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Active secondary yaw control to improve curving behaviour of a railway vehicle

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Abstract

Active primary / secondary suspensions have been proposed as a means to solve the trade-off between curving and stability which represents a key problem in the design of modern railway vehicles.

In particular, one concept proposed for active control of the vehicle's running behaviour is known as Secondary Yaw Control (SYC) and consists of applying a controllable yaw torque between the carbody and the two bogies. This concept has been studied in the past mainly to enhance the vehicle's curving ability.

This paper extends the idea by examining the implications of designing a bogie with soft yaw stiffness between the bogie frame and the wheelsets and using SYC to provide active stabilisation. To this aim, a state feedback control law is designed according to the LQR and LQG techniques.

The paper presents the general concept of active suspension control investigated and the control strategies applied. Then the effectiveness of the proposed actuation concept is investigated by means of numerical simulations performed on mathematical models of the passive and actively controlled vehicles implemented in a fully nonlinear multi-body simulator. Comparisons are performed and benefits assessed between the actively controlled vehicle and the passive one in terms of: non-linear stability in straight track running; and safety and wear in curves.

Key words : Active rail suspension, secondary yaw control, LQG control, stability, curving

1. Introduction

In modern rail vehicles, bogie design involves satisfying a stability constraint at maximum speed, at the same time achieving satisfactory curving performance, often resulting in the search for a trade-off between these conflicting requirements (Iwnicki Ed., 2006). In recent years, bogies incorporating active steering have been proposed as a means to overcome this design conflict, according to a variety of different control strategies and actuation concepts, as illustrated in some recent surveys (Bruni et al., 2007, Goodall et al. 2012).

Secondary Yaw Control (SYC), consisting of applying a controllable yaw torque between the body and the two bogies, is an attractive actuation concept as it implies a less radical impact on the mechanical design of the vehicle than other alternative actuation concepts (Bruni et al., 2007). SYC has been studied both theoretically and experimentally (Diana et al., 2002, Matsumoto et al. 2009, Simpson and Cole, 2011). In these studies, the passive vehicle was generally designed to have good intrinsic stability properties, while SYC was used to enhance curving capability through active steering. In this paper, a different concept is investigated, which consists of designing the passive vehicle to achieve satisfactory curving performances, then using SYC to obtain a satisfactory running behaviour in tangent track at high speed, in particular the same or higher non-linear critical speed as the passive vehicle with standard suspensions.

The desired curving performance of the vehicle is achieved by reducing substantially the primary yaw stiffness (PYS), i.e. the yaw stiffness of the suspension elements connecting the bogie frame and the wheelsets, allowing the wheelsets to assume a more radial position in curves and therefore reducing lateral contact forces and rail wear, while active stabilisation is introduced by SYC compensating for the poorer running stability properties of the bogie with reduced PYS.

On the basis of a linearized horizontal plane model of the bogie considering 7 degrees of freedom, a state feedback linear quadratic regulator (LQR) is designed for the SYC. In the first instance, full-state feedback control is assumed. Then, an LQG regulator is designed using a Kalman filter state estimator to replace the direct measurement of some state variables.

The effectiveness of the proposed application of the SYC concept is investigated by means of numerical simulation performed using a fully nonlinear multi-body model of the vehicle implemented in ADTreS, an in-house multi-body software specifically designed for the study of rail vehicle dynamics, developed at Politecnico di Milano.

2. The concept of Secondary Yaw Control

The active suspension concept considered in this work is known as “Secondary Yaw Control” (SYC). It is based on applying a yaw torque on the bogie by means of actuators mounted in longitudinal direction between the bogie frame and the carbody, see figure 1.

Past investigations of this actuation concept can be found in (Matsumoto et al 2009, Simson and Cole 2009, Simson and Cole 2011), mainly looking at active steering of the bogie, whereas in (Diana et al., 2002) SYC is used to mimic the behaviour of a passive yaw damper, taking advantage from the wide pass-band of the actuator so that higher levels of energy dissipation can be achieved at relatively high frequency of the hunting limit cycle (6-8 Hz), a case in which the efficiency of hydraulic yaw dampers is reduced by internal deformability effects.

In this paper, the use of SYC is proposed for the active stabilisation of the bogie, so that the yaw stiffness of the connection between the bogie frame and the wheelsets can be designed to be much lower than in a standard railway bogie. In this paper, we will indicate as ‘standard’ yaw suspension stiffness a value in the range normally used in modern vehicles with passive suspensions designed for a maximum service of 40 m/s approximately, and we will denote by ‘soft’ yaw suspension stiffness a value 10 times lower, see Appendix 1 for the bogie parameters used in this study.

Furthermore, as will be shown below, the control strategy chosen (LQR / LQG) also improves the quasi-static behaviour of the vehicle in a curve, resulting in better curving performances of the actively controlled vehicle compared to one with passive suspension for the same yaw stiffness.

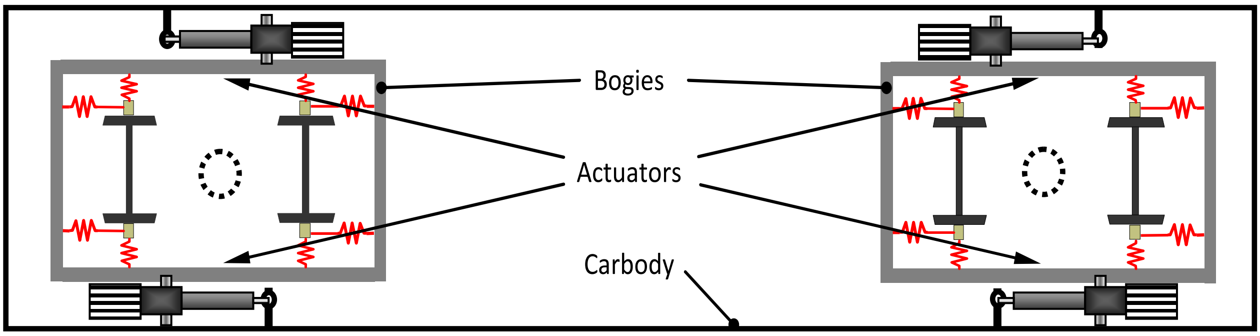

Fig. 1 Actuation principle for Secondary Yaw Control (SYC).

3. Linear vehicle model

The tuning of the gain matrix for the Linear Quadratic control and the definition of the Kalman filter observer for the LQG control were both based on the simplified linear model of the vehicle shown in Figure 2. The model represents the dynamics of a half-vehicle and consists of two wheelsets, one bogie and one half car body, based upon a typical modern passenger vehicle design for a speed of 40 m/s – parameter values for the standard passive arrangement are given in Appendix 1. Given that the focus of this work is on vehicle stability and curving, 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, for each body except for the half car body for which only the lateral displacement is modelled.

The primary suspension is modelled as 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 are considered in the range 0.10÷0.25.

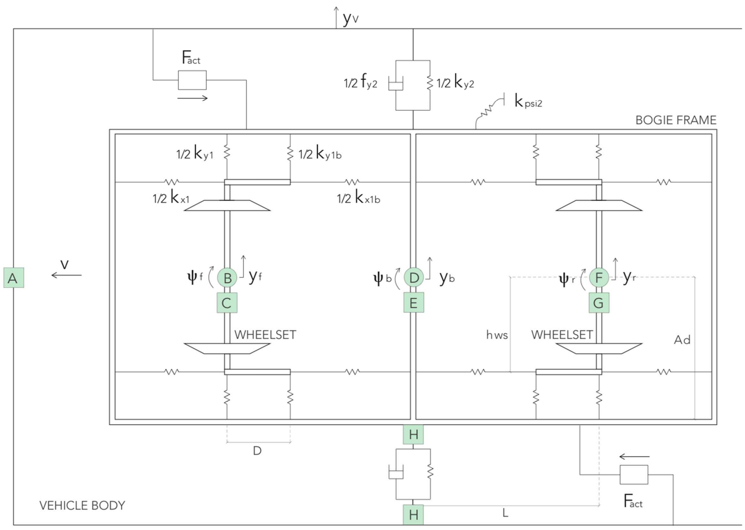

Fig. 2 The plan-view half-vehicle dynamic model and position of sensors used by the LQG controller. A: lateral body accelerometer; B, F: wheelset yaw rate gyroscope; C, G: lateral wheelset accelerometer; D: bogie yaw rate gyroscope; E: lateral bogie accelerometer; H: relative displacement between bogie and car body

4. Control strategies

In this work, two different control strategies are considered. The first one is a full state LQR controller, based on the assumption that all the states involved by the model in Figure 2 can be measured. The second is an LQG controller with state estimation, taking into account a realistic situation in which a reduced set of sensors is mounted on the vehicle.

4.1 Linear quadratic regulator (LQR) control

In the first instance, full-state feedback control is assumed for the linear bogie model described in Section 3 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, though minimising actuation requirements. This was achieved based on the state normalisation method, with the maximum expected values of the state variables taken from a set of simulations performed on the passive vehicle with nominal primary yaw stiffness. For the actuator force, the maximum allowed value was also obtained from simulations performed on the nominal passive vehicle, considering the maximum force generated by the yaw damper. A second stage of tuning was then performed by means of manual trial and error, to improve the final behaviour of the LQR controller.

More details concerning the weight tuning of the LQR can be found in (Prandi et al. 2015).

4.1 Linear Quadratic Gaussian (LQG) control

Following the tuning of the LQR , an LQG regulator was designed to take into account that in a real application not all the states of the system can be measured directly, and also to take into account that both the dynamics of the system and measurements are inevitably affected by noise.

In the LQG regulator, the direct measure of some state variables is replaced by the use of a Kalman filter state estimator receiving as input a reduced set of measurements.

Table 1 lists the measurements assumed to be available from the system. The position assumed for the sensors is shown in Figure 2 using alphabetic labels also listed in Table 1.

Table 1 - List of the sensors used by the LQG regulator.

Label	Measure	Sensor
A	$\ddot{y}_v$	Carbody lateral accelerometer
H	$y_v - y_b$	Relative distance between bogie and carbody
E	$\ddot{y}_b$	Bogie lateral accelerometer
D	$\dot{\psi}_b$	Bogie yaw rate (gyroscope)
C,G	$\ddot{y}_f, \ddot{y}_r$	Front and rear wheelset lateral accelerometer
B,F	$\dot{\psi}_f, \dot{\psi}_r$	Front and rear wheelset yaw rate (gyroscope)

The Kalman filter estimator considers track lateral irregularity as the source of process noise affecting the state of the system; the corresponding co-variance was defined as the variance of the lateral irregularity assumed in the numerical simulations described in Section 5.

The covariance matrix for measurement noise was defined assuming the noise occurring on the different sensors to be fully uncorrelated and assuming the variance of the measurement noise for each sensor to be 1% of the sensor’s measuring range (defined by the maximum values from the passive vehicle simulation).

Numerical experiments performed using the linear vehicle model described in Section 3 demonstrated the suitability of the Kalman filter for state estimation, replacing the direct state measurement in the LQG regulator, see (Prandi 2014 and Prandi et al. 2015) for more details.

5. Results

Numerical investigations were performed to assess the behaviour of the actively controlled vehicle with soft primary yaw suspension compared to a passive vehicle with standard primary yaw suspension stiffness. To this aim, a non-linear model of the vehicle was set-up using ADTreS, a multibody software developed at Politecnico di Milano for the study of rail vehicle dynamics.

Numerical simulations were directed to investigate non-linear stability in tangent track and the vehicle’s running behaviour in curves.

5.1 Non-linear stability

The behaviour of the vehicle in straight track was investigated in terms of its non-linear stability, i.e. the occurrence of periodic oscillations as the result of self-excited vibrations caused by wheel-rail contact forces. The method used to assess numerically the stability of the vehicle replicates the one proposed by the European standard EN14363 to verify vehicle stability based on line tests.

Simulations are for the vehicle running at constant speed in straight track, subjected to random excitation caused by track irregularities. The time history of the lateral bogie frame acceleration over one axle-box is considered as the output of the simulation. This is treated as follows:

i) a pass-band filter is applied on the signal with pass band $f_0^c \pm 2$ Hz, f_0^c being the frequency that corresponds to the harmonic component having largest amplitude in the signal;
ii) the sliding rms of the signal is computed over a 100 m window length which is updated at each 10 m step length
iii) the sliding rms values obtained are compared to a limit value defined as:

$$\ddot{y}^{+}{}_{\lim} = \frac{1}{2}\left(12 - \frac{m_b}{5}\right) \qquad (1)$$

the limit value being expressed in m/s^2 and being m_b the mass of the bogie.

Figure 3 shows the results obtained for the passive vehicle with standard primary yaw stiffness running at 40 m/s over an irregular track representing a spatial realization of the power spectral density defined by ORE/ERRI B176 for ‘low-level’ irregularities. The blue line shows the time history of the pass-band filtered lateral acceleration signal, the red line with crosses the sliding rms and the horizontal dashed line the limit value according to EN14363.

In this running condition the vehicle shows a stable running behaviour with the sliding rms values well below the limit. The maximum value of the sliding rms is 2.1 m/s^2.

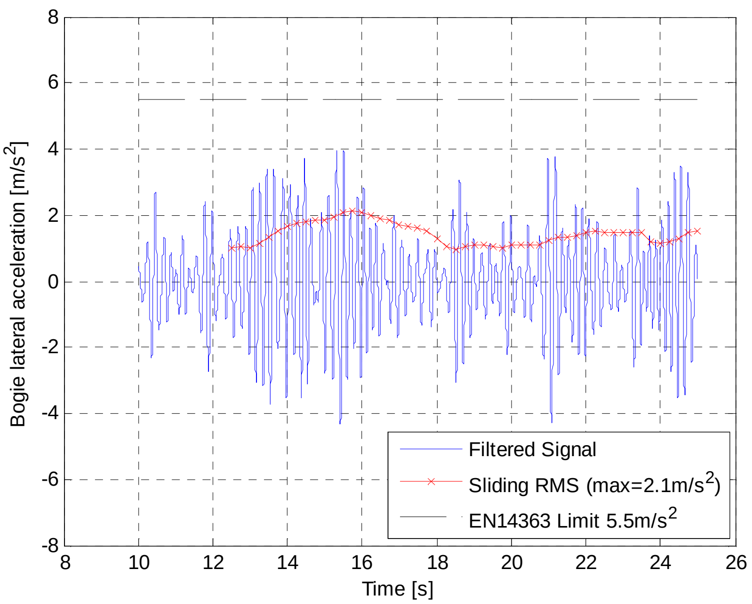

Fig. 3 Results of non-linear stability analysis for the passive vehicle with standard primary yaw stiffness. Time history of the pass-band filtered lateral acceleration signal (blue line), sliding rms (red line with crosses), EN14363 limit (dashed line).

Figure 4 shows the results obtained for the passive vehicle with soft primary yaw stiffness, considering the same running condition as in Fig. 3. The effect of lowering the primary yaw stiffness is apparent: the maximum value of the sliding rms is in this case 3.1 m/s^2 which is still below the limit, but is increased compared to the previous case by 50% approximately. Furthermore, large oscillations are observed in the time histories of the filtered acceleration (blue line), that instantaneously exceed the limit. In conclusion, the behaviour of the vehicle with soft primary suspension cannot be considered fully satisfactory from the point of view of running stability.

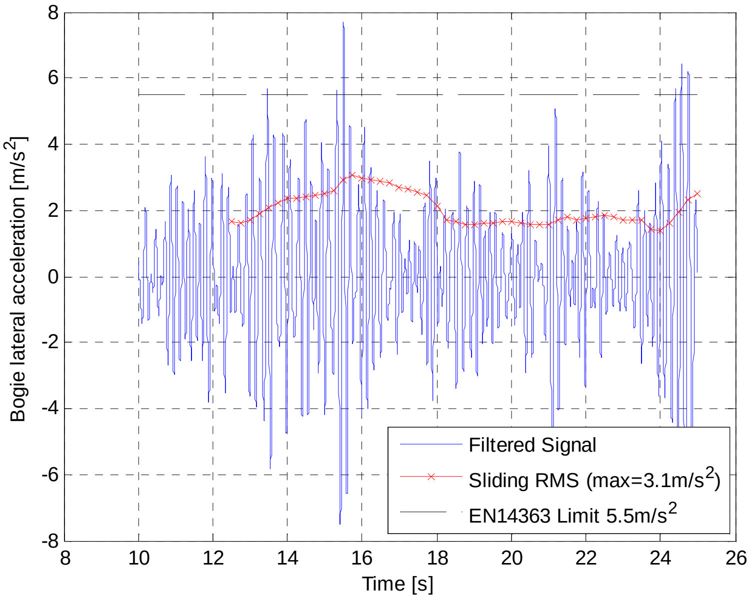

Fig. 4 Results of non-linear stability analysis for the passive vehicle with soft primary yaw stiffness. Time history of the pass-band filtered lateral acceleration signal (blue line), sliding rms (red line with crosses), EN14363 limit (dashed line).

Figure 5 shows the results obtained for the vehicle with soft primary yaw stiffness and active stabilisation, considering the same running condition as in Fig. 3. In the results shown, the LQR control strategy is applied with gains tuned as described in (Prandi et al, 2015). Comparing these results to the ones shown in Fig. 4 for the same vehicle in passive configuration, the advantages of active stabilisation can be assessed: the maximum value of the filtered lateral acceleration only slightly exceeds 3 m/s^2 (compared to almost 8 m/s^2 for the passive vehicle) and the sliding rms is correspondingly lower, with a max. value of 1.5 m/s^2, even lower than the value obtained for the passive vehicle with standard primary yaw stiffness.

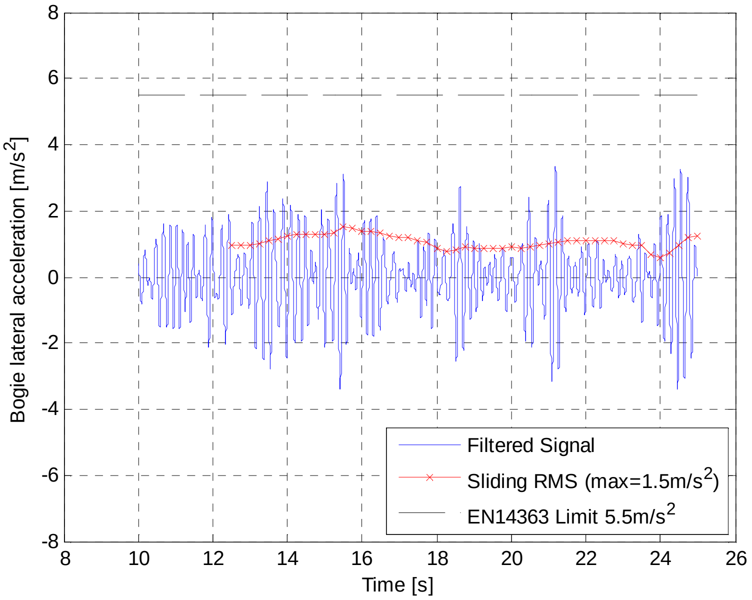

Fig. 5 Results of non-linear stability analysis for the vehicle with soft primary yaw stiffness and active stabilisation. Time history of the pass-band filtered lateral acceleration signal (blue line), sliding rms (red line with crosses), EN14363 limit (dashed line).

5.2 Curving behaviour

The curving behaviour of the railway vehicle was investigated considering the negotiation of a curve with 500 m radius and 150 mm super-elevation. The vehicle speed was set to 31.5 m/s, corresponding to a cant deficiency of 150 mm. To highlight the steady-state behaviour of the vehicle, this curving condition was simulated not considering track irregularity. The curving behaviour was assessed based on the following parameters:

- the Y/Q derailment coefficient, i.e. the ratio of the lateral Y over vertical Q components of the contact force, evaluated for the outer wheel of the leading wheelset. This wheel was chosen as it is the one for which the largest derailment coefficient is obtained;
- the track shift force, i.e. the sum of the lateral Y forces on the inner and outer wheels of the trailing wheelset in the rear bogie. This wheelset was chosen as it is the one producing the largest track shift force in full curve;
- the frictional energy dissipated by the tangential contact forces while the vehicle negotiates the entire curve, evaluated for all wheels in the vehicle.

The derailment coefficient and track shift force are related with the assessment of running safety, whereas the frictional energy is related with the amount of abrasive wear that is likely to occur on both the wheel and rail profiles as a consequence of curve negotiation.

Figure 6 shows the comparison of the Y/Q ratio for the passive vehicle with standard primary suspension, the passive vehicle with soft primary suspension and the active vehicle with soft primary suspension. The use of a soft primary yaw stiffness leads to a very significant decrease of the derailment coefficient compared to the vehicle with standard suspension, especially in full curve. The maximum value is approximately 0.25 for the vehicle with standard suspension (still far from the limit value of 0.8) and 0.12 for the two vehicles with soft suspension. No significant difference is observed for the vehicle with active control compared to the passive one having the same primary yaw stiffness.

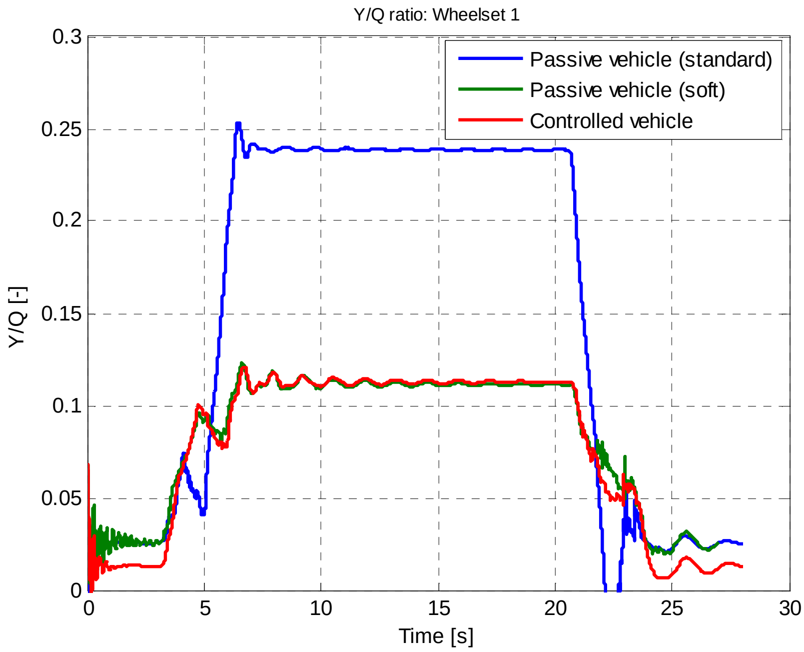

Fig. 6 Comparison of the Y/Q ratio on the external wheel of the first wheelset of the vehicle. Blue line: passive vehicle with standard primary suspension. Red line: passive vehicle with soft primary suspension. Green line: active vehicle with soft primary suspension.

Figure 7 shows the time histories of the track shift forces for the three vehicle configurations considered. The benefit of using a soft primary suspension is apparent, as the maximum track shift force is reduced from 16.9 kN to 12.4 kN for the passive vehicle. Note that the reduction of the track shift force is obtained for the same cant deficiency, i.e. for the same non-compensated centrifugal force acting on the whole vehicle. Therefore, the reduction of the track shift force in the vehicle with soft suspension is obtained through a more even distribution of the lateral contact forces on the four axles, which in turn is the consequence of the wheelsets taking a more radial attitude thanks to the softer suspension. The use of the LQR active control leads to a further redistribution of the lateral forces on the vehicle's axes and produces a further slight reduction of the maximum track shift force.

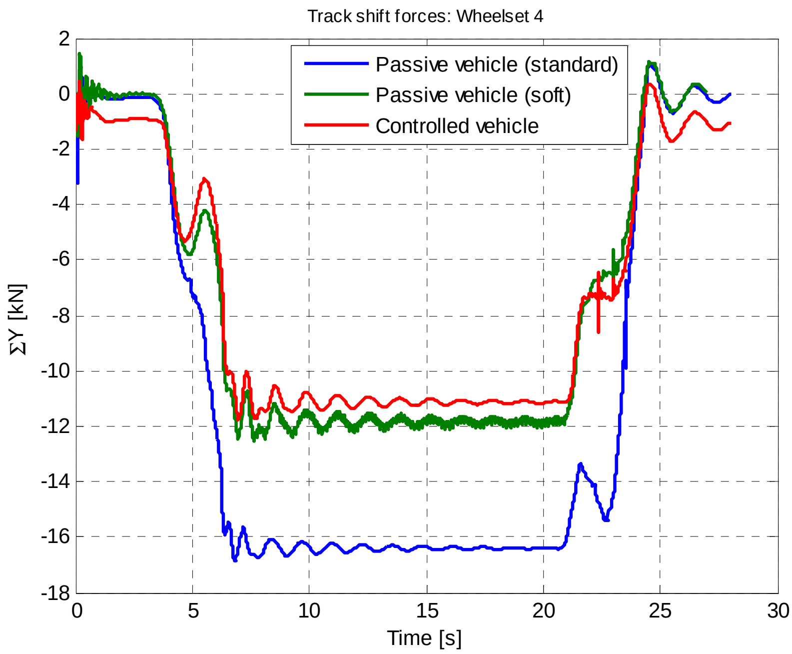

Fig. 7 Comparison of the track shift forces for the fourth wheelset of the vehicle. Blue line: passive vehicle with standard primary suspension. Red line: passive vehicle with soft primary suspension. Green line: active vehicle with soft primary suspension.

The frictional energy dissipated by the tangential contact forces during the whole curve negotiation is compared in Table 2 for the three vehicle configurations considered. For the passive vehicle with standard suspension, very large values of dissipated energy are obtained for the leading wheelsets of the front and rear bogies, as strong flange contact occurs on the outer wheels of these wheelsets. Given that the dissipated frictional energy is directly related with abrasive wear occurring at the wheel-rail contact, this result confirms that the vehicle with standard suspension would suffer from accelerated wear of the wheel profiles and, at the same time, would be highly aggressive to the infrastructure in terms of wear of the rail profiles.

Table 2 - Frictional energy dissipated by the tangential contact forces during the whole curve negotiation.

Wheelset	Wheel (O=outer, I=inner)	Passive with standard primary suspension	Passive with soft primary suspension	Active with soft primary suspension
1	O	132107 J	2271 J	2562 J
1	I	34500 J	4194 J	4619 J
2	O	4298 J	1554 J	1543 J
2	I	3918 J	1644 J	1618 J
3	O	111034 J	1338 J	2468 J
3	I	30493 J	2433 J	4531 J
4	O	6369 J	2772 J	1837 J
4	I	5525 J	2837 J	1840 J

The use of a soft yaw primary stiffness results in a very strong reduction (by a factor 60-100) of the frictional energy dissipated on the leading wheelsets of the front and rear bogies, and also in a significant reduction (by a factor 2-3) of the dissipated energy in the trailing wheelsets. The differences between the passive vehicle with soft suspension and the active vehicle are minor and do not show a unique trend, so it can be concluded that the use of LQR control during curve negotiation does not allow to further reduce wear effects on the wheels and on the rails.

It should be kept in mind that these results are obtained for one single curving condition while a more comprehensive investigation of curving behaviour for e.g. different curve radii and cant deficiency values is envisaged as an extension of this work.

6. Conclusions

In this paper, the use of secondary yaw control for improving the running behaviour of a railway vehicle with standard architecture (one carbody, two bogies and four wheelsets) was investigated. Differently from previous studies in which SYC was often used to provide active steering in bogies designed to have a stiff primary suspensions, active control is used here to provide active stabilisation for bogies having a soft primary suspension, with the implication that the curving behaviour will be inherently good due to the fact that the wheelsets are allowed to take a nearly-radial attitude.

A state feedback LQ regulator was designed for the active vehicle and an LQG version of the same was also designed using a Kalman filter-based state estimator to reduce the number of sensors needed to implement the control strategy. The Kalman filter observer is defined according to a linearized horizontal plane model of the bogie considering seven degrees of freedom (see Section 3).

The effectiveness of the proposed actuation concept is investigated by means of numerical simulations performed on mathematical models of the passive and actively controlled vehicles, implemented in a multi-body simulator. Comparisons are performed between the actively controlled vehicle and the passive one in terms of non-linear stability in tangent track and of running safety and wear in curves.

The proposed active control system succeeds in providing a fully satisfactory running behaviour of the vehicle with soft suspension in respect to non-linear stability in tangent track, which is found to be even better than for a passive vehicle having a standard primary yaw stiffness. Additionally, the vehicle with soft primary suspension exhibits good curving behaviour with low values of the running safety indicators (derailment coefficients, track shift forces) and reduces considerably the frictional energy dissipated across the whole curve, when compared to the vehicle with higher primary yaw stiffness. The use of active control in curve provides only minor benefits, which is to be expected, given that the control strategy is designed only to improve vehicle stability.

A more comprehensive investigation of the benefits of the proposed active control concept for other curving conditions including e.g. different curve radii and cant deficiencies is envisaged as a future extension of this work.

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Appendix 1: Vehicle parameters

Symbol	Value	Parameter
r_0	0.45 m	Nominal rolling radius
L	0.75 m	Half gauge
L	1.30 m	Semi-wheelbase
D	0.45 m	Primary bush arm length
h_{ws}	1.00 m	Primary suspension lateral semispacing
Ad	1.25 m	Semi-spacing of longitudinal dampers
f_{11}	10.0×10^6 N	Longitudinal creep coefficient
f_{22}	8.8×10^6 N	Lateral creep coefficient
f_{23}	13.7×10^3 N/rad	Spin creep coefficient
f_{33}	0 Nm/rad	Spin creep coefficient
m_v	30000 kg	Carbody mass
m_b	2500 kg	Bogie mass
m_w	1120 kg	Wheelset mass
I_b	2500 kg m^2	Yaw inertia of the bogie
I_w	730 kg m^2	Yaw inertia of the wheelset
I_{wy}	29.6 kg m^2	Pitch inertia of the wheelset
W	96.825 kN	Axle load
k_{y1}	1.00×10^6 N/m	Primary lateral stiffness
k_{x1}	1.00×10^6 N/m	Primary longitudinal stiffness
k_{y1b}	4.00×10^6 N/m	Bushing lateral stiffness (Standard)
k_{x1b}	14.00×10^6 N/m	Bushing longitudinal stiffness (Standard)
k_{y2}	280×10^3 N/m	Secondary lateral stiffness
f_{y2}	30×10^3 N/m	Secondary lateral damping
k_{psi2}	50×10^3 N/rad	Secondary yaw stiffness
f_{x2}	250×10^3 Ns/m	Longitudinal yaw damping
k_{y1b}	0.40×10^6 N/m	Bushing lateral stiffness (Soft)
k_{x1b}	1.40×10^6 N/m	Bushing longitudinal stiffness (Soft)