INTEGRATED TILT AND ACTIVE LATERAL SECONDARY SUSPENSION CONTROL

Ronghui Zhou *, Argyrios Zolotas * and Roger Goodall *

* *Control Systems Group, Department of Electronic and Electrical Engineering, Loughborough University, Loughborough, LE11 3TU, UK,*
{r.zhou, a.c.zolotas, r.m.goodall}@lboro.ac.uk

ABSTRACT

This paper describes a theoretical study on the integration of tilt and active lateral secondary suspension control issues relate to the system performance requirements, controller assessment approaches, modelling process and dynamics interaction analysis. Two dual-actuator control system configurations with classical decentralized controllers are presented. The work aims to improve the performance of a tilt controller based only upon local vehicle measurements by integrating the lateral active secondary suspension with the roll (tilt actuator). The effectiveness of the integrated control is illustrated via simulations and comparisons with previous modified nulling tilting control as well as the commercial precedence equivalent.

1. INTRODUCTION

Although tilting trains are now mature technologically and widely used in railway services throughout the world, they mainly employ the so-called 'precedence' control approach which is based upon providing tilt command signals from the vehicle in front [1]. There has however been a more rigorous study on original nulling-type tilt control approaches that utilize only local vehicle measurements in [2, 3].

In particular, the work in this paper presents results from current research on integrating lateral and tilt control for tilting train performance enhancement. We study two decentralized control solutions, the lateral actuator control loop and the tilt actuator control loop. For the lateral actuator control loop, there have been a number of suggestions from previous works, i.e. complementary filter, intuitive skyhook damping and nonlinear dual-kalman filter [4]. For the single tilt actuator control loop, a number of suggestions exist in recent literature, i.e. modified nulling-type on vehicle body measurements, local-command driven, model-based H-infinity and fuzzy control solutions, see [2, 5] and references within. These solutions based upon local vehicle measurements alone offer a more simplified framework as well as being more straightforward in terms of failure detection.

The paper discusses two types of decentralized control: a Conventional Dual-Actuator Control (CDAC, skyhook with complementary filters and modified nulling control) and a proposed New Dual-Actuator Control (NDAC) utilizing estimated vehicle body lateral acceleration for the lateral actuator and modified-nulling with true cant deficiency information for the tilt controller. The control input and measurement output selection is performed via Relative Gain Array (RGA) analysis.

The structure of the paper is as follows: Section 2 discusses the model of the tilting train, while Section 3 presents issues with performance specification and assessment. The input-output interaction analysis via RGA is in Section 4, and Section 5 discusses the control systems design issues and results. Conclusions are drawn in Section 6.

2. SYSTEM MODELING

The simplified mechanical configuration of the integration system is shown in Figure 1. Active Anti-Roll Bar (ARB) is utilised to tilt the vehicle body. The concept of active Anti-Roll Bar can be found in [6]. Compared with tilt mechanism, it is more simple, with small weight and low cost. Also, a lateral actuator is installed between the vehicle body and bogie in parallel with the original secondary damper. In this configuration, the actuators for the tilt suspension and lateral suspension can be easily fitted as an optional extra during manufacture. If the actuators lose control, the system can roll back to the non-tilting train with passive suspension system.

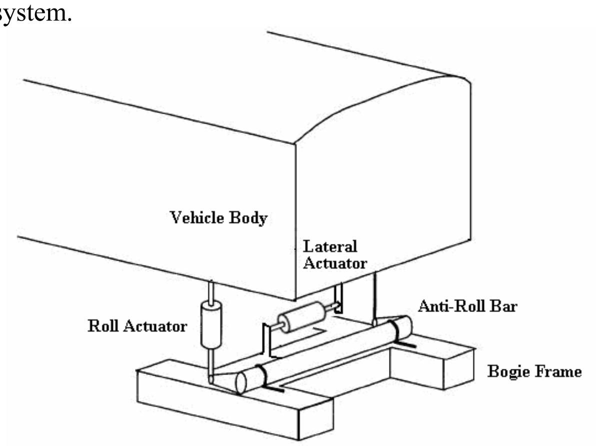


Figure 1: The integration of roll and lateral actuators

System design is based on a four degree-of-freedom end-view model which is illustrated in Figure 2. The lateral and roll degrees of freedom for both the body and the bogie systems are included in this model while the vertical degrees of freedom is ignored, although the effects of the roll stiffness and damping introduced by the vertical suspension are included. A rotational displacement actuator shown by δ_a is included in series with the roll stiffness. Moreover, a lateral actuator shown by F_a is installed in parallel with the original lateral damper between the bogie and body. Both the actuators are assumed to be ideal. For simplicity wheelset dynamics are ignored.

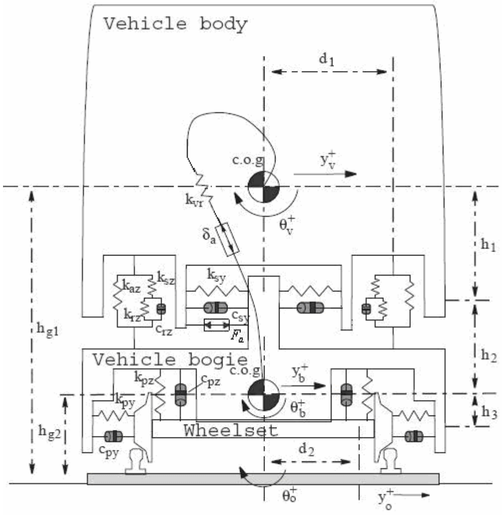


Figure 2: Model of tilting train with lateral actuator

The primary (bogie-wheelsets) lateral, primary vertical and secondary (body-bogie) lateral suspensions are modeled by pairs of parallel spring/damper combinations. A representation of a pair of air-springs is used to model the roll effect of the secondary vertical suspension. Via the Newtonian approach, the four degree-of-freedom end-view model is illustrated in Figure 2. The parameters used in this paper are explained and listed in [2, 3, 7]. The difference here is c_{sy} = 25000 (Ns/m).

Body lateral dynamics:

$$m_v\ddot{y}_v = -2k_{sy}(y_v - h_1\theta_v - y_b - h_2\theta_b) - 2c_{sy}(\dot{y}_v - h_1\dot{\theta}_v - \dot{y}_b - h_2\dot{\theta}_b) - \frac{m_v v^2}{R} + m_v g\theta_0 - m_v h_{g1}\ddot{\theta}_0 + F_a \tag{1}$$

Body roll dynamics:

$$\begin{aligned} i_{vr}\ddot{\theta}_v &= 2h_1k_{sy}(y_v - h_1\theta_v - y_b - h_2\theta_b) \\ &+ 2h_1c_{sy}(\dot{y}_v - h_1\dot{\theta}_v - \dot{y}_b - h_2\dot{\theta}_b) - k_{vr}(\theta_v - \theta_b - \delta_a) - F_a h_1 + m_v g(y_v - y_b) \\ &- 2d_1^2 k_{az}(\theta_v - \theta_b) - 2d_1^2 k_{sz}(\theta_v - \theta_r) - i_{vr}\ddot{\theta}_0 \end{aligned} \tag{2}$$

Bogie lateral dynamics:

$$\begin{aligned} m_b\ddot{y}_b &= 2k_{sy}(y_v - h_1\theta_v - y_b - h_2\theta_b) + 2c_{sy}(\dot{y}_v - h_1\dot{\theta}_v - \dot{y}_b - h_2\dot{\theta}_b) - 2k_{py}(y_b - h_3\theta_b - y_0) - 2k_{py}(\dot{y}_b \\ &- h_3\dot{\theta}_b - \dot{y}_0) - m_b(\frac{v^2}{R} - g\theta_0 + h_{g2}\ddot{\theta}_0) - F_a \end{aligned} \tag{3}$$

Bogie roll dynamics:

$$\begin{aligned} i_{br}\ddot{\theta}_b &= 2h_2k_{sy}(y_v - h_1\theta_v - y_b - h_2\theta_b) + 2h_2c_{sy}(\dot{y}_v - h_1\dot{\theta}_v - \dot{y}_b - h_2\dot{\theta}_b) - 2h_3\{k_{py}(y_b - h_3\theta_b - y_0) \\ &+ c_{py}(\dot{y}_b - h_3\dot{\theta}_b - \dot{y}_0)\} + k_{vr}(\theta_v - \theta_b - \delta_a) \\ &+ 2d_1^2\{k_{az}(\theta_v - \theta_b) + k_{sz}(\theta_v - \theta_r)\} - 2d_2^2(k_{pz}\theta_b \\ &+ c_{pz}\dot{\theta}_b) - i_{br}\ddot{\theta}_0 - F_a h_2 \end{aligned} \tag{4}$$

For the additional air-spring state:

$$\dot{\theta}_r = -\frac{k_{sz} + k_{rz}}{c_{rz}}\theta_r + \frac{k_{sz}}{c_{rz}}\theta_v + \frac{k_{rz}}{c_{rz}}\theta_b + \dot{\theta}_b \tag{5}$$

An 'end-moment' effect: $m_v g(y_v - y_b)$ is included in equation (2) which models the roll effect of the body weight due to the lateral displacement of its centre of gravity (c.o.g). Both the translation and rotation of the reference axes associated with curves are considered in the equations, the body lateral acceleration ($\ddot{y}_v$) used in the NDAC strategy is relative to the track reference while the measured body lateral acceleration ($\ddot{y}_{vm}$) is utilized in the CDAC strategy. The dynamic interactions between the lateral and roll motions are obvious from this model. Further details on the tilting train end-view model can be found in [2, 7].

The vehicle model and control systems are tested on a specified track including both deterministic (low frequency) and stochastic (high frequency) features. The deterministic track used is a curved track with a radius of 1000m and a maximum track cant angle ($\theta_0^{\max}$) of 6°, with a transition at the start and end of the steady curve. The stochastic track inputs represent the irregularities in the track alignment on both straight track and curves, and these are characterised by an approximate spatial spectrum equal to $(2\pi)^2\Omega_l v^2 / f_t$ ($m^2/(cycle/m)$) with a lateral track roughness (Ω_l) of $0.33\times10^{-8}\,m$ [2].

3. SYSTEM PERFORMANCE REQUIREMENT AND CONTROLLER ASSESSMENT APPROACH

3.1 Active suspension design requirement

The general active railway vehicle suspension structure is shown in Figure 3. The active suspension design is a multi-objective optimization process which needs to minimize the body acceleration on the straight track, consider the constraints for suspension deflection and system stability, optimize the curving performance and minimize the actuator power consumption (actuator force).

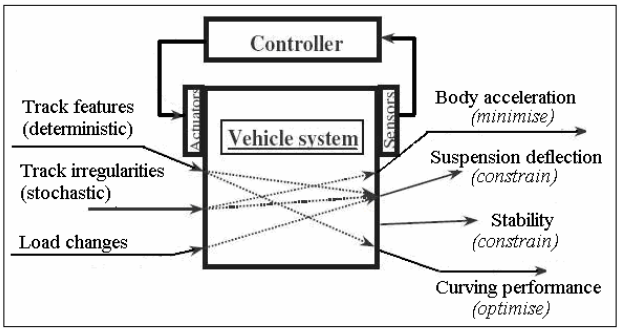


Figure 3: Design requirements for suspension system

3.2 Dual-actuator control system assessment approach

The controller design for the dual-actuator system needs to meet both tilting performance and active lateral suspension requirements [7]. The tilt controller assessment relies upon identifying how a tilting vehicle would ideally perform on the transition from straight to curved track and then quantifying the deviation of the actual response compared with this ideal.

In particular,

Deterministic performance criterion:

- Maintain appropriate curve transition comfort level for standing and seated passengers, it is qualified by P_{ct} value which provides the percentage of (both standing and seated) passengers who feel uncomfortable during the curve transition, and can be calculated with the measured body lateral acceleration, lateral jerk and roll rate.
- Minimize the integral of absolute error between actual measured body lateral acceleration ($\ddot{y}_{bm}^{actual}$) from the dynamics simulation compared with the ideal tilting case ($\ddot{y}_{bm}^{ideal}$).

$$\int |\ddot{y}_{bm}^{actual} - \ddot{y}_{bm}^{ideal}| \tag{6}$$

- Minimize the integral of absolute body roll velocity deviation between measured ($\dot{\theta}_{bm}^{actual}$) and ideal ($\dot{\theta}_{bm}^{ideal}$) responses [7].

$$\int |\dot{\theta}_{bm}^{actual} - \dot{\theta}_{bm}^{ideal}| \tag{7}$$

- Maintain lateral suspension deflection at levels provided by the single tilt actuator [3].

Stochastic performance criterion:

- Straight-track ride quality at no more than 7.5% worst compared to the non-tilting train equivalent at high speed; aim to provide the minimization of ride quality (in terms of passenger lateral acceleration measurement (assessed by its Root Mean Square value (RMS value)) by the lateral actuator.

Control effort:

- Maintain appropriate control forces in the actuators, in particular the lateral actuator force should be no more than 10 kN, for both deterministic and stochastic criteria.

Also four trade-offs exist in the design process:

(1) the design trade-off for the tilting controller is concentrated on the transition performance (P_{ct} value). If the loop bandwidth is low enough not to interfere with the lateral suspension, it is then too slow acting on the curve transition [1]. It is a critical problem for the local nulling tilting control system design.

(2) the additional trade-off for tilting controller performance between the curved track (P_{ct} value) and the straight track responses (RMS value). The tilting train runs at higher speed on the same rail infrastructure compared with non-tilting train which deteriorates the ride-quality on the straight track and introduces a trade-off for tilting controller performance between the curved track and the straight track responses.

(3) the trade-off for the lateral actuator controller between the ride quality on the straight track (RMS value) and curving suspension deflection [4].

(4) the trade-off for the lateral actuator control between the actuator power consumption and overall system performance. Large lateral actuator force improves the curving performance at the expense of higher power consumption.

The integration of active anti-roll bar system and active lateral secondary suspension system can help to improve the first two trade-off relationships. With approximate system configuration, the third trade-off can be optimized. The last trade-off is improved by installing the lateral actuator in parallel with the original damper rather than replacing the damper. However, the non-trivial design case of such multi-objective control problem is still evident.

4. INPUT-OUTPUT INTERACTION ANALYSIS VIA RGA

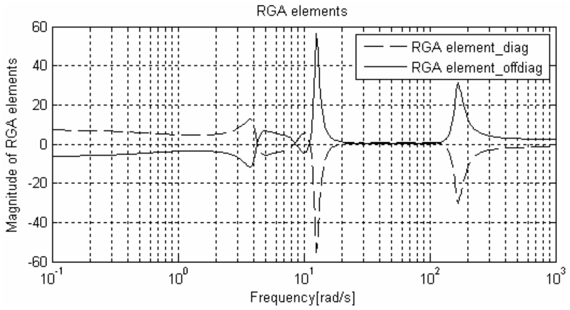
The integration strategy aims to attenuate the dynamics interaction between the vehicle body lateral and roll modes by adding a lateral actuator control loop. The conventional feedback signals for the controllers are the measured lateral body acceleration (for lateral actuator control) and the effective cant deficiency (for tilting control [3]). Further decoupling control can be achieved by choosing appropriate system configuration.

The body lateral acceleration ($\ddot{y}_v$) which is unaffected by the curving response can be utilised as a more effective feedback signal for the lateral actuator control. As described in [3], true cant deficiency ($\theta_d = v^2 / R - (\theta_0 + \theta_v)$) can help the tilting controller. Therefore, the combination of these two signals can significantly improve the dynamics interaction and thus the performance of the local control system.

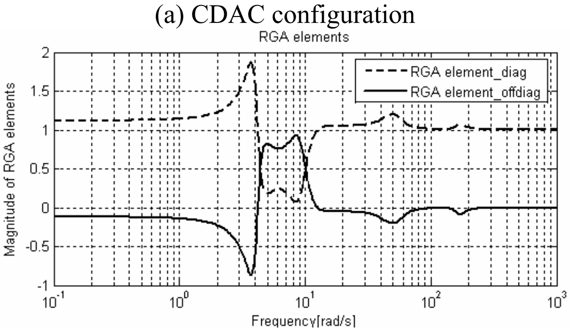
Relative Gain Array (RGA) is utilised to illustrate the efficiency of the I/O configuration strategy on the interaction attenuation. The principles of RGA are detailed in [9], but two rules which are useful for the design process are presented here:

(1) Large RGA elements at frequencies important for control indicate that the plant is fundamentally difficult to control due to strong interactions and sensitivity to uncertainty.

(2) Good pairings are such that the rearranged system, with the selected pairings along the diagonal, has an RGA matrix close to identity at frequencies around the closed-loop bandwidth. If the RGA matrix equal to identity matrix, then the selected input-output pairing can completely decouple the interaction.



(a) CDAC configuration



(b) NDAC configuration

Figure 4: RGA elements

Figure 4 shows the frequency-dependent RGA elements for the two types control system configuration (2×2 system with two actuator inputs and two outputs for the feedback control). The Conventional configuration has the larger RGA element (more than 5) indicating the difficulty of decentralised controller design due to the strong interaction. As shown in Figure 4(a).

The interaction can be significantly attenuated by the NDAC configuration as the RGA matrix approaches the identity matrix particularly at low frequencies (steady-state) and after 10 rad/s. The cut-off frequency of tilting controller in NDAC is designed to be 2.8 rad/s (detailed in Section 5.2) with the diagonal RGA elements around 1.3. The RGA elements are smaller than 1.3 at frequencies below 2.8 rad/s with the steady-state value equal to 1.1, as shown in Figure 4(b).

5. CONTROLLER DESIGN

The control system design based on the conventional system configuration and new combination strategy are introduced in this section:

5.1 Conventional decentralised dual-actuator control (CDAC)

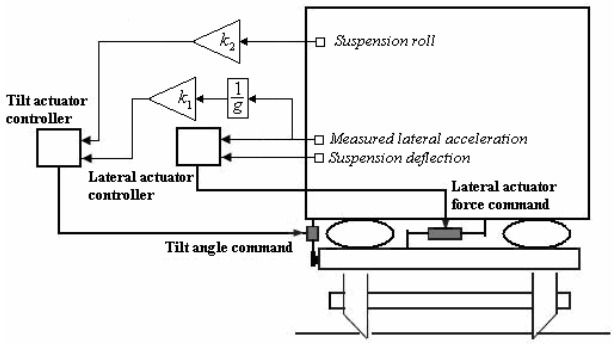


Figure 5: CDAC configuration

Figure 5 is the overall configuration of the Conventional decentralised dual-actuator control (CDAC).The skyhook damping strategy with complementary filter is employed to control the lateral actuator which is a better choice for easing the trade-off between the straight line ride-quality and the suspension deflection on curves compared with intuitive skyhook damping strategy [4]. Lateral accelerometer and displacement sensor are used to provide measured body lateral acceleration and suspension deflection respectively. A pair of complementary second order filters (High pass + Low pass = 1) with flat "Butterworth response" are utilised. The low pass filter (LP) combined with a derivative processes the suspension deflection, plus the high pass filter (HP) combined with an integrator processes measured body lateral acceleration, and together generate the lateral damping command and feed into the skyhook damper coefficient, which in turn feeds into the lateral actuator as the force command. More details about the complementary filters design can be found in [4].

The tilt actuator control loop design is the same as the modified nulling control strategy [3], $\ddot{y}_{vm}$

($\ddot{y}_{vm} = v^2/R - g(\theta_0 - \theta_v) + \ddot{y}_v$) is measured and used to drive the tilt actuator with introducing proportion of secondary suspension roll angle θ_{2sr} ($\theta_{2sr} = \theta_v - \theta_b$) to give partial tilt. k1 and k2 are set to 0.615 and 0.385 respectively for ensuring partial tilt with 60% passenger lateral acceleration compensation which constructs the effective cant deficiency ($-k_1\ddot{y}_{vm}/g - k_2\theta_{2sr}$). The controller structure is a simple PI.

The controller is tuned with Genetic algorithm (NSGA_II) [5, 10] and tested for the specified track with the vehicle speed at 58 m/s. The results are illustrated in Figure 8 and Table 1.

5.2 New decentralised dual-actuator control (NDAC)
Based on the analysis in Section 4, the combination of the true cant deficiency and the body lateral acceleration can significantly attenuate the dynamics interaction. However, measuring true cant deficiency and body lateral acceleration are not a practical solution because these signals related to the track, for which there is no a priori knowledge. Therefore, a Kalman-Bucy filter is employed to estimate these quantities. The overall configuration of the control system combined with the Kalman-Bucy filter is illustrated in Figure 6.

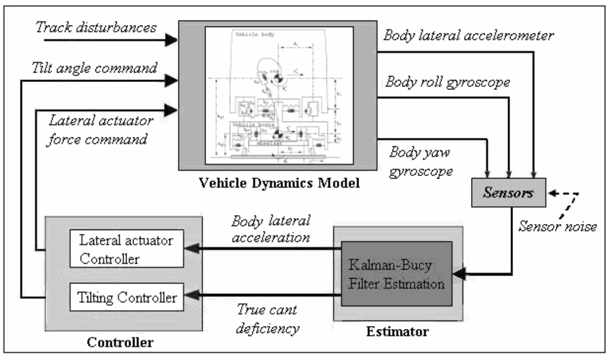
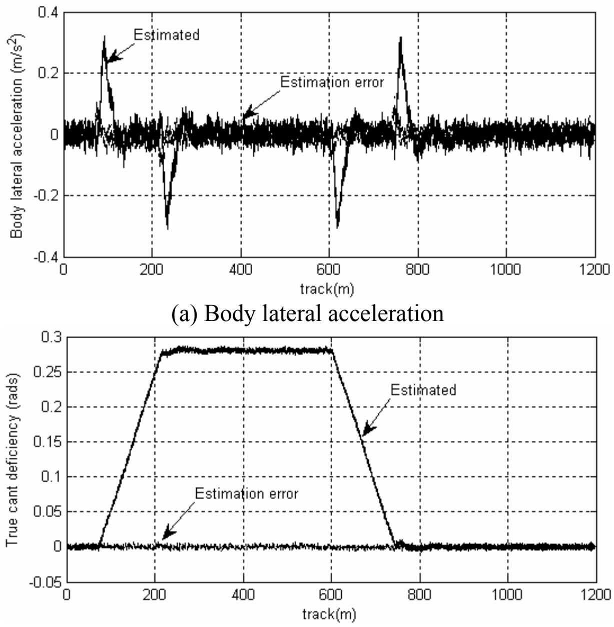


Figure 6: NDAC configuration

The inputs to the Kalman filter are three measurements and two control inputs. It has been found that only three body measurements were necessary for the Kalman filter design [2, 3]: vehicle body roll gyroscope (cant information), body lateral accelerometer (for cant deficiency information) and vehicle body yaw gyroscope (required only for extra information on the curvature $1/R$). The body roll gyroscope measures absolute roll rate ($\dot{\theta}_v + \dot{\theta}_0$). Figure 7 illustrates the estimation results on the curved track.

The intuitive skyhook damping control is utilised for the lateral actuator control,

$$K_{skyhook} = -C_s \times \frac{s^2}{s^2 + 2w_i\xi \cdot s + w_i^2} \times \frac{1}{s} \tag{8}$$



(a) Body lateral acceleration
(b) True cant deficiency
Figure 7: Estimation results

where C_s is the skyhook damping coefficient, w_i is the cut-off frequency with the value $2\pi \times 0.1\ rad/s$, damping ratio $\xi = 0.707$.

Approximate PID controller combined with a first order low pass filter (for attenuating high frequencies due to the derivative portion) is adopted for the tilting controller:

$$K_{PID} = (k_p + \frac{k_i}{s} + \frac{k_d s}{s/N+1}) \times \frac{w_c}{w_c+1} \tag{9}$$

where w_c is the cut-off frequency of the low pass filter with the value $0.25\ Hz$.

The controller parameters can be easily tuned manually and tested in the same track as the CDAC strategy and the results are illustrated in Figure 8 and Table 1.

6. DISCUSSION AND FUTURE WORK

The performance of the CDAC is better than the Modified Nulling Tilting Control (MNTC) [2] with smaller P_{ct} value and RMS value less than the RMS value in the passive situation. The trade-off for tilting control between the curved track performance and straight track ride quality is helped by this dual-actuator decentralized control strategy. However, the improvement of the ride-quality on the straight track is limited due to the skyhook damping increasing the lateral secondary suspension deflection. The curving performance improvement is limited by the lateral actuator force.

Both the P_{ct} value and RMS value of NDAC are slightly reduced compared to the employed commercial precedence control equivalent, which shows the possible benefit obtained by using this new dual-actuator control strategy. Also, this strategy does not increase the lateral suspension deflection. All the trade-offs have been optimized. However, the accuracy of the estimator will affect the overall control system performance.

The next step will test the integration strategy in a full-degree of freedom non-linear vehicle model with varying the track profile. Furthermore, the new dual-actuator control strategy will be implemented in a DSP-based controller.

REFERENCES

[1] R. M. Goodall. Tilting trains and beyond - the future for active railway suspensions. 1. improving passenger comfort. *Computing & Control Engineering Journal,* 10 (4) :153–160, 1999.

[2] A. C. Zolotas. *Advanced control strategies for tilting trains, Thesis (Ph.D.)* - Loughborough University, 2002.

[3] A. C. Zolotas, G. D. Halikias, and R. M. Goodall. A comparison of tilt control approaches for high speed railway vehicles. *In Proc ICSE 2000,* pages 632–636, 2000.

[4] H. Li and R. M. Goodall. Linear and non-linear skyhook damping control laws for active railway suspensions. *Control Engineering Practice,* 7(7): 843–850, 1999.

[5] H. Zamzuri, A. C. Zolotas, and R. M. Goodall. Tilt control design for high-speed trains: a study on multiobjective tuning approaches. *Vehicle System Dynamics,* 46(01): 535–547, 2008.

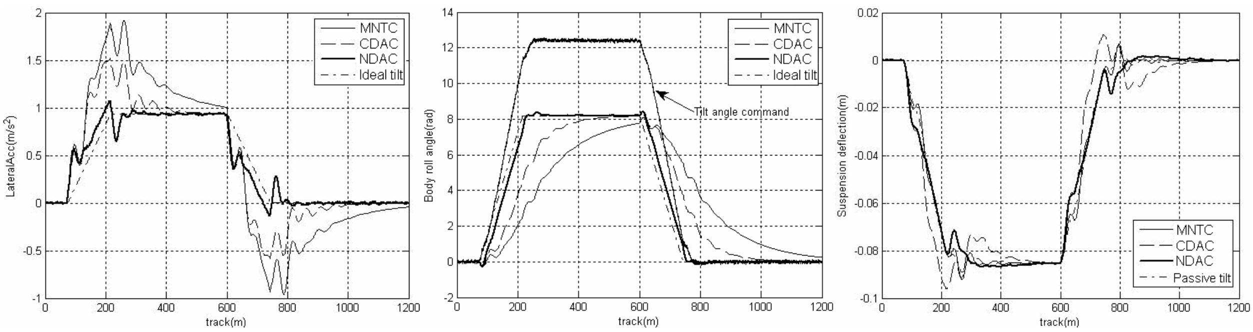
[6] J. Pearson, R. M. Goodall, and I. Pratt. Control system studies of an active anti-roll bar tilt system for railway vehicles. *Journal of Rail and Rapid Transit,* 212:43–60, 1998.

[7] A. C. Zolotas and R. M. Goodall. Modelling and control of railway vehicle suspensions. *Lecture notes in control and information sciences,* (367): 373–412, 2007.

[8] R. M. Goodall. Assessment of the performance of tilt system controllers. *RAILWAY ROLLING STOCK,* 2001 (4):231–239, 2001.

[9] S. Skogestad and Ian Postlethwaite. *Multivariable feedback control: analysis and design, volume 2nd ed.* John Wiley, Chichester, 2005. ISBN 047001167x.

[10] N. Srinivas and Kalyanmoy Deb. Multiobjective optimization using nondominated sorting in genetic algorithms. *Evolutionary computation,* 2:221–248, 1994.



(a) Measured lateral acceleration (b) Body tilt angle (c) Lateral suspension deflection

Figure 8: Time domain simulation results (on the curved track)

Table1: Control system configuration assessment @ 58(m/s)

Deterministic (CURVED TRACK)	CDAC	NDAC	MNTC	CPC
Pct (P-factor) - standing (% of passengers)	59.8	44.9	71.4	47.6
- seated (% of passengers)	17.9	11.9	22.6	13.5
Maximum lateral suspension deflection (mm)	96.1	86.9	89.0	89.0
Stochastic (TRACK IRREGULARITIES)				
Passenger comfort - R.M.S. passive (%g)	3.78	3.78	3.78	3.78
(actual vs ideal) - R.M.S. active (%g)	3.50	3.28	3.99	3.31
- degradation (%)	-7.41	-13.23	5.80	-12.12
Maximum lateral actuator force (N)	9796	4000	n/a	n/a

Notes: MNTC……….The Modified Nulling Tilting Control [2, 7]
CPC…………The Commercial precedence tilting control with passive secondary suspension [2, 3, 7]