a brake slip control model.

Using a dump valve to bleed off excess pressure at a predetermined rate until the wheel spins up again is substantially more successful in principle. This strategy might be called Hold-Dump-Hold and is shown in Figure 8.13. However, although successful at preventing wheel lock, it does not optimise stopping distance and it does not deal well with low-to-high friction transitions.

Closing the dump valve and reopening the hold valve reconnects the master cylinder to the system and allows the reintroduction of pressure at a controlled rate. This is illustrated in Figure 8.14. Note that for the simplified friction model and idealised system shown here the strategy manages to discover the ideal brake

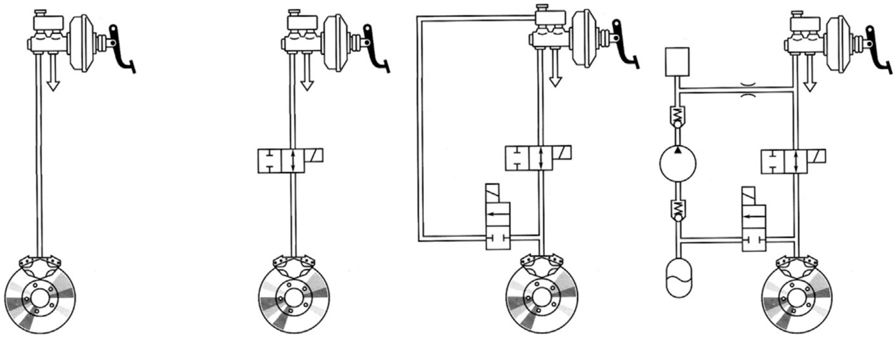

FIGURE 8.11

Ascending levels of brake hardware complexity — from left to right: simple push through, hold valve, dump and hold valves, dump/pump/hold valving. Illustrations based on those in Driving-safety Systems.

(Bosch, 1999.)

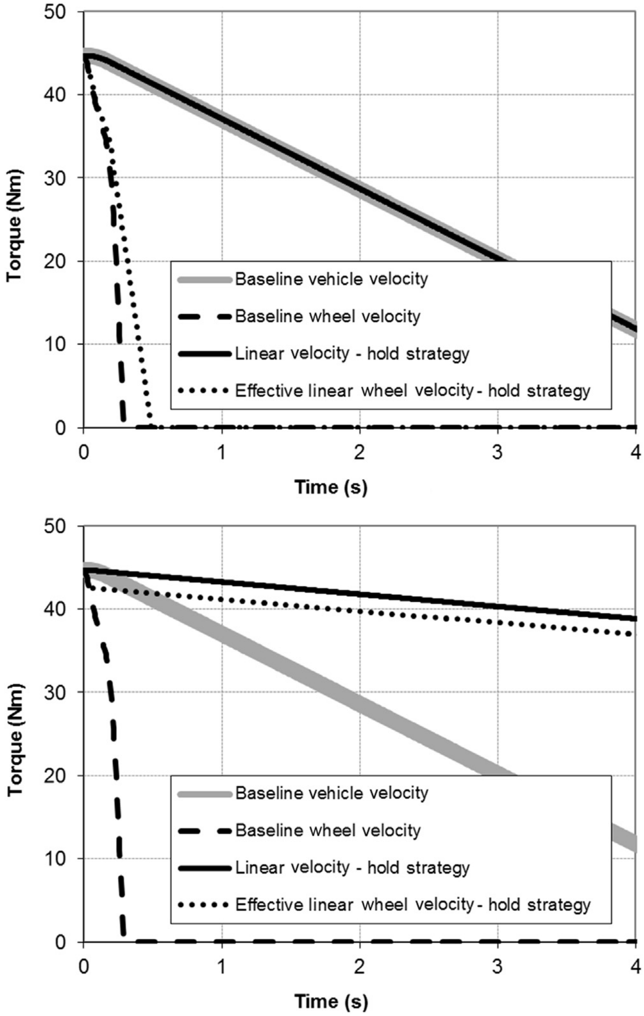

FIGURE 8.12

A comparison of unsuccessful hold-only strategies — too late (top) and too early (bottom); the hold threshold varies by 5% between the two.

pressure to prevent wheel lock and achieve deceleration that is indistinguishable from the baseline system; in real systems the nonlinear friction characteristics of the tyre combined with transport delays in the system mean that even on a smooth, consistent surface the system continues to cycle with the familiar ABS 'pulsing'. It can be readily seen that repeated cycles deplete the fluid from the master cylinder and so a pumped return is often employed. Both the cycling and the pumped return can give haptic sensations that surprise operators.

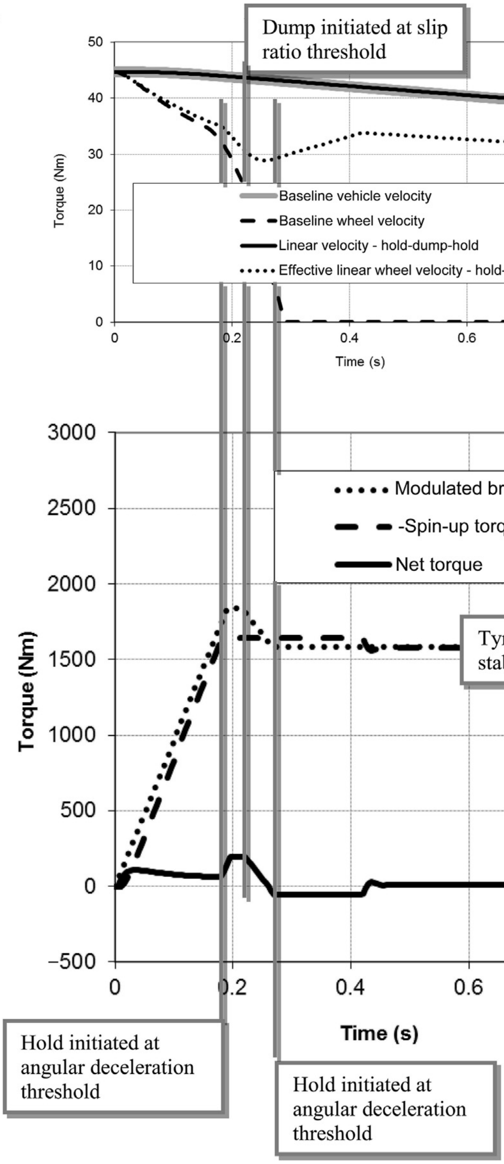

FIGURE 8.13

A hold-dump-hold strategy with the simplified 2 DoF vehicle braking model.

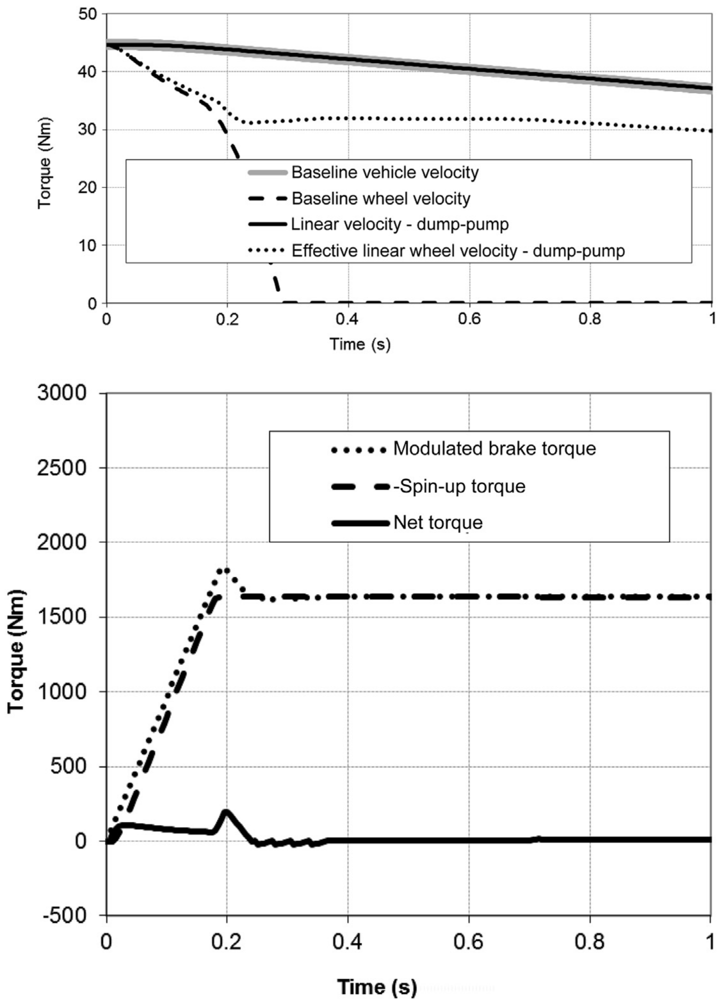

FIGURE 8.14

A hold-dump-hold-pump strategy with the simplified 2 DoF vehicle braking model.

The modelling in MBS of more realistic ABS algorithms (van der Jagt et al., 1989) is more challenging as the forward velocity and hence slip ratio is not directly available for implementation in the model; this of course is also the challenge to implementing the algorithms in real life. The implementation of such a model allows the angular velocity of the wheel to be factored with the rolling radius to produce an output commonly referred to as 'wheel speed' by practitioners in this area. A plot of wheel speed is compared with vehicle speed in Figure 8.15, where the typical oscillatory nature of the predicted wheel speed reflects the cycling of the brake pressure during the activation of the ABS model in this vehicle simulation.

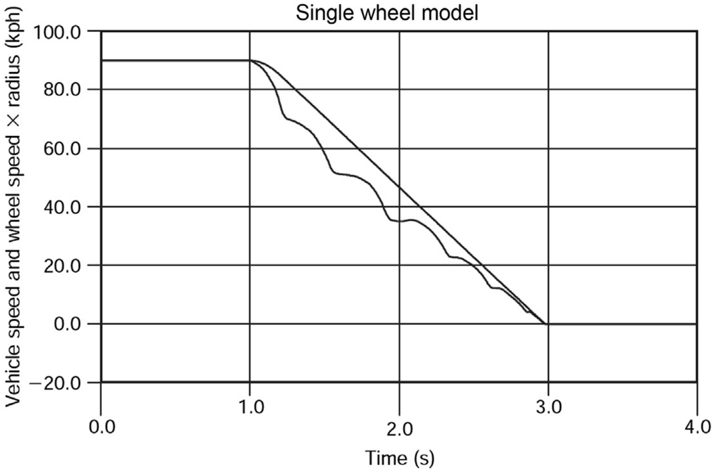

FIGURE 8.15

Plot of vehicle speed and wheel speed during from ABS braking simulation.

In 1995 Mercedes introduced the Bosch 'Electronic Stability Program' (ESP) on the Mercedes S-Class. This system applies the brakes asymmetrically and without the driver's foot applied to the brake pedal, in order to exert restoring yaw moments on the vehicle. While previous ABS strategies had an element of open-loop control to minimise the build-up of yaw moment for asymmetric braking situations, ESP proceeds to the next logical step of closed-loop control of yaw rate. Desired yaw rate is mapped through testing on the vehicle and compared with actual yaw rate. Conceptually this is similar to the driver control model described in Chapter 7. However, a fundamental difference is that the brake actuators are not always controlled as proportional devices. To suppress understeer, brake torque is typically applied at the inside rear wheel in a manner substantially proportional to the magnitude of the understeer. For oversteer, a different strategy is employed. First, a threshold is set below which no intervention occurs and second, the system introduces a defined amount of brake torque for variable lengths of time to trim oversteer. It is also typical that engine torque will be reduced under these conditions. This gives rise to the typical 'lumpy' sensation of the operation of ESP and the belief for many drivers that it is a relatively crude system. In truth, it is extremely elaborate, but the requirement not to burn out the brakes during spirited driving means this 'threshold' control is necessary.

Both ABS and ESP control can be difficult to model for reasons of the discontinuities described. In the real world, finite hydraulic bandwidth imposes its own smoothing on the system but the operation of ABS systems remains subject to a repeated series of 'impact' noises during sharp hydraulic transients. Only the very latest electrohydraulic braking technology succeeds in avoiding this. Cosimulation between multiple modelling codes is the answer suggested by the industry for all these problems although this is not preferred for a variety of reasons, described in Chapter 6. Another alternative is so-called ‘hardware-in-the-loop’ simulation that uses a real physical brake system with pressure sensors to feed the behaviour of the brake system back into the model. The model computes the effects on the vehicle and feeds vehicle state information out to ‘virtual sensors’ that replace the real sensors on the vehicle, making its decisions and affecting the real brake system to complete the loop. Although potentially accurate, this type of modelling is more suited to confirmation work as shown on the right-hand side of Figure 7.45.

The biggest difficulty in real-world systems is the accurate discernment of the vehicle speed under conditions when the wheel speeds are only loosely connected to it and each other. This is usually referred to as the ‘reference speed’ problem. Another significant difficulty is the need to control the vehicle by varying longitudinal slip ratio when the actuator is a torque-control device. On ice, for example, 10 Nm may be enough to lock the wheel whereas on dry pavement, 10 Nm will produce barely any change in slip ratio. For both these reasons, ABS and ESP strategies are more complex than might first be imagined; differences in effectiveness and implementations largely come down to the sophistication of the reference speed calculation and friction estimation.

8.2.3 Active steering systems

In 1986, Nissan launched their High Capacity Active Steering (HICAS) system on the rear axle of the R31 Skyline Coupé. Honda followed with a mechanical system on the Prelude and Mitsubishi were close behind in the Lancer, using an electronically controlled system like the Nissan. Nissan used the notion of a compliant sub-frame with its location controlled by a hydraulic actuator, while Honda and Mitsubishi essentially duplicated the front steering system at the rear of the vehicle, using a central rack. The objectives of the systems were clear and there is a great deal of literature published on the subject (Ro and Kim, 1996). Four-wheel steer seeks to control the body slip angle and rate, and hence has a profound effect on driver impressions of the vehicle. It also has the comparatively trivial effect of tightening the turning circle at low speeds. Honda has subsequently dropped four-wheel steer, declaring that ‘advances in tyre technology have rendered it unnecessary’. Mitsubishi kept it until 1999 on their 3000 GTO model, while Nissan continue with a development now called Super-HICAS system on the Skyline.

A difficulty with four-wheel steer is that, while it makes the vehicle feel excellent in the linear region (through enhanced body slip angle control), as the handling limit approaches the flat nature of the tyre side-force-versus-slip-angle curve means that its ability to improve vehicle control disappears. In this sense, it is possibly the worst type of system – enhancing driver confidence without actually improving limit capability. Four-wheel steer is easily simulated in MBS modelling, with an additional part to represent the rear steering rack and forces applied to it according to a control law as with other systems.

BMW announced ‘Active Front Steer’ in conjunction with ZF for the 2004 model-year 5-Series. Consideration of the behaviour of competition drivers, particularly rally drivers, suggests the potential for this system is high and that it offers a much more continuous control than brake-based systems. Although onerous, the potential failure modes have clearly been overcome. It also offers the chance to overcome the problem of increased control sensitivity at high speeds by reducing the yaw rate gain progressively with speed. It does not suffer any of the problems of rear-wheel steering in terms of limit control.

There are several patents for an ‘Active Toe Control’ system for application to all four wheels. This is primarily an on-centre modifier for the vehicle and is intended to complement the torque distribution systems they are also known for. Such a system is postulated in Lee et al (1999) and has been prototyped on a research vehicle. The interaction between such a system and a torque distribution system is explored in (He et al., 2003).

Modelling active steer systems is in principle identical to modelling a driver. An external force is applied to the steering system – either the rack or column in response to an algorithm. The algorithm is typically similar to that in use for a brake-based stability system in that it will calculate a target yaw rate and then use the steering system to attempt to achieve that yaw rate.

8.2.4 Active camber systems

Milliken Research Associates produced an active camber Corvette in the mid-1980s with closed-loop control of yaw rate following the success of the ‘camber racer’, the latter recently run again at Goodwood on modern motorcycle tyres. Mercedes have also produced the F400 Carving that runs an active camber system. Although attractive in function, the increased package requirements in the wheelhouse make active camber something that will probably not be applied to road cars.

8.2.5 Active torque distribution

Nissan’s Skyline R32 Coupé had the somewhat bewilderingly named ATTESA-ETS (Advanced Total Traction Engineering System for All-Electronic Torque Split) system in 1989. The Porsche 959 had similar technology in the same year, although without the lengthy acronym. Since then, systems of torque redistribution according to handling (as against traction) priorities have remained elite in production cars. The author’s (Harty) work at Prodrive revived interest with several low-cost Active Torque Dynamics (ATD) systems suitable for a wide variety of vehicles in the period 1999–2004, but commercial restrictions prevent discussion of which, if any, achieved production status. In essence, traditional driveline technology systems seek to minimise wheelspin, whereas these systems seek to connect drive torque distribution to vehicle handling via closed-loop feedback. From a modelling point of view, they are simple to implement in that some state variables are declared (typically a target yaw rate and current state, similar to ESP systems) and the actuation forces are implemented accordingly.

There exists a large amount of confusion surrounding torque distribution, which frequently exercises students. It is commonly declared, with some confidence, that 'a locked differential produces a 50/50 torque split'. The origins of this statement are unknown and certainly do not bear scrutiny. Where a traditional open differential of the bevel or epicyclic type does indeed produce a torque split of known proportions, a locked differential — in essence a shaft — produces an equality of speed outputs. How these speeds turn into torques is governed entirely by the wheel to which they are attached — it is a function of vertical load, friction coefficient and slip state. The latter is influenced by vehicle states and, of course, the differential locking condition. These comments apply whichever differential is under consideration.

Figure 8.16 shows a comparison at the rear axle of a vehicle in four different states, combinations of an open and locked differential, applied drive torque and no applied drive torque. The tractive force arrows shown are dimensionally representative of torques in the half shafts, since no other mechanism is applying a force to the tyre.

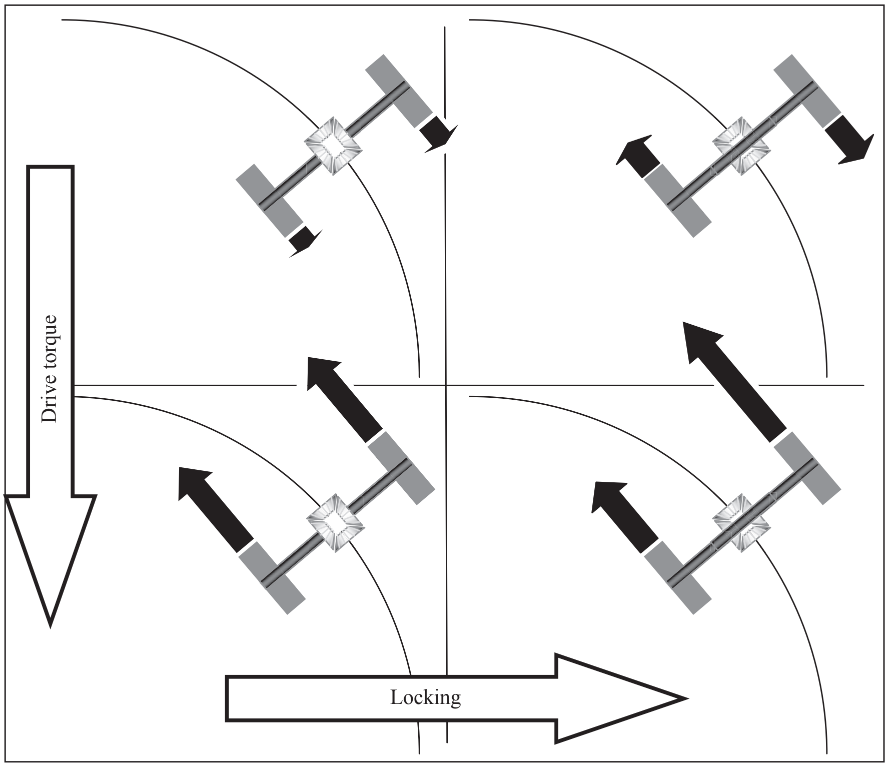

FIGURE 8.16

Locked differentials do not result in equal torque distribution — this is a common myth.

In the upper left quadrant of Figure 8.16, in the presence of an open differential and the absence of driveline torque, the only forces present at the contact patch are the rolling resistance. The outer wheel may be expected to have a slightly higher rolling resistance due to its higher load, associated with weight transfer as discussed in Chapter 7. The individual wheel rotational speeds are related to their individual radius of turn and rolling radius.

In the lower left quadrant of Figure 8.16, in the presence of an open differential and with driveline torque, the forces at the contact patch may be expected to be the sum of the rolling resistance and drive torque. When the drive torque is large and neglecting any frictional torque in the differential, the forces are broadly equal; this is the fundamental purpose of the differential. The individual wheel speeds are the sum of the wheel speeds from the upper left quadrant and the wheel speeds required to deliver the slip ratio, which in turn delivers the required reaction torque to the sideshaft such that it is in equilibrium. Note that a lightly laden inner wheel and a high drive torque may exceed the tyre's ability to generate such a reaction torque, in which case no equilibrium state will be reached and the sideshaft (and hence wheel) rotational speed will continue to increase until torque is interrupted — typically by an attentive driver, an engine speed limiter or a traction control system.

In the upper right quadrant of Figure 8.16, with a locked differential imposing equal rotational speeds across the axle and in the absence of driveline torque, it can be seen that something approximating the average of the speeds from the upper left quadrant is imposed on both wheels. Since this is slower than the free-rolling speed for the outer wheel, it constitutes a negative slip ratio and produces a rearward-acting force at the contact patch. Since it is faster than the free-rolling speed for the inner wheel, it constitutes a positive slip ratio and produces a forward-acting force at the contact patch. Together these produce a yaw moment on the car in opposition to the turn; it is common experience that 'stiff differentials blunt turn-in'. This effect can be used in active systems to introduce an amount of yaw damping to a vehicle. It works with both cross-axle differentials and with the centre differential in an all-wheel-drive vehicle.

In the lower right quadrant of Figure 8.16, with a locked differential imposing equal rotational speeds across the axle and in the presence of driveline torque, positive slip ratios are induced on both wheels. The slip ratios are such that, while satisfying the constraint of equal speeds in both shafts, the total reaction torques sum to the applied shaft torques. The weight difference between the two wheels results in more torque for a given slip ratio, as described in Chapter 5. The geometric effects of the turn tend to slightly increase the slip ratio on the inner wheel but for realistic turn radii these effects are relatively small compared to weight transfer effects. The net result tends to add yaw moment with driveline torque for a cross-axle differential. This effect is readily observable in both front- and rear-wheel drive vehicles when a comparison can be made with and without cross-axle locking.

A locked differential does not result in a 50/50 torque split under most imaginable circumstances. The persistence of the myth that it does is somewhat baffling.

A fully locked drive axle has an unfortunate tendency to spin up both wheels simultaneously, which robs the vehicle of side force on the drive axle by broadly collapsing the cornering stiffness of both rear tyres simultaneously. From the discussions of anti-lock brake control, it can be seen that the runaway phenomenon in the presence of excess drive torque can be very rapid indeed. For rear-wheel-drive vehicles in particular, locked differentials spin the vehicle while open differentials spin a wheel. It can be imagined that spinning the vehicle is a rather more onerous failure mode and it is true that limited slip differentials (which may be regarded as some way along the spectrum to locked) have a reputation for being difficult to drive at the limit. A control system for a proportional limited slip differential that can recognise the vehicle states and reduce the differential locking in response to excess yaw rate is extremely effective, in the author's experience, at limiting vehicle departure behaviour both on split friction and uniform friction surfaces.

Driveline devices that impose a left-right speed difference have also been constructed. The Mitsubishi Active Yaw Control, Super Active Yaw Control and Super All Wheel Control (AYC, S-AYC and S-AWC respectively) use a clutch and additional gear wheels to deliver a limited slip differential in which the 'fully locked' state results in a fixed speed ratio between one sideshaft and the other — typically differing by the order of 10%. The requirement for symmetric control results in a second clutch and a second pair of gear wheels. Honda's Super Handling All Wheel Drive (SH-AWD) and BMW's Dynamic Cornering Performance systems use similar arrangements in principle.

Such a device might be known generically as an 'asymmetric overspeeding differential' and can be represented in a multibody model using the topology shown in Figure 8.17. A three-element coupling equation represents the open differential, an

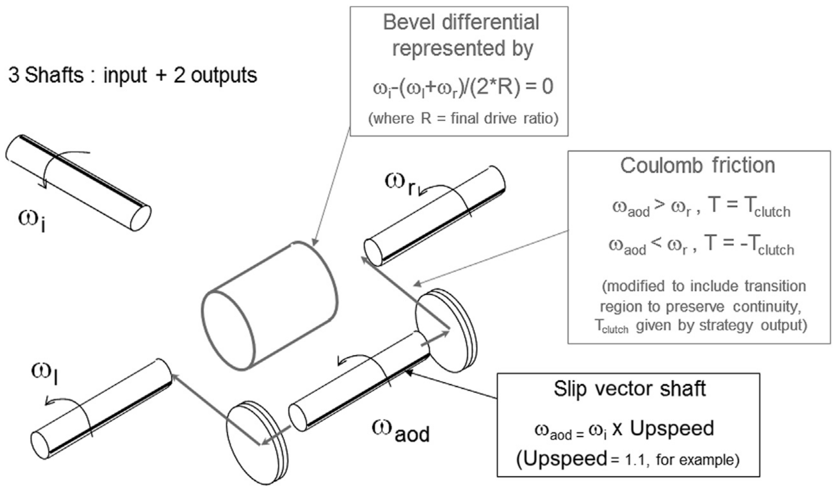

FIGURE 8.17

A modelling scheme for an asymmetric overspeeding differential — right shaft speedup.

additional spin degree of freedom represents the AOD coupling shaft. An additional mass is created, constrained to spin at some multiple of the input speed. In the example shown, a frictional torque using the Coulomb model is then engaged which is scaled by a solution variable T_{clutch}, which in turn is given as the output from some control strategy. When T_{clutch} is high, the upsped AOD shaft is effectively locked to the right-hand shaft. The conventional bevel differential enforces the speed relationship in the model, ensuring that the left-hand shaft is downsped by the same amount to preserve the average speed relationship. Additional details can be added for frictional bias torques and so on, to suit the level of available data. Despite being widely described as Torque Vectoring, this approach is really 'slip vectoring'; the torque distribution is not really enforced. Nevertheless on homogenous surfaces the effect is to deliver a yaw moment that may be used advantageously to manage the vehicle states.

For the rear axle, any use of an asymmetric device tends to reduce the capacity of the axle as a whole and thus it is easier to add yaw moment than to subtract it. The opposite is true for the front wheels. To the author's knowledge there has only been one front axle asymmetric overspeeding device, the Honda Active Torque Transfer System (ATTS) fitted to the Prelude SH in the late 1990s. 'SH' allegedly stood for 'Super Handling'.

A similar principle can be applied between the axles. The Honda SH-AWD system runs an epicyclic overspeed — extremely similar to the electric overdrive unit on 1970s Triumphs in principle — on the propeller shaft to the rear axle. If, say, a 10% overspeed is invoked then the centre differential, when open, simply processes to accommodate the now 10% difference in overall axle ratios. If the centre differential were to be fully locked, the rear axle would have a 10% overspeed compared to the front. This overspeed has an interesting effect on the lateral behaviour of the car; in order to understand it we must return to the tyre characteristics discussed in Chapter 5. To illustrate the effects, a sample tyre longitudinal force curve is presented in Figure 8.18. The curve is normalised to simplify the illustration. A notional 'lateral capacity fraction' can be calculated by presuming a uniform maximum frictional force is available and subtracting the longitudinal force such that the Pythagorean sum of lateral and longitudinal is unity. This sounds complex in words but is simple mathematically:

$$\frac{\widehat{F}_y}{\mu F_z} = \sqrt{1 - \frac{F_x}{\mu F_z}} \tag{8.9}$$

For a slip ratio beyond that at which maximum longitudinal force is generated, the lateral capacity is presumed zero. Normalised longitudinal force and this notional 'lateral capacity fraction' are plotted together in Figure 8.18 against longitudinal slip ratio.

Examining this graph closely, it can be seen for two similarly laden tyres that one with 10% slip ratio has approximately half the lateral capacity of one with 0% slip ratio — a slip ratio difference of 10% between them. If these are the outboard tyres on a vehicle it can be seen that the ability to exert a yaw moment on the vehicle is

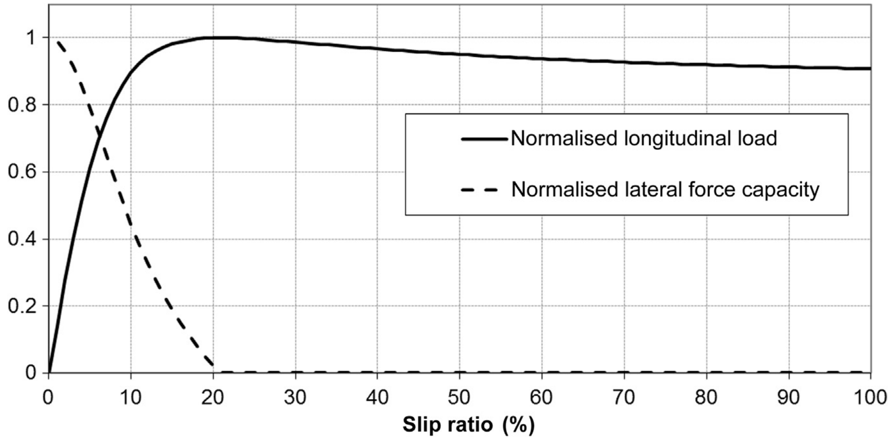

FIGURE 8.18

Normalised Longitudinal Force and Lateral Force Capacity against Slip Ratio.

substantial – this is the effect of slip vectoring and makes a rear-wheel-drive car yaw tail out under power without gross wheelspin. What is interesting is that if in a four-wheel drive driveline, the same slip ratio difference is kept but the slip ratios are now 10% and 20%, then the relative difference in lateral force capacity is of the same order; this means the yaw moment on the vehicle is broadly independent of the mean slip level between 5% and 15% – a substantial torque window. It means that absolute speed control is not vital for interaction with yaw moment, only the inter-axle speed difference. If the mean speed comes up further, to say 15% on one wheel and 25% on the other, lateral force capacity broadly converges and so the consequence of a gross excess driveline torque does not destabilise the vehicle but rather uniformly removes yaw moment as both tyres overspeed.

It is generally true that manipulating side forces via lateral force capacity gives a larger yaw moment than by exerting it directly left–right on the vehicle since generally wheelbase exceeds track. For this reason it is rarely necessary to run such a large speed difference as a left–right device; the first Prodrive prototype slip-vectoring vehicle ran an 11% speed difference but the next and subsequent vehicles dropped to a more typical 5% or so – the values used by other manufacturers.

It can be seen in the Honda-style configuration that the overspeed at the rear can have the effect of reducing lateral force capacity at the rear axle – the Prodrive ATD systems also used this effect – and thus can add yaw moment. When coupled with an epicyclic centre differential having a notional front-biased torque split, a simple proportional clutch across it delivers a front torque bias when open and a rearward torque bias when closed. Using this, a simple linear yaw rate demand model can generate a target yaw rate curve, with the control law for the centre differential responding to insufficient yaw rate by tightening and an excess yaw rate by slackening from its current position. In this manner, the driveline can be set up never to destabilise the car – in other words to decouple the steering from the throttle.

Overall driving impressions of such systems are, on the whole, 'remarkably unremarkable' with the drive experience uncorrupted by drive torque.

One demonstration of the Prodrive system involved wedging the steering wheel with the driver's leg and driving a continuous drift on a 60-m snow circle, managing the line of the car using throttle inputs but leaving the car to look after its yaw rate on the rough, rutted surface of the snow circle by controlling the centre differential. The task of managing the car was so undemanding that the author could fold his arms and eat an apple while doing it. A comparison data set with four-wheel drives but no closed-loop torque vectoring is shown in Figure 8.19; the duration is some 70 s, showing this was no cherry-picked small data segment. At a speed of around 60 mph, this represented of the order of a mile of continuous control. The uncontrolled car had a sophisticated all-wheel-drive system (including hydraulic centre differential and asymmetric rear axle) and a substantially faster steering ratio than the controlled car, but crucially no loop closure around vehicle yaw rate. It is noted with interest that since 2007 the Mitsubishi S-AWC system has had yaw rate feedback added compared to the AYC and S-AYC systems, in which it was absent.

Nothing about the learning with geared slip-vectoring systems precludes the same ideas being applied with individually controlled wheel motors. In particular, in-wheel motors such as those produced by Protean Electric Ltd at the time of writing offer high fidelity control and very accurate speed sensing, allowing, for example, a left and right motor to be 'software coupled' for relative speed and managed for mean torque to mimic an asymmetric overspeeding functionality that

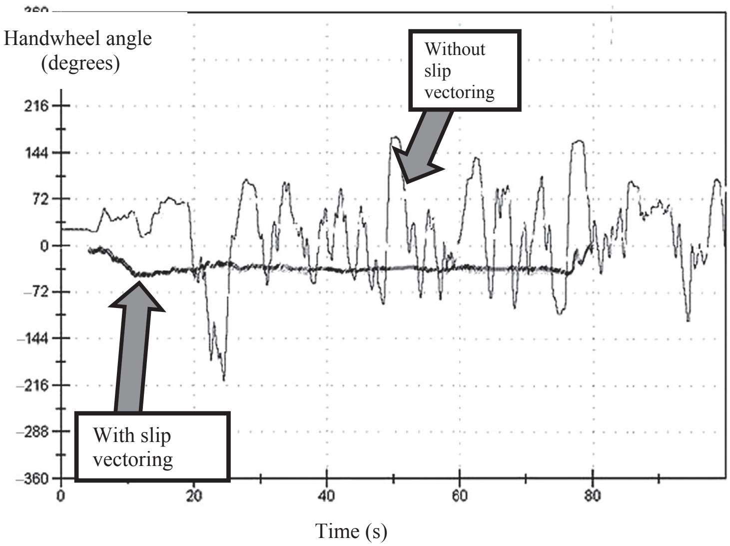

FIGURE 8.19

A comparison between the presence and absence of closed-loop slip vectoring driving on a 200-m snow circle in Arvidsjaur, Sweden — author (Harty) driving both vehicles.

needs no knowledge of friction coefficient. Key to unlocking these benefits is the realisation that what is in play is ‘slip vectoring’ and not ‘torque vectoring’.

8.3 Which active system?

The North American market is currently resisting complexity in its best-selling light truck segment (Pickups and SUVs), since the appeal from the vehicle manufacturer’s perspective is the low cost of production, giving margins that are allowing the vehicle manufacturers to remain in business during lean times. Nevertheless, competition from offshore competitors is making the US consumer expectations for vehicle performance and quality increase steadily. In Europe, an already sophisticated consumer is expecting steadily improving dynamic performance from their vehicle, while in Asia the predilection for ‘gadgets’ on vehicles has seen this market lead the way in terms of satellite navigation and so on. All three major world vehicle markets have their own reason for needing to improve vehicle dynamic performance over and above the level that can be achieved using conventional, passive technology. However, no-one wants to be first to market with a system that initially gives them a cost penalty and runs the risk of delivering benefits that the customer does not notice — and is therefore not prepared to pay for.

This appears to have been a lesson learned by the Japanese manufacturers — notably Toyota with their remarkable Soarer, who have largely withdrawn from the active system war and concentrated significantly on hybrid drivelines. In deciding which active system to employ, some note needs to be taken of the fact that different systems have different levels of authority under different driving regimes. In terms of systems that intervene directly — that is to say by generating ground plane forces, as distinct from modifying vertical forces in the manner of an active suspension — then different systems have differing strengths. Figure 8.20 presents the available ‘understeer and oversteer’ moments — which is to say moments out of the turn and into the turn, respectively — for a number of different systems. The characteristics are arrived at by considering wheel loads for a point mass vehicle with a finite footprint (wheelbase and track) and a centre of gravity above the ground. Each system is then imagined to modify the ground plane forces up to the friction limit or system capacity limit, whichever is the lower. For the electronically controlled limited slip differential, tractive power is presumed added to remove the presence of braking slip from the inside wheel. The different ‘authority envelopes’ can be clearly seen. Oversteer moment is available to correct an understeer error state and vice versa. Thus it can be seen that the steering is by far the most powerful actuator on the car for controlling oversteer, but brake-based stability systems are a close second. Steering is ineffective at suppressing understeer in any normally balanced car. Of particular interest is the somewhat limited envelope of the asymmetric overspeeding differential despite its complexity. In particular, in the oversteer case when understeer moment is needed, it is typically reduced in capacity by unloading of the inside rear wheel.

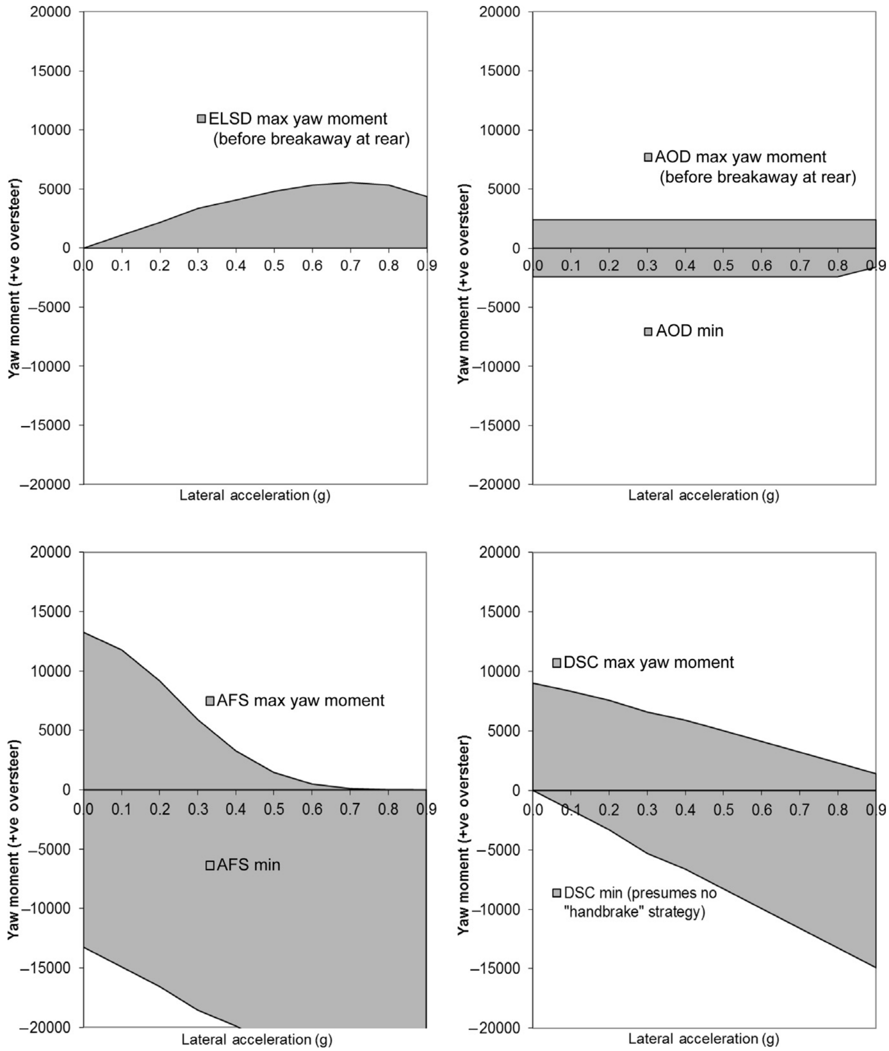

FIGURE 8.20

An illustrative authority envelope comparison between an electronic limited slip differential (ELSD) (top left) and asymmetric overspeeding differential (AOD) (top right), a full authority steering system (AFS) (bottom left) and a brake-based system (bottom right) with no 'handbrake' mode.

Any 'full spectrum' dynamic intervention system is likely to need more than one set of actuators. For this reason, predictive work is needed on a case-by-case basis to sort the useful from the gimmick in terms of vehicle dynamics controls. The author's expectations for the use of active systems on vehicles are given in Table 8.2. Three driving regimes are identified, which may be described as 'normal', 'spirited' and

Table 8.2 Which Active System – and Why?

Name	Lateral Acceleration	Steering Rate	Guiding Principles	Applicable Active System
Normal	0–0.3 g	0–400°/s	• To steer the car, steer the wheels. • Ride matters.	Active rear toe, active front steer, adaptive dampers.
Spirited	0.3–0.6 g	400–700°/s	• Intelligently combine drive and steer forces to deliver control without retardation. • Control body motion.	Active torque distributrion, *active rear toe, active front steer*, *adaptive dampers*
Emergency	0.6 g+	700+°/s	• Use brakes to reduce kinetic energy of the vehicle. • Minimise wheel load variation for maximum grip and control.	Brake-based system, *adaptive dampers*, active anti-roll bars.

Italics denote duplication – i.e. reuse.

'accident avoidance'. These may be loosely classified as being 0 to 0.3 g lateral acceleration, 0.3 to 0.6 g lateral acceleration and over 0.6 g lateral acceleration respectively. They are also characterised by driver inputs, particularly in terms of handwheel rates — 0 to 400°/s, 400 to 700°/s and 700 to 1200°/s, respectively. These classifications are entirely arbitrary and based on empirical observations. The upper bound is observed to be broadly independent of the nature of the driver, which is to say that motorsport drivers are not observed to be able to put in higher handwheel rates than the rest of us. It's fair to say they have a better idea what handwheel rate to choose than many of us, though.

The results in Table 8.2 are not in any sense definitive but the authors considered views in the light of the available evidence. It is hoped that the readers of this text will be part of the group of engineers discovering just how good these judgements are.