A NOVEL CONTROL STRATEGY FOR ACTIVE STEERING OF RAILWAY BOGIES

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Abstract

This paper presents a novel active steering strategy for railway bogies to improve performances on track curves. The vehicle configuration and requirements of wheelset steering on curves are explained. It then presents the design of a controller to control the attack angles and lateral deflections of the wheelsets such that the desired steering conditions are met. The controller is based on the measurement of forces or deflections of the primary suspensions. To assess the robustness of the controller, parameter variations and nonlinearities in the system are considered. The simulation results are given to show that the proposed steering strategy can significantly improve the vehicle performance on curves.

1 Introduction

A trade-off between the high speed stability and low speed curving for railway vehicles has been a difficult challenge for many years [1]. There have been many investigations on possible mechanical solutions such as cross-bracing, body steering and primary yaw damping, to improve this trade-off [2]. However the contradiction between stability and curving remains. More recent studies have indicated that active control technology provides a solution for this problem [3], as it offers a much greater design flexibility that is not possible with pure mechanical approaches. A number of active steering strategies have been studied, including the concept of yaw relaxation [4], the control of yaw angles between the bogie and the wheelsets [5], and the control of lateral position and/or the yaw moment of the wheelsets [6]. An assessment of various control strategies is given in [7].

However, there are a number of practical issues associated with the design of real active steering controllers. Some active strategies require feedback signals such as the wheelset angle of attack and the wheel-rail lateral deflections, but a direct measurement of these signals would be very difficult and prohibitively expensive in practice. State estimation techniques can be applied to estimate these signals, but these 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 such as creep coefficients and wheelset conicity may lead to poor controller performance and undesired steering actions.

This paper presents a practical solution for the active steering of bogie vehicles. 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. The control strategy is robust against parameter variations, and it requires only the measurement of suspension forces, which can alternatively be easily replaced by measuring suspension deflections.

2 Modelling and Control Requirements

2.1 Vehicle Configuration

Figure 1 shows the arrangement of wheelsets and actuators within a railway vehicle bogie. Only one bogie and half the vehicle body are shown for clarity.

Figure 1. Plan-view diagram of a half rail vehicle

Two wheelsets are mounted onto a common (bogie) frame via primary suspension connections and the bogie is connected to the body via secondary suspensions. No longitudinal springs or dampers in the secondary suspensions are shown, although...
in practice some form of longitudinal connection is needed to transmit traction and braking forces from the bogie to the vehicle body. The inclusion of the bogie between the body and the wheelsets is justified in passive vehicles for two main reasons. One is that it reduces the distance between the wheelsets, and this in turn reduces the misalignment between the wheelset and the track on curves. The other reason is that it de-couples the problems of achieving adequate running stability and providing ride comfort, which depend on the primary and the secondary suspensions respectively. Two actuators are located between wheelsets and the bogie frame to provide separate control of the two wheelsets for the implementation of active steering.

2.2 Modelling

The stability of a rail vehicle and the performance on curved tracks are dominated by contact forces at the wheel-rail interface, the characteristics of which are complex and vary with external conditions. The models used by railway engineers in the vehicle design and verification are highly non-linear and complex. However, measures for linearisation and model reduction are applied in practice and simplified models are usually used in the control design, the more complex non-linear models being used for the dynamic analysis, although it is normal to test the controller using the full complexity model once developed. This study uses a design model based upon a linearised plan-view half-vehicle model [8, 9]; it includes the lateral and yaw motions of the wheelsets and of the bogie frame, but only and the lateral motion of the body. The half-vehicle model is justifiable because the interactions between the two bogies of a vehicle are relatively weak due to the large mass of the body frame. A state-space form can be readily derived as shown in equation 1, the detail of which can be found in [5]. The model is largely valid as far as there is no severe flange contact between the wheel and rail. Note that all state variables are related to the local track references and all symbols and parameters are given in the Appendix. The dynamics of the two electro-mechanical actuators are also modelled.

\[ \dot{x} = Ax + Bu + Pw \] (1)

where the state vector is defined as:
\[ x = [y_l, y_r, y_{gb}, \phi_l, \phi_r, \psi_{gb}, y_{fl}, y_{fr}, \phi_{fl}, \phi_{fr}]^T \]

the control input vector is:
\[ u = [\theta_l, \theta_r]^T \]

and the disturbance vector is:
\[ w = [\theta_l, \theta_r, 1/R_l, 1/R_r, 1/R_{fl}, 1/R_{fr}, y_{fl}, y_{fr}]^T \]

2.3 Steering Requirements

The requirements for active steering control are very demanding. The overall aim of a steering control is to minimise the creep forces at the wheel-rail contact. The contact forces (or creep forces) are caused by so-called “creepages” between the wheel and rail surfaces which are small relative velocities resulted from elastic deformation of the steel at the point of contact. The creep forces cause both the wear and noises at the contact surfaces and must be reduced as much as possible, but this must be achieved without compromising the vehicle stability. Also some creep forces in the lateral direction are desirable in order to counter-balance the centrifugal force on curves. A perfect steering condition is specified as [7]:

- Equal lateral forces on all wheelsets
- Zero longitudinal forces on all wheelsets (or equal force for the two wheels of each wheelset in the presence of traction or braking)

The first condition has an effect of minimising the tracking shifting force, whereas the second condition attempts to minimise the wear and noise.

3 Control Design

In general, feedforward control strategies offer the advantage of not interfering with the stability of the vehicle, but are sensitive to the system uncertainties such as parameter inaccuracy and/or model deficiencies. Conversely, appropriate feedback strategies are able to deal with the uncertainties and to deliver more robust performances, but care must be taken in the controller design to avoid any effect upon the vehicle stability. A feedback strategy is developed and presented in this study.

The steering strategy is based on the observation that conventionally there is a symmetrical mechanical arrangement for primary suspensions and the wheelsets: therefore, if the two wheelsets in a bogie can be steered to behave the same on curves, then both the longitudinal and lateral creep forces (in magnitude and direction) between the wheelsets would be equalised. The combination of the two equalised conditions leads naturally to zero longitudinal creep forces as there would be no other forces to counter-balance the total torque were they not zero.

To carry out the proposed control design, let \( y^- = [y_l, y_r] \), \( \phi^- = \phi_l - \phi_r \), \( T = T_l - T_r \), \( y^+ = [y_{fl}, y_{fr}] \), \( \phi^+ = \phi_{fl} + \phi_{fr} \), and \( T^+ = T_{fl} + T_{fr} \). It is also assumed that the two wheelsets are on a constant curve and no track irregularity is considered. From the half vehicle model, the following expressions can be derived.

\[ m_w \dot{y}^- = -f_{12} \left( \frac{\dot{y}^-}{v_s} - \phi^- \right) + (F_{gl} - F_{gl}^-) \] (2)

\[ I_w \ddot{\phi}^- = -f_{11} \left( \frac{\dot{y}^-}{v_0} \lambda - \frac{\dot{\phi}^-}{v_s} I_g \right) + (T_{gl} - T_{gl}^-) + T^- \] (3)

\[ I_g \ddot{\phi}_g^- = I_x (F_{gl} + F_{gl}^-) - (T_{gl} + T_{gl}^-) - T^+ \] (4)

\[ m_w \dot{y}^+ + m_{gb} \dot{y}_{gb} + m_{fl} \dot{y}_{fl} = -f_{12} \left( \frac{\dot{y}^+}{v_s} - \phi^+ \right) + MC_d \] (5)
The latter can be a more feasible alternative as the primary suspensions are in general stiff and the movements are small and are typically restricted to be within the range of 8-10mm. There are certain shortcomings related to the deflection measurement in particular the reliance of the controller on the knowledge of the suspension stiffness, but those will not be the focus of this study.

**4 Simulation Results**

Two different vehicle speeds and corresponding track conditions are used in the performance assessment. The first is the vehicle speed of 50m/s. A curved track with radius of 1250m connected to straight tracks on either end via a transition of 100m is selected to study the control performance on curves. The curved track is canted inwards by 6 degrees to reduce the lateral acceleration experienced by the passengers (a normal feature of railway track). To ensure desired steering performance at different operation speeds, the controller is also assessed at a lower speed of the vehicle (25m/s) on a tighter curved track (R=310m) where the steering control is more critical. The length of the track transitions is 50m and the cant angle is again 6 degrees.

In nominal conditions, where all key vehicle parameters and wheel-rail contact conditions are known, the steering controller has shown to be effective in delivering the control objectives defined in section 2.

**4.1 Nominal Condition**

Figures 3 and 4 give the lateral and longitudinal creep forces at the wheel-rail contact point(s) for the 'nominal case' at the speed of 50m/s, where the solid lines represent the results for the leading wheelset and the dotted lines for the trailing wheelset. The figures also compare the results with different control gains, where the gain $k_i^{-}$ has different quantities of 0, 0.5, and 4.

The simulation results indicate clearly that when appropriate control gains are used, the perfect steering condition is achieved on constant curves – the longitudinal creep forces at the both leading and trailing wheelsets are zero, and the lateral creep forces at the wheelsets are equal each.
contributing towards half of the curving force. Note that, if the steering actions are not adequate (e.g. see the case of \(k_i^r=0\) in Figures 3 and 4), not only do the longitudinal creep forces increase (even though they may be equalised as indicated in Figure 4), but also the track shifting forces are unbalanced between the two wheelsets. The use of an adequate gain \(k_i^r\) is essential to ensure a desirable steering performance of the closed-loop system.

![Figure 3. Lateral creep forces at the wheelset-rail contact](image)

![Figure 4. Longitudinal creep forces at the wheel-rail contact](image)

4.2 Performance Robustness

Railway vehicles are subject to large parameter variations and non-linearities [10]. The creep coefficient (the ratio between the creep force and creep) is dependent on the contact conditions and the normal load on the wheels; the conicity (the slope of a wheel at the contact point with rail) varies with the lateral displacement of the wheelset (in practice non-linear profiles are used for the wheel tread and rail surface). Also it is very difficult to get accurate values for other vehicle parameters such as stiffness constants of the suspensions and steering linkages. It is therefore essential that the proposed controller should remain effective and deliver a robust performance where and when such adverse changes occur.

In this study, a reduction of 50% in creep coefficient, an increase of 80% in the yaw stiffness of the suspension and in the linkage stiffness are considered. Also included in the assessment is a standard pair of non-linear wheel/rail profiles used by the railway industry [11].

To provide a quantitative evaluation of the steering performance in different conditions, two performance indices are used to measure the average deviations of the creep forces from those expected in the perfect steering condition [11]. Equations (9) and (10) define in terms of root-mean-square values the level of undesirable creep forces of the leading wheelset in the longitudinal and lateral directions (in the latter case compared with the balanced quasi-static curving force. Similar equations can be derived for the trailing wheelset, and the total index of the bogie in either direction is the root mean square of those from the two wheelsets. A larger lateral index would indicate that the lateral creep forces of leading and trailing wheelsets are dynamically and/or statically unbalanced. Similarly, a large longitudinal index indicates a poor steering performance as the longitudinal creep force is increased. A zero value of the indices would be ideal, but in practice this is not possible particularly due to transient responses.

\[
\bar{F}_{dl} = -\frac{T_s}{T_e} \frac{1}{\sup_{t \in [0,T_e]} F_d(t)} \left( \frac{1}{T_s} \int_{t}^{t+T_s} \left| F_d(t) \right|^2 dt \right)^{1/2}
\]

(9)

\[
\bar{F}_{df} = -\frac{T_s}{T_e} \frac{1}{\sup_{t \in [0,T_e]} |F_d(t)|} \left( \frac{1}{T_s} \int_{t}^{t+T_s} \left| F_d(t) - F_d(t) \right|^2 dt \right)^{1/2}
\]

(10)

Table 1 gives a comparison of the performance indices. Compared with those in the nominal condition, there are no significant increases when the parameters are different from the design values and there is even a reduction of the creep forces in some cases. The differences are mainly due to increased/reduced creep forces on curve transitions. On the constant curve the perfect curving condition is largely maintained. The controller is less robust when the non-linear wheel-rail profiles are taken into account with noticeable increases in both longitudinal and lateral creep indices, which are more than doubled at the lower speed of 25m/s.

<table>
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<th>(v_s=25\text{m/s})</th>
<th>(v_s=50\text{m/s})</th>
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<td></td>
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<td>reduced by 50%</td>
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<td>reduced by 80%</td>
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Table 1. Performance indices of longitudinal and lateral creep forces (in percentage)
5 Conclusions and Further Work

The paper has shown that the proposed control strategy is able to steer a railway bogie around curved tracks in a manner that minimises not only the creep forces (hence the wear and noises) at the wheel-rail interface, but also the lateral creep forces necessary for providing the curving force are evenly distributed between the wheelsets.

The control scheme requires only the measurement of forces in the primary suspensions, which can also be replaced by the measurement of suspension movements. By taking advantage of the symmetrical mechanical arrangements of the suspensions, the controller has been shown to deliver almost perfect steering performance even when large variations of key parameters in the system occur although the performance is less robust for non-linear profiled wheel and rail.

Acknowledgement

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References


Nomenclature

\[ b_{px}, b_{py} \] longitudinal, lateral damping ratios of the primary suspension, per wheelset (14, 24 \( kNs/m \))
\[ b_y \] lateral damping of the secondary suspension, per bogie (40 \( kNs/m \))
\[ B_{px}, B_{py} \] equivalent yaw damping of primary suspensions at per wheelset, at the bogie
\[ C_d \] cant deficiency
\[ F_{g1}, F_{g2} \] Forces due to primary suspensions at the leading and trailing wheelsets.
\[ F_{d1}, F_{d2} \] lateral creep forces at leading, trailing wheelsets.
\[ f_{11}, f_{22} \] longitudinal and lateral creep coefficients (10 MN)
\[ g \] gravity constant (9.8 m/s^2)
\[ I_{g}, I_{b} \] wheelset, bogie yaw inertia (766, 3200 kg/m^2)
\[ k_{px}, k_{py} \] longitudinal lateral stiffness of the primary suspension, per wheelset (1.9, 9.4 MN/m)
\[ k_y \] lateral stiffness of the secondary suspension, per bogie (0.49 MN/m)
\[ l_x, l_y \] half wheelbase and half spacing of the steering links (1.225, 1.0 m)
\[ l_{g} \] half gauge of wheelset (0.75 m)
\[ m_w, m_g, m_b \] wheelset, bogie, half vehicle mass (1376, 3477, 17230 kg)
\[ r_0 \] Wheel radius (0.45)
\[ R \] radius of the curved track.
\[ R_y, R_z \] radius of the curved track at leading and trailing wheelsets.
\[ T_{g1}, T_{g2} \] Yaw torque due to primary suspensions at the leading and trailing wheelsets.
\[ T_a, T_t \] control torque for leading and trailing wheelsets respectively.
\[ T, T^* \] difference and sum of the control torques.
\[ T_{c1}^{crp}, T_{c2}^{crp} \] creep torque at leading and trailing wheelsets vehicle travel speed.
\[ v \] wheel-rail deflections at the leading and trailing wheelsets.
\[ y_{rl}, y_{rt} \] track lateral irregularity at the leading and trailing wheelsets.
\[ y_{gl}, y_{gt} \] lateral displacement of the bogie, half body.
\[ y^*, y^+ \] difference and sum of the lateral displacements in lateral and yaw modes respectively.
\[ \lambda \] Wheel conicity at the contact point (nominal value 0.2, range 0.05-0.4)
\[ \phi, \phi^*, \phi^+ \] attack angles of leading and trailing wheelsets.
\[ l_{bl} \] yaw angle of the bogie.
\[ \theta, \theta^* \] cant angle of the track.
\[ \theta_1, \theta_2 \] cant angle of the curved track at the leading and trailing wheelsets.