Running dynamics of railway vehicles equipped with torsionally flexible axles and partially independently rotating wheels

A. Bracciali, G. Megna
Department of Industrial Engineering, University of Florence, Italy

ABSTRACT: The paper discusses the fully non-linear behaviour of railway vehicles whose axles are considered as torsionally flexible. Critical speeds of the ERRI Wagon available in the library of the selected multibody software are investigated with different axle stiffness in order to evaluate any possible reduction in the dynamic characteristics at high speed. All these analyses are applicable to a novel wheelset arrangement where individually supported wheels are connected by a torsionally flexible shaft where the wheels can be easily changed and the axle subjected to rotational bending is replaced by two fixed stub axles. In a further development, the connecting shaft is equipped with a torque limiter. Steering and stability are discussed in order to identify any potential drawbacks in this arrangement.

1 INTRODUCTION

Independently rotating wheels (IRWs) are often used in trams and in all vehicles that require a low floor arrangement. This layout is very interesting as it allows to access the wheels for maintenance without the need to lower the wheelset as in the case of conventional wheelsets. Nevertheless, the absence of the torsional constraint between the wheels leads to premature wear of wheel flanges, as the bogie tends to run skewed and with one or more wheel flanges in continuous contact with rail gauge corner. The gravitational stiffness proves to be insufficient to restore the central position of the wheelsets.

This evidence, clear already in the ‘70s of the last century (Becker, 1970, Dukkipati, 1992, Wickens, 2006), explains why no IRW-equipped vehicles are used in conventional railways.

In order to overcome the drawbacks of the classical IRW arrangement while allowing a dramatically improved maintainability of conventional vehicle wheelsets, the fully passive *AIR Wheelset* (Apparently Independently Rotating Wheels Wheelset) was developed and patented (AB Consulting, 2015).

It consists of two wheels supported on hollow supports by a specific bearings arrangement which is able to withstand both lateral and vertical loads acting on each single wheel. This function is critical as the absence of a conventional axle does not provide any equalization of the lateral forces acting on the two wheels. The design makes use of bearings recently developed for inboard bearings high-speed vehicles.

Two versions of the *AIR Wheelset* are available, a motor one and a trailed one. In both arrangements the wheels are connected by a shaft passing through the hollow supports. In the case of the trailed *AIR Wheelset*, the connection can be made through friction torque limiters that allow finite rotations between the wheels in case the set torque limit is exceeded. A comprehensive description of the *AIR Wheelset* can be found in (Bracciali, 2015a), from which Figure 1 is extracted.

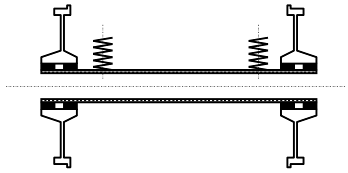
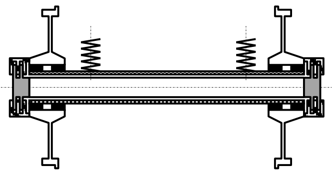
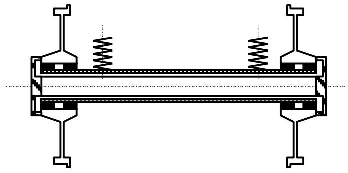
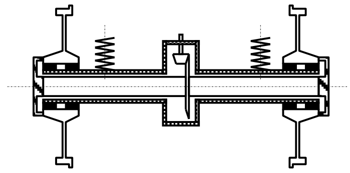
Maximum torque	Trailed *AIR Wheelset*	Motor *AIR Wheelset*
Zero (free)		Not applicable
Limited by torque limiter		Not applicable
Rigid joint Limited by wheel-rail adhesion limit		

Figure 1. Possible arrangements of the *AIR Wheelset* (from Bracciali, 2015a).

As long as the connecting shaft introduces a finite torsional stiffness which can be much lower than that of a conventional axle, in this paper a fully non-linear analysis is conducted in order to evaluate the dynamic behaviour of a vehicle equipped with the torsionally flexible axle in tangent and curved track. The behaviour of the vehicle when equipped with the torque limiter is analysed as well, in order to highlight and inadmissible worsening in the running characteristics of the vehicle.

In order to get reproducible results based on a well-known and established vehicle model, all the simulations were performed using the software package VI-Rail v. 16.0 by using as a basis the “ERRI wagon” library vehicle. It is a passenger car equipped with anti-yaw dampers, two bogies and a total of four wheelsets.

2 COMPARISON PARAMETERS AND ASSESSMENT CRITERIA

The correct comparison between different vehicle solutions requires the definition of the parameters to be considered, such as displacements, forces, critical speed, etc.

In the present research it was chosen to use the criteria defined in the European Standard on the acceptance of running characteristics of railway vehicles (EN14363, 2005), that is used to assess running safety and track fatigue of almost all vehicles, either conventional or tilting.

As long as the wheel arrangement analysed in this paper does not affect the static behaviour of the vehicle in terms of response to track twist (vertical unloading of the wheels), the part relative to this item (chapter 4 of the EN14363 standard) is not considered further here.

The following quantities were therefore considered:
- critical speed in straight track;
- track shifting forces ΣY in straight track, for both low and high speed with the appropriate track defectiveness, to be compared to the Prud’Homme limit;
- track shifting forces ΣY and quotient Y/Q of the leading wheel (to be compared to the Nadal limit) in curves with alignment defects.

A low pass filtering at 20 Hz and a sliding mean over 2 m and 0.5 m step were applied to the forces resulting from simulations. The 99.85 and 0.15 percentiles of track shifting forces and *Y*/*Q* ratio were then compared to the corresponding limit values.

The basic assumption is that the original vehicle (ERRI Wagon) is fully compliant with the requirements of the standard, so the comparison was focused on the differences between the new designs and the library vehicle.

It is worth to highlight that, obviously, all the results shown in this research are vehicle dependent, in the sense that they may sensibly change if the vehicle is markedly different. This may be particularly true for the effect of the torque limiter, that should be carefully tuned on each vehicle characteristics.

3 THE EFFECTS OF AXLE TORSIONAL FLEXIBILITY

3.1 *Introduction*

It is in known from the literature that the instability of a vehicle increases by decreasing the torsional stiffness of the axle and that therefore unstable conditions can be reached at much lower speeds.

The first papers on the subject are those published by (Hadden, 1977) and by (Doyle, 1977). In these works the dynamics of the vehicle is derived starting from the equations of dynamic equilibrium considering all the effects as linear, both those arising from the vehicle architecture and from the wheel-rail contact. Eigenvalue analysis led to the conclusion that the first natural frequency decreases not linearly reducing the axle torsional stiffness (Figure 2).

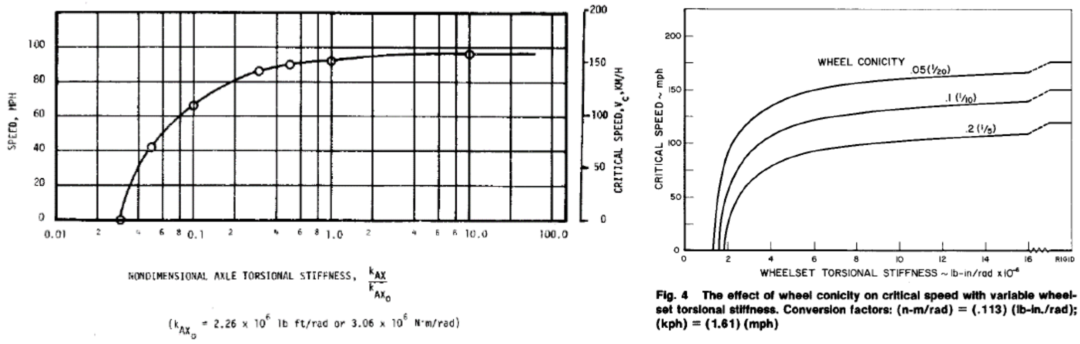


Figure 2. Effect of axle torsional stiffness on critical speed on nominal 11-DOF model (left, Hadden 1977). Effect of axle torsional stiffness on critical speed with varying wheel conicity (right, Doyle 1977).

Although very interesting, these papers were only much later complemented by the work from (Ahmed, 1989), that analysed also eigenmodes (mode shapes), showing that that instability is *not* a classical bogie (or truck, in the American terminology) hunting, but a *primary hunting* or *carbody instability* (Figure 3). Against this instability, which is characterized by low frequencies and by lateral movements which typically do not saturate the lateral clearance between the wheel flange and the railhead, damping of the secondary suspension is particularly effective. This is in fact the approach that was selected in this research, trying to achieve the advantageous characteristic shown in Figure 3 that shows that critical speeds related to bogie instability increase by decreasing the torsional stiffness of the axle.

After the review paper (Dukkipati, 1992), no other relevant contributions can be found in the last twenty-three years.

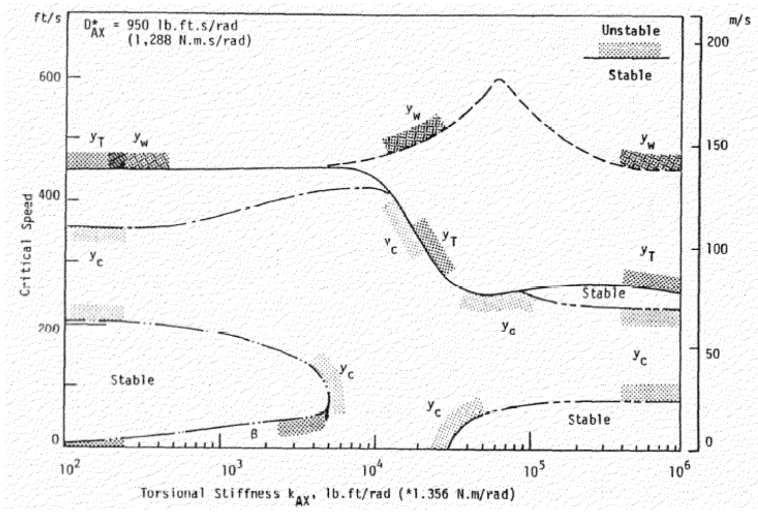


Figure 3. Critical speed areas as a function of torsional stiffness of the axle for a given value for damping (Ahmed, 1989). The lower right area corresponds to the curve shown in the previous figure.

3.2 *Critical speed of the ERRI Wagon with torsionally flexible axle on perfect straight track*

As already mentioned, the analysis of the stability of a vehicle equipped with torsionally flexible axles is based on the ERRI wagon (Figure 4) by introducing first a finite stiffness between the wheels, corresponding to a relatively common axle diameter, and then by introducing a longer axle, which is more similar to the axle that is used in the arrangement of the new design wheelset (Bracciali, 2015a).

Rigid bodies mass [kg]		
Bogie Frame	Wheelset	Carbody
2615	1503	32000
Axle loads [kN]		
Front Bogie	Rear Bogie	
111	108	
Main dimensions [mm]		
Bogie Wheelbase	Pivot Pitch	
2560	19000	

Figure 4. Main features of the ERRI wagon.

The analysis of the stability of the ERRI wagon is done in this chapter by carrying out simulations on perfect track without torque limiters that will be introduced in the next chapter. All the analyses were performed for a vehicle with new S1002 wheel tread profiles running on a track with 60E1 rails inclined by 1:40 and standard track gauge of 1435 mm. Adhesion limit was set to μ=0.4 as a normal practice in vehicle dynamics simulation.

The vehicle was modelled with the software package VI-Rail v. 16.0 first to generate the reference value for the critical speed on perfectly straight track. Fully non-linear critical speed with torsionally flexible axles only were then calculated with an integration time step of 10^{-3} s.

In order to verify the critical speed of the reference vehicle when the torsional stiffness of the axles is not infinite, a specific template was developed by the software supplier (VI-grade GmbH) to introduce an arbitrary torsional stiffness between the wheels.

Reminding that the torsional stiffness of a round bar is $k_t = G\,Jp\,/\,l$, where $G=E/(2(1+\nu))$ and $Jp=\pi/32\,(D^4\text{-}d^4)/D$ where E=Young's modulus (=206 GPa for steel), ν=Poisson's ratio (=0.29 for steel), l=length of the axle subjected to torque and D,d=outer and inner diameter of the axle, several possible axles were used in stability simulations. A "rigid" axle with very high torsional stiffness was also considered to verify that it gives the same critical speed of the standard VI-Rail wheelset and to check the model reliability.

Instability was induced starting the simulation from very high speed, then the speed was reduced in steps of 5 m/s, maintaining each step for a few seconds to identify whether oscillations were decreasing or not. The speed at which the lateral oscillations of the wheelset stop is identified as the maximum stable speed.

Figure 5 shows the simulated axles and the critical speeds obtained with all wheelsets of the ERRI wagon equipped with the same axles. A non-negligible difference between the critical speeds of a rigid wheelset and a conventional one with finite torsional stiffness can be observed (90 m/s vs. 85 m/s). The critical speed for axles with torsional stiffness lower than 1 MNm/rad is very low, but considering that no flange contact occurs and the low frequency of the wheelset lateral oscillation involved, this motion is a car body instability (swaying). The hunting critical speed related to bogie instability decreases, but its reduction is rather small.

						c= original ERRI wagon anti-yaw dampers				
						Flange contact		Non-flange contact		No motion
Axle type	**D**	**d**	**l**	**k_t [Nm/deg]**	**k_t [MNm/rad]**	**$v_{crit,\ min}$ [m/s]**	**f_{crit} [Hz]**	**v_{sway} [m/s]**	**f_{sway} [Hz]**	**$v_{stable,\ max}$ [m/s]**
Standard VI-Rail package				∞	∞	90	4.8	/	/	85
0: "Rigid"				10^9	57000	90	4.8	/	/	85
1: Conventional	160	0	1500	59774	3.42	85	4.6	/	/	80
5: Conventional hollow	160	80	1500	56038	3.21	85	4.6	/	/	80
2: Long	160	0	2000	44831	2.57	85	4.6	80	3.8	75
6: Long hollow	160	80	2000	42029	2.41	85	4.6	80	3.3	75
3: Long and thin	140	0	2000	26279	1.51	80	4.4	75-65	3.3-3.1	60
7: Long thin hollow	140	80	2000	23477	1.35	80	4.4	75-65	3.5-2.9	60
4: Long and very thin	120	0	2000	14185	0.81	75	4.2	70-25	3.5-1.7	20
8: Long very thin hollow	120	80	2000	11383	0.65	70	3.7	65-15	3.3-1	10

Figure 5. Critical speed as a function of axle torsional stiffness, sorted in descending stiffness order.

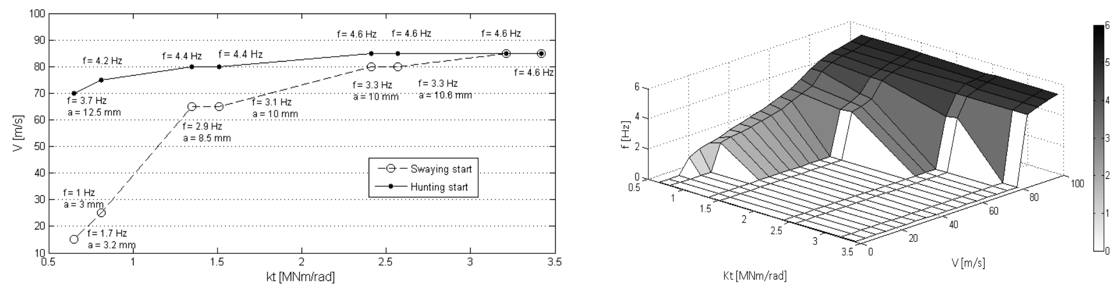


Figure 6. Limits of stable regions for the first wheelset of the ERRI Wagon with frequencies and peak-to-peak amplitudes of oscillation (left); frequencies of lateral oscillations of the first wheelset (right).

3.3 *Running safety of the ERRI Wagon with torsionally flexible axle on defective straight track*

Running safety was evaluated according to EN 14363 on defective straight track. Track geometry defects are statistically described according to (ERRI B176 RP1, 1989). Vehicle dynamics is conventionally studied with “large amplitude track defects” at slow speed (v ≤ 160 km/h) and with “small amplitude track defects” at high speed (v > 160 km/h). This makes sense considering that high speed tracks are normally better maintained than conventional tracks.

The resulting track shifting forces on both wheelsets of the first bogie are compared to the Prud’homme limit, that according to EN 14363 is $\Sigma Y = k_1 * (10+2Q_0/3) = 1*(10+111/3) = 47$ kN, and plotted vs. the running speed in Figure 7. As long as syawing movements do not lead to high lateral forces, these diagrams consider in practice only bogie hunting where large track shifting forces are obtained. This explains why these simulations are in good agreement with the trend observed for bogie stability on perfect track.

It can be observed that the use of torsionally flexible axles leads to degraded safety conditions, in the sense that the Prud'homme limit is reached at lower speeds with the same vehicle. For example, the first wheelset reaches the limit between 80 and 90 m/s with torsionally rigid axle and between 60 and 70 m/s with torsionally flexible axle.

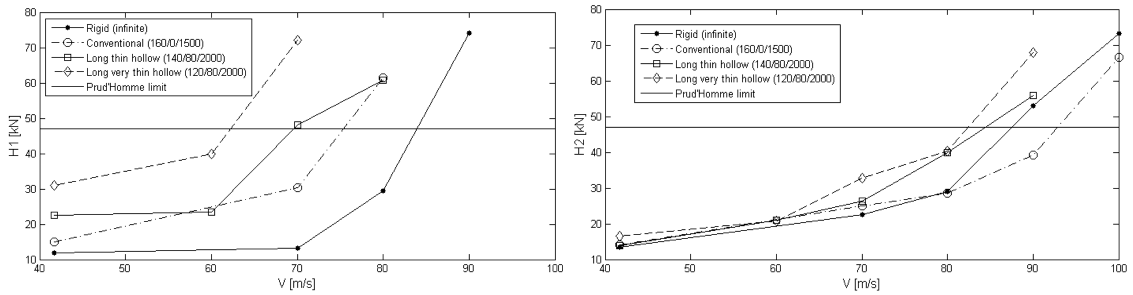


Figure 7. Track shifting forces of front wheelset (left) and rear wheelset (right).

3.4 *Effect of anti-yaw dampers on stability of the ERRI Wagon with torsionally flexible axle*

The reduction of hunting critical speed due to the decreasing of torsional stiffness can be compensated by increasing the anti-yaw damping coefficient. A comparison of the damping behaviour of different anti-yaw dampers and their effect on stability of the ERRI wagon is shown in Figure 8.

It can be seen that the original ERRI wagon dampers have a much softer behaviour at low speeds while their effect increases noticeably with speed. Other dampers, used in high-speed trains, are much "harder" at low speeds and their behaviour is almost saturated at higher speeds.

Stability speed, intended in this case as the maximum speed in intervals of 5 m/s that can be run without oscillations, is plotted in Figure 8 vs. axle torsional stiffness. An additional case of two original ERRI dampers in parallel was considered, similarly to the practice of some modern high speed trains where two anti-yaw dampers in parallel per side are installed. It can be seen that for torsionally very soft axles, stability against bogie hunting can be achieved at very high speed only with the harder dampers.

An undesired feature, not shown below, is that at speeds lower than 70 m/s a further carbody swaying movement appears with axles with very low torsional stiffness (< 1 MNm/rad). It is stressed the concept that this behaviour is vehicle-dependent and that different results may be obtained in other situations. This nevertheless proves that bogie hunting can be effectively tackled by using appropriate anti-yaw dampers without major modifications to an existing vehicle and carbody swaying can be avoided in any case by using a torsional stiffness greater than 1 MNm/rad.

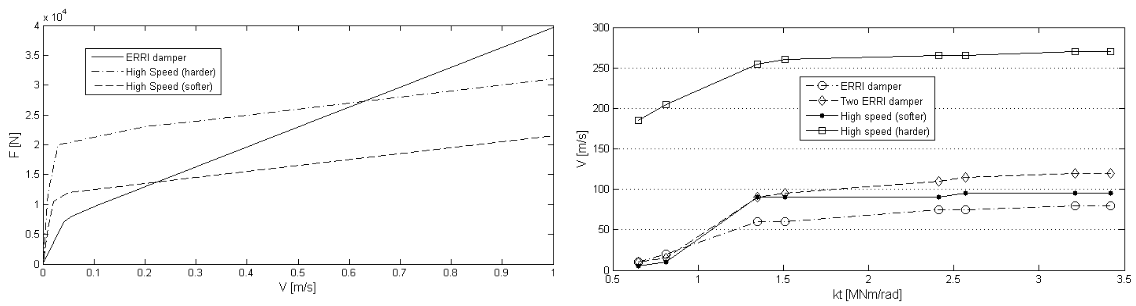


Figure 8. Behaviour of different types of anti-way dampers used in the calculation (left). Stability speeds of the first wheelset (right).

3.5 *Effect of anti-yaw dampers on running safety of the ERRI Wagon with torsionally flexible axle*

The effect of modern anti-yaw dampers is beneficial also with respect to track shifting forces. As shown in Figure 9, the application of greater damping coefficients always leads to lower track shift forces for a given speed or, similarly, to higher speeds to reach the Prud'Homme limit. As a result, changing the anti-yaw dampers of the ERRI wagon leads to a more stable and more track friendly vehicle, increasing running safety.

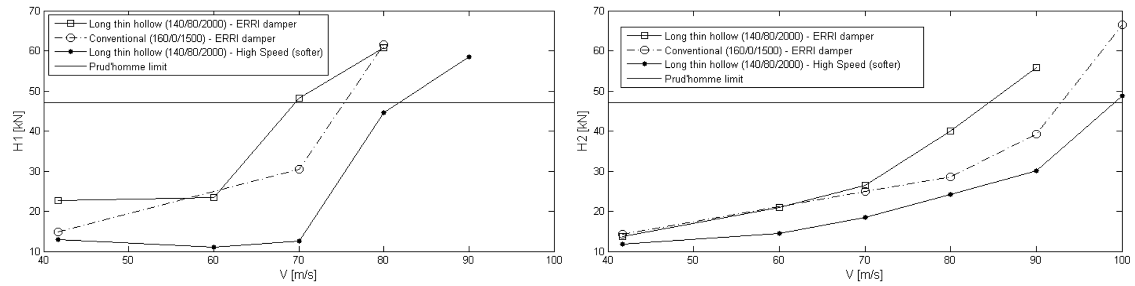


Figure 9. Track shifting forces for the first (left) and the second (right) wheelset.

3.6 *Curving behaviour of the ERRI Wagon with torsionally flexible axle*

The curving behaviour of the torsionally flexible axle is identical to that of the torsionally rigid axle, with very limited differences during the transition. This is obvious considering that the vehicle with torsionally flexible wheelsets negotiates the curve exactly as a conventional vehicle does, with the only difference that the two wheels of the same wheelset will roll with a small constant angular difference due to axle flexibility.

The equation that links speed v, a_{nc}, R and h is

$$a_{nc} = \frac{v^2}{R} - \frac{gh}{s} \tag{1}$$

Normally the highest non-compensated acceleration is set to a_{nc}=1 m/s^2, corresponding to a cant deficiency of 153 mm; as the maximum superelevation is in the same order, for a train standing in a curve with such superelevation the resulting cant excess is 153 mm, i.e. a_{nc}=-1 m/s^2.

In this work three basic conditions were therefore considered, i.e. the maximum (a_{nc}=1 m/s^2), nought (a_{nc}=0 = equilibrium) and minimum (a_{nc}=-1 m/s^2) non-compensated acceleration on six curve radii (R= 300, 548, 1000, 1430, 2000, 3300 m). Resulting speeds are shown in Table 1. They cover the full range of operations of conventional and high-speed vehicles.

Table 1. Running speeds [km/h] for simulated kinematic conditions (h=160 mm).

Curve radius [m]	a_{nc}=-1 m/s^2	a_{nc}=0	a_{nc}=+1 m/s^2
300	13	64	89
548	18	86	121
1000	25	116	163
1430	29	139	195
2000	35	165	230
3300	45	212	296

The stationary behaviour of the vehicle on curved track was studied, checking the derailment ratio Y/Q of the external wheel of the first wheelset, the track shifting forces H=ΣY of all the wheelsets, the angle of attack of the first wheelset, the attitude angles of both bogies and the longitudinal forces on all wheels and the resulting torques. These results are not shown here as they were only computed as the basis for the comparison with the wheelset equipped with torque limiter (see next chapter).

3.7 *Torques acting on the axle and curving behaviour of the torsionally flexible axle*

The torque M transmitted through the axle can be estimated from the longitudinal forces X_{i1} and X_{i2} acting at the wheel-rail contact of both wheels of the i-th wheelset. These two forces have an average value $X_{i,av}= (X_{i1} + X_{i2})/2$, which is the resistance to rolling and rotationally accelerates the whole wheelset, while the semi-difference $\Delta X_{i,av}= (X_{i1} - X_{i2})/2$ generates both the torque on the axle $M_y= \Delta X_{i,av}\, r$, where r is the wheel radius, and the steering torque $M_z= \Delta X_{i,av}\, s$, where s is the wheelset gauge.

Torques acting on front and rear axles of the front bogie are shown in Figure 10. The trend of the curves can be justified by observing that running on a curved track with small radius results in the first wheelset negotiating the curve with the highest angle of attack independently from the a_{nc} value, saturating the adhesion with high lateral forces and leaving therefore small room for longitudinal forces. The front wheelset is therefore laterally displaced towards the high rail and the rear wheelset almost runs centred. For higher curve radii, the first bogie tends to run less skewed although tangential forces remain important.

This trend strongly depends on the adhesion coefficient that, as mentioned, was set to μ=0.4 according to the practice used in vehicle dynamics simulation and assessment. Other analyses will be shown in a companion paper which focuses on the effect of different wheel-rail contact conditions on the torque transmitted through the axle (Bracciali, 2015b) and will not be described here further. It can be anticipated that the diagram trend changes noticeably for low curve radii and high adhesion coefficients, a rather common situation in metros.

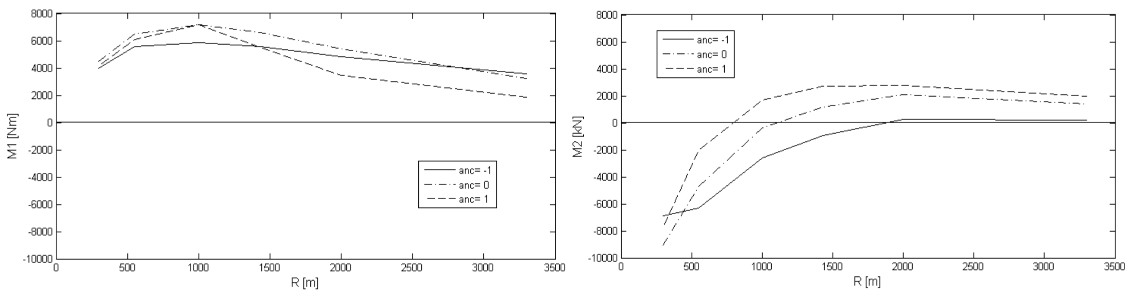


Figure 10. Torque acting on the first (left) and the second (right) axle of the front bogie of the ERRI wagon for a_{nc}=-1, 0 and 1 m/s^2. Coefficient of friction μ=0.4.

3.8 *Running safety of the ERRI Wagon with torsionally flexible axle on defective curved track*

While the stationary behaviour is independent from the torsional flexibility of the axle, the dynamic behaviour is not, as shown in paragraph 3.3. So, the effect of torsional stiffness was studied on a curved and defective track. Defects amplitude were applied as for tangent track. The same curve radii of the previous paragraph were considered for only a_{nc}= 1 m/s^2. The values of track shifting forces ΣY and Y/Q ratio remain below the safety limits for all cases, except for the curve with R = 3300 m. This is quite reasonable considering that in this case the speed is around 300 km/h, i.e. greater than the critical speed for almost all axles. Therefore, an increase of anti-yaw damping coefficient is necessary to allow the vehicle to run safely. Maximum values of the post-processed signals of ΣY and Y/Q are plotted in Figure 11 for two types of axle (the most flexible and the most rigid) and three types of anti-yaw dampers (ERRI dampers and both high-speed dampers).

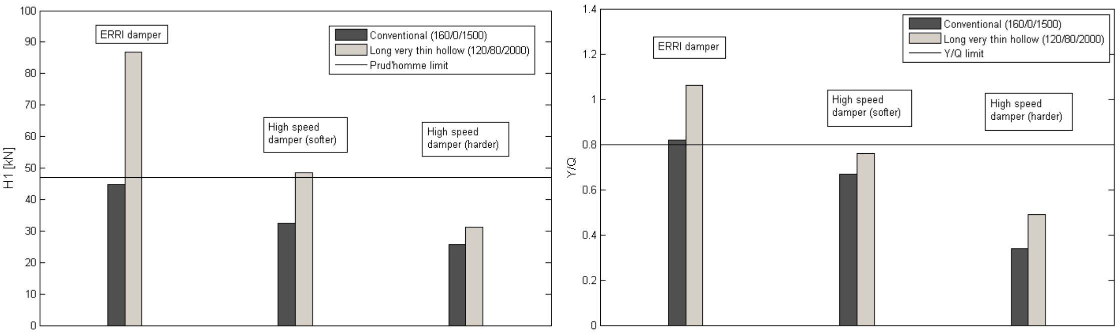


Figure 11. Track shifting forces (left) and Y/Q ratio (right) for the first wheelset.

4 THE EFFECTS OF TORQUE LIMITER

4.1 *Introduction*

Considering now the *AIR Wheelset* equipped with friction torque limiter, two main effects arise when the torque limit M_{lim} is exceeded: the wheels are capable to perform finite relative rotations and the longitudinal forces at the wheel-rail contact are limited. To investigate the consequences of these effects on the dynamic behaviour of the vehicle on both tangent and curved track, a limit value for the torque of 4000 Nm was chosen and implemented in the vehicle model resulting in a maximum longitudinal force $X_{max}= M_{lim}/r= 4000/0.46 \cong 8700$ N.

This arbitrary value used for the setting of the torque limiters may be changed according to the service operated by the vehicle (Bracciali, 2015b). This degree of freedom can be seen as a distinct advantage of the *AIR Wheelset*.

4.2 *Critical speed of the ERRI Wagon with torque limiter on straight track*

To investigate the stability of the torque limited wheelset solution, simulations on tangent track in presence of a localized defect were performed. The torque acting on the axle assumes large values, especially when the vehicle runs in proximity of the critical speed. Consequently, the longitudinal forces are limited by the intervention of the torque limiter. As the longitudinal forces are the cause of hunting movements, lower maximum values imply a better stability of the vehicle without affecting the self-steering capabilities at lower speeds. In fact, as shown in Figure 12, the lateral oscillations of the vehicle are characterized by frequencies lower than the case without torque limiter.

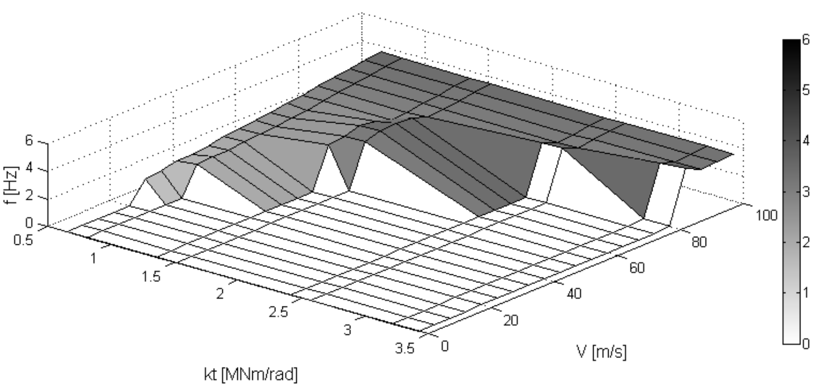


Figure 12. Frequencies of the lateral oscillation of the first wheelset equipped with the torque limiter.

4.3 *Running safety of the ERRI Wagon with torque limiter on defective straight track*

A running safety analysis on tangent track confirms that a large improvement in the maximum reachable speed can be obtained with the introduction of the torque limiter, as lower track shifting forces can be observed on both first and second wheelsets. Figure 13 show these values for a conventional axle type with and without torque limiter. The gain in the maximum speed slightly reduces if axles with lower torsional stiffness are considered, but the advantage remains notable.

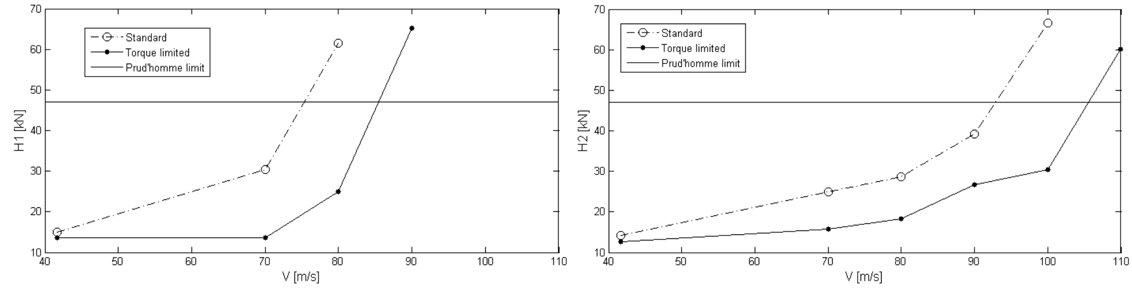


Figure 13. Track shifting forces of a standard wheelset and the torque limited wheelset. Both first (left) and second (right) axles are shown.

4.4 *Effect of anti-yaw dampers on stability of the ERRI Wagon with torque limiter*

The increase of the anti-yaw damping coefficient seen in paragraph 3.4 holds also for *AIR Wheelsets* equipped with torque limiter.

4.5 *Effect of anti-yaw dampers on running safety of the ERRI Wagon with torque limiter*

The reduction of the dynamic component of ΣY on wheelsets with torque limiter can be further increased by the use of more efficient anti-yaw dampers, as shown in paragraph 3.5.

4.6 *Curving behaviour of the ERRI Wagon with torque limiter*

As the trend of the torque acting on the axles is known (Figure 10), a limit value for the transmissible torque M_{lim} can be chosen depending on the geometry of the curved track on which the vehicle will run. Nevertheless, the specific trend for the front wheelset does not give the possibility to choose a value that let the torque limiter intervene only for small curve radii, similarly to the rear wheelset.

As already explained in the paragraph 3.7, more investigations considering also different values of adhesion coefficient are included in (Bracciali, 2015b), where the effect of reduction of the longitudinal forces is shown. It leads to higher reachable speed on tangent track but is a drawback on curved track as it limit also the steering capability of the bogie. Therefore, the stationary behaviour was compared to the results obtained in the paragraph 3.6 for a standard wheelset. As usual, particular attention was paid for ΣY and Y/Q, whose trend is reported in Figure 14.

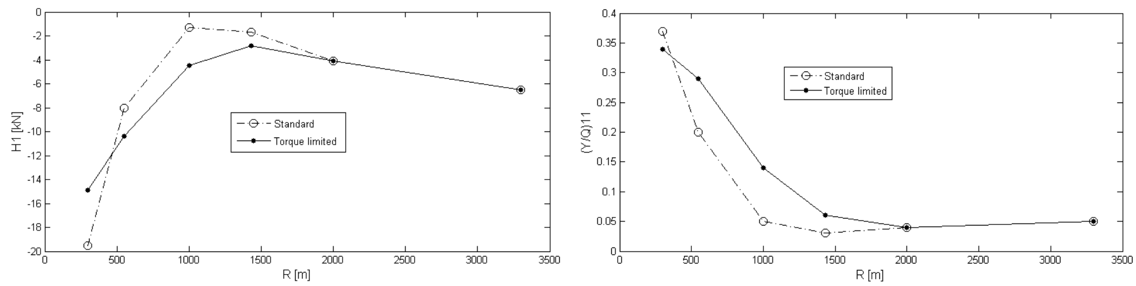


Figure 14. Track shifting force ΣY of the first wheelset (left) and Y/Q (right) of the external wheel of first wheelset for a vehicle running with a_{nc}=1 m/s^2.

As expected, slightly higher values can be found but the maximum values that occur for a curve radius of R=300 m are lower for the torque limited wheelset. This effect can be associated with the inability of a standard wheelset to provide enough steering in very sharp curves, while the *AIR Wheelset*, allowing the wheels to rotate independently, can perform a better curve approach.

4.7 *Running safety of the ERRI Wagon with torque limiter on defective curved track*

Although the increase in lateral forces on curved track is quite small for a vehicle equipped with torque limiter, it is important to evaluate ΣY and Y/Q also when the vehicle runs on defective track, as the torsional flexibility further influences running dynamics. As a special case, it was observed that the safety of the standard vehicle running with a_{nc}=1 m/s^2 on a curved track with R= 3300 m can be obtained only with anti-yaw dampers with the highest damping. For this reason a running safety analysis on curved track was repeated (Figure 15) in the case of an average flexibility axle.

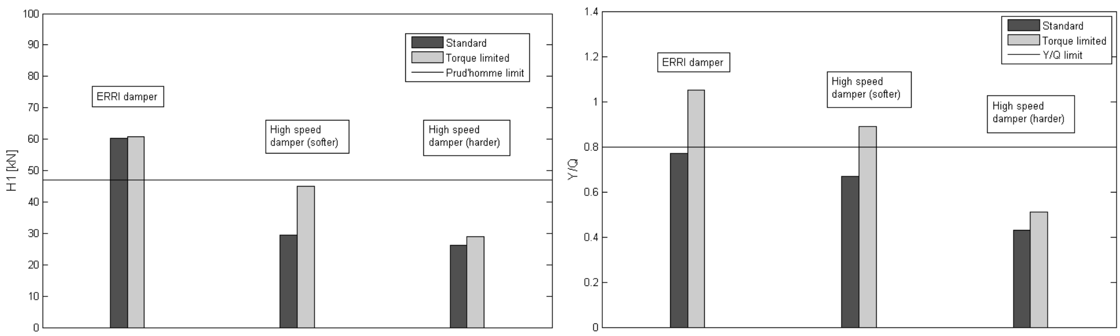


Figure 15. Comparison of ΣY of the first wheelset (left) and Y/Q of the first external wheel (right) for a standard wheelset and a wheelset equipped with torque limiter and an average flexible axle (D=140 mm, d= 80 mm, l=2000 mm).

5 CONCLUSIONS

The simulations described in this paper show how a novel wheelset arrangement where independent wheels are connected through a shaft of proper stiffness and the torque transmitted by the shaft is limited by suitable devices exhibits interesting behaviour in terms of vehicle dynamics when applied to a standard passenger car.

The full non-linear problem was studied on different track geometries, both ideal and defective. It was in particular shown that the use of a longer and smaller shaft connecting the wheels leads to no particular “derating” of the vehicle operation conditions. In fact, the forecasted reduction of critical speeds is not as large as expected and it can be easily compensated by using modern anti-yaw dampers. Swaying movement resulting by the application of the softest shafts can be easily tackled by the use of a shaft with torsional stiffness greater than 1 MNm/rad.

The effect of torque limiter was evaluated in a set of different kinematic conditions during runs in tangent and curved track. It is shown how wheel-rail contact forces change with the introduction of such torque limiter, affecting the running dynamics of the vehicle especially on tangent track, where great improvements in terms of stability and maximum reachable speed can be obtained. On the other hand higher lateral forces can be observed in curved track, but it was demonstrated how they do not compromise the running safety of the vehicle, as the track shifting forces and derailment ratio are always compliant to EN14363 standard provided that the proper anti-yaw dampers are used.

It can be concluded that the use of the *AIR wheelset* does not lead to any unacceptable worsening of the running safety of a passenger car. Although this result is vehicle-dependent and therefore cannot be generalized, the outcomes of the present research are encouraging the use of wheelsets with torque limiters and flexible axles. Their advantages in terms of maintainability and track friendliness will very likely compensate the larger simulation efforts needed to ensure safety of this new class of wheelsets.

REFERENCES

AB Consulting sas di Andrea Bracciali & C. 2015. Railway Wheelset With Partially Independent Wheels. World patent PCT/IB2015/051855.

Ahmed, A.K.W. & Sankar, S. 1989. An examination of railway vehicle stability pertaining to torsional flexibility of the wheelsets. *The 1989 ASME Design Technical Conferences, Montreal, Canada*.

Becker, P. 1970. On the use of individual free rolling wheels on railway vehicles. *Eisenbahn Technische Rundschau* 19, 11.

Bracciali, A. 2015a. Apparently Independently Rotating Wheelset - a possible solution for all needs?. *The Stephenson Conference - Research For Railways, 21 - 23 April 2015, Institution of Mechanical Engineers, London.*

Bracciali, A. & Megna, G. 2015b. Contact Mechanics Issues Of A Vehicle Equipped With Partially Independently Rotating Wheelsets. Accepted for presentation to *CM2015 (10th International Conference on Contact Mechanics of Wheel / Rail Systems), Colorado Springs, Colorado, USA, 30 August 2015.*

Doyle, Jr., G.R. & Prause, R.H. 1977. Hunting Stability of Rail Vehicles with Torsionally Flexible Wheelsets. *Journal of Engineering for Industry, Transaction of the ASME*: 10-17.

Dukkipati, R. V. & Narayana Swamy, S. & Osman, M.O.M. 1992. Independently Rotating Wheel Systems for Railway Vehicles: A State of the Art Review. *Vehicle System Dynamics* 21(1): 297-330.

EN 14363 2005. *Railway applications - Testing for the acceptance of running characteristics of railway vehicles - Testing of running behaviour and stationary tests*. CEN: Brussels.

ERRI B176 RP1 1989. *Bogies with steered or steering wheelsets, Report No. 1, Specifications and preliminary studies, Volume 2, Specifications for a bogie with improved curving characteristics*. Utrecht.

Hadden, J.A. & Law, E.H. 1977. Effects of Truck Design on Hunting Stability Of Railway Vehicles. *Journal of Engineering for Industry, Transaction of the ASME*: 162-171.

Wickens, A. H. 2006. A History of Railway Vehicle Dynamics, in S. Iwnicki (ed.) *Handbook of Railway Vehicle Dynamics*. Taylor and Francis.