Railway bogie stability control from secondary yaw actuators

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ABSTRACT: The idea of active control based upon applying a controllable yaw torque between the body and the two bogies has been studied previously, mainly to try and provide enhanced curving capability. This paper extends the idea by examining the opportunities for using secondary yaw actuators to stabilise a bogie having very soft yaw stiffness between the bogie frame and the wheelsets, the objective being to take advantage of the good curving performance offered by the soft primary yaw stiffness.

1 INTRODUCTION

Secondary yaw dampers are often fitted to many passenger vehicles; they provide additional damping to the bogie kinematic modes such that lower yaw stiffness between the bogie frame and the wheelsets (i.e. primary yaw stiffness, PYS) can be used which provides better curving performance. The idea of replacing these with active elements in order to apply a controllable yaw torque between the body and the two bogies has been studied both theoretically and experimentally, mainly to try and provide enhanced curving capability (Diana et al. 2002, Matsumoto et al. 2009, Simson and Cole 2011]. The advantage is that in this location it is relatively straightforward to replace dampers with actuators, but also they are in a more favourable, low-vibration environment. Figure 1 shows a typical installation and Figure 2 illustrates an active device in position.

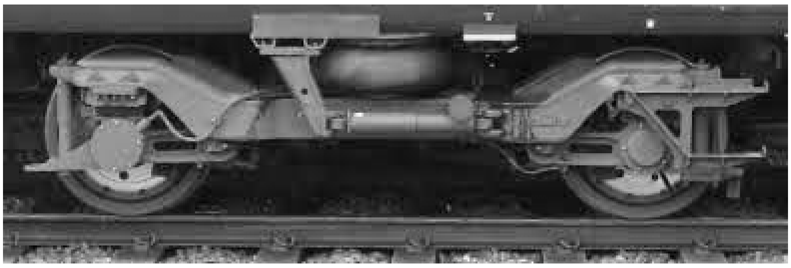

Figure 1 A secondary yaw damper on a bogie

This paper extends the idea by examining active stabilisation strategies for a bogie with very soft primary yaw stiffness (PYS) between the bogie frame and the wheelsets. The low PYS means that curving will intrinsically be good, but the bogie will be unstable during operation, hence the use of active control to provide stability. A plan-view, half-vehicle dynamic model is described that is sufficient to represent the key dynamic characteristics; this is suitably representative but is also not overly complicated so that it can be used to develop stability control strategies.

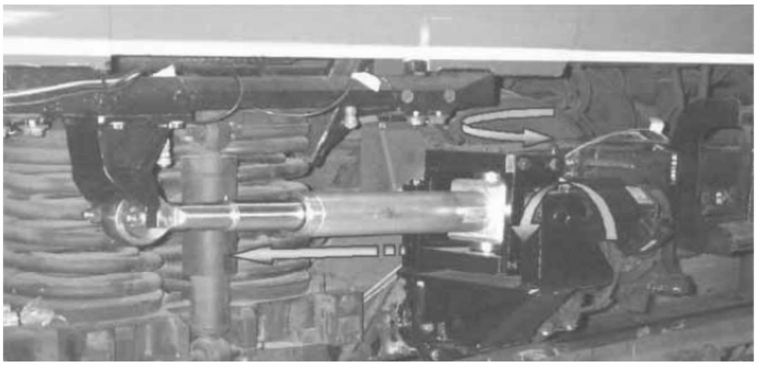

Figure 2 Active yaw damper (Braghin et al 2006)

The research was aimed towards providing step-change improvements in curving performance, and consistent with this aim the actively-controlled configuration studied has used a “target” PYS value which is 10% of baseline PYS assumed for the passive vehicle. This value is typically what will arise due to the shearing effect of the primary vertical suspension. In addition to the simplified vehicle model that has been used to design the control strategies, the performance has been evaluated using a complex Multi-Body System (MBS) model.

2 VEHICLE MODELLING

2.1 *Dynamic model*

The vehicle model used for control design and development is shown in Figure 3. It represents the dynamics of a half-vehicle and consists of two wheelsets, one bogie and one half car body. Given that the focus of this work is on vehicle stability, the model is restricted to consider the motion of the vehicle in the horizontal plane. Two degrees of freedom are introduced, the lateral displacement and yaw rotation (except for the half car body for which only lateral is modelled).

The primary suspension consists of linear springs and dampers connecting the wheelsets and the bogie frame. The secondary suspension is also modelled by means of linear springs and dampers. For the passive vehicle two yaw dampers are placed symmetrically on the two sides of the bogie, and these are replaced by actuators for the active system. The wheels are assumed to have conical shape, and different conicity values have been considered in the range 0.10-0.25.

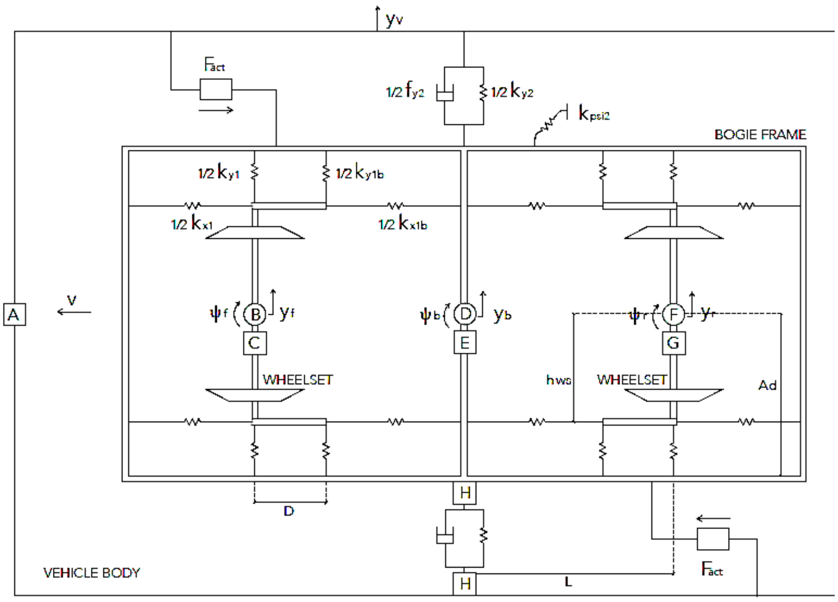

Figure 3 Plan-view half-vehicle dynamic model, including sensor positions

2.2 *Assessment approach*

Three types of track input have been used as part of the study:

- a 10mm lateral step to provide an initial assessment
- a spatial realization of the ORE/ERRI ‘low level’ power spectrum to provide representative lateral irregularities
- a high speed curve example, chosen for initial assessment of curving performance to have a transition length of 100 m, full curve length of 450 m, curve radius of 500 m and cant of 150 mm. At a speed of 31.5m/s this gives a cant deficiency acceleration of 1.0 m/s^2.

A variety of tests have been undertaken both to check the linearized plan-view model and to provide a performance benchmark. Figure 4 is an example showing the yaw rate response of the front wheelset for a 10mm lateral step input on the track with the PYS set to 50% and 10% of the baseline value (left and right graphs respectively).

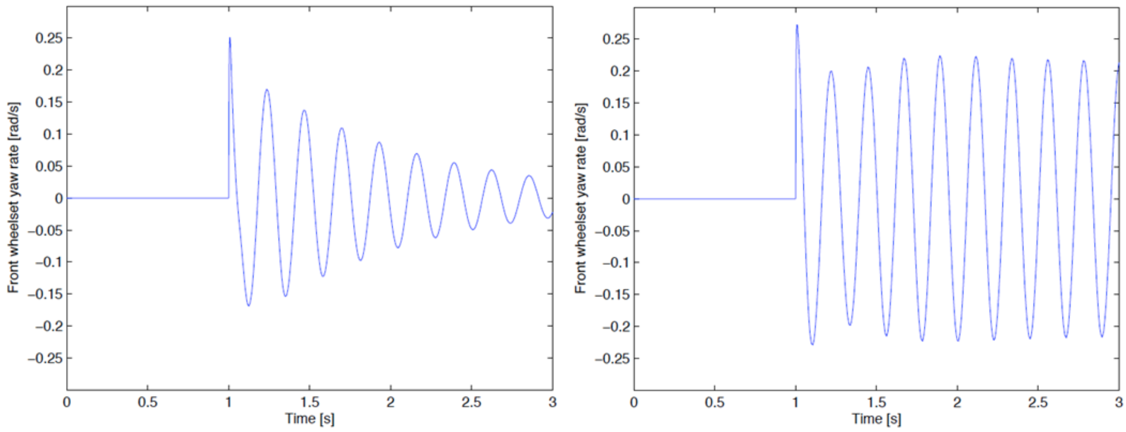

Figure 4 Yaw rate step response for front wheelset, 50% and 10% PYS

Table 1 quantifies the stability by listing the lowest damping ratio under a variety of conditions. It can be seen that, using 100% PYS (i.e. the normal passive value), for speeds up to 50 m/s with the higher conicity of 0.25 a reasonable level of stability is achieved. However as soon as the PYS is reduced, firstly to 50% and then to the target value of 10%, the dynamic response becomes unstable, even at the lower speed of 40m/s.

Table 1 Lowest damping ratios for passive vehicle

Speed	Conicity	PYS	Damping ratio
40 m/s	0.15	100%	18%
50 m/s	0.15	100%	14%
40 m/s	0.25	100%	12%
50 m/s	0.25	100%	9%
40 m/s	0.25	50%	4%
40 m/s	0.25	10%	(Unstable)

The linearized model is not only used to provide an initial assessment of stability and straight track performance, but also as the design model required for controller development. As explained before, a full MBS-based model is used to provide a more accurate simulation model that can be used for performance assessment.

3 CONTROL STRATEGIES

The objective of the control law is to enable a large reduction of the PYS for an actively-controlled vehicle, while preserving the same running behaviour on straight track and the same (or higher) critical speed as a passive vehicle that has higher PYS. To this aim, different control strategies are considered. Strategies based upon “classical” type approaches were investigated, but the complexity of the control loop meant that design was difficult using these simpler methods. This paper therefore considers linear quadratic (LQ) optimal control in which full state feedback is assumed in order to establish a baseline for performance improvement, followed by an LQG strategy in which a state estimator is included so as to allow for sensing practicalities.

3.1 *Control with full state feedback (LQR)*

In the first instance, full-state feedback control is assumed for the linear system described in Section 2, and the performance index is defined as a weighted integral of the state and input values. Weight tuning was performed to ensure stability requirements to be met by the active vehicle with reduced primary yaw stiffness, whilst also minimising actuation requirements. This was achieved by setting initial weights as the inverse square of the state variables' expected values, taken from a set of simulations performed on the passive vehicle with nominal primary yaw stiffness. For the actuator force, an expected value was also obtained from simulations performed on the nominal passive vehicle, based upon the maximum force generated by the yaw damper. A second stage of tuning was then performed empirically to achieve a good design tradeoff for the final behaviour of the LQ regulator.

The final tuning produced the gain matrix that is given below:

K_{rn} . [1.02; 0.32; 0.62; 0.14; 1.14; 3.65; 0.56; 0.7; 0.45; 0.71; 0.37; 0.54; 0.86; 0.1], where K_{rn} has a value of 6.078 kN/unit. With this setting the maximum force exerted by the actuator is 7.6 kN, which is consistent with the level of force required for other active suspension solutions.

An interesting result was that one of the tuned weightings had a value notably higher than all the others, this being associated to the bogie yaw rate (i.e. the element in column 6 of the vector), which points out that the bogie yaw rate is the most important component used in the feedback; intuitively this is correct because in the passive situation the anti-yaw damper provides stability by working only on the bogie yaw rate. The tuning also resulted in low weighting values for the lateral displacements for the wheelsets and the body. Figure 5 compares the lateral displacements of the active solution with 10% PYS with the baseline 100% PYS passive configuration, and shows that a similar dynamic performance is achieved.

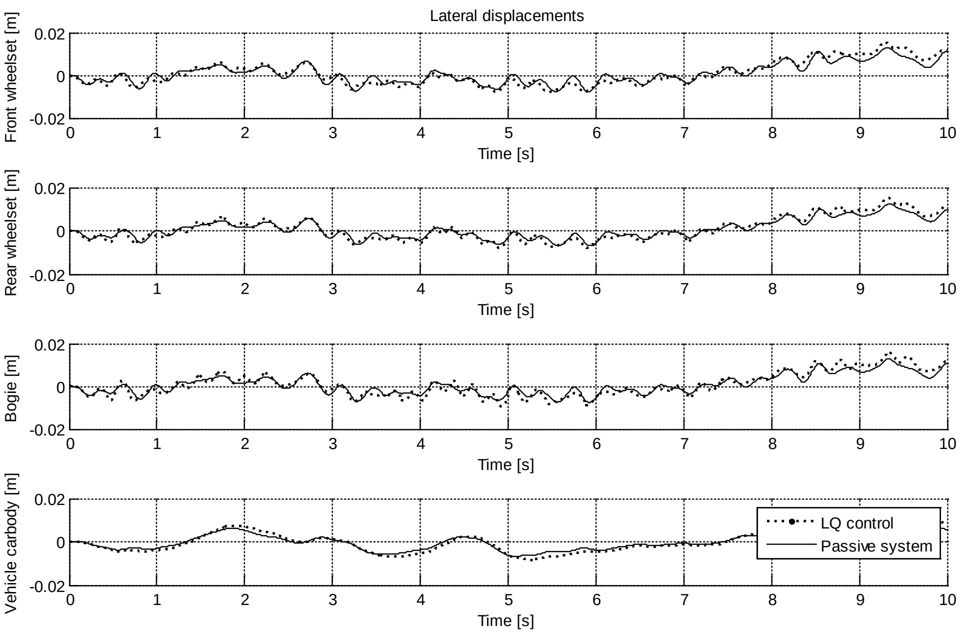

Figure 5 Lateral displacements for the passive and active situations

This LQR control strategy constitutes the natural first step for an optimal control, and also provides a comparison for the next step, i.e. control with state estimation.

3.2 *Control with state estimation (LQG)*

The Kalman filter is a state observer which is the most appropriate estimation technique when the system is a subjected to stochastic disturbances. An LQG regulator was designed using a Kalman filter state estimator to replace the direct measurement of some state variables, using the sensors defined by Figure 3 and listed below. The estimated outputs replace the direct state feedback, and the control loop is completed using the same gain matrix that was calculated for the LQR design (previous sub-section).

Table 2 List of sensors

Label	Measure	Sensor
A	$\ddot{y}_v$	lateral body accelerometer
H	$y_v - y_b$	relative displacement between bogie and car body
E	$\ddot{y}_b$	lateral bogie accelerometer
D	$\dot{\psi}_b$	bogie yaw rate gyroscope
C,G	$\ddot{y}_f, \ddot{y}_r$	lateral wheelset accelerometer
B,F	$\dot{\psi}_f, \dot{\psi}_r$	wheelset yaw rate gyroscope

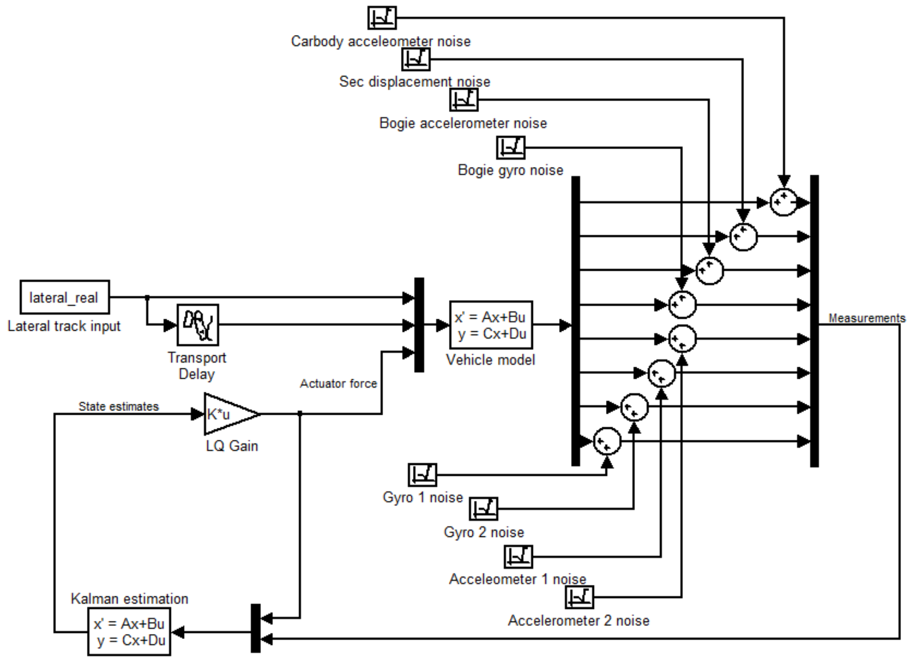

Figure 6 LQG control scheme including Kalman Filter

Gaussian white noise has been added to the measurements with variances taken as 1% of the full scale value for each sensor, and the corresponding values used to specify the measurement noise for the Kalman filter design. Information regarding the process noise is also needed, and this has been calculated from the variance of the lateral disturbances used in the simulation. Of course, this depends on the characteristics of the track, and the alternative would be to derive a variance average from several lateral disturbance examples in order to produce a more general value.

Figure 7 compares the “real” and measured signals to illustrate the performance of the Kalman filter, here shown for the bogie yaw displacement and yaw rate. It can be seen that there is some variation in the low frequency content, particularly for the displacement estimate, but a small divergence such as this is common with Kalman filter implementations and does not cause a problem in terms of system performance. Other state estimates have smaller errors, and simulations show that the LQG approach, using the estimated values instead of the “real” values, gives a performance similar to that achieved using basic LQR control.

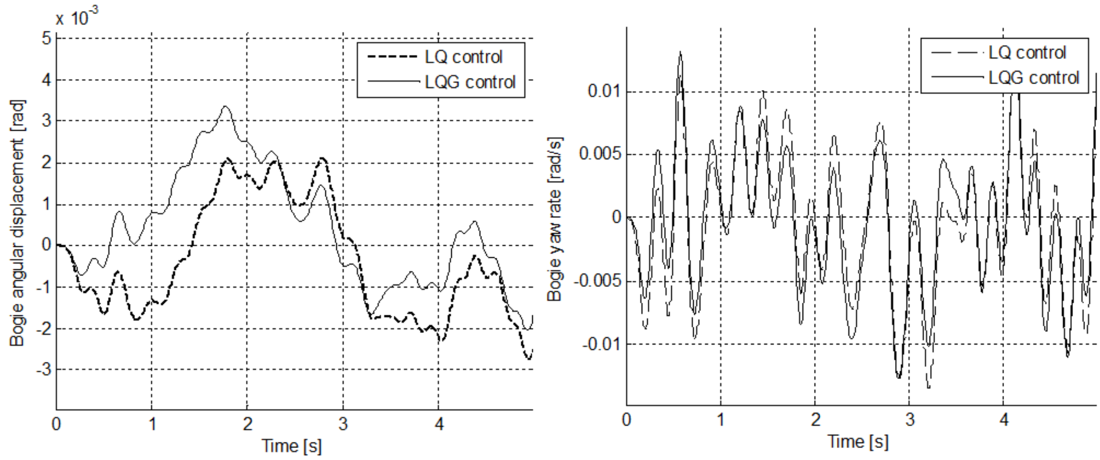

Figure 7 Examples of comparison between actual and estimated values

4 SIMULATION RESULTS USING NON-LINEAR MBS MODEL

This section presents results from a multi-body non-linear simulation of the LQR control strategy that was developed on the simplified linear design model, in particular to provide results on curved track for which non-linearities in the wheel-rail contact are important. Comparisons between the linearized and MBS results have shown broadly comparable overall values for the key variables but with some differences in dynamic detail. The MBS simulation provides a consistent comparison between the following configurations: basic passive, passive with reduced PYS and the active model (also of course with reduced PYS). In both passive cases the normal secondary yaw damper is included.

4.1 *Straight track (stability) assessment*

The same lateral irregularity data as before are used to assess straight track operation. With a non-linear model poor dynamic performance now results in a limit cycle oscillation that is constrained by the non-linear wheel-rail profile, and so performance has been assessed on the basis of an appropriate European norm (EN14363 2005) which defines limits regarding the maximum lateral acceleration of each wheelset. Table 3 gives the RMS values for the three situations, from which it can be seen that the active solution provides an improvement compared with the baseline passive response. The simulations also show that the passive model with lowered primary yaw stiffness exceeds the limit imposed by the norm, which is of course a consequence of the low stability predicted by the linear simulations, and so the remainder of this section concentrates upon comparing the active with the baseline passive configurations.

Table 3 Straight track lateral wheelset RMS acceleration values

Passive – 10% PYS	Passive – 100% PYS	Active – 10% PYS
3.1 m/s^2	2.1 m/s^2	1.5 m/s^2

4.2 *Curved track performance*

For curved track the following three criteria have been assessed using the curve information listed in sub-section 2.2: the first two are safety-related, whereas the third is used to quantify track damage:

- Y/Q ratio
- Maximum lateral track shift force
- Wear index

The simulations show that the passive model with baseline PYS, as expected, has a bigger Y/Q ratio than the other two models, although the value of 0.24 for the simulated curve is perfectly acceptable. Both the active and the passive with the lowered PYS show a more favourable Y/Q of 0.11.

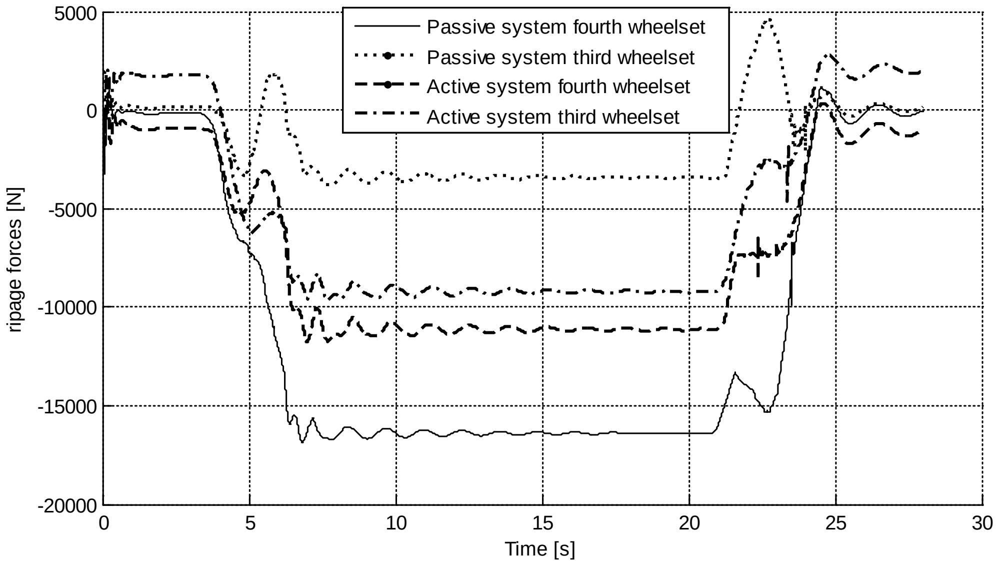

Figure 8 Comparison of track shifting forces for the passive and active configurations

Figure 8 compares the track shifting forces and shows the significant reduction for the trailing wheelset. This reduction arises because the effect of the low PYS is to equalise the forces between the leading and trailing wheelsets, the sum of which is defined by the curving conditions. This result itself is an indication of improved curving performance, but Table 4 quantifies the improvement by listing the energy dissipation for the eight wheels of the vehicle through the curve. The total wear for the leading bogie is reduced by a factor of 15, a very significant improvement.

Table 4 Energy dissipation (kJ)

Wheel	Passive	Active
1 R	132.11	2.56
1 L	34.50	4.62
2 R	4.30	1.54
2 L	3.92	1.62
3 R	111.03	2.47
3 L	30.49	4.53
4 R	6.37	1.84
4 L	5.53	1.84
Total	328.25	21.02

CONCLUSIONS

The paper has demonstrated that an active secondary yaw actuator to replace the conventional yaw damper has enabled stability to be achieved with the primary yaw stiffness reduced to one tenth of the value for a conventional bogie. The initial hypothesis of better curving performance has been verified and quantified by the simulation tests performed on the different models in different situations. Simple linearized models have been used for control design and development, and a more complex non-linear MBS simulation model has evaluated the overall benefits.

More research has been undertaken to quantify in more detail the advantages brought by the actively-stabilised vehicle while running in a curved track, in terms of reduced lateral forces and wear of the wheel and rail profiles (Alfi et al 2015), and further study is required to validate the performance under a wider variety of operational conditions, also to consider the practicalities of sensing and actuation. Nevertheless the proposed concept offers a potentially important option for achieving substantial improvements in bogie performance.

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APPENDIX Symbols and parameter values

Symbol	Value	Parameter
r_0	0.45[m]	Rolling radius
λ	0.15-0.25	Conicity
l	0.75[m]	Half gauge
L	1.3[m]	Semi wheelbase
D	0.45[m]	Primary bush length
h_{ws}	1[m]	Primary suspension lateral semi spacing
A_d	1.25[m]	Semi spacing of longitudinal dampers
f_{11}	10e6[MN]	Longitudinal creep coefficient
f_{22}	8.8e6[MN]	Lateral creep coefficient
f_{23}	13.7e3[MN]	Spin creep coefficient
m_v	30000[kg]	Carbody mass
m_b	2500[kg]	Bogie mass
m_w	1120[kg]	Wheelset mass
I_b	2500[kgm^2]	Yaw inertia of the bogie
I_w	730[kgm^2]	yaw inertia of the wheelset
I_{wy}	29.61[kgm^2]	pitch inertia of the wheelset
W	96824.7[N]	Wheelset load
k_{y1}	1[MN/m]	Primary lateral stiffness
k_{x1}	1[MN/m]	Primary longitudinal stiffness
f_{y1}	0[Ns/m]	Primary lateral damping
f_{x1}	0[Ns/m]	Primary longitudinal damping
k_{y1b}	250[kNs/m]	bush lateral stiffness
k_{x1b}	250[kNs/m]	bush longitudinal stiffness

f_{y1b}	0[Ns/m]	Bush lateral damping
f_{x1b}	0[Ns/m]	Bush longitudinal damping
k_{y2}	280[kN/m]	Secondary lateral stiffness
f_{y2}	30[kNs/m]	Secondary lateral damping
k_{psi2}	50[kNm/rad]	Secondary yaw stiffness
f_{x2}	250[kNs/m]	Longitudinal yaw damping