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Active yaw damper for the improvement of railway vehicle stability and curving performances: simulations and experimental results

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To further increase passenger train comfort and handling performances, a mechatronic approach to the design of railway vehicles is necessary. In fact, active systems on board a railway vehicle allow to push design barriers beyond those encountered with just passive systems.

The article deals with the development of an electro-mechanical actuator to improve the running behaviour of a railway vehicle, both in straight track and curve. The main components of the active system are a brushless motor and a mechanical transmission, used to apply a longitudinal force between the carbody and the bogie of the vehicle. The actuator is operated in force control. Different control strategies were developed for straight track running, where the aim is to increase the vehicle critical speed, and for curve negotiation, where the goal is to reduce the maximum values of track shift forces.

A mathematical model of the railway vehicle incorporating the active control device has been developed and used to optimise control strategies and hardware set-up of the active device and to estimate the increase in operating performances with respect to a conventional passive vehicle.

The active control device has then been mounted on an ETR470 railway vehicle, and its performances have been evaluated during in-line tests in both straight and curved tracks.

Keywords: Mechatronic train; Active yaw damper; Stability; Curve negotiation; Real-time control

1. Introduction

Tilting trains, such as the Italian ETR470/480 or the Swedish X2000, have achieved a great commercial success, especially in Europe, because of their capability to run at high-speeds on both standard lines and high speed tracks [1]. To achieve these results, tilting trains need to combine high steering performances with high stability requirements. However, it is well known [2, 3] that these two needs are in conflict with each other: the chosen value of design parameters (such as the stiffness of the primary suspensions, the yaw dampers, etc.) is optimised either for steering or stability performances. It is therefore often suggested [4] that an increase in performances of these railway vehicles will be achievable only by means of actively controlled systems that may adjust the vehicle's behaviour to accommodate for different track features.

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To this end, two approaches are possible: the first one is to re-consider completely the design of the whole vehicle in order to reach a full integration between the mechanical design and the extensive application of controlled devices. The second one is to introduce active controlled devices into existing railway vehicles.

In the long term, the first approach is likely to produce a new generation of light and mechanically simple vehicles with high performances and lower total life cycle costs. Nevertheless, the introduction of actively controlled sub-systems impacting the ride safety of the vehicle into existing vehicles, as the active steering of the bogies or wheelsets, could be used to test the safety and reliability of these systems.

The research carried out at the Mechanical Engineering Department of Politecnico di Milano has led to the design and testing of an active device for a tilting carbody vehicle, designed to be mounted between the bogie and the carbody, in the same position as traditional yaw dampers. For this reason the device has been called 'active yaw damper' (or, based on the Italian acronym, AASA).

As better explained in section 3, this active control system applies two different control logics in order to optimise the vehicle's performances from a stability point of view (when running on a straight track at high speed) and from a steering point of view (when running on a curved track). Regarding the straight track running, the actuator is controlled in order to behave as an ideal viscous damper, *i.e.* the applied force is opposite to the carbody–bogie relative speed and proportional to its absolute value. The use of an actuator based on a brushless motor allows to apply this counterforce even at high frequencies (up to 8 Hz) that characterise the hunting instability of the bogie with respect to the carbody [5, 6].

On curved tracks, the railway vehicle speed is limited by the comfort of the passengers, by safety considerations and by the wear of wheels and rails. Usually, comfort requirements are the most bounding, but for tilting trains, these requirements are less severe because the carbody's inclination reduces the cant deficiency (lateral acceleration) felt by the passengers, thus allowing to increase the vehicle's speed. Safety considerations, as well as wear problems, then become the bottle neck to a further increase in service speed. Therefore, the active yaw damper can be used to reduce the derailment coefficient (the ratio Y/Q between the lateral and vertical contact force components) during curve negotiation or, instead, to re-balance track shift forces. For the train used during the experimental campaign, the unbalance of track shift forces is more critical. Thus, the control strategy during curve negotiation was developed to achieve the second goal.

2. Description of the active damper

As already said, in the proposed layout, the four traditional yaw dampers on each railway vehicle are replaced by four AASA devices. Thus, the AASA devices have longitudinal axes (figure 1). However, the possibility of mounting the AASA devices transversally with respect to the track was also analysed in the present work. This second solution (section 5.1) allows us to obtain better performances than the first one but would require a thorough re-design of the bogie.

The active system is made up of a centralized control unit, one local control unit per actuator, four actuators, local sensors (for each actuator) and global sensors (to determine the dynamics of the whole railway vehicle).

Each actuator is an electric brushless motor connected to a recirculating ball-bearing screw in order to obtain a linear actuator. The nominal torque of the motor is 30 Nm and the base speed is equal to 3000 r.p.m. Its peak torque is equal to 77 Nm and the maximum speed is 7000 r.p.m.

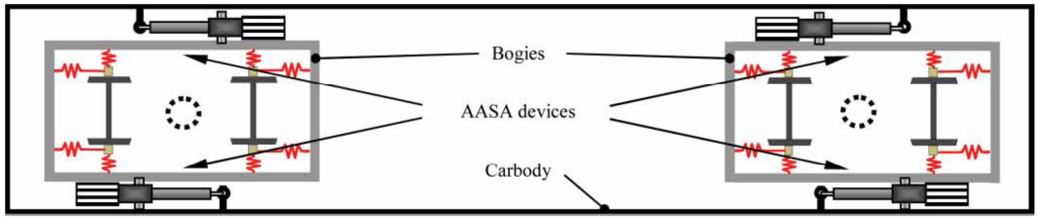


Figure 1. Scheme of the layout of the four AASA devices.

One side of the actuator (point A in figure 2(a)) is connected to the bogie, whereas the other side (point B in figure 2(a)) is attached to the carbody. The connection between actuator and bogie is obtained through a spherical joint in order to allow tilting. The connection between the actuator and the carbody, instead, is obtained through a rubber joint and a bracket that works as a mechanical fuse: in case of failure of the actuator, the bracket should brake, thus avoiding a rigid connection between bogie and carbody. The mechanical fuse, however, introduces a deformability (see section 5.1) that has to be taken into account to optimise the damping behaviour.

The actuators are operated in force control through a reference signal F_{ref}. This force signal is converted into a current signal by the local control units to drive the motors. F_{ref} is, in general, the sum of two different contributions: one for straight track running F_{ref_S} and one for curved track running F_{ref_C}. These two contributions are generated at two different levels of the control logic: while F_{ref_S} is determined locally by each actuator (independently of all other actuators), F_{ref_C} is calculated by the centralised control unit. This two layer control strategy is schematically shown in figure 3 and will be described in detail in section 3.2.

In order to determine the reference force signal F_{ref}, the active yaw damper has to know the current working conditions of the bogie. This is done by interfacing the control system with several sensors: an inductive velocimeter, placed between points A (bogie) and B (carbody) in figure 2(a) and thus measuring the relative longitudinal speed v_{rel} between bogie and carbody, and a piezoaccelerometer, placed at the connection of the active damper on the bogie and having axis parallel to the longitudinal direction (a_x), are used by the control logic during straight track running. During curved track running, signals coming from a piezoaccelerometer, placed on the bogie and having axis parallel to the lateral direction (a_y), and from a gyroscope, bogie-fixed and having vertical axis (thus measuring the yaw angular speed of the bogie $\dot{\sigma}$), are combined with the forward vehicle speed signal v and fed into the real-time control system.

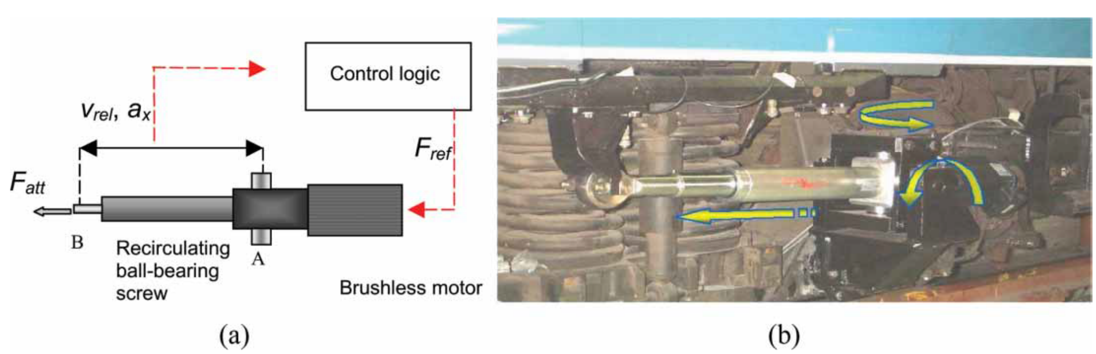


Figure 2. The active yaw damper: the actuator's scheme (a) and the active damper on the test vehicle (b).

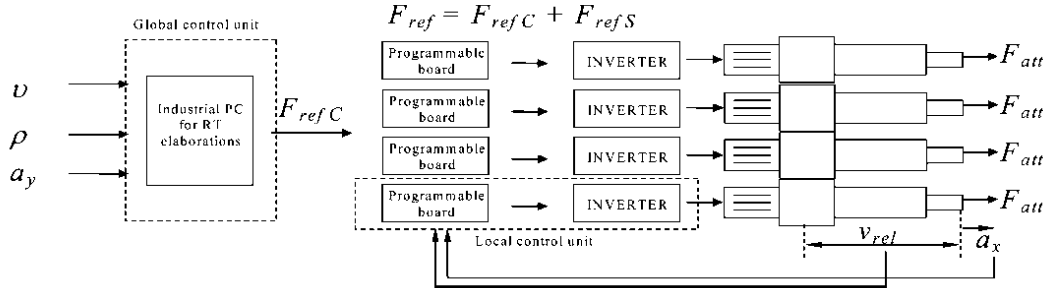


Figure 3. Scheme of the experimental set-up of the test vehicle for tangent and curved tests.

3. Control strategies

The control strategy aims at improving the performances of the railway vehicle both at high speed and on curved tracks. Thus, the reference value of the force is given by the sum of two different contributions

$$F_{\mathrm{ref}} = F_{\mathrm{ref_C}} + F_{\mathrm{ref_S}} \tag{1}$$

where $F_{\mathrm{ref_C}}$ is referred to curve negotiation and $F_{\mathrm{ref_S}}$ is referred to straight track running (stability). These two reference forces are very different and therefore require different control strategies.

3.1 *Control strategy to increase the critical speed*

The maximum speed of the vehicle in tangent track is limited by the occurrence of hunting instability, which is a self-excited combined lateral and yaw oscillation of the bogie caused by the action of wheel–rail contact forces typically occurring at high speeds. One possible way to suppress hunting instability is to use a damping force between the bogie and the carbody. In a passive vehicle this task is performed by oil dampers. However, in many cases, these devices prove to be rather inefficient, because of their internal flexibility [7], and therefore do not allow to raise sufficiently the vehicle's critical speed.

The reference force $F_{\mathrm{ref_S}}$ for straight track running is determined to achieve an ideal viscous damper, thus increasing the vehicle stability threshold: $F_{\mathrm{ref_S}}$ is proportional to the carbody–bogie relative speed and opposite to it. Thus, the inductive velocimeter signal v_{rel} is used. Moreover, to compensate for the delays of the measurement set-up, of the control logic, and the phase shifts introduced by the deformability of the system and by the inertial forces on the transmission, the reference force $F_{\mathrm{ref_S}}$ has an additional term proportional to the longitudinal acceleration a_x of the bogie

$$F_{\mathrm{ref_S}} = -(c_v v_{\mathrm{rel}} + m_v a_x) \tag{2}$$

c_v being the viscous damping gain and m_v the phase gain of the control action itself.

3.2 *Control strategy to improve curve negotiation*

During curve negotiation at high values of non-compensated lateral acceleration, large values of lateral contact forces may occur at the wheel–rail interface, with negative effects on ride safety. As already explained in the introduction, the so-called 'track shift forces', *i.e.* the total lateral force applied by each wheelset to the track, are particularly important for the kind of vehicle considered in this article. Large values of track shift forces are deemed to be dangerous for ride safety because they may lead to permanent deformations of track geometry. UIC 518

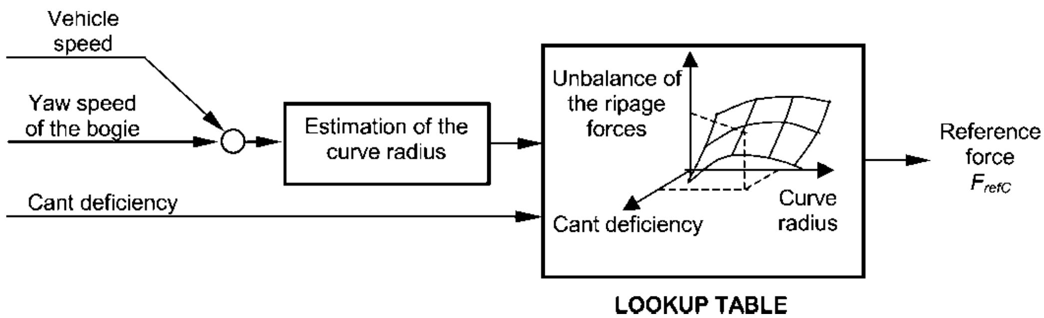


Figure 4. Scheme of the control strategy for the determination of the reference force $F_{\mathrm{ref_C}}$ during curve negotiation.

code limits the maximum allowed value of non-compensated lateral acceleration of the vehicle in order to guarantee that the track shift force limit given by Prud'homme's criterion [8] is never exceeded in any running condition.

Owing to the reasons described in [3], it may happen that a non-uniform distribution of track shift forces on the two wheelsets of the same bogie takes place. In this case, the maximum vehicle speed in curve will be reduced to keep the largest of the two forces below the Prud'homme's limit regardless of the value of the force on the other axle. Thus, a non-uniform distribution of track shift forces on the two axles is very unfavourable because it will lead to lower vehicle's operation speed.

The control strategy adopted during curve negotiation is therefore aimed at balancing the track shift forces on the two axles of the same bogie by applying a steering action. The value of the steering action is computed as a function of the non-compensated lateral bogie acceleration and the radius of the curve, by interpolating a bi-dimensional table that may be built on the basis of measurements performed on the vehicle in passive configuration or by using a numerical model of the vehicle (section 3.2). Figure 4 shows a schematic representation of the control strategy along a curved track: measuring the non-compensated lateral bogie acceleration and estimating the curve radius (as described subsequently), it is possible to determine from the lookup table the ΔF_{rip} force and thus the reference force $F_{\mathrm{ref_C}}$ necessary to balance the track shift forces on the two axles of the considered bogie.

In order to determine the curve radius R, the railway vehicle speed signal v (available from train instrumentation) as well as the bogie yaw speed signal $\dot{\sigma}$ (measured by a gyroscope on the bogie) are low-pass filtered and then fed into the real-time control board (together with the non-compensated lateral acceleration measured by a servo-accelerometer) that estimates the curve radius and performs the table lookup.

3.2.1 Calculation of the lookup table. In order to determine the steering action to be applied to the considered bogie, the track shift forces on the two axles have to be known. This can be done either by directly measuring such forces or by estimating them. The first solution requires the adoption of instrumented wheelsets, thus leading to very high installation costs. The second solution, instead, does not require instrumented wheelsets, thus being much cheaper, but the control logic is less robust. If the estimation of the track shift forces on the two axles is carried out online, *i.e.* while the train is running, other sensors than the ones listed in section 2 may be necessary. If, instead, the estimation is changed into the interpolation of previously stored data, no additional sensors are necessary and the computational effort necessary to determine these forces is very small. The lookup table can either be based on experimental data or on numerical results. Owing to the fact that no experimental track shift

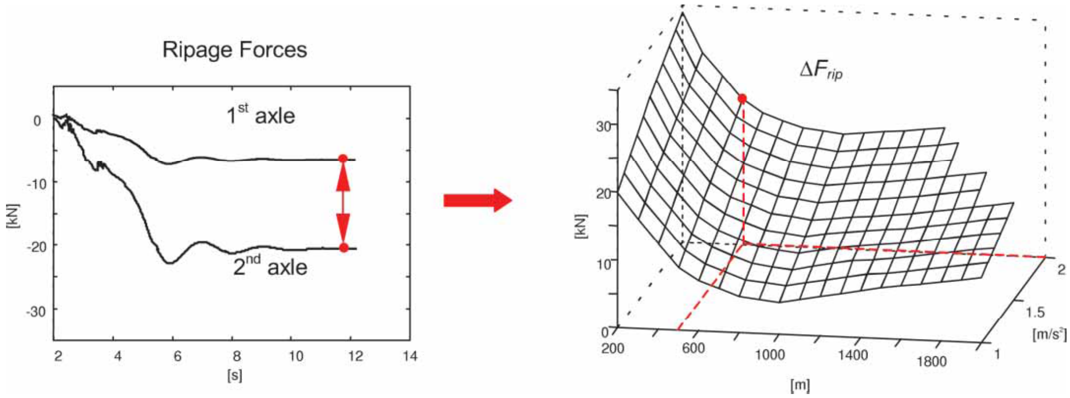


Figure 5. Scheme of the construction of the lookup tables.

force data of two adjacent wheelsets were available, the simulation approach was followed using a validated multibody vehicle model [9].

In order to numerically determine the lookup table, several simulations of the vehicle's running performances have been carried out considering wheel and rail profiles with different levels of wear, different values of friction between wheels and rails, different vehicle speeds (lateral accelerations), and different values of curve radii. These dynamic simulations allow us to determine the whole time histories of the ripage forces on the various axles of the vehicle. The steady-state value is identified and saved in the lookup table. It should be noticed that there are several lookup tables for the different bogies, for the friction coefficient value and for the wear level. Figure 5 schematically describes this procedure.

4. Mathematical model of the active damper

In order to investigate the potentials of the active yaw damper concept, a mathematical model of the ETR470 railway vehicle equipped with the electro-mechanic actuators was developed. The model is based on a mathematical description of the passive vehicle that was developed during previous research carried out, among others, by the authors and was thoroughly validated by comparison with service measurements [9].

The full vehicle is modelled using a multibody approach. The carbody and the two bogie frames are introduced as rigid bodies, whereas the deformability of the four wheelsets is introduced by modal superposition approach. Primary and secondary suspensions, including bump-stops, are modelled by means of linear and non-linear elastic and viscous elements.

A detailed non-linear representation of wheel–rail contact forces accounting for the real geometry of wheel and rail profiles is adopted [9]. Thereby, the model allows us to obtain realistic results even in flanging conditions, *i.e.* for large relative wheel–rail lateral displacements that take place during curve negotiation or during the hunting motion. The presence of non-linearities implies that the equations of motion of the vehicle have to be integrated in the time domain.

The active yaw dampers are modelled taking into account the electronic drive, the brushless motor, the recirculating ball-bearing screw transmission and the flexible connections between the actuator and the vehicle. Thus, for each actuator, six state variables are introduced. A detailed description of the active yaw damper's mathematical model is reported in [10] together with a validation based on comparisons with laboratory measurements.

5. Numerical results along a straight track and along a curved track

Using the mathematical model described in section 4, several numerical simulations were performed to assess the performances of a railway vehicle equipped with the active yaw dampers. The obtained results were compared to those of a standard passive vehicle.

5.1 *Evaluation of the active yaw damper performances on a straight track*

In order to numerically determine the stability threshold of the vehicle, the non-linear vehicle model described in section 4 is used to reproduce the tests performed on the line. The stability threshold is determined by repeating the vehicle ride simulations at increasing speeds until the RMS value of the lateral acceleration of the bogie exceeds a reference value assumed equal to $4\,\mathrm{m/s^2}$. In the simulations, severely worn wheel profiles were considered together with rail profiles measured on the line in some of the most critical sections. The c_v control gain was determined through numerical simulations in order to obtain the best stability performances in any working condition ($c_v = 200{,}000\,\mathrm{N\,s/m}$). During such simulations, the deformability of the connection to the carbody was neglected. The m_v control gain, instead, was assumed to be equal to zero (the m_v control gain was tuned during line tests as described in section 6.1).

In figure 6(a), the results obtained for the passive vehicle arrangement are compared with different configurations of the actively controlled vehicle:

– active yaw damper placed in the longitudinal direction, low stiffness of the connection to the carbody (k_c value in the legend);
– active yaw damper placed in the longitudinal direction, high stiffness of the connection to the carbody;
– active yaw damper placed in the transversal direction, low stiffness of the connection to the carbody,

Comparing the results obtained for different arrangements with the longitudinal positioning of the actuators (arrangement used during in-line tests), it can be seen that the stiffness of the connection plays a very important role: using a low stiffness, the performances of the actively controlled vehicle are comparable to those of the passive one. The reason is that, because of the relatively large deformation of the connecting flange, the signal measured by the velocimeter

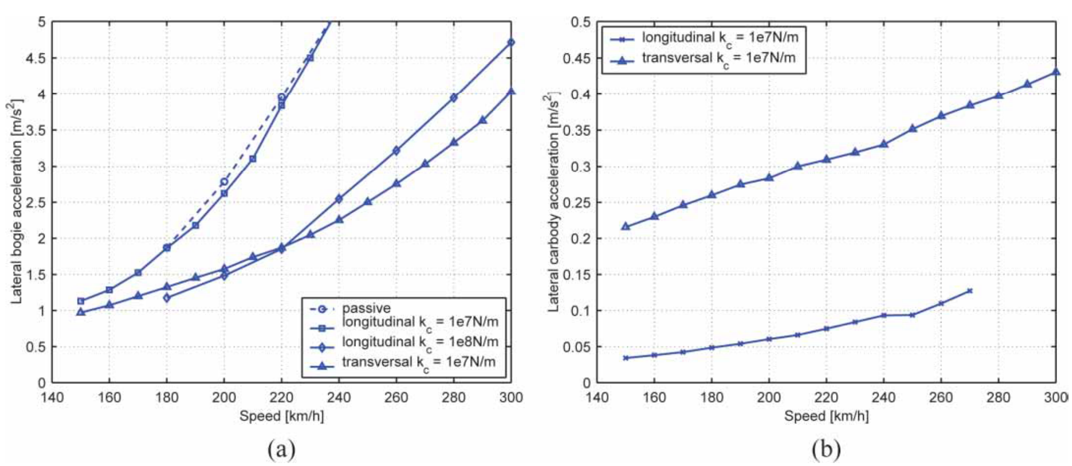


Figure 6. RMS value of the lateral bogie acceleration (a) and lateral carbody acceleration (b) for different vehicle arrangements.

and used to define the reference force is much lower than the actual relative velocity between the carbody and the bogie. Instead, if a stiffer connection is used, an increase in the critical speed of ∼40 km/h can be obtained with the active device.

Figure 6(a) also allows us to draw some conclusions about the optimal positioning of the actuators on the vehicle: the transversal positioning allows us to greatly increase the vehicle stability threshold, even with a low stiffness of the connection. This is due to the fact that the motion of the bogie during hunting instability is a combination of lateral and yaw oscillations. Therefore, for a given amplitude of the hunting motion, the energy dissipation is higher when the damper is placed in the transversal direction. In fact, in this case, both the lateral and yaw speed of the bogie contribute to the relative speed v_{rel} used as feedback signal by the active damper. However, the positioning of the active damper in the lateral direction leads to a higher RMS value of the lateral acceleration of the carbody, which means reduced passengers' comfort. This is shown in figure 6(b), where the RMS value of the lateral carbody acceleration is plotted versus speed for the lateral and longitudinal positioning of the actuator.

In conclusion, the transversal positioning of the actuator seems to be more effective from the point of view of vehicle stability but needs to be carefully studied with respect to comfort issues. These results also show that the integration of the active yaw dampers with an active lateral suspension of the carbody could be very promising.

5.2 *Evaluation of the active yaw damper performances on a curved track*

In agreement with the considerations done in section 3.2, the evaluation of railway vehicle's performances along a curved track is based on the examination of the lateral components of the contact forces.

Figure 7 shows a graphical representation of the adopted lookup table for the front (7a) and rear (7b) axles of the front bogie. Figure 8 shows the time histories of track shift forces for the two axles of the leading bogie during a curve negotiation, with radius equal to 800 m and non-compensated lateral acceleration equal to 1.8 m/s^2.

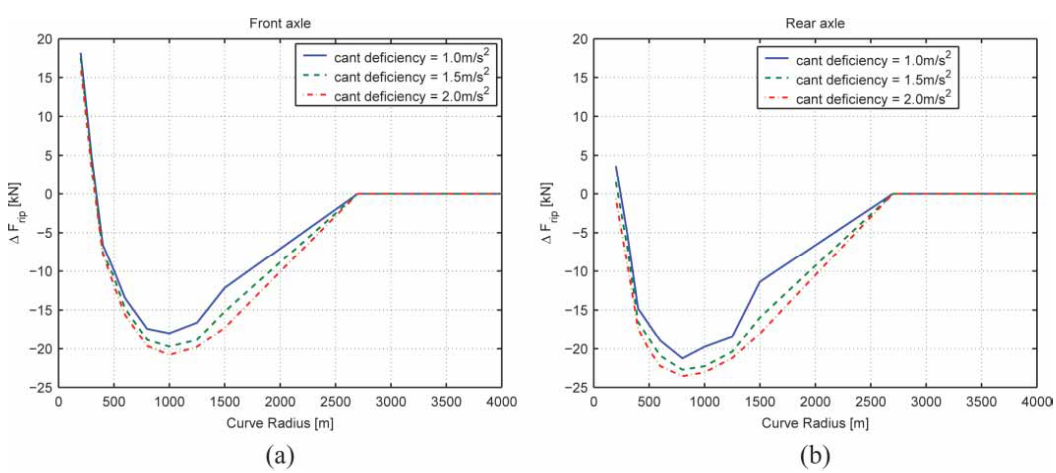


Figure 7. Graphical representation of the adopted lookup table for the front (a) and rear (b) axles of the front bogie.

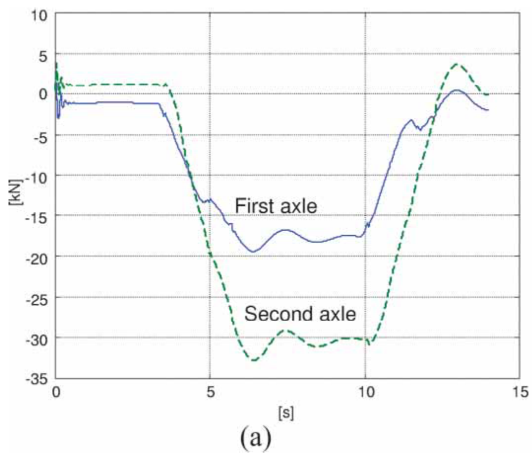


Figure 8. The simulated ripage forces on the front bogie ($v = 46\,\mathrm{m/s}$, $R = 800\,\mathrm{m}$, $a_y = 1.8\,\mathrm{m/s^2}$) with traditional oil dampers (a) and the active steering device (b).

This figure shows the capabilities of active control on vehicle's curving performances: for the standard passive vehicle arrangement (8a), the unbalance of track shift forces between leading (first) and trailing (second) axles is clearly visible. By the use of active control (8b), this unbalance can be almost completely compensated, thus reducing the maximum value of the track shift force by $\sim$8 kN.

In order to obtain a more general picture of the effectiveness of active control in curving conditions, several simulations were performed using different combinations of curve radius, cant deficiency, wheel–rail profiles and track irregularity. For each simulation, the maximum value of track shift force (averaged over a 2 m long distance of a full curve) is reported in figure 9 as a function of the non-compensated lateral acceleration. According to UIC 518 standard, a linear regression is used to define the maximum allowable lateral acceleration. It is clearly visible that the value of the track shift force obtained by the linear regression method for the actively steered vehicle at a lateral acceleration of $2.5\,\mathrm{m/s^2}$ (43.3 kN) is much lower than that for the passive vehicle (51.2 kN).

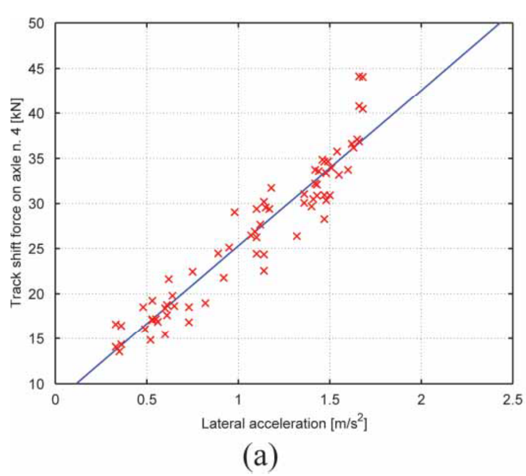


Figure 9. Correlation between lateral acceleration and maximum track shift forces with traditional oil dampers (a) and the active steering device (b).

6. Line tests in tangent and curved tracks

Together with Alstom Ferroviaria and Trenitalia, an experimental campaign was carried out on the Italian high-speed line 'Direttissima' and a standard line to test the AASA devices in real working conditions. The test train (an ETR470 Pendolino) was composed of three vehicles, one of them having the traditional oil dampers replaced by four AASA devices. In order to obtain a low critical speed of the vehicle under test, the wheels were re-profiled, reproducing a severely worn profile with a high conicity.

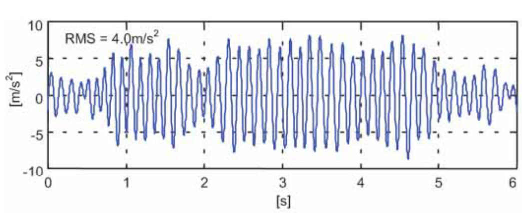
The experimental campaign has been divided into two parts: a first phase during which the control logic in straight track running was tested (changing the values of the equivalent damping c_v and mass m_v at increasing vehicle speed) and a second phase during which the steering logic was checked without actually applying the determined steering torque because of safety reasons.

6.1 *Tangent track tests*

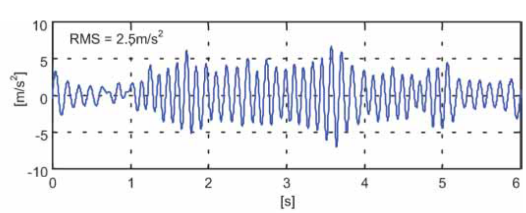
The tests performed on the high-speed line were dedicated to the evaluation of the increase in the vehicle's critical speed because of the use of the AASA devices instead of traditional oil dampers. The control gains were set to the following values: $c_v = 200{,}000\,\mathrm{N\,s/m}$ and $m_v = 500\,\mathrm{kg}$. As already said, m_v value was tuned during line tests to obtain the best vehicle stability performances.

To this end, particularly relevant is the time history and the RMS value of the lateral bogie acceleration over one axlebox. Figure 10 shows the time history of the lateral bogie acceleration, at the same speed (210 km/h) and the same position along the line, for the two vehicle arrangements (passive and active). It can be observed that the use of the active yaw damper leads to a reduction in the amplitude of bogie lateral vibrations (RMS value of $2.5\,\mathrm{m/s^2}$ against $4.0\,\mathrm{m/s^2}$), but the presence of a limit cycle cannot be completely suppressed.

Table 1 shows the RMS values of the lateral bogie acceleration at different speeds and positions along the line, allowing us to compare in more detail the performances of the vehicle in passive and active configurations. The controlled vehicle, when compared with the passive one, shows a 20% average reduction of the lateral vibration. These results are quite encouraging, considering the fact that it was the first time the active set-up was tested in-line. However, the active device is not able to completely suppress instability. As already pointed out, in order to obtain better performances, the effect of local deformability in the connection between the actuator and the carbody has to be correctly taken into account or, as an alternative, a different positioning of the actuators (*e.g.* transversal instead of longitudinal) should be considered.



(a)



(b)

Figure 10. Measured lateral bogie acceleration (time history) with traditional oil dampers (a) and the active yaw damper (b).

Table 1. RMS values of the lateral bogie acceleration: comparison between the passive and active vehicle arrangements.

Speed (km/h)	RMS acceleration passive dampers (m/s^2)	RMS acceleration active dampers (m/s^2)	Reduction (%)
200	2.78	2.45	12
200	2.11	1.64	22
200	2.36	1.58	33
210	3.47	2.98	15
210	3.29	2.76	16
210	2.54	1.95	23

6.2 *Curved track tests*

Line tests on the standard (not high speed) line were carried out to assess the performances of the controlled vehicle during curve negotiation. At this stage of the research, the test train was run on the standard line, with the control system (real-time board) activated to determine the reference forces but with the actuators switched off. Therefore, all the outputs of the control system could be acquired, but the vehicle was actually running with a passive configuration. Despite this fact, the tests allowed us to quantify the accuracy and robustness of the logic with respect to the estimation of curve radii and to indirectly evaluate the performances of the active control, as explained in the following sections.

The capabilities of the proposed methodology for curve radius estimation is displayed in figure 11: the estimated curve radius (solid line) is compared, at some points, with the nominal curve radius values (in bold) known from the track geometry. The maximum difference between the estimated curve radius' values and the real ones is <25 m.

An estimate of track shift forces for the controlled vehicle can be derived from the actual measurements of track shift forces (in passive configuration, as mentioned earlier). In fact, assuming that the force applied by the actuators would be exactly the same as the reference one and the application of a yaw torque by the AASA actuators would not change the longitudinal wheel–rail contact forces, it is possible to determine the time histories of the track shift forces for a 'virtually' controlled vehicle by adding to the measured track shift forces the reference forces estimated by the real-time board. Figure 12 allows us to compare the actual track shift forces (12a) with the ones obtained by the described post-processing (12b). Within the earlier cited assumptions, it can be stated that the actuators always allow us to reduce the unbalance

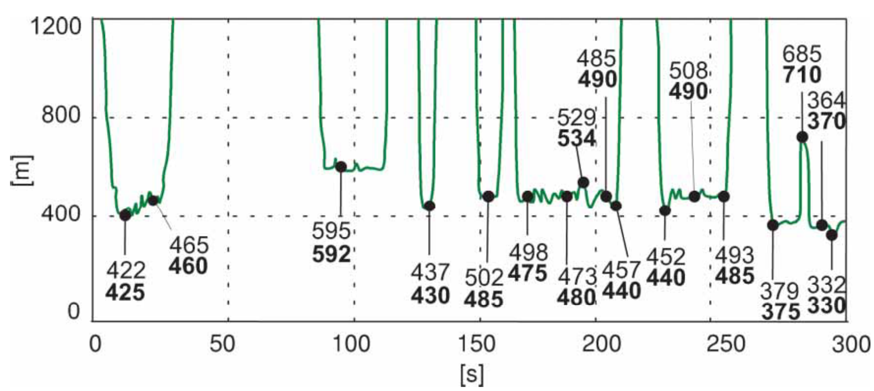


Figure 11. Time history of the estimated curve radius; comparison between some estimated (normal) and corresponding tabulated (bold) curve radius values.

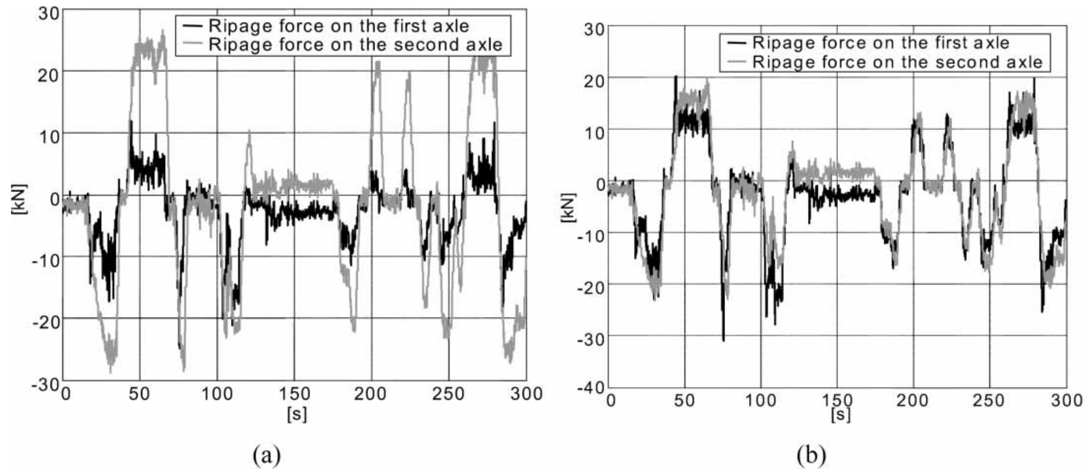


Figure 12. Time history of the ripage forces on the first and second axles of the vehicle with traditional (a) and 'virtually' controlled (b) yaw dampers (law pass filtered, $f_{cut} = 10\,\text{Hz}$).

of track shift forces and, in many cases, the force distribution is very well balanced, with significant decrease in the maximum value of track shift forces.

7. Conclusions

An application of active control concepts to increase the running performances of a railway vehicle has been presented. The proposed active device is based on an electro-mechanical actuator, and strategies for vehicle dynamics control in tangent track and curve have been developed.

The applicability and effectiveness of the proposed solution has been investigated by means of in-line tests and numerical simulations. The results obtained show the possibility to significantly raise the critical speed of the vehicle and to negotiate curves at higher values of cant deficiency. At the present stage of research, further experimental investigations are needed to optimise the set-up of the actuator, with particular reference to the connection with the carbody. Another subject for further research is the optimisation of the actuator positioning and its integration with other mechatronic devices like active lateral suspensions.

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