11 Active Suspensions

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CONTENTS

I. INTRODUCTION

It is clear from the preceding chapters that the subject of railway vehicle dynamics has developed principally as a mechanical engineering discipline, but an important technological change is starting to occur through the use of active suspension concepts. The use of advanced control has been common for many decades in the power electronic control of traction systems, and it is now firmly established as the standard technology which has yielded substantial benefits, but its application to suspensions is much more recent. Although the term "active suspension" is commonly taken to relate to providing improved ride quality in fact, it is a generic term which defines the use of actuators, sensors, and electronic controllers to enhance and/or replace the springs and dampers that are the key constituents of a conventional, purely mechanical, "passive" suspension; as such it can be applied to any aspect of the vehicle's dynamic system.

II. BASICS OF ACTIVE SUSPENSIONS

Vehicle dynamicists have been aware of active suspensions for some time, with major reviews having been undertaken in 1975, 1983, and 1997,[1–3] but so far they have only found substantial application in tilting trains — which can now be thought of an established suspension technology. However, there are two other major categories: active secondary suspensions for improved ride quality, and active primary suspensions for improved running stability and curving performance. The sections which follow in this chapter deal with these three categories in turn: tilting, active secondary, and active primary suspension, but first there are a number of general principles and considerations which need to be explained.

A. Concepts

The general scheme of an active suspension is shown in diagrammatic form in Figure 11.1. The input/output relationship provided by the suspension, which in the passive case is determined solely by the values of masses, springs, dampers, and the geometrical arrangement, is now dependent upon the configuration of sensors and actuators, and upon the control strategy in the electronics (almost invariably now involving some form of software processing). For all the three categories it will be seen that the introduction of active control enables things to be achieved that are either not possible or extremely difficult with a passive suspension.

B. Active and Semi-Active

The greatest benefits can be achieved by using fully-controllable actuators with their own power supply, such that the desired control action (usually a force) can be achieved irrespective of the movement of the actuator. Energy can flow from or to the power supply as required to implement the particular control law. This is known as a "full-active" suspension, but it is also possible to use a "semi-active" approach in which the characteristic of an otherwise passive suspension component can be rapidly varied under electronic control — see Figure 11.2. Semi-active

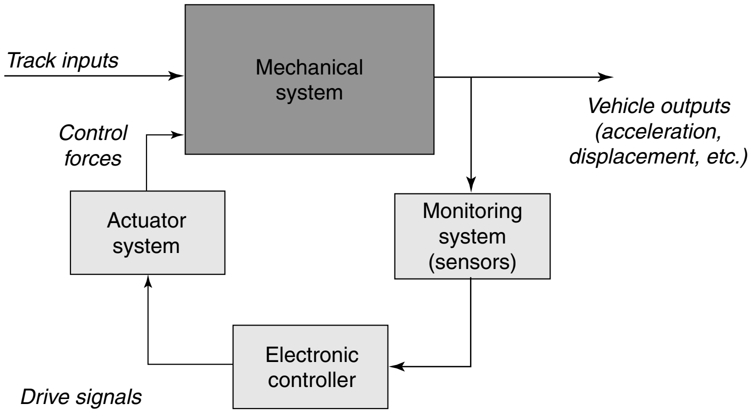

FIGURE 11.1 Generalised active suspension scheme.

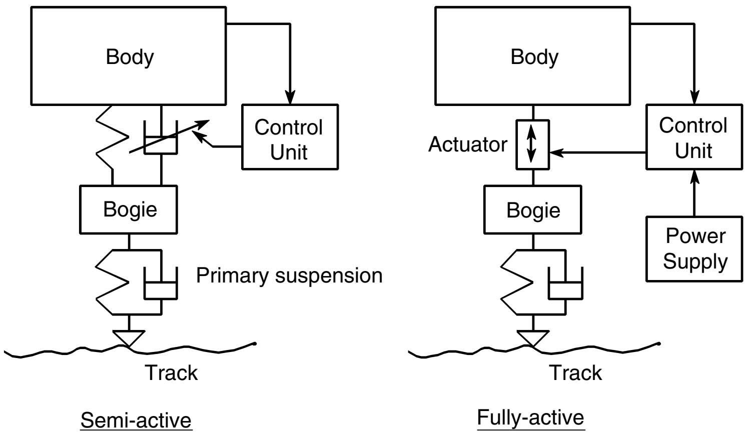

FIGURE 11.2 Semi- and full-active control.

suspensions usually use controllable dampers of some kind, although the concept is not restricted to dampers.[a]

The benefit of the semi-active approach compared with full-active is one of simplicity, because a separate power supply for the actuator is not needed. The disadvantage of a semi-active damper is that the force remains dependent upon the speed of damper movement, which means that large forces cannot be produced when its speed is low, and, in particular, it cannot develop a positive force when the speed reverses because it is only possible to dissipate energy, not inject it. Figure 11.3 clarifies the limitation by showing areas on the force–velocity diagram that are available for a semi-active damper based upon its minimum and maximum levels, whereas an actuator in a full-active system can cover all four quadrants. This limitation upon controllability restricts the performance of a semi-active suspension to a significant degree.[4]

Closely related is an option known variously as semi-passive, adjustable passive or adaptive passive, in which the characteristics are varied on the basis of a variable which is not influenced by the dynamic system being controlled (e.g., as a function of vehicle speed).

[a] An interesting option would be the use of an electronically-controllable spring to provide a semi-active suspension, but as far as the authors are aware, no such device has been invented.

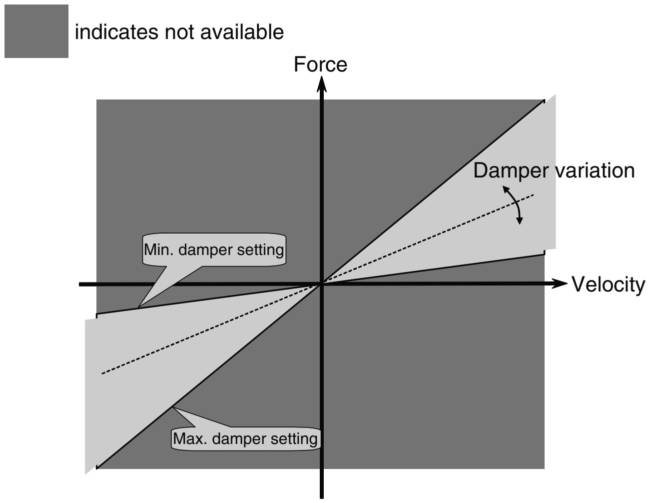

FIGURE 11.3 Force–velocity diagram for semi-active damper.

C. Design Considerations

For designing active suspension systems such as these, an important difference arises compared with passive suspensions. A conventional suspension is designed with as accurate a model as possible so that the computer simulation can predict the on-track performance effectively. The designer then adjusts the values of the suspension components based upon well-understood expectations for the particular vehicle configuration until the required performance is achieved. However, for an active suspension, it is important to distinguish between the design model and the simulation model: the former is a simplified model used for synthesis of the control strategy and algorithm, whereas the latter is a full-complexity model to test the system performance, i.e., as used for conventional suspensions. The importance of having an appropriately simplified design model is less profound when "classical" control design techniques are being used, although even here key insights arise with simplified models; the real issue arises when modern model-based design approaches are being used, either for the controller itself or for estimators to access difficult or impossible to measure variables, in which case the controller and/or estimator assumes a dynamic complexity equal to or greater than that of the design model. Since a good simulation model of a railway vehicle will usually have more than a hundred states, a controller based upon this model would at best be overly complex to implement, at worst impossible because some of the states may be uncontrollable or unobservable.

There are formal methods for reducing the model complexity, but often engineering experience will provide a suitable abstraction. For example, there is a relatively weak coupling between the vertical and lateral motions of rail vehicles and, depending on the objectives, only selected degrees of freedom need to be included in the design model. Common simplifications are based around a vehicle model that is partitioned into side-view, plan-view, and end-view models: the side-view model is concerned with the bounce and pitch degrees of freedom, and can be used for active vertical suspensions; the plan-view model deals with the lateral and yaw motions, and can be used for active lateral suspensions and active steering/stability control; the end-view model covers the bounce, lateral, and roll motions, and can be used for the design of tilting controllers.

It is, of course, essential that such modelling software can support the integration of the controller into the mechanical system. This can be achieved within a single package, but, there is

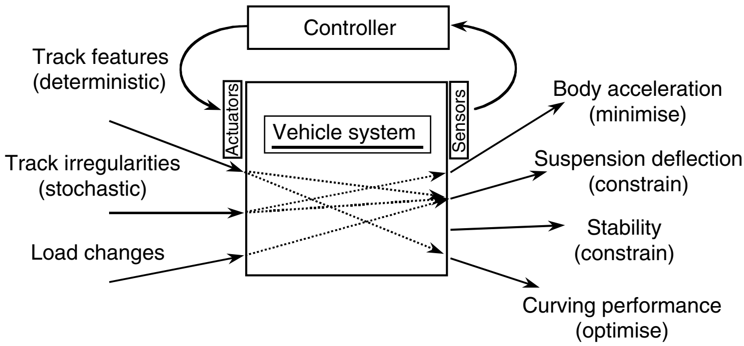

FIGURE 11.4 Design process.

a strong argument for distinct but well-integrated software, i.e., one of the many MBS dynamics packages in combination with a control design package such as Matlab/Simulink®. Ideally, there should be a number of interface possibilities: controllers designed using the simplified design model need to be exported into the MBS package for simulation purposes; equally it is often valuable to be able to export a complex but linearised model from the MBS package for further controller evaluation using the targeted analytical tools provided for controller design; and finally, running the two packages simultaneously in a co-simulation mode is also important because this avoids the need for conversion and export, although the data transfer process must be robust.

A final point is illustrated by Figure 11.4, which emphasises the multi-objective nature of the design process. There are a variety of input types and output variables that must be considered, and each output will be affected by different combinations of inputs. The design will require an optimisation involving constraints. For example; an active secondary suspension design must minimise the frequency-weighted accelerations on the vehicle body without exceeding the maximum suspension deflection; an active primary suspension must optimise the curving performance whilst maintaining adequate levels of running stability on straight track; etc.

III. TILTING TRAINS

The earliest proposals for tilting trains go back into the first half of the 20th century, but it was not until the 1960s and 1970s that experimental developments were aimed towards producing operational trains for prestigious high-speed routes. These emerged as the Talgo Pendular in Spain (1980), the APT in the UK and the LRC in Canada (1982), the first ETR 450 Pendolino trains in Italy (1988), and the X2000 in Sweden (1990). A similar pattern occurred in Japan, although the developments there were aimed at the regional/narrow-gauge railways rather than the high-speed Shinkansen. The 1990s saw tilting mature into a standard railway technology, with applications extending throughout most of Europe and Japan, and all the major rail vehicle manufacturers now offer and supply tilting trains for regional and high-speed applications.

A. Concept and Equations

Tilting trains take advantage of the fact that the speed through curves is principally limited by passenger comfort, and not by either the lateral forces on the track or the risk of overturning, although these are constraints that cannot be ignored. Tilting the vehicle bodies on curves reduces the acceleration experienced by the passenger, which permits higher speeds and provides a variety of operational benefits. The principles and basic equations related to tilting are relatively straightforward and are explained here in a manner that focuses upon the operational advantages.

There are two primary decisions that need to be made. The first is what maximum tilt angle is to be provided (θ_{tilt}), a decision based upon mechanical design of the vehicle, especially taking gauging issues into account. The second decision is what cant deficiency the passengers should experience on a steady curve (θ_{active}), which clearly is of primary importance to comfort. Given these two decisions, and the cant deficiency that applies for the passive (nontilting) case (θ_{passive}), it is possible to derive an equation for the increase in speed offered by tilt. Note that, although the curve radius and the acceleration due to gravity appear in the basic acceleration equations, they disappear when the equation is dealing with the fractional or percentage speed increase:

$$\text{speed increase} = \frac{V_{\text{active}} - V_{\text{passive}}}{V_{\text{passive}}} = \left\{\sqrt{\frac{\sin(\theta_{\text{cant}} + \theta_{\text{tilt}} + \theta_{\text{active}})}{\sin(\theta_{\text{cant}} + \theta_{\text{passive}})}} - 1\right\} \times 100\% \qquad (11.1)$$

Although in principle the cant deficiency could be fully compensated by the tilting action, i.e., to make $\theta_{\text{active}} = 0$, in practice this is not sensible either from the operational or the ride comfort viewpoint. It is possible to recognise this by introducing a "cant deficiency compensation factor" (K_{CD}), an important design parameter in the tilt controller, the choice of which will be discussed later.

$$K_{\text{CD}} = 1 - \frac{\theta_{\text{active}}}{(\theta_{\text{active}} + \theta_{\text{tilt}})}, \text{ i.e., } \frac{\theta_{\text{tilt}}}{\theta_{\text{active}} + \theta_{\text{tilt}}} \qquad (11.2)$$

Consider some examples: track cant is usually 6°, and typically 6° of cant deficiency is applied for a nontilting train. Applying 9° of tilt and a cant deficiency of 6° for the tilting train, the calculation indicates a speed-up of 32% with a compensation factor of 60%. In this particular case, the passengers nominally experience the same comfort level on curves (although the passive vehicle will usually roll out by a small angle, typically less than 1°, so in practice tilting will give a small reduction in the curving acceleration). Another example might be where the tilting cant deficiency is reduced to 4.5°, perhaps to offer an improved ride comfort; using a slightly smaller tilt angle of 8°, the speed-up falls to 24% with a compensation factor of 64%.

Speeds on curves may, therefore, be theoretically increased by around 30% or more with tilting trains. However, the performance on curve transitions as well as the steady curves is important from a comfort point of view, and the comfort level can be predicted using a method described by a European standard.[5] It is based on an empirically-based method in which the percentage of passengers (P_{CT}) that are likely to feel uncomfortable during the curve is determined from the lateral acceleration ($\ddot{y}$), the lateral jerk ($\dddot{y}$), and the body roll velocity ($\dot{\vartheta}$) experienced during the transition. Details of the method are given in the quoted reference, including the way in which the three measurements should be made. Equation 11.3a gives the appropriate empirically-derived equations, and the constants which must be used to calculate the P_{CT} factor, a separate calculation for seated and standing passengers derived from either simulated or measured performances of the vehicle at the entry to a curve Table 11.1 lists.

$$P_{\text{CT}} = |(A\ddot{y} + B\dddot{y} - C)|_{\leq 0} + D\dot{\vartheta}^{E} \qquad (11.3a)$$

There is also the issue of motion sickness. In contrast to the curve transition comfort level, which may be considered on a curve-by-curve basis, motion sickness is a cumulative effect, which comes as a consequence of a number of human factors, the exact nature of which is not fully understood. Again, the effect is aggravated on highly curvaceous routes with rapid transitions.[6] The degree to which the curving acceleration is compensated for by the tilting action is an important factor, but once this has been optimised, the only other mitigation measure is operating at lower speed.

TABLE 11.1
Constants for P_{CT} Equation

Condition	A	B	C	D	E
Standing	28.54	20.69	11.1	0.185	2.283
Seated	8.97	9.68	5.9	0.120	1.626

B. Mechanical Configurations

Broadly speaking, there are four mechanical arrangements which are possible to provide the tilting action.

The first is passive or pendular tilt, in which the secondary suspension is raised to around roof level in the vehicle: the vehicle centre of gravity is then substantially below the suspension and the body naturally swings outwards, reducing the lateral curving acceleration experienced by the passengers. This is a technique pioneered in the Talgo trains — the air springs are raised by means of vertical pillars at the vehicle ends, an arrangement made much easier by the articulated configuration of the trains.

A second approach is to achieve tilt directly by applying active control to the secondary roll suspension. One method which has been tried in both Europe and Japan is to apply differential control to the air springs, but this may cause a dramatic increase in air consumption and generally has not found favour, although one Japanese development has achieved it by transferring air between the air springs using a hydraulically-actuated pneumatic cylinder.[7] The alternative method of direct control of the roll suspension is by means of an active anti-roll bar (stabiliser), and this is applied in Bombardier's regional Talent trains. This uses the traditional arrangement consisting of a transversely-mounted torsion tube on the bogie with vertical links to the vehicle body, except that one of the links is replaced by a hydraulic actuator, and thereby applies tilt via the torsion tube.

The previous two arrangements are very much minority solutions, because most implementations use a tilting bolster to provide the tilt action. An important distinction is where this bolster is fitted compared with the secondary suspension, which leads to the third and fourth of the arrangements. With the tilting bolster above the secondary suspension, the increased curving forces need to be reacted by the secondary lateral suspension; since a stiffer lateral suspension is not consistent with the higher operating speed of a tilting train, in practice, either an increased lateral suspension movement or some form of active centring method is needed to avoid reaching the limits of travel.

The final arrangement has the tilting bolster below the secondary suspension, thereby avoiding the increased curving forces on the lateral suspension, and this is probably the most common of all schemes, the necessary rotation being achieved using either a pair of inclined swing links, or a circular roller beam. Typical schemes with inclined swing links and with a roller beam are shown in Figure 11.5.

Actuators to provide tilt action have seen significant development since the early days of tilt. Some early systems were based upon controlling the air springs (i.e., intrinsically pneumatic actuation), but it was more normal to use hydraulic actuators because these tend to be the natural choice for mechanical engineers. However, experiments with electro-mechanical actuators in the UK in the 1970s, in Switzerland in the 1980s, and in Germany in the 1990s, paved the way for a progressive change away from the hydraulic solution. Electric motors controlled by solid-state power amplifiers drive screws fitted with high-efficiency ball or roller nuts to convert rotary to linear motion. They are less compact than hydraulic actuators at the point of application, but, overall, they provide significant system benefits and they are now employed in the majority of new

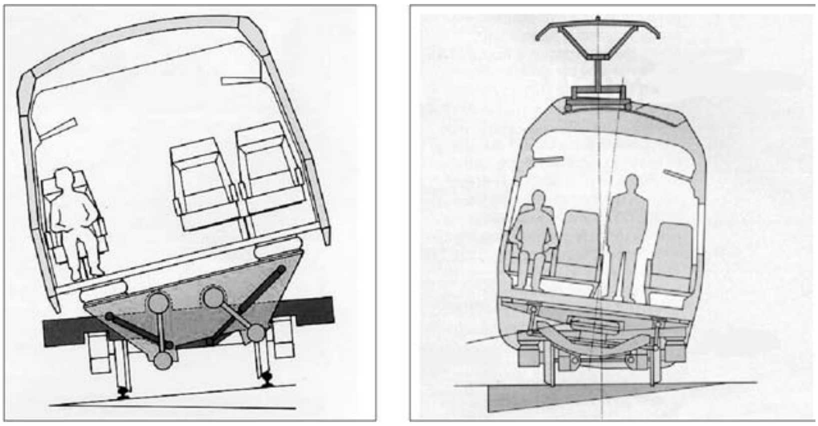

FIGURE 11.5 Tilt below secondary schemes.

European tilting trains. Interestingly, Japanese tilting technology has tended to use pneumatic actuators.

C. Control: Strategies and Assessment

This section explains some of the essential control approaches that are possible to achieve effective tilting action, and then discusses how the performance of particular controllers can be assessed.

1. Control Approaches

The most intuitive control approach is to put an accelerometer on the vehicle body to measure the lateral acceleration that the tilt action is required to reduce, yielding the "nulling" controller shown in Figure 11.6(a). The accelerometer signal is used to drive the actuator in a direction that will bring it towards zero, i.e., a classical application of negative feedback. Implementation of the required value of K_{CD} can be achieved with a modification of the basic nulling controller to give a partial tilt action by including a measure of the tilt angle in the controller, as shown by the dotted arrow on the figure. However, there is a difficulty with this scheme due to interaction with the lateral

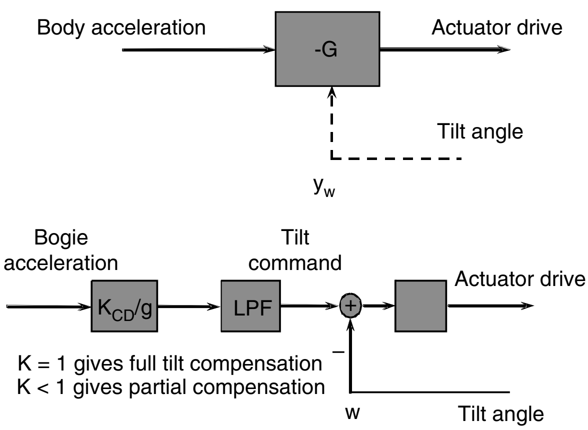

FIGURE 11.6 Tilt control methods.

suspension: the roll and lateral modes of the vehicle body are strongly coupled in a dynamic sense, and it can be shown that if the loop bandwidth is low enough not to interfere with the lateral suspension, it is then too slow-acting on the curve transition.

Figure 11.6(b) shows the next solution: the dynamic interaction problem can be avoided by putting the accelerometer on a nontilting part, in other words the bogie. This will then tell how much tilt is needed to reduce the lateral acceleration on the vehicle body, and can be multiplied by the factor K_{CD} which determines what proportion of the lateral acceleration is to be compensated; $K_{CD} = 1$ gives 100% compensation, not a good idea for motion sickness reasons, and typically 60 or 70% compensation is used (as mentioned above). This "tilt angle command signal" then provides the input to a feedback loop which uses a measurement of the tilt angle.

Unfortunately there is still a problem, because the accelerometer on the bogie is not only measuring the curving acceleration, but also the pure lateral accelerations due to track irregularities. With the accelerometer on the vehicle body, these accelerations are reduced by the secondary suspension, but they are much larger when the accelerometer is on the bogie. Consequently, it is necessary to add a low-pass filter (LPF) to reduce the acceleration signals caused by the track irregularities, otherwise there is too much tilt action on straight track resulting in a worse ride quality. However, to apply sufficient filtering, there is also too much delay introduced at the start of the curve, so the full lateral curving acceleration is felt for a short time, even though it reduces to an acceptable level once properly on the curve.

Figure 11.7 shows the next step: the signal from the vehicle in front is used to provide precedence, carefully designed so that the delay introduced by the filter compensates for the precedence time corresponding to a vehicle length. In effect, this scheme is what most European tilting trains now use; sometimes roll and/or yaw gyros are used to improve the response, and normally a single command signal is generated from the first vehicle and transmitted digitally with appropriate time delays down the train.

The signal from the bogie-mounted accelerometer is essentially being used to generate an estimate of the true cant deficiency of the track's design alignment, the difficulty being to exclude the effects of the track irregularities. An obvious development is to feed the vehicle controllers with signals from a database which defines the track, instead of from the accelerometer. Both the position of the vehicle along the track and the curve data contained in the database need to be known accurately for this approach to work effectively, but it is likely that such systems will become the norm in the future.

Japanese tilting trains often use a balise on the track ahead of the curve to initiate the tilting action, a technique which helps to mitigate the relatively slow response of the pneumatic tilt actuators.

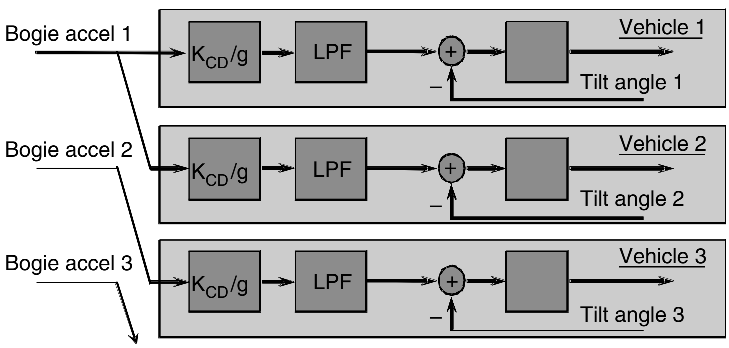

FIGURE 11.7 Precedence tilt control scheme.

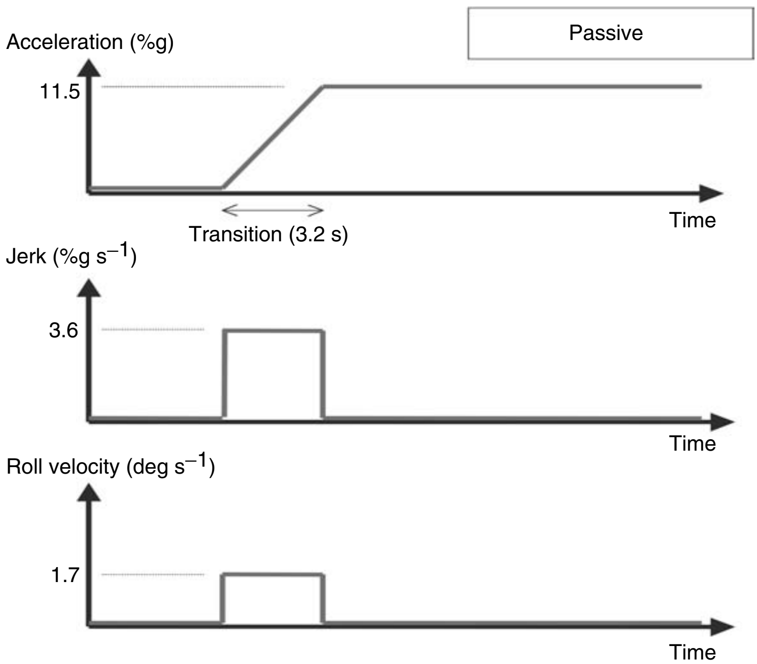

FIGURE 11.8 "Ideal" passive transition responses.

2. Assessment of Controller Performance

It is clear that what happens in the steady curve is important, however, the dynamic response during the transition must also be considered. In an ideal tilt control strategy, the tilt angle of the body should rise progressively, perfectly aligned both with the onset of curving acceleration and the rising cant angle, and the difficulties in achieving this kind of response have been explained above. Since the principal benefit of tilt is to be able to operate at higher speeds without degradation in

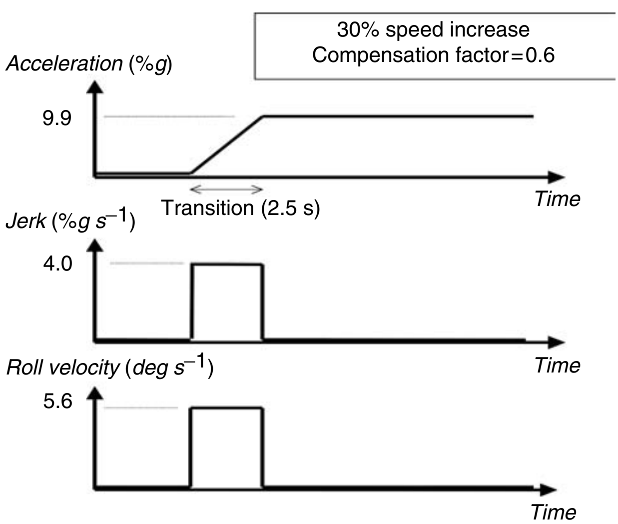

FIGURE 11.9 "Ideal" tilting responses.

passenger comfort, from a design point of view there are two issues: how well does the tilting vehicle perform on straight track, and how well does it perform on curve transitions?

The accelerometer-based control strategies means these two issues must, in practice, be traded off against each other — if the tilt action is fast to give good transition performance, in general, the straight track ride quality may be degraded. Qualitatively, a good tilt controller responds principally to the deterministic track inputs, and as much as possible ignores the random track irregularities. In order to assess different tilt control strategies in an objective manner, it is necessary to define appropriate criteria and conditions.

The straight track performance can be dealt with using a criterion of degrading the lateral ride quality by no more than a specified margin compared with the nontilting response, a typical value being 7.5%. Note that for assessing the tilt controller performance, this comparison must be made at the higher speed. Of course, a comparison of ride quality with a lower speed vehicle is also needed,

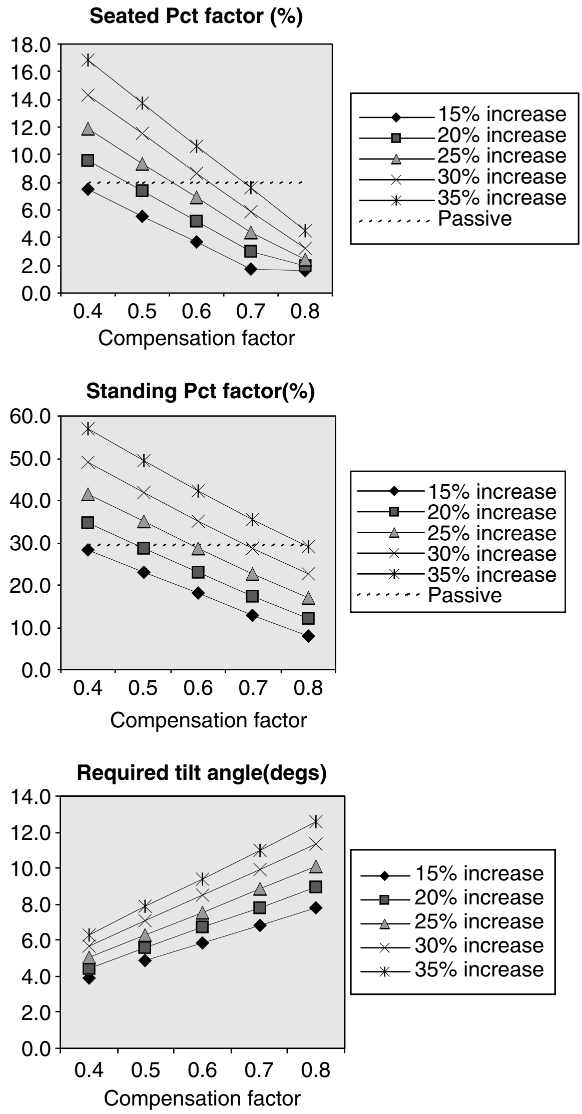

FIGURE 11.10 Comfort factors and tilt angle results.

but achieving a satisfactory ride quality at elevated speeds will require either an improved suspension or a better quality track, i.e., not a function of the tilt controller.

The curve transition response has to be separated into two aspects. Firstly, the fundamental tilting response, measured by the P_{CT} factors as described previously, must be as good as a passive vehicle at lower (nontilting) speed, otherwise the passenger comfort will inevitably be diminished, no matter how effective the tilt control is. It is possible, therefore, to introduce the idea of "ideal tilting" where the tilt action follows the specified tilt compensation perfectly, defined on the basis of the fundamental tilt system parameters — the operating speed (increase), maximum tilt angle, and the cant deficiency compensation factor. This combination of parameters can be optimised using the P_{CT} factor approach for deterministic inputs in order to choose a basic operating condition, and this will give "ideal" P_{CT} values (one for standing, one for sitting).

Consider, for example, the ideal transition responses for passive and tilting trains shown in Figure 11.8 and Figure 11.9, where the transition length gives a time of 3.2 sec for the passive vehicle and both cant and cant deficiency are 6°. (The passive response also includes the effect of a "passive roll-out" of 1°, but this is obviously vehicle-dependent.) Figure 11.9 shows the corresponding acceleration, jerk, and roll velocity graphs for a particular tilting condition, i.e., 30% higher speed with a compensation factor of 0.6, but of course similar diagrams can be developed for other conditions.

The three graphs in Figure 11.10 show the results of P_{CT} calculations undertaken with speed-up factors of between 15 and 35% and compensation factors from 40 to 80%, where the dotted horizontal lines show the values for the slower nontilting train, plus the corresponding tilt angle requirement. In this case, with a relatively slow transition, increasing the compensation factor improves the comfort level, although this is not necessarily the case with faster transitions; however, it can be seen that a larger tilt angle is required.

The other consideration is that it is necessary to quantify the additional dynamic effects which are caused by the suspension/controller dynamics as the transitions to and from the curves are encountered, which can be quantified as the deviations from the "ideal" response mentioned in the previous paragraph. These deviations relate to both the lateral acceleration and roll velocity, although the former is likely to be the main consideration. The performance in this respect will depend upon detailed characteristics of the controller, such as the filter in the command-driven scheme and the tuning parameters in the tilt angle feedback loop. It is clear that the deviations need to be minimised, but at present there is no information regarding their acceptable size, although the values derived for a normal passive suspension can be used as a guide.

D. Summary of Tilting

It should be emphasised that, although tilting seems in many ways to be a rather simple concept, it requires considerable care in practice and has taken a number of years to introduce reliable operational performance, and tilting controllers still need adjustment for specific route characteristics. It is likely that the state-of-the-art will continue to be developed in the years to come.

IV. Active Secondary Suspensions

A. Concepts and Requirements

For the secondary suspensions, active controls improve the vehicle dynamic response and provide a better isolation of the vehicle body to the track irregularities than the use of only passive springs and dampers. Active control can be applied to any or all of the suspension degrees-of-freedom, but, when applied in the lateral direction, will implicitly include the yaw mode, and in the vertical direction will include the pitching mode. (Controlling in the roll direction is of course equivalent to tilting, which is essentially a particular form of active secondary suspension, but of sufficient

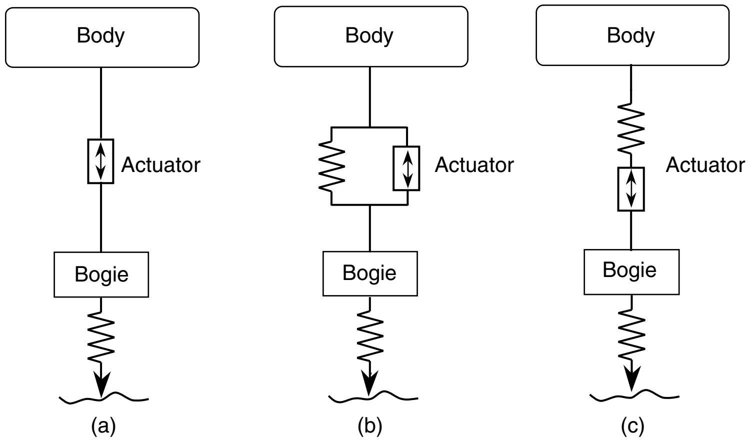

FIGURE 11.11 Active secondary suspension actuator configurations.

importance to have its own section.) The improved performance can be used to deliver a better ride quality, but this is not directly cost-beneficial and so normally will be used to enable higher train speed whilst maintaining the same level of passenger comfort. The other possibility is to provide the same ride quality on less well aligned track, in which case the cost-benefit analysis needs to take account of the reduced track maintenance cost.

B. Configurations

Active secondary suspensions can be used in the lateral and/or vertical directions and a number of actuator configurations are possible as illustrated in Figure 11.11.

Actuators can be used to replace the passive suspensions as shown in Figure 11.11(a) and the suspension behaviour will be completely controlled via active means. In practice, however, it is more beneficial that actuators are used in conjunction with passive components. When connected in parallel, as illustrated in Figure 11.11(b), the size of an actuator can be significantly reduced as the passive component will be largely responsible for providing a constant force to support the body mass of a vehicle in the vertical direction or quasi-static curving forces in the lateral direction. On the other hand, fitting a spring in series with the actuator, as shown in Figure 11.11(c), helps with the high frequency problem caused by the lack of response in the actuator movement and control output at high frequencies (see Section IV.C.6, Actuator response), and in practice a combination of a parallel spring for load-carrying and a series spring to help with the high frequency response is the most appropriate arrangement. The stiffness of the series spring depends upon the actuator technology: a relatively high value can be used for technologies such as hydraulics that have good high frequency performance, and a softer value for other technologies which means that achieving a high bandwidth is more problematic.

The other option is to use actuators mounted between adjacent vehicles, although the improvement of ride quality is less significant and, in general, the design problem is more difficult because the complete train becomes strongly coupled in a dynamic sense via the actuators.

C. Control Strategies

1. Sky-Hook Damping

There are different control approaches possible for active suspensions. A high bandwidth system, which deals with the random track inputs caused by irregularities, can be used to improve suspension performance largely through the provision of damping to an absolute datum.

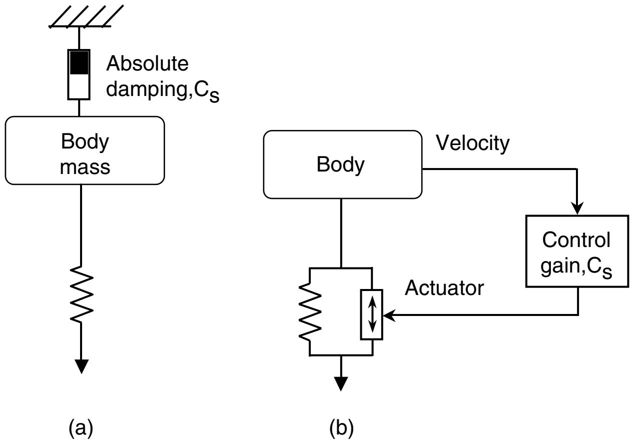

FIGURE 11.12 Sky-hook damping.

The principle of absolute damping is depicted in Figure 11.12(a), where a damper is connected from the mass to the sky, hence the term "sky-hook" damping. For practical implementations, the principle of the sky-hook damping can be realised by the arrangement shown in Figure 11.12(b). The feedback measurement is provided from a sensor mounted above the suspension on the body and the control demand is fed to the actuator which is placed between the vehicle body and the bogie.

A comparison between the passive and the sky-hook damping of a simple (one-mass) system illustrates the potential advantages of the active concept very well. For a passive damper, a higher level of modal damping can only be achieved at the expense of increased suspension transmissibility at high frequencies, as shown in Figure 11.13. For the sky-hook damper, however, the high frequency responses are independent of the damping ratio, and the transmissibility is significantly lower than that of the passive damping at all frequencies concerned. This is also the consequence of applying optimal control, as described in Ref. 8.

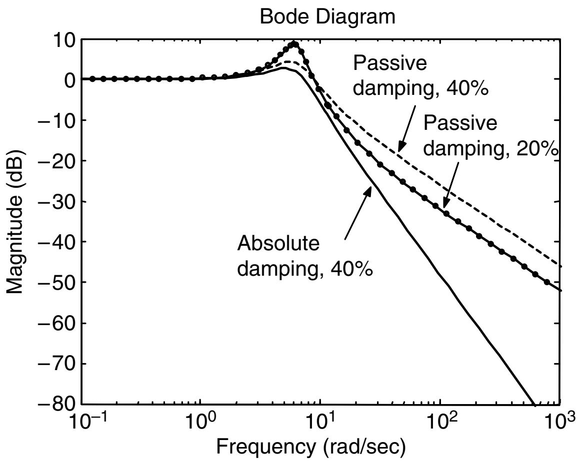

FIGURE 11.13 Comparison of passive and absolute damping.

The equation which implements the skyhook control law is simple, i.e.,

$$F_a = -C_s \frac{dz}{dt} \tag{11.3b}$$

where C_s is skyhook damping coefficient and F_a is the actuator force.

This yields the transfer function for a simple single-mass suspension as follows:

$$\frac{z}{z_t} = \frac{K}{K + sC_s + s^2M} \tag{11.4}$$

where K [N/m] and M [kg] are the spring constant and mass.

The equivalent transfer function for the passive suspension with a conventional damper having a coefficient C (Ns/m) is

$$\frac{z}{z_t} = \frac{K + sC}{K + sC + s^2M} \tag{11.5}$$

from which it can be seen that the high frequency response is $\propto 1/f$ for the passive suspension, compared with $\propto 1/f^2$ for the active skyhook suspension, the overall effect of which was seen in Figure 11.13.

Skyhook damping gives a profound improvement to the ride quality for straight track operation, however, it creates large deflections at deterministic features such as curves and gradients. Although this can be accommodated in the control design, e.g., by filtering out the low frequency components from the measurements which is largely caused by track deterministic features,[9] it is recognised that reducing the deterministic deflections to an acceptable level will compromise the performance achievable with "pure" skyhook damping. In fact, the absolute velocity signal that is required for skyhook damping will usually be produced by integrating the signal from an accelerometer, and so, in practice, it will also be necessary to filter out the low frequency components in order to avoid problems with thermal drift in the accelerometer — a typical scheme is shown in Figure 11.14. In practice, the integrator and high-pass filter will normally be combined to provide a "self-zeroing" integration effect.

Whilst the use of a high-pass filter can eliminate the quasi-static suspension deflections due to the large quasi-state force of the skyhook damping on gradients or curves, it is less effective in reducing the transient suspension travel on track transitions, and in the selection of the filter cut-off frequency there is a difficult trade-off between the ride quality improvement of the vehicle body and the maximum movement of the suspension.

There are a number of possible solutions proposed to overcome the problem. The complementary filtering approach, as shown in Figure 11.15, uses a relative damping force at the low frequency range in addition to the sky-hook damping at high frequencies, which results in a much improved trade-off. There are also Kalman filter based strategies where the effect of the track deterministic input can be minimised or the track features are directly estimated.[10] A typical trade-off comparison between different control approaches is given in Figure 11.16, in this case for the vertical suspension of a vehicle running onto a gradient.[9]

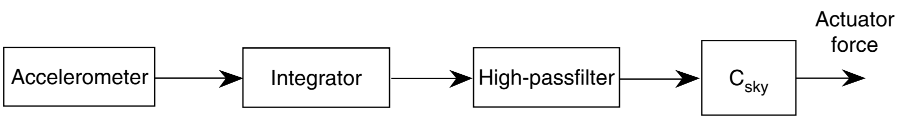

FIGURE 11.14 Practical implementation of skyhook damping.

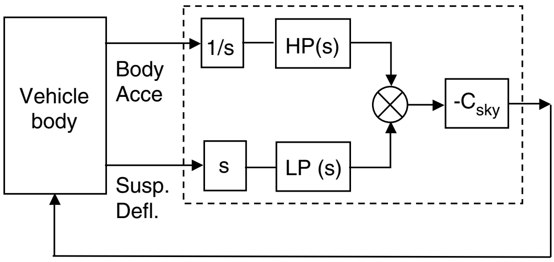

FIGURE 11.15 Complementary filters.

2. Softening of Suspension Stiffness

Another strategy is to create a softer suspension by controlling the actuator to cancel part of the suspension force produced by the passive stiffness. The control equation is of a simple form as shown in Equation 11.6, but note that positive feedback is used to reduce the overall stiffness to a value of $(K - K_s)$. The corresponding transfer function is not given because it is a trivial change to what was given for the passive suspension.

$$F_a = +K_s(z - z_t) \tag{11.6}$$

3. Low-Bandwidth Controls

Active secondary suspensions can also be used to provide a low bandwidth control, which is similar to tilting controls in that the action is intended to respond principally to the low frequency deterministic track inputs. In low bandwidth systems, there will be passive elements which dictate the fundamental dynamic response, and the function of the active element is associated with some low frequency activity. A particular use of the concept is for maintaining the average position of the suspension in the centre of its working space, thereby minimising contact with the mechanical

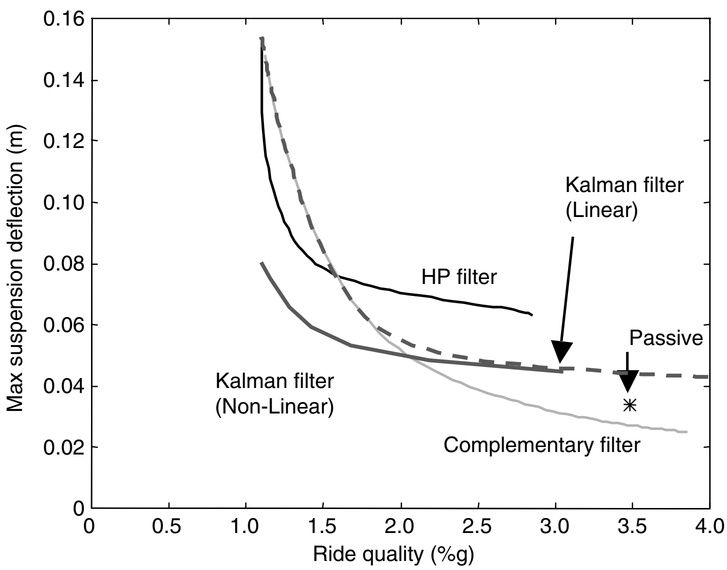

FIGURE 11.16 Trade-off between ride quality and suspension deflection.

limits of travel, and enabling the possibility of a softer spring to be used.[11,12] This is a powerful technique for the lateral suspensions because curving forces are large, and without centring action there may sometimes be significant reductions in ride quality whilst curving.

The idea of active levelling (or centring for a lateral suspension) can be achieved using the equation.

$$F_{\mathrm{a}} = -K_{\mathrm{L}} \int (z - z_{\mathrm{t}})\mathrm{d}t$$

The suspension transfer function becomes

$$\frac{z}{z_{\mathrm{t}}} = \frac{K_{\mathrm{L}} + Ks + Cs^2}{K_{\mathrm{L}} + Ks + Cs^2 + Ms^3} \tag{11.7}$$

The integral action changes it from second to third order, the effect of which is less obvious, but it can readily be shown that the suspension deflection $(z - z_{\mathrm{t}})$ is zero in response to an acceleration input from the track, and it is this characteristic that corresponds to the self-levelling effect.

4. Modal Control Approach

For a conventional railway vehicle with two secondary suspensions between the body frame and the two bogies, it is possible to use local control for each suspension, i.e., the measurement from the sensor(s) mounted above either of the bogies is fed to the controller which controls the actuator on the same bogie. However, the tuning of control parameters may be problematic, as interactions between the two controllers via the vehicle body will be inevitable. To overcome this difficulty, a centralised controller for both suspensions may be used to enable independent control of the body modes.

Figure 11.17 shows how the lateral and yaw modes of a vehicle body can be separately controlled by using active suspensions in the lateral direction, and a similar scheme can be applied to actuators in the vertical direction to control the bounce and pitch modes. The output measurements from the two bogies are decomposed to give feedback signals required by the lateral and yaw controllers, respectively, and the output signals from the two controllers are then recombined to control two actuators at the two bogies accordingly. In this way, it is possible to apply different levels of control, in particular to reduce the suspension frequency and add more damping to the yaw (or pitch) mode, which is less susceptible to the low frequency deterministic inputs.

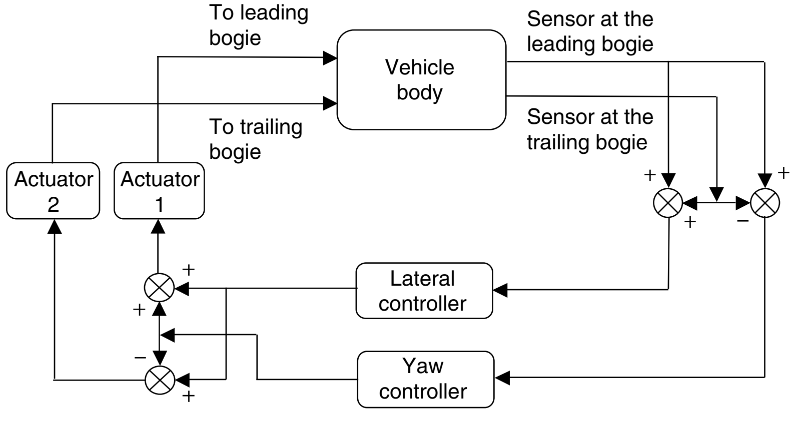

FIGURE 11.17 Modal control diagram.

5. Model-Based Control Approaches

Increased system complexity also encourages the use of mathematically rigorous design approaches such as optimal control, which enables a trade-off between ride quality and suspension deflection to be formally defined and optimised.[13] Equation 11.8 gives a typical cost function which is minimised in the design of an optimal controller to reflect the suspension design problem. Suitable choices of the weighting factors q_1, q_2, and r (on the body acceleration a_b, suspension deflection x_b and actuator force F_a) enable an appropriate design trade-off to be achieved.

$$J = \int (q_1 a_b^2 + q_2 x_d^2 + r F_a^2) \mathrm{d}t \tag{11.8}$$

6. Actuator Response

In order to implement the control laws, for example, those listed in the previous subsection, it is necessary to have force control. However, very few actuator types inherently provide a force and so an inner force feedback loop is required, but it is important to appreciate that dynamics of this actuator force loop need to be significantly faster than is immediately obvious. The physical explanation can be seen from Figure 11.18, which is a generalised scheme of a force-controlled actuator.

The force command to the actuator would be generated by an active suspension controller, not shown here because it is useful to consider what happens even with a zero force command, which should in principle leave the suspension response unchanged compared with the passive suspension. The track input will impact upon the dynamic system, and this will cause actuator movement which the force control loop must counteract in order to keep its force as close as possible to zero. Remembering that the actuator will be connected across the secondary suspension, its movements at low frequencies will be small as the vehicle follows the intended features of the track, but relatively large at high frequencies as the suspension provides isolation by absorbing the track irregularities. How well the actuator generates the force required of it in the presence of the high frequency movement depends upon the characteristics of the actuator, and it is not possible to generalise. A more detailed analysis reveals that a force loop bandwidth in the region of 20 Hz will still yield noticeable degradations in the acceleration p.s.d. on the suspended mass at around 4 Hz, but this analysis is beyond the scope of this handbook because it is a detailed control engineering issue. However, studies of this problem can be found in Ref. 14.

7. Semi-Active Control

The basis of controlling a semi-active system is to replicate, as far as possible, the action of sky-hook damping.[4] Most semi-active control strategies are based upon achieving the demanded force

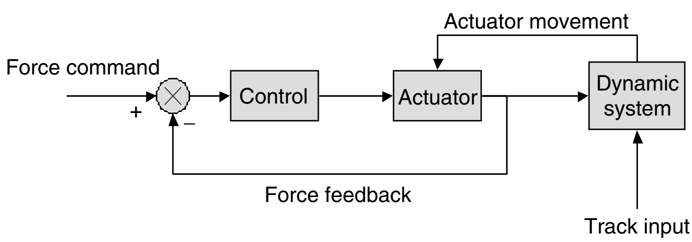

FIGURE 11.18 Actuator force control.

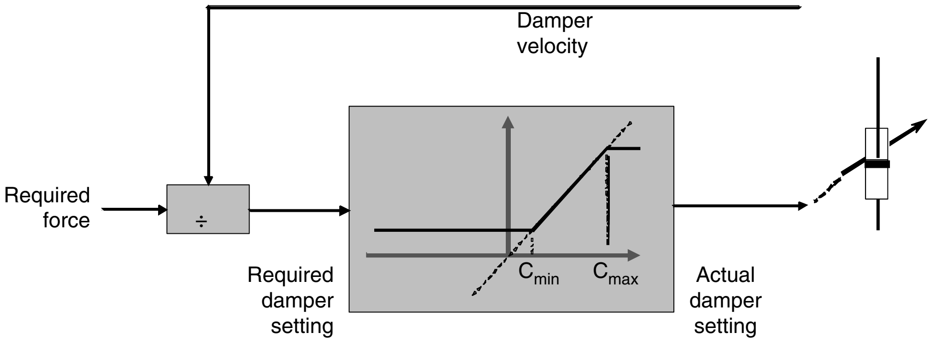

FIGURE 11.19 Controller for semi-active damper.

as closely as possible, but the actual damper setting is constrained to be between C_{min} and C_{max} — Figure 11.19 shows the control concept. To achieve operation in the upper left and lower right quadrants of the force–velocity diagram of Figure 11.3, for example, which would require a negative damper setting, the semi-active controller will simply apply C_{min}. As with full-active sky-hook damping, this would potentially create large deflections in response to deterministic features; of course a semi-active damper cannot create the necessary forces, but prefiltering, as shown in Figure 11.14, is still required to ensure an effective control law.

Extra performance benefits are realised by adopting a modal approach, similar to that shown in Figure 11.17, but achievable improvements in ride quality depend upon both the minimum damper setting and the speed of response of the control action — valve switching speeds significantly less than 10 msec are needed to ensure effective implementation.

D. Examples

1. Servo-Hydraulic Active Lateral Suspension

The first full-scale demonstration of an active railway suspension was an active lateral secondary suspension using hydraulic actuators.[15] An actuator was fitted in parallel with the lateral secondary air suspension at each end of the vehicle, as can be seen in the left hand side of Figure 11.20. The performance obtained from a comprehensive series of tests is shown on the right, from which it can be seen that a large improvement in ride quality was obtained — a 50% reduction compared with the passive suspension.

The controller used a modal structure, shown in Figure 11.21, that provided independent control of the vehicle's lateral and yaw suspension modes using the complementary filter technique.

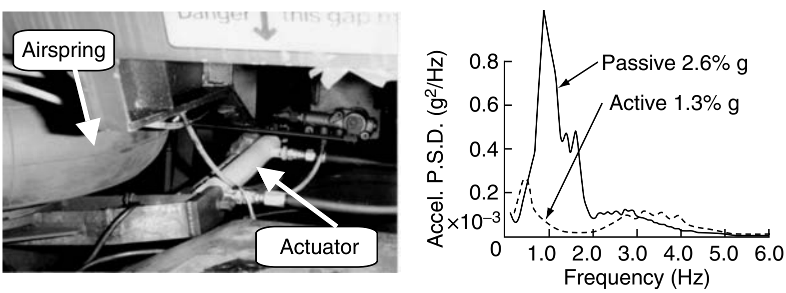

FIGURE 11.20 Servo-hydraulic actuator and experimental results for active lateral suspension.

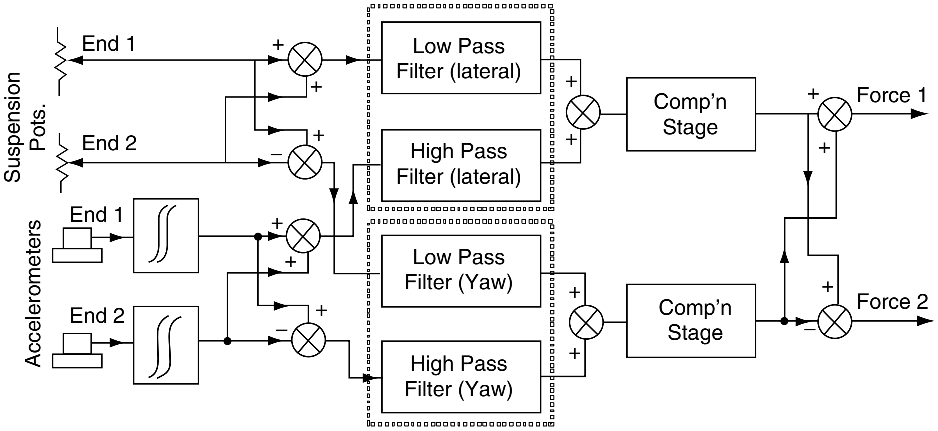

FIGURE 11.21 Controller for servo-hydraulic active lateral suspension.

Although hydraulic actuators provide a high bandwidth when used in normal applications, fast-acting force control loops (not shown in the diagram) were included to overcome the difficulty outlined above in the "actuator response" subsection, and to ensure adequate high frequency performance. Even with these inner loops, it can be seen that there is a small degradation above 3 Hz compared with the passive response.

2. Shinkansen/Sumitomo Active Suspension

The first commercial use of an active suspension was developed by Sumitomo for the East Japan Railway Company on their series E2-1000 and E3 Shinkansen vehicles, introduced in 2002.[16]

The main object of the control was the lateral vibration, i.e., closely related with riding comfort, the aim being to reduce by more than half the lateral vibration in the frequency range from 1 to 3 Hz. A pneumatic actuator system was adopted which has the advantage of easy maintenance and low cost, and is installed in parallel with a secondary suspension damper (see Figure 11.22). The damper is electronically-switched from a soft setting when active control is enabled, to the normal harder setting for passive operation.

An H-infinity controller was designed to provide robust vibration control using measurements from body-mounted accelerometers. It provides independent control of the yaw and lateral/roll

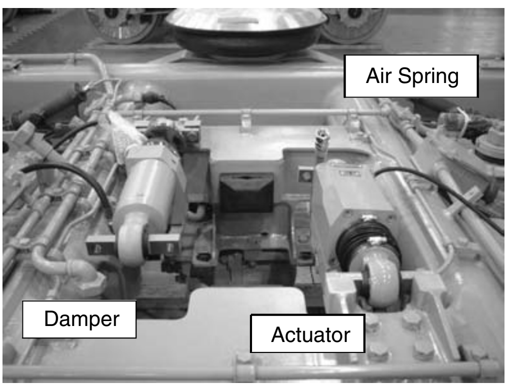

FIGURE 11.22 Actuator installation in bogie.

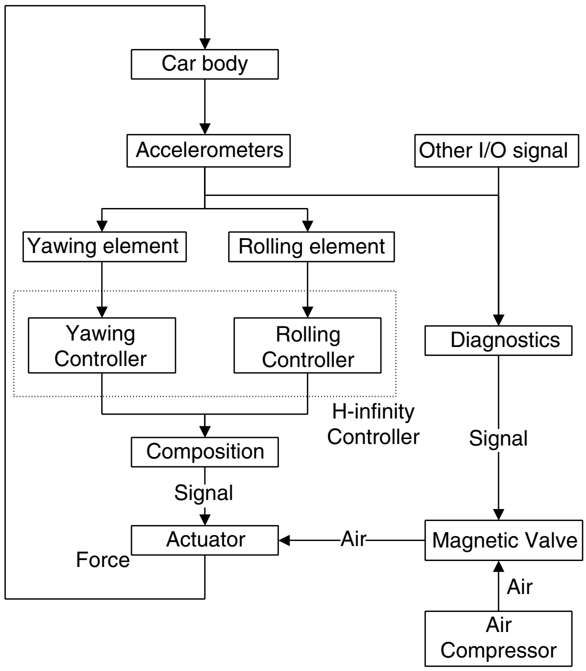

FIGURE 11.23 Overall scheme of control algorithm.

modes, with the yaw controller driving the two actuators in opposition, and the lateral/roll controller driving them in the same direction. Figure 11.23 is a diagram of the overall control scheme.

It was shown that improvements of between 5 and 9 dB in acceleration level were achievable (44–64% reduction); initially, it was a problem to achieve this kind of improvement in tunnel sections, and it was necessary to design a special controller that was switched in for use in tunnels.

V. ACTIVE PRIMARY SUSPENSIONS

A. Concepts and Requirements

Although active control could be applied to vertical primary suspensions, in fact, there seems little to be gained from such an application. The main area of interest therefore relates to controlling the wheelset kinematics through the active primary suspensions. The important issue here is the trade-off between running stability (critical speed) and curving performance, which with a passive suspension is difficult, as has been outlined in earlier chapters. Various methods of passive mechanical steering to create radial alignment of the wheelsets on curves have been attempted with some improvement. However, the idea of using active control for the wheelset steering is relatively new and, therefore, mainly theoretical studies are described in this section.

There are two types of railway wheelset. As has been explained, a solid-axle wheelset consists of two coned or otherwise profiled wheels joined rigidly together by a solid-axle, which has the advantage of natural curving and self-centring, but when unconstrained exhibits a sustained oscillation in the lateral plane, often referred to as "wheelset hunting." The structure of an independently-rotating wheelset is very similar to that of solid-axle wheelset except that two wheels on the same axle are allowed to rotate freely. The release of the rotational constraint between the two wheels significantly reduces the longitudinal creepage on curves, but it loses the ability of natural curving and centring.

The control objectives for active primary suspensions are largely related to the wheelset configurations. For the solid-axle wheelset, the controller must produce a stabilisation effort for the kinematic mode and it must also ensure desirable performance on curves. For the independently-rotating wheelset, there is a weak instability mode which needs to be stabilised. However, more critically, a guidance control must be provided to avoid the wheelset running on flanges.

B. Configurations

A number of actuation schemes are possible for implementing active steering. One of the obvious options is to apply a controlled torque to the wheelset in the yaw direction. This can be achieved via yaw actuators, as shown Figure 11.24(a), or, in practice, very likely by means of pairs of longitudinal actuators. Alternatively, actuators may be installed onto a wheelset in the lateral direction, as shown in Figure 11.24(b), but a drawback of the configuration is that the stabilisation forces also cause the ride quality on the vehicle to deteriorate. For the independently-rotating wheelset, there is a possibility of controlling the wheelset via an active torsional coupling between the two wheels, as illustrated in Figure 11.24(c). A more radical approach proposed is to remove the axle from the wheelset and to have two wheels mounted onto a wheel frame, as shown in

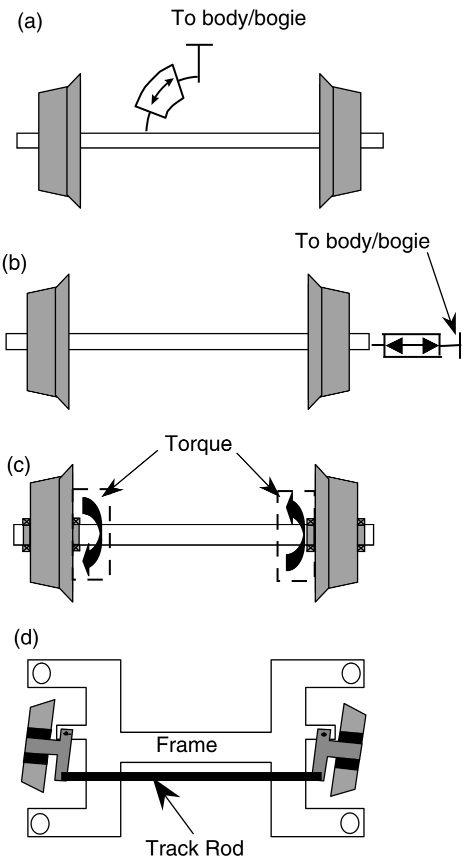

FIGURE 11.24 Actuation configurations for active steering.

Figure 11.24(d). It is then possible to apply a lateral force between the frame and the wheels to steer the wheel angle directly via a track rod, much like the steering of a car.

Similar to active secondary suspensions, the actuators for the primary suspensions can be used in combination with passive components. The passive stiffness can then be used primarily to provide the stabilisation function, whereas the actuator is used to produce an appropriate steering action on curves.

C. Control Strategies

The control development for active primary suspensions ranges from separate design for stability and steering to integrated design approaches, as presented below.

1. Stability Control — Solid-Axle Wheelset

The focus is on the stabilisation of the kinematic oscillation associated with the railway wheelset, but the control is ideally achieved in a way that it does not interfere with the natural curving and centring of the wheelset. One effective control technique is so-called active yaw damping, where a yaw torque from an actuator, as shown in Figure 11.24(a), is proportional to the lateral velocity of the wheelset.[17] The stabilising effect of the control technique can be shown using a linearised wheelset model given in Figure 11.25. It is clear from the figure that an unstable mode exists and that the inclusion of the active control loop produces positive damping to the mode. It can also be shown, using the figure, that an alternative and equally effective control method is to apply a lateral force proportional to the yaw velocity of the wheelset, a technique known as active lateral damping.[17] Both control techniques are difficult to realise using conventional passive components, but are relatively straightforward to implement with active means using sensors, controllers, and actuators.

2. Stability Control — Independently Rotating Wheelset

An independently-rotating wheelset can still be unstable, even though the torsional constraint between the two wheels on the same axle is removed — a very effective measure that significantly reduces the longitudinal creep forces at the wheel–rail interface. The instability of an independently-rotating wheelset has been reported in Refs 17,18 and it is caused by the need of a longitudinal creep (albeit small) to rotate the wheels. However, the instability is much weaker compared to the kinematic oscillation of a solid-axle wheelset, and a high level of damping can be

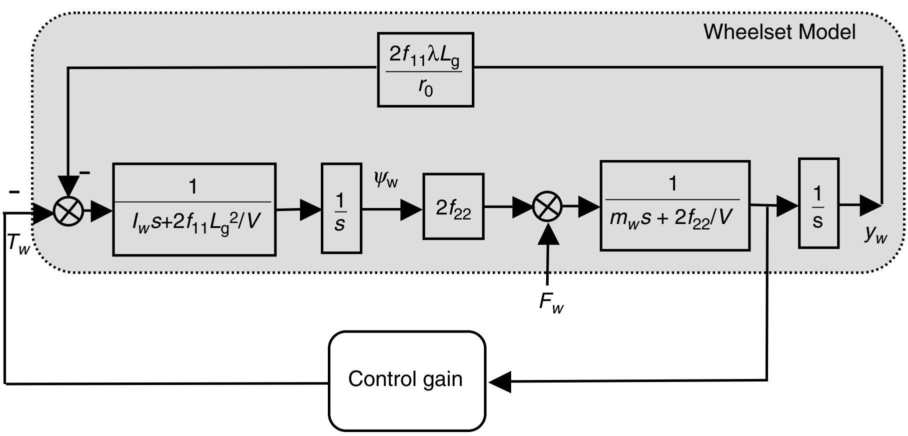

FIGURE 11.25 Active yaw damping.

attained with either a passive yaw damper or an active yaw moment control.[19,20] The latter is achieved by applying a yaw torque proportional to the lateral acceleration of the wheelset.

3. Steering Control — Solid-Axle Wheelset

When the stabilisation is obtained passively, or there are (passive) elements in the system that interfere with the natural curving action of the solid-axle wheelset, a steering action may be actively applied to provide a low bandwidth control that will eliminate, or at least reduce, the adverse effect on curves. Ideally, an active steering is required to achieve equal longitudinal creep between the wheels on the same axle (or zero force if no traction/braking) and equal creep forces in the lateral direction between all wheelsets of a vehicle. The first requirement is obviously to eliminate unnecessary wear and damage to the wheel–rail contact surfaces. The second requirement is concerned with producing and equally sharing the necessary lateral force to balance the centrifugal forces caused by the cant-deficiency.

A number of steering strategies are possible.[21] It can be readily shown that the perfect steering can be achieved if the angle of attack for two wheelsets (in addition to the radial angular position) can be controlled to be equal, and the bogie to be in line with the track on curves. This idea can be implemented by controlling the position of each actuator, such that the wheelset forms an appropriate yaw angle with respect to the bogie. As indicated in Equation 11.9 and Equation 11.10, the required yaw angle is determined by the track curve radius (R), cant-deficiency (defining the necessary lateral force F_c for each wheelset), the creep coefficient (f_{22}), and semi-wheelbase (l_x).

$$\varphi_{\text{leading}} = \sin^{-1}\left(\frac{F_c}{2f_{22}}\right) - \sin^{-1}\left(\frac{l_x}{R}\right) \approx \frac{F_c}{2f_{22}} - \frac{l_x}{R} \tag{11.9}$$

$$\varphi_{\text{trailing}} = \sin^{-1}\left(\frac{F_c}{2f_{22}}\right) + \sin^{-1}\left(\frac{l_x}{R}\right) \approx \frac{F_c}{2f_{22}} + \frac{l_x}{R} \tag{11.10}$$

Alternately, a yaw torque can be applied such that it cancels out the effect of the longitudinal stiffness of the primary suspension which forces the wheelsets away from the pure rolling. As long as the cancellation occurs at frequencies significantly lower than that of the kinematic mode, the steering strategy will not compromise the stability. This can be realised by either measuring the relative yaw angle between the individual wheelset and the bogie and compensating for the primary forces, or by controlling the forces and/or moments of the primary suspension.[22,23]

4. Guidance Control — Independently Rotating Wheelset

For the independently-rotating wheelset, a different kind of steering action is required. The longitudinal creep is no longer an issue, which is solved by the introduction of the extra degree of freedom in the relative rotation between the two wheels. However, a guidance control becomes necessary to ensure that the wheelset will follow the track without running on flanges. To provide the necessary guidance action, it is obvious that the relative displacement between the wheelset and the track (i.e., the wheel–rail deflection) is the natural choice of feedback and the control design should then be straightforward. Sensing possibilities for the measurement vary from electromagnetic, eddy current to video imaging or optical techniques, but the potentially high cost and low reliability are the main obstacles for practical applications. Instead, angles between adjacent vehicles, as well as the vehicle body yaw rate, have been used as an indirect measurement of the track curvature and a steering action is applied to control the wheelset yaw motion.[24,25]

Another guidance method is to control the relative rotational speed between the two wheels.[19] Although there is no "hard" connection between the two wheels on an IRW axle, a control action can be formulated such that the actuator will steer the axle to achieve the zero-speed difference or a speed bias defined by the track curvature. This approach adds a damping effect between the two wheels via the active means, however, it does not result in the stiff connection of the solid axle wheelset which forces the two wheels to be at the same angular position (rather than velocity) at all times.

5. Integrated Control Design

The approach to design separate controllers for the stability and steering/guidance is a pragmatic solution, and the integration of the two parts is, in general, not a problem as the two functions can be separated in the frequency domain. On the other hand, modern model based control techniques provide a more effective means to deal with the multi-objective nature of a complex control problem, although the control structures tend to be more dynamically complex. H_2 optimal controls have been proposed to either maintain the natural curving of solid-axle wheelset or to provide the missing curving action for independently-rotating wheels.[26] Also, robust H_∞ controls have been studied to tackle the problem of parameter variations, such as the conicity and creepage deviating from their nominal values during operation.[27] The stability can be guaranteed in the design process and the focus is then on the other key issues such as curving performance, uncertainty, sensing, and actuation requirements.[27]

6. Assessment of Control Performance

At low speeds, the performance of active primary suspensions is measured by the reduction of creep forces and wear at the wheel–rail interface, and the focus is primarily on curved tracks where severe wear/noise may occur in passive vehicles. Many proposed active steering schemes deliver similar performances on constant curves, although the responses in transitions will be somewhat affected by different control design, which is less critical as track transitions are generally short. Compared with passive suspensions or even the radial steering (where wheelsets are mechanically forced to take a radial angle on curves), actively steered wheelsets provide significant performance improvements, as shown in Figure 11.26. The data has been obtained from a railway bogie with conventional solid-axle wheelset(s) and with much softer passive suspensions.[21] The creep forces produced in non-active cases would be much worse for vehicles with stiffer suspensions. F_x (w1) is the longitudinal creep force of the leading wheelset of the bogie; F_x (w2) is that of the trailing wheelset; and F_y (w2 − w1) is the difference in lateral creep forces between two wheelsets. Note that, while the longitudinal creep is undesirable except for traction purposes and should be reduced as much as possible, a certain level of the creep in the lateral direction will be inevitable in order to

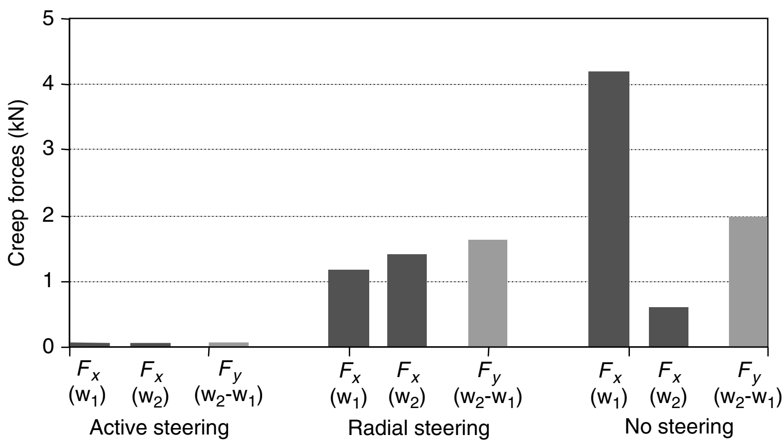

FIGURE 11.26 Steering performance comparison.

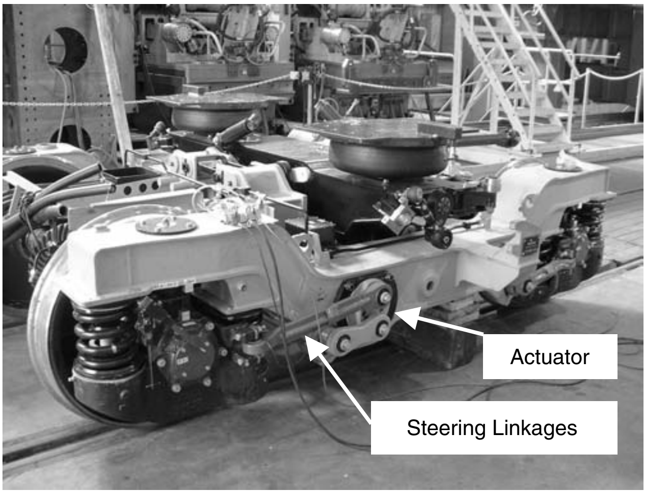

FIGURE 11.27 Actively controlled bogie.

produce a force to balance the cant-deficiency on curves. Therefore, the steering performance in the lateral direction is best assessed by examining the difference in the lateral forces, and a zero difference will indicate that track shifting forces at the two wheelsets are well balanced.

The performance of the active primary suspensions at high speeds is concerned with the running stability, and the level and speed of control effort required to control the wheelset kinematic mode and to cope with high frequency track irregularities. Those are affected more by wheelset and actuator configurations than by specific control strategies. In general, the solid-axle wheelset is much more demanding than the independently-rotating wheels as the latter arrangement allows the free rotating of the two wheels and is hence more readily adaptable to track positions.

D. Examples

This example presents an implementation and full size experiment of active control for railway wheelsets, the first example of its kind in the world.[28] Figure 11.27 shows a photograph of the

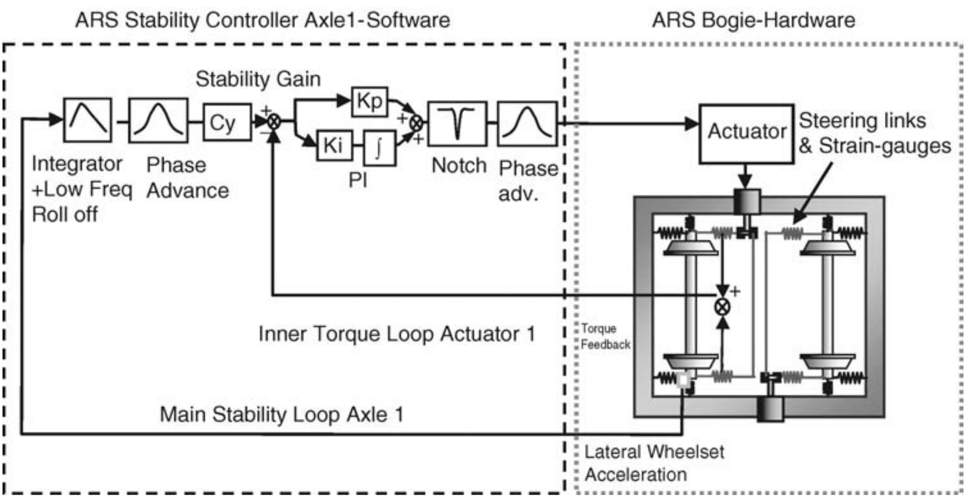

FIGURE 11.28 Stability control loop.

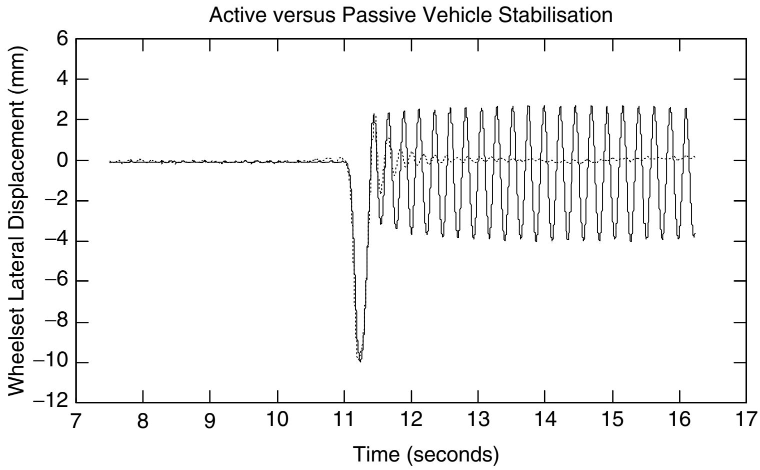

FIGURE 11.29 Rig stability test result — active and passive control.

actively controlled bogie, which is a modified version of a Bombardier VT612 bogie. The bogie has a soft primary suspension and no secondary yaw dampers. In fact, the only stiffness in the longitudinal direction is due to the shear stiffness of the vertical suspensions. Removing the secondary yaw dampers offers significant advantages in terms of the vehicle's weight and comfort, however, once removed, stability and consequently high-speed operation are significantly compromised. Without active control, the modified bogie can reach a critical speed of around 90–100 km/h.

Active control is applied by means of two electrically-driven actuation mechanisms that apply independent yawing actions to each wheelset. Two a.c. servo-motors act through gearboxes, from which steering linkage mechanisms transfer the control action to the wheelsets. Control strategies for the stability and steering are designed separately,[21,28] but the two are brought together through an integration process to ensure there are no adverse interactions. Additional measures in the control loops are needed for reasons of practicality such as sensing and actuation, which is particularly important for the stability control due to the requirement of a high bandwidth control. Figure 11.28 shows the stability controller for one of the wheelsets, where an inner loop is added to ensure a fast dynamic response of the actuator to the torque demand from the stability control loop. The controller for the second wheelset is the same.

A fully actively controlled bogie was tested on a full size roller rig in Munich, Germany. Extensive stability tests and track file tests were performed and the controller successfully operated at speeds in excess of 300 km/h. Figure 11.29 shows results for both active control and the passive vehicle, and illustrates clearly the effectiveness of active stabilisation.

VI. TECHNOLOGY

Technology of control concerned with the practicalities of implementation, the controller, sensors, and actuators, is an important issue. Satisfactory performances and costs are obviously essential, but, more critically, the safety and reliability requirements must be met before any applications can be considered.

A. Sensing and Estimation Techniques

A large variety of suitable sensors is available, and the key aspects here relate to the conflict between the control requirements and practical issues such as the reliability and cost. In general, the sensing for the active control of primary suspensions is more problematic than that for the secondary suspensions. The measurement of the wheelset movements, in particular those relative to the track, is highly desirable in order to control the wheelset effectively, but mounting effective sensors on the wheelset is extremely difficult and costly because of the harsh vibration environment.

To enable a practical and cost-effective implementation of the active control schemes, model-based estimation techniques such as Kalman filters provide a very valuable alternative to the direct measurement. Figure 11.30 shows the principle of a model-based estimator. The measured output from the sensors is compared with the output from a mathematical model of the vehicle, and any deviations will produce a corrective action via the gain matrix to compensate for inaccuracies in the model and/or sensors. Estimated state variables, or some of the variables, are then used as the feedback signals for the controller as shown by the dotted line in the figure. The use of only inertial sensors on the wheelsets and bogies/body was first proposed and proved to give excellent results,[29] but it is also possible to remove the sensors from the wheelsets and replace them with bogie-based displacement sensors to provide the primary suspensions deflection.[30]

The use of more sophisticated equipment may become economically feasible in the future to measure directly some essential feedback signals and/or track features, e.g., by using track database and Global Positioning System (GPS), whereby the estimation may be simplified and its robustness improved.

B. Actuators

The provision of high reliability actuation of sufficient performance is one of the main challenges in active suspensions. Capital cost of the total system is certainly important, but ease of installation, maintainability and maintenance cost, reliability and failure modes must all have essential inputs into the process of choosing and procuring the actuator system.

Actuator technologies, which are possible for active suspensions, are servo-hydraulic, servo-pneumatic, electro-mechanical, and electro-magnetic. Servo-hydraulic actuators themselves are compact and easy to fit, but when the power supply is included, the whole system tends to be bulky and inefficient, and there are important questions relating to maintainability. Pneumatic actuators are a possibility, particularly since the air-springs fitted to many railway vehicles can form the basic actuator, but the compressibility of air leads to inefficiency and limited controllability. Electro-mechanical actuators offer a technology with which the railway is generally familiar, and the availability of high performance servo-motors and high efficiency power electronics are favourable indicators. However, they tend to be less compact, and the reliability and life of the mechanical components needs careful consideration. Electro-magnetic actuators potentially offer an extremely

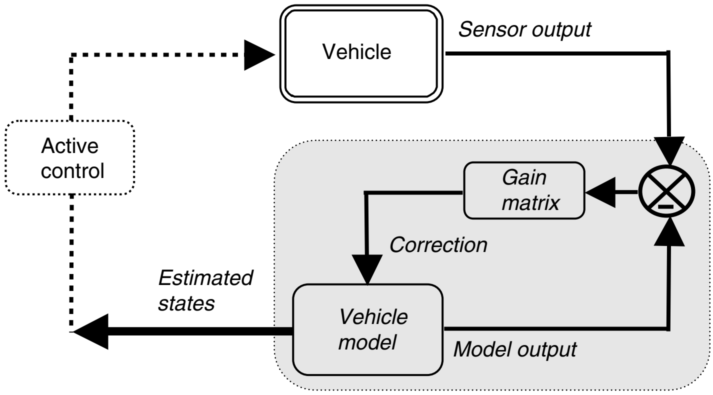

FIGURE 11.30 Block diagram of a model-based estimator.

high reliability and high performance solution, but they tend to be very bulky and have somewhat limited travel.

C. Controllers and Fault Tolerance

The availability of remarkable quantities of computing power means that the controller is unlikely to be a limiting factor in the implementation, although issues such as reliability, ruggedness etc. cannot be ignored.

More importantly, the issues of safety and reliability will have to be addressed satisfactorily. Any new technology must demonstrate that it can cope with component fault(s) without compromising passenger safety, and satisfy that any component fault would not lead to the system failure. On the other hand, the reliability and availability are of great importance to rail operators in order to maintain an effective operation of a rail system. Therefore, any active steering scheme must also meet the necessary standards of reliability.

Traditionally, mechanical components are used for the wheelset stabilisation and they are generally accepted as “safe.” Safety is assured by having all safety-critical mechanical components designed as far as is practicable not to fail. This is achieved through a combination of conservative design, careful quality control during manufacture, and rigorous maintenance procedures during operation. However, failure modes in sensors and electronics are less definable, so it becomes necessary to reconsider the approach. Having a proven mechanical back-up for an active system, which takes over in the case of an electronic system failure, is one solution, but in the longer term this is not appropriate because it will detract from benefits. The alternative is a fault-tolerant active system based upon functional and/or analytical redundancy.

There are many fault tolerant studies carried out in many other applications, especially in the aerospace and process industry, but so far, very little is reported for railway vehicles except for a couple of studies on fault-tolerant sensing and fault detection for the provision of measurement data, and so this is an important area for development of both standards and technological solutions.

VII. LONG TERM TRENDS

This chapter has covered the range of possible active railway vehicle suspension systems, from present-day tilting trains through to more speculative options that are, at the moment, little more than theoretical possibilities. Whether the active (secondary) suspensions are being used for improved passenger comfort, or active primary suspensions are being used to control the wheels and wheelsets, it should be clear to the reader that the use of active elements enables substantial performance improvements, improvements which are not possible with purely mechanical or passive solutions.

Active railway suspensions therefore represent an emerging and important technology, offering the railway industry a large variety of commercial and operational opportunities, although there are of course a number of major technical challenges. It is almost inevitable that the concepts will become progressively incorporated into railway vehicles, although it is less clear how quickly this will happen. However, many other industrial devices and systems which have started to replace or enhance mechanically-based products using electronic control concepts have never looked back, and so, almost certainly, the already-established tilting technology is just the starting point for active railway suspensions.

NOMENCLATURE

a_b: body acceleration (m/sec^2)
C: coefficient of a passive damper (Nsec/m)
C_s: sky-hook damping coefficient (Nsec/m)

F_w: Lateral control force
F_x: Longitudinal creep force
F_y: Lateral creep force
T_w: Yaw control torque
y_w: Wheelset lateral displacement
ψ_w: Wheelset yaw angle
F_a: actuator force (N)
F_c: lateral force on curved track (N)
f_{22}: lateral creep coefficient (N)
J: optimisation index
K: spring constant (N/m)
K_{CD}: can deficiency compensation factor
K_L: control gain of active leveling
K_s: control gain of suspension stiffness softening
l_x: semi wheelbase (m)
M: body mass (kg)
q_1: weighting factor for optimisation
q_2: weighting factor for optimisation
R: Curve radius (m)
r: weighting factor for optimisation
V_{active}: Vehicle speed in active case (m/s)
$V_{passive}$: Vehicle speed in passive case (m/s)
x_d: suspension deflection (m)
z: body displacement (m)
z_t: track displacement (m)
θ_{active}: cant deficiency in active case
θ_{cant}: cant angle of the track
$\theta_{passive}$: cant deficiency in passive case
θ_{tilt}: tilt angle
$\varphi_{leading}$: required yaw angle at the leading wheelset (rads)
$\varphi_{trailing}$: required yaw angle at the trailing wheelset (rads)

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