ACTIVE STEERING CONTROL FOR RAILWAY BOGIES BASED ON DISPLACEMENT MEASURMENTS

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Abstract: This paper presents an active steering control strategy that improves the performance of rail vehicles on track curves. The control scheme is developed to explore the usually symmetrical arrangement of bogie suspensions such that only simple and practical sensors are needed. The vehicle configuration and requirements of wheelset steering on curves are explained, before the detailed control design is described and the performance is assessed. To examine the robustness of the controller, parameter variations and nonlinearities in the system are considered. Simulation results are given to show that the proposed steering strategy can achieve the desired steering conditions, and significantly reduce wear and noise at the wheel-rail interface and minimise tracking shifting forces.

Keyword: Active steering, Railway bogie, Wheelset, Displacement sensors

1. INTRODUCTION

A railway bogie is a complex mechanical system, which presents a difficult design challenge. For many years, railway engineers have always had to balance carefully conflicting requirements of the high speed stability and low speed curving performance for railway vehicles (Wickens, 1978) which has led to even more complex bogie structures in order to improve this trade-off (Illingworth and Pollard, 1982). More recently, mechatronic approaches using active control have been shown to offer a modern alternative to solve the problem which has not been available previously. The new technology has been shown to be capable of achieving much beyond what is possible with the traditional passive means, because there is a much increased design flexibility and the possibility to supply or absorb energy to or from a controlled object (Kortum et al, 1998). There have been many successful experiments and commercial operations of active controls on railway vehicles, mainly for secondary suspensions and tilting controls (Goodall, 1997).

However, the latest studies have indicated that the most significant gains may be obtained through the use of active control for primary suspensions to control the dynamic behaviour of railway wheelsets (Mei and Goodall, 2003). Functionally, there are two different control tasks of active wheelset control. One is to provide a stability control for the inherently unstable kinematic mode of a wheelset, but it must be done in a manner that the control does not interfere with the natural curving action of the wheelset (Mei and Goodall, 2003). The other task is to steer the wheelset actively and to restore the self curving ability that is hampered by passive suspensions used in railway vehicles (Shen and Goodall, 1997; Perez et al, 2002; Shen et al, 2003). This paper is concerned with the development of a novel control strategy for active steering, with a particular focus on practicalities. In general, the knowledge of wheelset variables such as the angle of attack and wheel-rail lateral deflection is highly desirable for active steering, but a direct measurement of these movements would be very difficult and prohibitively expensive in practice. State estimation techniques can be applied to estimate these signals, which would substantially increase the complexity of the controller. Tackling the system uncertainties is another issue that needs to be addressed, as a lack of accurate knowledge of key parameters related to wheel-rail contact mechanics may lead to poor controller performance and undesired steering actions.

The proposed controller deals with the above design issues, by taking full advantage of the symmetrical mechanical arrangement in the use of suspensions in conventional railway bogies. A control strategy has been presented previously which requires the measurement of suspension forces (Shen et al, 2004), and this paper presents a control scheme which overcomes the problem of worsened performances due to variations or uncertainties of the suspension stiffness when the force measurement is replaced by the measurement of suspension deflections.

2. RAILWAY BOGIE AND MODELLING

A picture of a typical railway bogie is shown in Figure 1. There are two wheelsets connected to a (bogie) frame via suspensions in longitudinal, lateral and vertical directions. The wheelsets are normally of solid-axle type, where two wheels are rigidly mounted onto a common axle. The wheel treads are profiled such that the wheelset follow the track naturally. However the wheelset alone also exhibits a sustained kinematic oscillation in the lateral plane commonly referred as the "*wheelset hunting*". This is overcome on conventional railway vehicles with the use of stiff suspension connections in the horizontal plane, which on the other hand degrade the ability of the wheelset to curve and results in severe wear of the wheels and rails. The longitudinal spacing of the two wheelsets is normally quite small, which is a design feature to improve the performance on curves. However substantial contact forces can still be generated, and hence there is the desire for additional steering control.

Figure 1. A typical railway bogie

The contact forces between the wheel and rail arise from so-called "creepages" between the wheel and rail, these being small relative velocities which occur because of elastic deformation of the steel at the point of contact and which apply in both the longitudinal and the lateral directions. The creepages result from both the actual wheelset motion and the effects of the changing wheel radius at the contact point. The corresponding dynamic motion of the wheelset is a strongly coupled effect occurring in the lateral and yaw directions (Wickens, 2003).

A half-vehicle model is used in the study to develop and assess the steering controller. The model is linearised for the control design as common practice in the study of railway vehicle dynamics, although nonlinearities associated with the wheel-rail profiles are considered in the control assessment. It includes all main motions relevant to the study, as defined in equations 1-7. There are 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. The model is therefore 14th order overall, and is a highly coupled complex MIMO system. Two actuators are located between the wheelsets and the bogie frame to provide separate control of the two wheelsets for the implementation of active steering.
The parameters for the model are given in Appendix and are representative of a modern high-speed railway vehicle.

$$m_w \ddot{y}_{w1} + \left[C_y \left(\dot{y}_{w1} - \dot{y}_g - l_x \dot{\psi}_g \right) + K_y \left(y_{w1} - y_g - l_x \psi_g \right) \right] + \left[\frac{2 f_{22}}{V} \dot{y}_{w1} - f_{22} \psi_{w1} \right] = m_w \left(\frac{V^2}{R_1} - g \theta_{c1} \right) \quad (1)$$

$$I_w \ddot{\psi}_{w1} + \left[C_x l_y^2 \left(\dot{\psi}_{w1} - \dot{\psi}_g - \dot{\left(\frac{l_x}{R_1} \right)} \right) + K_x l_y^2 \left(\psi_{w1} - \psi_g - \frac{l_x}{R_1} \right) \right] + \left[\frac{2 f_{11} l_g^2}{V} \dot{\psi}_{w1} + \frac{2 f_{11} \lambda\, l_g}{r_0} \left(y_{w1} - y_{t1} \right) - \frac{2 f_{11} l_g^2}{R_1} \right] = T_{w1} \quad (2)$$

$$m_w \ddot{y}_{w2} + \left[C_y\left(\dot{y}_{w2} - \dot{y}_g + l_x\dot{\psi}_g\right) + K_y\left(y_{w2} - y_g + l_x\psi_g\right)\right] + \left[\frac{2f_{22}}{V}\dot{y}_{w2} - f_{22}\psi_{w2}\right] = m_w\left(\frac{V^2}{R_2} - g\theta_{c2}\right) \quad (3)$$

$$I_w\ddot{\psi}_{w2} + \left[C_x l_y^{\,2}\left(\dot{\psi}_{w2} - \dot{\psi}_g + \dot{\left(\frac{l_x}{R_2}\right)}\right) + K_x l_y^{\,2}\left(\psi_{w2} - \psi_g + \frac{l_x}{R_2}\right)\right] + \left[\frac{2f_{11}l_g^{\,2}}{V}\dot{\psi}_{w2} + \frac{2f_{11}\lambda\, l_g}{r_0}\left(y_{w2} - y_{t2}\right) - \frac{2f_{11}l_g^{\,2}}{R_2}\right] = T_{w2} \quad (4)$$

$$m_g\ddot{y}_g + \left[C_y\left(2\dot{y}_g - \dot{y}_{w1} - \dot{y}_{w2}\right) + K_y\left(2y_g - y_{w1} - y_{w2}\right)\right] + \left[C_{sc}\left(\dot{y}_g - \dot{y}_v\right) + K_{sc}\left(y_g - y_v\right)\right]$$
$$= m_g\left(\frac{V^2(R_1+R_2)}{2R_1\cdot R_2} - g\frac{\theta_{c1}+\theta_{c2}}{2}\right) \quad (5)$$

$$I_g\ddot{\psi}_g + \left[C_x l_y^{\,2}\left(2\dot{\psi}_g - \dot{\psi}_{w1} - \dot{\psi}_{w2} + \dot{\left(\frac{l_x}{R_1}\right)} - \dot{\left(\frac{l_x}{R_2}\right)}\right) + K_x l_y^{\,2}\left(2\psi_g - \psi_{w1} - \psi_{w2} + \frac{l_x}{R_1} - \frac{l_x}{R_2}\right)\right]$$
$$+ \left[C_y l_x\left(2l_x\dot{\psi}_g - \dot{y}_{w1} + \dot{y}_{w2}\right) + K_y l_x\left(2l_x\psi_g - y_{w1} + y_{w2}\right)\right] = T_{w1} + T_{w2} \quad (6)$$

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

The second set of terms on the left hand side of equations 1 and 3 represents the lateral suspension forces at the leading and trailing wheelsets ($F_{\text{w1\_susp\_y}}$, $F_{\text{w2\_susp\_y}}$) and the third set of terms gives the lateral creep forces at wheel-rail contact ($F_{\text{w1\_creep\_y}}$, $F_{\text{w2\_creep\_y}}$). The second set of terms in equations 2 and 4 is for the yaw torques due to the longitudinal suspensions connecting the two wheelsets to the bogie frame ($T_{\text{w1\_susp\_x}}$, $T_{\text{w2\_susp\_x}}$) and the third set is due to the longitudinal creep forces ($T_{\text{w1\_creep\_x}}$, $T_{\text{w2\_creep\_x}}$). The second and third sets of terms in equation 5 represent the total lateral forces of the primary and secondary suspensions respectively. The second and third terms in equation 6 are the total torques due to the longitudinal and lateral primary suspensions; and the second term in equation 7 is due to the forces of the secondary suspension. The term at the right hand side of equations 1, 3, 5 and 7 correspond to the curving accelerations. Finally, T_{w1} and T_{w2} in equations 2, 4 and 6 are the control inputs at the two wheelsets.

3. CONTROL DESIGN

The overall aim of a steering control is to minimise the creep forces at the wheel-rail contact on curves. The creep forces cause both the wear and noise at the contact surfaces and must be reduced as much as possible, although some creep forces in the lateral direction are necessary in order to produce lateral forces on the curves. A perfect steering condition is specified as (Shen et al, 2003):

- Condition 1 - Equal lateral forces on all wheelsets
- Condition 2 - Zero longitudinal forces on all wheelsets

The first condition has the effect of minimising the tracking shifting force, whereas the second condition attempts to minimise the wear and noise. However those must be achieved without compromising the vehicle stability.

As the steering control is mainly concerned with performance on constant curves, the design can be based on a much simplified model in quasi-steady state only. There will be the same track inputs (curvature, cant etc) at the two wheelsets. An analysis of the bogie model has shown that the steering condition 1 can be achieved by controlling the sum of the two control inputs to obtain a zero difference between the lateral suspension forces at the two wheelsets, and that the second condition can be met if the difference between the two control inputs is controlled to match the difference in torque between the two suspensions in the yaw direction (Shen et al, 2004), i.e:

$T_{w1} + T_{w2}$ ➔ $F_{\text{w1\_susp\_y}} - F_{\text{w2\_susp\_y}} = 0$ (8)

$T_{w1} - T_{w2}$ ➔ $T_{\text{w1\_susp\_x}} - T_{\text{w2\_susp\_x}} = T_{w1} - T_{w2}$ (9)

The control scheme requires a reliable measurement of all primary suspension forces, which is not always desirable for practical reasons. An alternative is to use displacement sensors to measure suspension deflections. An obvious advantage is that the measurement is relatively easy to achieve, because the movement of the primary suspensions is small and typically limited to be less than 10*mm* in both lateral and longitudinal directions. A drawback, however, is that the effectiveness of a part of the controller (as expressed in equation 9) will be closely dependent upon an accurate knowledge of the suspension stiffness for the controller.

The problem can be overcome by exploring the symmetrical bogie configuration, where the suspension stiffness values do not have to be precisely known, but they can be expected to be the same or similar. Based on this observation, the controller can be formed such that the ratio of the two control inputs is kept the same as the ratio of the yaw torques due to the suspensions in the longitudinal/yaw direction and hence the ratio of the suspension deflections. It is also clear that the other part of the controller (as expressed in equation 8) is little affected as a zero difference in force is equivalent to that in suspension deflection. It can then be readily shown that, if the above two control objectives are achieved, the perfect steering conditions will be met. The controller is defined in equations 10 and 11, and a schematic diagram is given in Figure 2. The controller is designed to work on curves. On straight tracks, there will be no control actions required from either of the actuators. A discontinuity offset as shown in Figure 2 is used to avoid possible computation errors (divided by zero) in the control algorithms in this case, and to ensure zero control demands.

$$T_{w1} + T_{w2} = G_i \cdot \int \left(\Delta y_{w1} - \Delta y_{w2} \right) dt \qquad (10)$$

$$T_{w1} \cdot \Delta\psi_{w2} + T_{w2} \cdot \Delta\psi_{w1} = 0 \qquad (11)$$

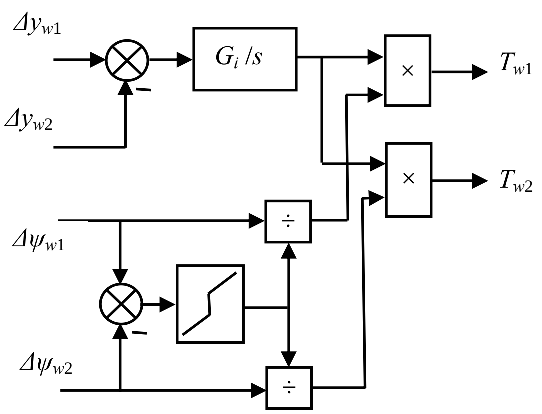

Figure 2. Control structure

4. PERFORMANCE EVALUATION

To evaluate properly the performance of the proposed controller, a number of additional features are added to the model defined in equation 1-7. An active stability control is used in parallel with the active steering controller (Pearson et al, 2004). The bogie would be unstable without the added control as it has very soft primary suspensions and no secondary yaw dampers. Dynamics of two electro-mechanical actuators for implementing the controller are included in the model. Typical nonlinear wheel-rail profiles as shown in Figure 3 are considered using a look-up table which defines the difference in wheel radius as a function of wheel-rail lateral deflection for each wheelset. Furthermore, uncertainties and variations of some key bogie/control parameters such as the creep coefficients, the suspension stiffness and the stiffness of the steering links are defined in Table 1 and included in the assessment.

Table 1. Parameter variations

Parameter	Changes
Creep coefficients	- 50%
Yaw stiffness	+ 80%
Steering linkage stiffness	- 80%

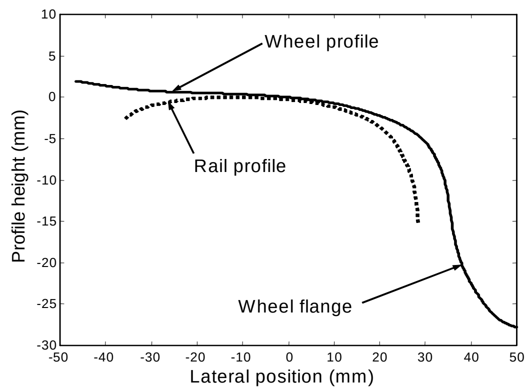

Figure 3. Non-linear Wheel and Rail Profiles

The vehicle performance is assessed at the speed of 50 *m/s* on a track section consisting of:- straight (100*m*) + transition (100*m*) + constant curve (500*m*, radius R=1250*m*) + transition (100*m*) + straight (200*m*). The curved track is canted by 6^0, giving a corresponding curving acceleration of $1 m/s^2$.

Figures 4 and 5 compare the lateral and longitudinal creep forces at the wheel-rail contact between the no steering and active steering cases.

In the lateral direction, the sum of the creep forces of the two wheelsets is the same for both no steering and active steering cases, because it is required to counter-balance the curving force. However if there is no steering condition, this total force is distributed unequally between the two wheelsets. This should be avoided as it increases the maximum force acting on and potentially shifting the track. In the longitudinal direction, it is possible to achieve a zero creep by the means of active steering as shown in Figure 5. Without a proper steering control, creep forces are clearly present resulting in undesirable wear and noise at the wheel-rail interface. It must be emphasised that the creep forces and the uneven distribution of the no steering case would be much higher on a conventional vehicle which is stabilised passively as the primary suspensions would have to much stiffer for high speed operations.

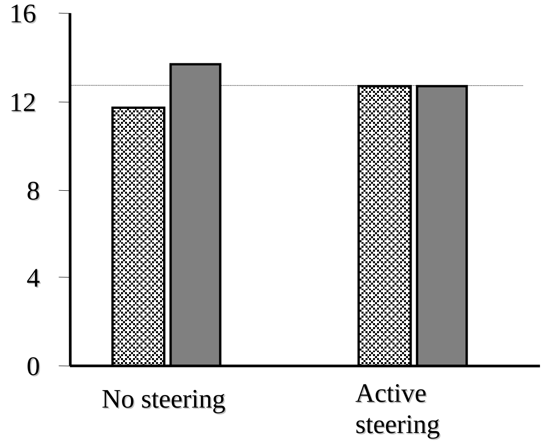

Figure 4. Lateral creep forces at the leading and trailing wheelsets (*kN*)

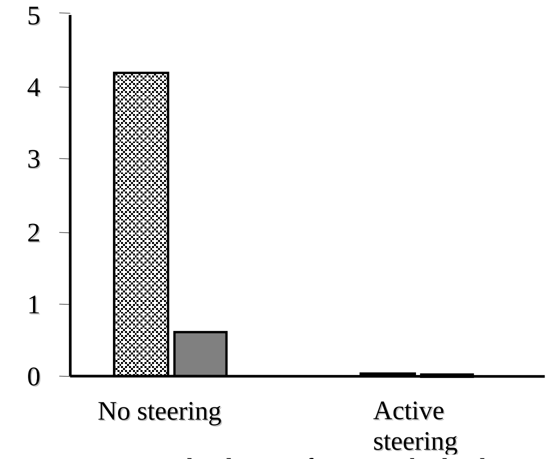

Figure 5. Longitudinal creep forces at the leading and trailing wheelsets (*kN*)

The robustness of the active steering controller against parameter uncertainties and variations is demonstrated in Figures 6 and 7, where a performance index is used in the assessment to indicate relative deviations from the perfect steering requirements (Shen, et al, 2003).

In the nominal condition, the deviations of the lateral and longitudinal creep forces are about 2.6% and 2.2% respectively, which are largely due to the performance on curve transitions. Track transitions are much more difficult to deal with in control design, but in general it is not considered essential to provide additional steering control for further improvements because they are short in length and travelling time.

When parameter variations or non-linear wheel-rail profiles are considered, the change in the lateral creep force is less than 0.5% from that in the nominal case. In the longitudinal direction, there is a larger increase in creep and creep forces in some conditions which is up to a further 2% from that in the nominal condition. Nevertheless, the creep forces in both directions are still significantly lower than those without the active steering control.

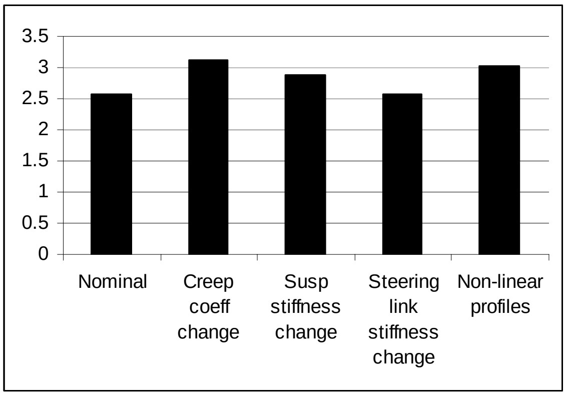

Figure 6. Relative deviations of lateral creep force (rms, in % of curving force)

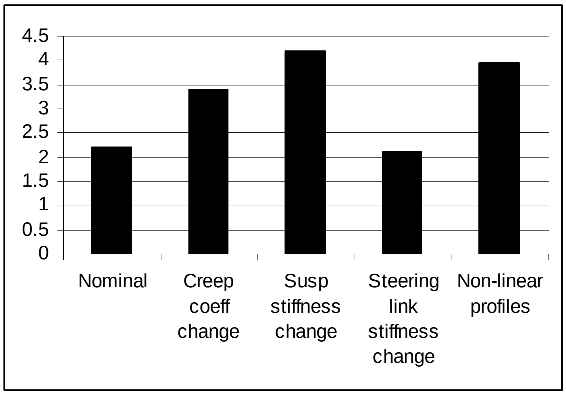

Figure 7. Relative deviations of longitudinal creep force (rms, in % of curving force)

5. CONCLUSIONS

The paper has proposed a novel control strategy and shown the controller is able to steer a railway bogie around curved tracks in a manner that not only reduce significantly the creep forces (hence the wear and noise) at the wheel-rail interface, but also minimise the track shifting forces on curves by evenly distributing the lateral creep forces between the wheelsets to provide the curving force.

The control scheme requires only the measurement of primary suspension deflections which have a maximum travel of less than 10mm. By taking advantage of the symmetrical mechanical arrangements of the suspensions, the controller has been shown to achieve near perfect steering performance even when there are large variations of key parameters in the system and/or real non-linear wheel-rail profiles are considered.

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APPENDIX. SYMBOLS AND PARAMETERS

C_x,	Longitudinal damping of primary suspension
C_x,	Lateral damping of primary suspension
C_{sc}	Lateral damping of secondary suspension
f_{11}	Longitudinal creep coefficient (10 *MN*)
f_{22}	Lateral creep coefficient (10 *MN*)
g	Gravity constant (9.81 m/s^2)
I_w	Wheelset yaw inertia (700 $kg\ m^2$)
I_g	Bogie yaw inertia (700 $kg\ m^2$)
K_x	Longitudinal stiffness of primary suspension
K_y	Lateral stiffness of primary suspension
K_{sc}	Lateral stiffness of secondary suspension
l_g	Half gauge (0.75 *m*)
l_x	Half wheel-base
l_y	Half space of longitudinal suspensions
m_g	Bogie mass (3447 *kg*)
m_v	Body mass (34,460 *kg*)
m_w	Wheelset mass (1250 *kg*)
R	Track curve radius
R_1, R_2	Track curve radius at the two wheelsets
r_0	Nominal wheel radius (0.45 *m*)
T_{w1}, T_{w2}	Control torque at the two wheelsets
V	Vehicle travel speed
y_{t1}, y_{t2}	Lateral track displacements at the two wheelsets
y_g	Bogie lateral displacement
y_v	Body lateral displacement
y_{w1} ,y_{w2}	Lateral displacements of the two wheelsets
θ_{c1}, θ_{c2}	Track cant angles at the two wheelsets
ψ_g	Yaw angle of the bogie
ψ_{w1}, ψ_{w2}	Attack angles of the two wheelsets
λ	Conicity at the wheel-rail contact