CONTROL FOR RAILWAY VEHICLES

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Contents

Summary

General aspects of control for railway vehicles are described. Some fundamental issues such as vehicle configurations, modelling, track inputs and performance requirements are first introduced, before two main control subjects are discussed which are traction control and vehicle dynamics control. Control of traction power to drive railway vehicle includes both the provision of positive tractive effort to move the train forward (traction) and that of negative tractive effort to stop the train (braking), which must be

achieved in a manner that is effective, efficient, safe and reliable. Modern railway vehicles are equipped with electric traction. Because of advances in power electronics, early resistor control of DC motors was replaced by thyrister/GTO/IGBT chopper control and the modern trend is that DC traction motors are being replaced by AC motors with inverter controls.

Control of vehicle dynamics is concerned with vehicle stability and running performance, which covers tilting and secondary suspensions for the passenger comfort and/or the increase of operational speed, and primary suspensions for stability and curving performance. Appropriate dynamic characteristics are conventionally provided through the choice of passive components such as springs and dampers, but more attractive options of active control are discussed here to reflect recent developments in the area. Traditionally, controls of traction and vehicle dynamics are largely two separate disciplines and they have gone through different development routes in the past. However the two functions interact through the wheel-rail interface and there are extra benefits to be gained if they are considered as a system with an integrated control.

1. Introduction

Control for railway vehicles can be broadly partitioned into two categorises. One is concerned with the control of vehicle dynamics which deal with issues such as vehicle stability, vehicle running behaviour and passenger ride comfort. The other is the control for the provision of traction and braking that drive the railway vehicles.

The development of railway vehicles and vehicle dynamics control has brought progressively greater comfort and convenience for passengers and benefit to the railways. Early four-wheel (two axle) vehicles with simple suspension components between the body and wheels were of very crude structures by today's standard, offering little comfort and suffering from physical wear and tear. The understanding of vehicle dynamics was limited and the control was basic.

The invention of bogie (or truck) in 1809 was a big step forward in the vehicle design, which improved ride-quality, achieved running stability without fear of wheel-wear on curves and enabled vehicles to operate at much high speeds. Bogie vehicles are now a standard configuration for railway passenger vehicles throughout world. A key feature in the use of bogies is that it allows a two-level suspension to be designed. The primary suspension components between the bogie and wheels are chosen mainly for the running stability, and the secondary suspension between the bogie and the vehicle body is used to provide ride comfort for passengers. However, the consequence has been greater mechanical complexity with increased weight (hence reduced efficiency) and all functions of the vehicle dynamics are realised and adjusted by a suitable choice of mechanical components.

Railway engineers have now turned their attention back to the idea of mechanically simple vehicle configurations, but with the vehicle dynamics control largely provided through the use of mechatronic and electronic components. It is recognised that active controls can achieve far beyond what is possible with mechanical suspensions and the trend is that future railway vehicles will be of much lighter and simple structure with much better performance and enhanced functions.

The progress of railway traction has been even more inspiring. The steam locomotives were the main supply of traction from 1830 through to the early of the 20th century and they are still used in some parts of the world. Control of traction power is realised via the regulator valves and steam entry ports.

By the end of the 19th century, railway engineers started to appreciate the superior traction capabilities and efficiency of electric traction. Before power electronics became available, electric traction was largely provided by d.c. motors and the motor control was achieved through the use of contactors and resistors.

The simple resistor control was later superseded in DC traction by chopper control using power switching devices such as thyristors and lately IGBTs, with which any desired value of current can be maintained in the motor at the appropriate level and the full performance of the motor can be realised without appreciable losses in the control. The use of electronic control enables many other functions such as regulation of traction effort, improved use of adhesion and control of slipping to be incorporated.

More recently, AC motors are being preferred for traction applications with obvious advantages such as less maintenance, lower cost and higher power/weight ratio. This has been made possible largely due to the advances in the power electronics and microprocessors and development in the motor control techniques. Modern variable speed AC drives are now able to offer similar performances to their DC counterpart.

Following sections will present all aspects of control on railway vehicles, including vehicle configurations; traction and braking control; suspension; guidance control and system integration. Effort will be given to cover as much as possible the modern technology and recent development in the control for railway vehicles. The emphasis is upon passenger vehicles, although some of the ideas could also apply for freight vehicles.

2. Overview of Railway Vehicle, Vehicle Models and Track Inputs

Railway vehicles carrying passengers and/or goods run on tracks, which provide the support and guidance functions. The interface between the two is established at contact point(s) at the wheels and rail surface, and both the vehicle configuration and the track input greatly influence how vehicles behave.

2.5. Vehicle Configurations

Conventional bogie vehicles are the most commonly used configuration and Figure 1 gives simplified side-view and end-view diagrams of a typical bogie vehicle. In this vehicle configuration, the body frame is supported by two bogies via secondary suspensions either directly or via bolsters. On each bogie, there are four wheels connected via primary suspensions. The wheels are arranged in two pairs, where each pair is rigidly connected via a common axle (known as the solid-axle wheelset) such that the two wheels have to rotate at the same speed. Suspensions are provided in the vertical, lateral and longitudinal directions, but extra stiffness is often added in the roll direction. These suspensions mainly comprise passive springs and dampers connected in parallel and/or series, but airbags providing better performance are commonly used as secondary (vertical) suspensions on modern passenger vehicles.

The weight of the bogies forms a high proportion of the total weight of a bogie vehicle. To reduce the tare (unladen) weight, there are vehicles with the articulated configuration. In these a common bogie supports the two ends of adjacent vehicles as shown in Figure 2. Two conventional bogies are still needed at either end of a train-set. For a train-set of *N* vehicles, only *N+1* bogies are required for this arrangement instead of *2*x*N* for a train-set with bogie vehicles.

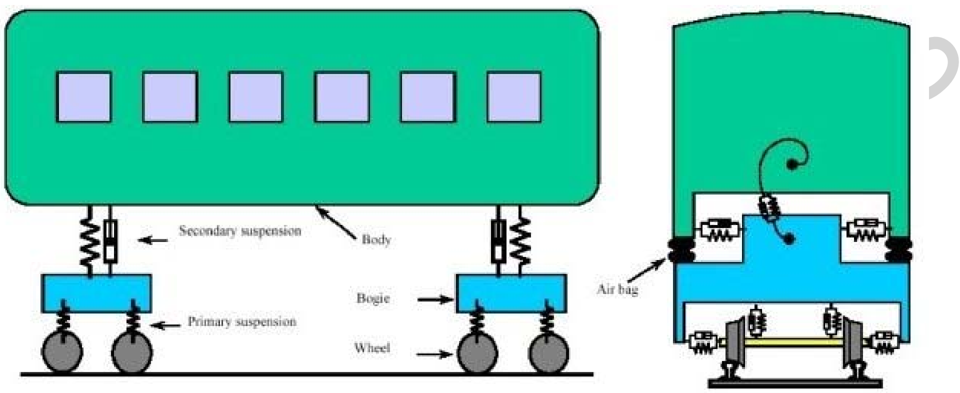

Figure 1. Simplified side-view and end-view diagrams of a bogie vehicle

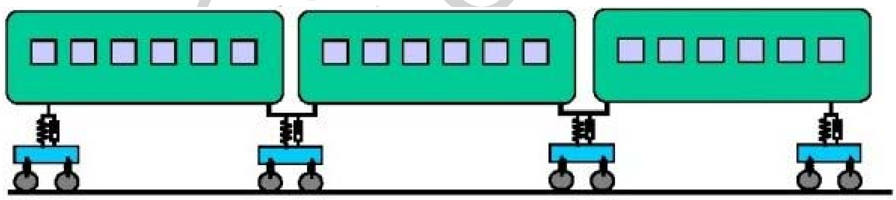

Figure 2. Simplified side-view diagram of articulated vehicles

The configuration of a two-axle vehicle is mechanically much simpler than the bogie vehicle and Figure 3 shows a simplified diagram of the configuration. Each bogie is replaced by a much lighter frame, where the wheelset is mounted. Primary suspensions may not be necessary as they have relatively less effort on the vehicle stability because of the lighter wheelset frame. Compared with the bogie vehicle, a two-axle vehicle is much lighter partly because of the removal of bogies and partly because it is normally shorter.

The problem with this configuration is that it is extremely difficult to obtain the stability and required ride quality at high speeds and to achieve good curving performance at low speed using traditional passive solutions. However with recent advances in active control technology for railway vehicles, it is highly conceivable that the two-axle vehicle and its articulated variants could be preferred for future trains.

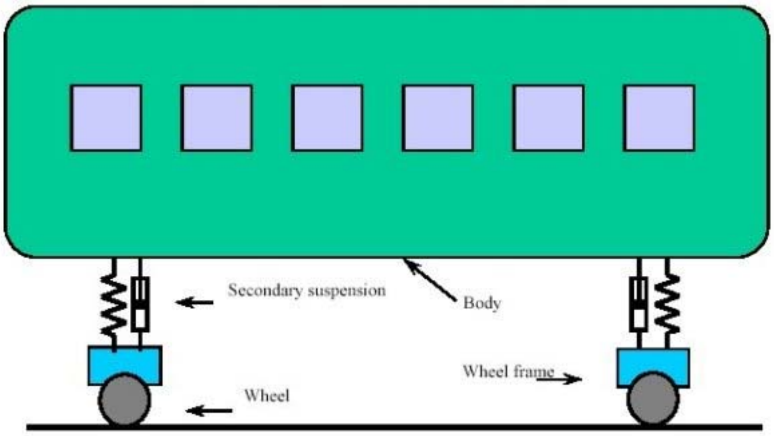

Figure 3. Simplified side-view diagram of a two-axle vehicle

2.6. Railway Track Inputs

The track inputs to rail vehicles comprise two distinct types: the deterministic and random inputs. The deterministic track inputs are intended features representing the design alignment associated with gradients and curves. Figure 4 illustrates a typical gradient and a curved track. On either end of the constant gradient/curve, there is normally a transition section connected to level/straight track with the length equivalent to about 1-3 seconds for designed operation speed. On curves, the track is also elevated at one side in the roll direction to give a 'cant' angle that coincides with the change of curve radius so that the lateral force experienced on the vehicle is reduced.

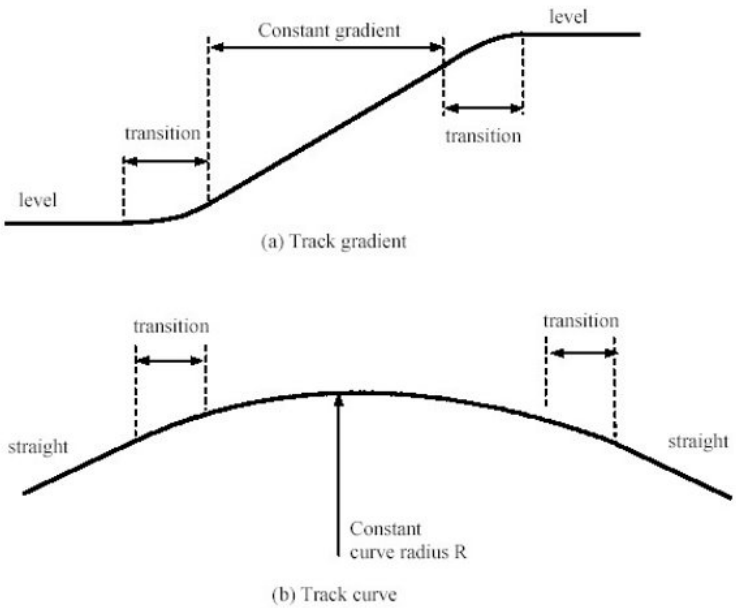

Figure 4. Track Gradient and Curves

The random inputs are track irregularities representing the deviations from intended alignment. Track irregularities exist in both vertical and lateral directions. Two more dominant irregularities are those of 'Alignment' and 'Vertical Surface Profile'. Alignment is the average of the lateral positions of two rails (often referred to as the centre line), and vertical surface profile is the average elevation of the two rails. There are also two irregularities with relatively lower magnitude:- irregularities of gauge and cross level. Gauge is the horizontal distance between two rails. Cross-level is the difference between the elevations of two rails. Depending on the magnitude of those irregularities, track is partitioned into different classes and an operating speed limit is imposed. Alignment variation is one of the major causes of lateral vibration in railway vehicles, whereas gauge plays an important role in the lateral stability. On the other hand, the vertical profile and cross-level are the main excitation sources of vertical vibration. There are other track features such track switches and joints, which produce undesirable pulse inputs to vehicles.

The characteristics of random track inputs may be represented using their power spectrum density. In the vertical direction, equation 1 is widely accepted as a simple and appropriate expression of the spatial spectrum of railway track, where A_{v} is the coefficient of vertical track roughness and f_{s} is the spatial frequency. In fact the 'accepted' vertical spectrum generally has 3rd and 4th order terms, but these do not affect things until perhaps 5-10 Hz. Therefore for secondary suspension applications it is possible to use the simplified expression, but to some extent this is expediency because it makes it white noise as a velocity. When the speed of a vehicle V_{t} is considered, this may be readily converted into the temporal spectrum as given in equation 2, where f_{t} is the temporal frequency.

The lateral track irregularities have a different structure to that in the vertical direction. Equation 3 gives the spatial power spectrum and equation 4 is the temporal expression. A_{l} is the coefficient of lateral track roughness. The velocity spectrum is not 'white noise' and there is a steady roll-off with the increase of frequency.

$$S_{\mathrm{vS}}(f_{\mathrm{s}}) = \frac{A_{\mathrm{v}}}{f_{\mathrm{s}}^2} \qquad m^2 (cycles/m)^{-1} \tag{1}$$

$$S_{\mathrm{vT}}(f_{\mathrm{t}}) = \frac{(2\pi)^2 \cdot A_{\mathrm{v}} \cdot \mathrm{V_t}}{f_{\mathrm{t}}^2} \qquad (m/s)^2 (Hz)^{-1} \tag{2}$$

$$S_{\mathrm{lS}}(f_s) = \frac{A_{\mathrm{l}}}{f_{\mathrm{s}}^3} \qquad m^2 (cycles/m)^{-1} \tag{3}$$

$$S_{\mathrm{lT}}(f_t) = \frac{(2\pi)^2 \cdot A_{\mathrm{l}} \cdot \mathrm{V_t}^2}{f_{\mathrm{t}}^3} \qquad (m/s)^2 (Hz)^{-1} \tag{4}$$

2.7. Modelling of Vehicle Dynamics and Wheel-Rail Contact

Railway vehicles are dynamically-complex multi-body systems. Each mass within the system has six dynamic degrees of freedom corresponding to three displacements (longitudinal, lateral and vertical) and three rotations (roll, pitch and yaw). Each degree of freedom results in a second-order differential equation and hence 6xN differential equations will be necessary to represent the system mathematically, where N is the number of masses. For a conventional bogie vehicle, there are seven main masses (one vehicle body, two bogies and four wheelsets) and therefore a total of 42 second-order differential equations will be required if no constraints are considered. In addition, wheel-rail contact presents a highly non-linear dynamic/kinematic problem adding extra complexity to the already complex system.

For applying control to complex systems such as these it is important to distinguish between the design model and the simulation model, the former a simplified model used for synthesis of the control strategy and algorithm, the latter a more complex model to test fully the system performance. For example, there is a relatively weak coupling between the vertical and lateral motions of a vehicle and, depending on the objectives, only selected degrees of freedom need to be included in the design model. To study the vertical response, it would be adequate to include the bounce, pitch and sometimes roll degrees of freedom of the components. For the lateral response, the lateral, yaw and sometimes roll degrees of freedom are sufficient. In studies of the longitudinal dynamics, the longitudinal, pitch and roll degrees of freedom may be included in the model. As a common practice, the vehicle model is partitioned into side-view, plan-view and end-view models. The side-view model is concerned with the bounce, pitch and longitudinal degrees of freedom; the plan-view model deals with the lateral and yaw motions and the end-view model covers the bounce and roll motions.

In the modelling, Newton's law is applied to every degree of freedom of vehicle body and bogies, and external forces/torques are applied through suspension components. For design purposes the suspensions can be largely considered as linear components, and can readily be generated using control design software such as Matlab. More thorough modelling for simulation will require the inclusion of non-linearities due to factors such as dead-band, hardening/softening and Coulomb friction elements, and nowadays will normally be undertaken using one of a number of 3D modelling packages which will incorporate the full complexity and non-linearity, including effects such as body flexibility that are essential for properly assessing ride quality, for example. It is of course essential that such modelling software can support the integration of the controller into the mechanical system.

Modelling of the wheel-rail contact is quite complex and consequently an introduction to the basic mechanics is included here. Although the true interaction at the contact patch is complex and non-linear, it is a common simplification to consider the effects separately for the three translational directions with linearised creep coefficients, and using this assumption properties can be described by wheel creepage and creep force generated at the contact interface. Railway wheel experiences longitudinal (γ_1), lateral (γ_2) and spin (ω_3) creepages which are defined as relative translational and rotational velocities between the wheel and rail as described in equations 5-7. V_x^{w}, V_y^{w} and Ω_z^{w} represent longitudinal, lateral and rotational velocities of the wheel. V_x^{r}, V_y^{r} and Ω_z^{r} represent longitudinal, lateral and rotational velocities of the rail.

$$\gamma_1 = \frac{\left(V_x^{\,w} - V_x^{\,r}\right)}{V_t} \qquad (5)$$

$$\gamma_2 = \frac{\left(V_y{}^{w} - V_y{}^{r}\right)}{V_t} \tag{6}$$

$$\omega_3 = \frac{\left(\Omega_z{}^{w} - \Omega_z{}^{r}\right)}{V_t} \tag{7}$$

The creepages result in creep forces that will be applied to both the wheel and track and the relationship between the creepages and the creep forces has been investigated by many researchers and engineers alike for a variety of situations. The simplest linear relationships for the longitudinal (F_1), lateral (F_2) and spin (M_3) creep forces are given in equations 8-10, which are applicable for small creepage cases. f_{11}, f_{22}, f_{23} and f_{33} are creep coefficients.

$$F_1 = -f_{11} \cdot \gamma_1 \tag{8}$$

$$F_2 = -f_{22} \cdot \gamma_2 - f_{23} \cdot \omega_3 \tag{9}$$

$$M_3 = f_{23} \cdot \gamma_2 - f_{33} \cdot \omega_3 \tag{10}$$

The total creep force available at the wheel-rail contact is limited by the coefficient of friction at the contact interface and the effect of creep saturation needs to be considered especially for powered wheels. The coefficient of friction is largely dependent on the conditions of the wheel and rail surfaces. In dry conditions with no surface contamination, the coefficient can be as high as 0.4, i.e. the saturation level is at 40% of the normal force applied. But the coefficient may be significantly lower (0.05-0.07) if the contact surface is contaminated with water, snow and leaves etc. In addition, the coefficient is also adversely varied with the total creepage although the dependency is smaller than with the surface conditions.

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Biographical Sketches

T. X. Mei received a B.Sc. and a M.Sc. degree from Shanghai Institute of Railway Technology in 1982 and 1985 respectively. He also received a M.Sc. from Manchester University in 1991 and a Ph.D from Loughborough University in 1994. He held an academic position in the department of Electrical Engineering at Shanghai Institute of Railway Technology between 1985 and 1989, where he was involved in a variety of research and teaching activities. From 1993 to 1998, he worked in industry in UK (with three different companies:- Linear Motors Ltd., Brush Traction and GEC Alsthom), where he carried out and successfully completed a number of industrial R&D projects. In 1998-2000, he was a senior research engineer at the Department of Electronic and Electrical Engineering at Loughborough University. From 2001, he is a Lecturer of Control Engineering in the School of Electronic and Electrical Engineering at the University of Leeds. His research is concerned with a variety of practical applications of advanced control, including motor drives, traction control, vehicle dynamics, contact mechanics, excitation control, fault-tolerant systems, system integration and real time implementations. He is a Member of the IEE and a Chartered Engineer.

Roger Goodall graduated from Peterhouse, Cambridge, in 1968. After working for two years for one of the GEC companies he joined British Rail's R & D Division in Derby, where he was involved in a variety of control-related projects connected with the railway industry. In 1982 he took up an academic position in the Department of Electronic and Electrical Engineering at Loughborough University, and is currently Professor of Control Systems Engineering. His research is concerned with a variety of practical applications of advanced control, usually for high performance electro-mechanical systems. He holds a number of research grants from EPSRC, the EC and industry concerned with active railway vehicle suspensions, advanced data fusion architectures for aerospace applications, and targetted processor architectures for implementation of high-performance controllers. He is a Fellow of both the IEE and IMechE, and has received a number of awards from both these institutions for his published work.