CONTROL SYSTEM DESIGN METHODOLOGY OF AN ACTIVE STABILISATION SYSTEM FOR A HIGH SPEED RAILWAY VEHICLE

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Keywords: active control, railway vehicles, suspension design, stability control

Abstract

This paper describes a control system design methodology used to produce an experimental active stability system for a high speed railway vehicle. The complete design process is described, starting with generating mathematical models of the system for control system design and for system validation. The control strategy is briefly outlined along with the necessary refinements required for practical implementation. Some brief results for the stability controller functioning on an experimental vehicle during a testing phase are presented.

1. Introduction

The successful implementation and testing of practical control strategies can be a complex and difficult process, even when a structured approach is followed. If a structured approach is not followed then the consequences may range from merely poor controller performance to catastrophic failures, and some problems may even remain hidden and may not be discovered until much later, possibly in the production phase, resulting in costly system redesigns.

The system requirements for modern high-speed railway vehicles are both diverse and stringent. The vehicles must run on a variety of tracks, negotiate a range of curves, provide good comfort levels, minimise track damage and minimise wheel/rail wear, to specify a few. These are becoming increasingly difficult to meet with conventional engineering approaches, however the integration of electronics with mechanical components has provided the opportunity to implement mechatronic solutions that can meet the ever-increasing requirements.

The running stability of a railway vehicle is a complex issue involving both multi-body dynamics and wheel-rail contact mechanics [7]. Typically a railway vehicle is stable at low speeds, but as the speed is increased a point is reached at which the vehicle becomes unstable (the instability is called 'hunting'). The coning of the wheelsets, which provides guidance, is also the source of this dynamic instability, and variations in the coning angle of the wheelsets (conicity) have to be considered as well as the full range of vehicle operating speeds.

The vehicle studied in this paper has no secondary yaw dampers (i.e from the bogie to the vehicle body), which conventionally help to provide stability at high speeds. However, these dampers have an unwanted effect of transmitting high frequency vibration to the vehicle body, and this has secondary effects on the design of the vehicle body (stiffness and consequently weight). The dampers themselves are also significantly heavy and are one of the least reliable of the suspension components. Removing the secondary yaw dampers therefore offers significant advantages in terms of the vehicle's weight and comfort, however once removed stability and consequently high-speed operation are significantly compromised.

An active stability system is proposed which provides vehicle stability at high speed without adding significant weight penalties. Control is applied by means of two electrically-driven actuation mechanisms. The mechanical arrangement, details of which are not included here, enables yawing action to be applied via a series of linkages independently to each wheelset. Although the focus of this paper is the stability controller, later stages of the project will investigate the integration and design of an active steering controller [6]. The aim of the steering controller is to provide good tracking performance when the vehicle is in tight curves and to minimise wheel/rail wear. It will utilise the same hardware (actuators, sensors and controller) and will require additional software only.

2. System Modelling

There are two requirements for system modelling: the first is for design models suitable for developing the controllers, and the second is for simulation models suitable to predict the complete behaviour of the vehicle and the control system. The design models tend to be simpler (relatively), linear and aim to embody the fundamental dynamics only, whereas the simulation models are complex and non-linear. For the generation of the control system models, SIMULINK has been used, which also enables the controllers to be designed and analysed using specialist control techniques and functions; classical tools such as Nichols and Bode plots, and more sophisticated model-based approaches such as optimal and robust control.

Figure 1 shows the plan-view half-vehicle design model used for this study - the parameters for the model are given in Appendix 1 and are representative of a modern high-speed railway vehicle. It includes seven degrees-of-freedom, i.e. lateral and yaw modes for each wheelset and for the bogie frame, and a lateral mode for the vehicle body (defined by equations 1-7). The model is therefore 14th order overall, and is a highly coupled complex MIMO system. It can be represented in state space form by equation 8 [3,4].

$$m_w \ddot{y}_{w1} + \left(\frac{2f_{22}}{V_s} + C_s\right)\dot{y}_{w1} + K_s y_{w1} - 2f_{22}\theta_{w1} - C_s \dot{y}_g - K_s y_g - C_s L_v \dot{\theta}_g - K_s L_v \theta_g = m_w\left(\frac{V_s^2}{R_1} - g\theta_{c1}\right) \tag{1}$$

$$I_w \ddot{\theta}_{w1} + \frac{2f_{22}L_g^{\,2}}{V_s}\dot{\theta}_{w1} + \frac{2f_{11}\lambda L_g}{r_0} y_{w1} = \frac{2f_{11}L_g^{\,2}}{R_1} - \frac{2f_{11}\lambda L_g}{r_0} y_{t1} + T_{w1} \tag{2}$$

$$m_w \ddot{y}_{w2} + \left(\frac{2f_{22}}{V_s} + C_s\right)\dot{y}_{w2} + K_s y_{w2} - 2f_{22}\theta_{w2} - C_s \dot{y}_g - K_s y_g + C_s L_v \dot{\theta}_g + K_s L_v \theta_g = m_w\left(\frac{V_s^2}{R_2} - g\theta_{c2}\right) \tag{3}$$

$$I_w \ddot{\theta}_{w2} + \frac{2f_{22}L_g^{\,2}}{V_s}\dot{\theta}_{w2} + \frac{2f_{11}\lambda L_g}{r_0} y_{w2} = \frac{2f_{11}L_g^{\,2}}{R_2} - \frac{2f_{11}\lambda L_g}{r_0} y_{t2} + T_{w2} \tag{4}$$

$$\begin{aligned} m_g \ddot{y}_g + (2C_s + C_{sc})\dot{y}_g + (2K_s + K_{sc})y_g - C_s \dot{y}_{w1} - K_s y_{w1} - C_s \dot{y}_{w2} - K_s y_{w2} - C_{sc}\dot{y}_v - K_{sc} y_v \\ = m_g V_s^2\left(\frac{1}{2R_1} + \frac{1}{2R_2}\right) - m_g g\left(\frac{\theta_{c1}}{2} + \frac{\theta_{c2}}{2}\right) \end{aligned} \tag{5}$$

$$I_g \ddot{\theta}_g + 2L_v^{\,2} C_s \dot{\theta}_g + 2L_v^{\,2} C_s \theta_g - L_v C_s \dot{y}_{w1} + L_v C_s \dot{y}_{w2} - L_v K_s y_{w1} + L_v K_s y_{w2} = -(T_{w1} + T_{w2}) \tag{6}$$

$$m_v \ddot{y}_v + C_{sc}\dot{y}_v + K_{sc} y_v - C_{sc}\dot{y}_g - K_{sc} y_g = m_v V_s^2\left(\frac{1}{2R_1} + \frac{1}{2R_2}\right) - m_v g\left(\frac{\theta_{c1}}{2} + \frac{\theta_{c2}}{2}\right) \tag{7}$$

$$\dot{\mathbf{X}} = \mathbf{A}\cdot\mathbf{X} + \mathbf{B}\cdot\mathbf{u} + \mathbf{\Gamma}\cdot\mathbf{w} \tag{8}$$

where

$$\mathbf{X} = \begin{bmatrix} \dot{y}_{w1} & y_{w1} & \dot{\theta}_{w1} & \theta_{w1} & \dot{y}_{w2} & y_{w2} & \dot{\theta}_{w2} & \theta_{w2} & \dot{y}_g & y_g & \dot{\theta}_g & \theta_g & \dot{y}_v & y_v \end{bmatrix}^T$$

$$\mathbf{w} = \begin{bmatrix} 1/R_1 & \theta_{c1} & y_{t1} & 1/R_2 & \theta_{c2} & y_{t2} \end{bmatrix}^T$$

$$\mathbf{u} = \begin{bmatrix} T_{w1} & T_{w2} \end{bmatrix}^T$$

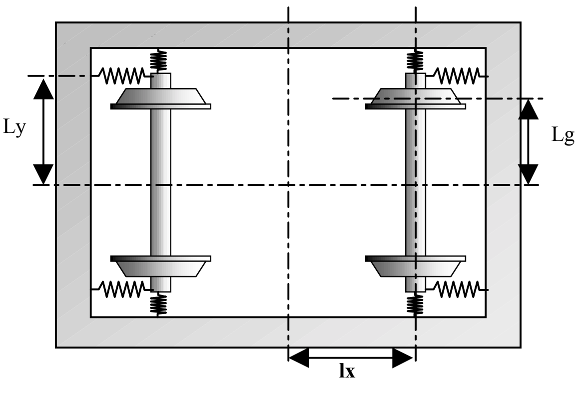

Figure 1 Design Model (Plan View Bogie - Simulink)

The main dynamic modes for the wheelset and bogie lie in the range from 2-10Hz, and there is also a body mode at a little under 1Hz.

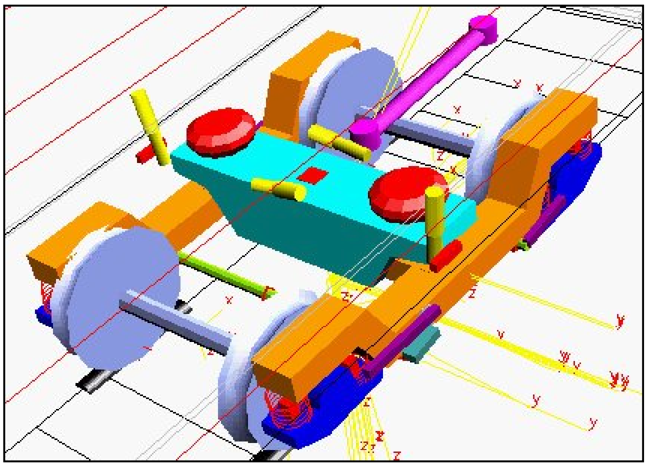

Figure 2. Simulation Model (Full Vehicle - Simpack)

For the generation of the simulation models a non-linear multibody software package called SIMPACK has been used.

This package is able to model, with high degrees of accuracy, particular features of railway vehicles such as:- the non-linear wheel/rail contact geometry, non-linear characteristics of some suspension components, large displacements and rotations between bodies etc. Figure 2 shows the leading bogie of the vehicle model generated by SIMPACK.

The models were validated against each other, by comparing eigenvalues and by the comparison of time histories in response to discrete inputs such as steps in the track. The models were also validated using experimental data obtained from the vehicle. This included a modal analysis of the bogie and also a stability test of the passive vehicle.

3. Control Strategy Design

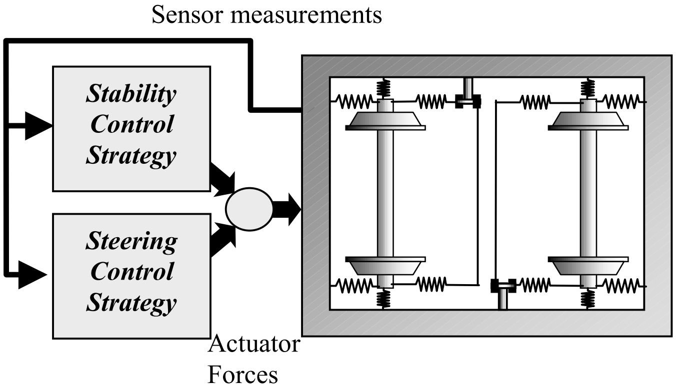

Figure 3 Basic Controller

A control-oriented analysis of a single wheelset [1] has shown that, whereas passive damping does not assist with stability, two types of active damping will give stability for a solid axle wheelset. The first active damping approach is to apply a lateral force proportional to the yaw velocity of each wheelset (hence termed active lateral damping). The second approach, used for the controller studied in this paper, is to apply a yaw torque between the bogie and each of the wheelsets proportional to the lateral velocity of the wheelsets (hence termed active yaw damping). This approach also enables the stability control actuators also to be used for steering purposes: stability action is applied at higher frequencies (>2Hz), whereas steering is a low frequency requirement (<0.5Hz).

For a railway vehicle the control design is more complicated than a single unconstrained wheelset because the two wheelsets are inter-coupled via the bogie frame. Therefore the calculation for the damping values (i.e. gains in the controllers) can only be used as a general guidance and in some cases it may be necessary to introduce phase compensation terms in the feedback loops.
The primary design objective for the stability controllers is to provide at least 20% damping across all modes at a vehicle speed of 65m/s and with a conicity of 0.3. A secondary, but nonetheless essential, design goal is to provide at least 5% damping across all modes at a higher conicity of 0.5.

The initial controller design and assessment used the models generated in Simulink, since these are relatively simple but embody the fundamental vehicle dynamics. For the final tuning and assessment of the controllers in realistic operating conditions before being implemented on the experimental vehicle, it was necessary to use the non-linear vehicle models in Simpack. Two methods were used to perform this task:
i) linear, but complex, models were exported from Simpack into Simulink to enable the controller performance to be tested and simulated in Simulink using complex linear models.
ii) the controller was simulated in Simulink, while the non-linear vehicle dynamics were simulated in Simpack. In this case the two packages are linked using co-simulation [2] as shown in Figure 4. Time history results are shown in Figure 5.

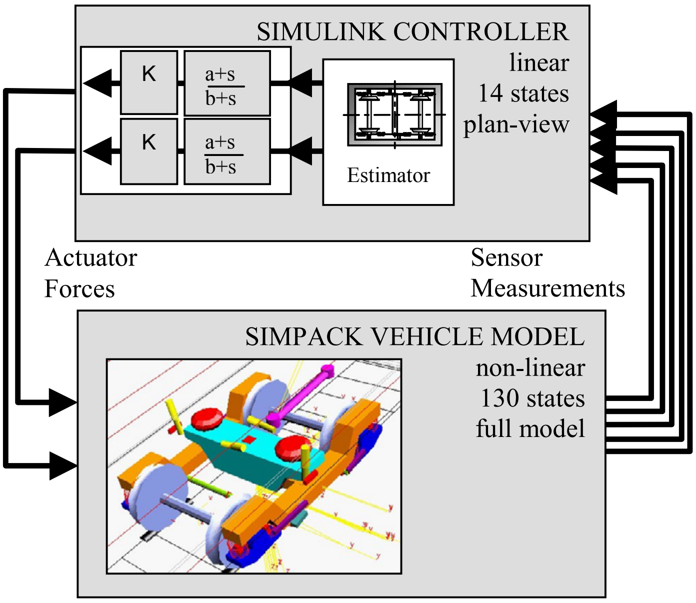

Figure 4 Control Algorithm Validation

Comprehensive simulation studies have been performed of the various control options. These studies have included stability tests and straight track tests using recorded track data.

The controller software, written in C, was validated by first replacing the Simulink Controller with the C-code in the simulations, and secondly by performing a frequency response analysis of the DSP running the controller code and comparing it to one taken of the Simulink Controller (Figure 6).

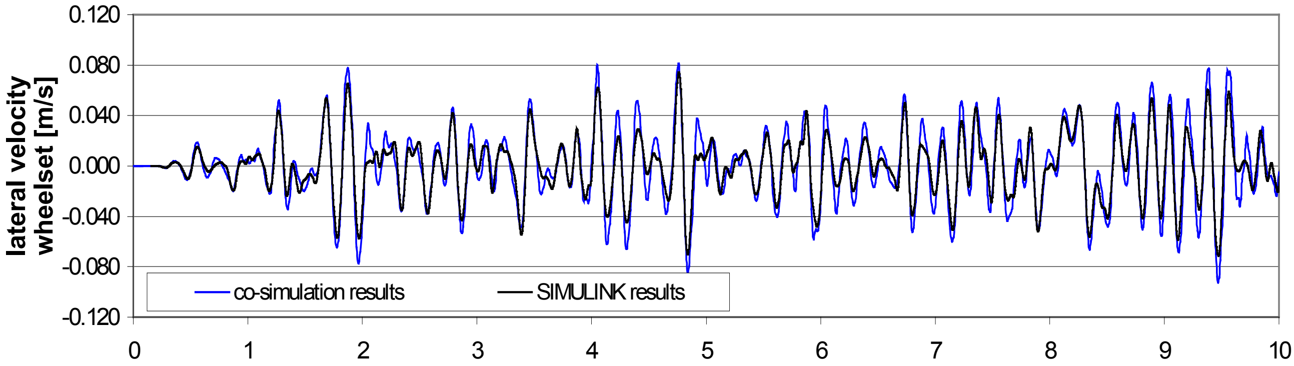

Figure 5 Co-simulation results

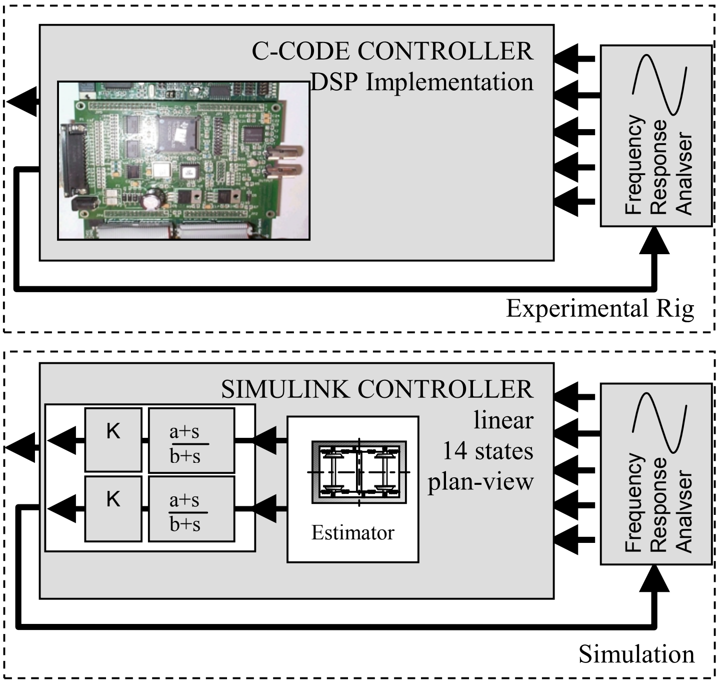

Figure 6 Software Validation

4. Experimental Results

Following the simulation study a period of testing on an experimental train (Figure 7) has been conducted on a full size roller rig. During this testing phase extensive stability tests and track file tests have been performed and the controller has successfully stabilised the vehicle at speeds in excess of 300km/h.

Figure 8 shows three results for the stability test (a 1-cosine input onto the leading axle with an amplitude of 7.5mm). The top result shows both the actively controlled vehicle and passive vehicle at 100 km/h (this vehicle in a passive configuration was unstable at speeds above 100km/h). The next two results show the stabilising effect of the controller at speeds of 200km/h and 230km/h.

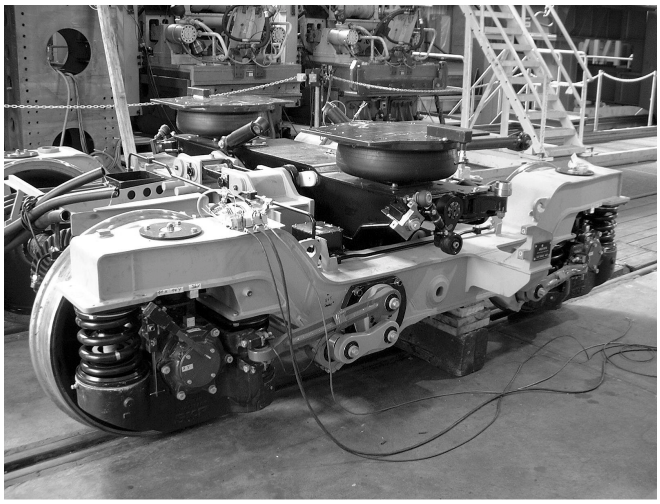

Figure 7 Experimental Bogie

5. Conclusions

This work discusses the process of developing models and control algorithms that can be implemented with a high degree of confidence. Control system models and vehicle simulation models have been used extensively, and the use of co-simulation techniques have proven to be a powerful tool for the implementation of practical control systems on complex multi-body systems. The design process has been validated by the successful implementation of an active stability system on an experimental vehicle.

Acknowledgements

The authors wish to acknowledge the support of Bombardier Transportation.

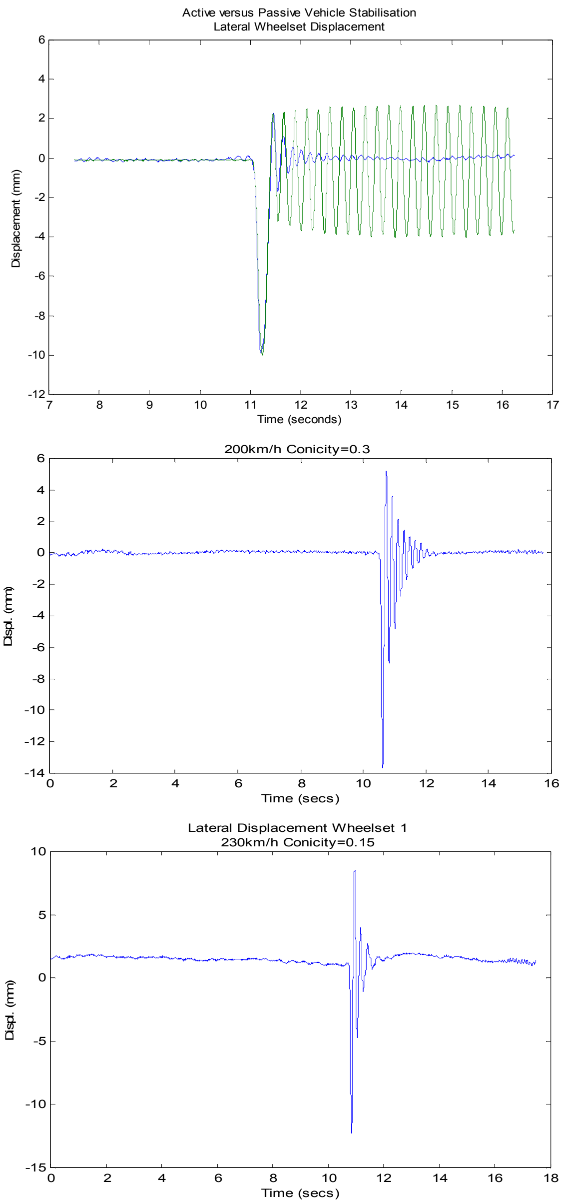

Figure 8 Rig Stability Test Results

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Appendices

Variables

y_{w1}, y_{w2}, y_g, y_v	Lateral displacement of leading, trailing wheelset, bogie frame and vehicle body
θ_{w1}, θ_{w2}, θ_g, θ_v	Yaw displacement of leading, trailing wheelset, bogie frame and vehicle body
V_s	Vehicle travel speed (83.3 *m/s* or 300 *km/hour*)
R_1, R_2	Radius of the curved track at the leading and trailing wheelsets
θ_{c1}, θ_{c2}	Cant angle of the curved track at the leading and trailing wheelsets (typically 6^0)
y_{t1}, y_{t2}	Track lateral displacement (irregularities)
T_{w1}, T_{w2}	Controlled torque for leading and trailing wheelsets respectively

Parameters

r_0, λ	Wheel radius (0.445 *m*) and conicity (0.3) respectively
m_g, I_g	Bogie frame mass (3447 *kg*)and yaw inertia (3200 $kg\ m^2$) respectively
K_s, C_s	Lateral stiffness (511 *kN/m*) and damping per wheelset (37 *kN s/m*) respectively
m_v	Vehicle mass (34,460 *kg*)
K_{sc}, C_{sc}	Secondary lateral stiffness (471 *kN/m*) and damping per wheelset (12 *kN s/m*) respectively
f_1, f_2	Longitudinal and lateral creepage coefficients (10 and 10 *MN*)
g	Gravity (9.8 m/s^2)
Im	Motor moment of inertia (0.00115 kgm^2)
Ra	Motor armature resistance (0.112 Ohm)
La	Motor armature inductance (9.04e-4 H)
Kt	Motor torque constant (0.537 Nm/A)
Kv	Motor back emf constant (0.435 V/rad/s)
Cm	Motor-gearbox shaft Damping (0.0084 Nm/rads)
$Ig1$	Gearbox moment of inertia (motor end) (3.864e-4 kgm^2)
n	Gear ratio (1/87)
Kg	Gearbox drive stiffness (1.131102e6 Nm/rad)
Cg	Gearbox drive damping (7540.7) Nms/rad