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TRACK FRIENDLINESS OF AN INNOVATIVE FREIGHT BOGIE

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Abstract: The design of freight running gear has always been a rather steady field. Due to the very demanding tasks as the high ratio between laden and tare weight (up to 5:1), the low cost production and the necessity of easy maintenance procedures, many potential innovations have never found application in the freight wagon market. The design of the Y25 bogie does not allow speed higher than 100 km/h in laden conditions and adequate track friendliness. The paper introduces a new concept of bogie for freight wagons, focusing the attention on the dynamic behaviour of the vehicle and its interaction with the track.

Keywords: freight bogie, track friendliness, inboard frame bogies, railhead damage, Tgamma, Variable Usage Charge.

1. Introduction

The current practice for freight operation follows two main directions: trains with high axle-load (30÷35 t/axle) that run at low speed (60 km/h) and trains with lower axle-load (20÷25 t/axle) running at higher speed (100÷120 km/h).

The first case is related to “heavy haul” operations that are very common in countries where very long distances have to be covered especially to carry raw materials. For this kind of operation, lines are almost fully dedicated to the operation of these heavy trains, and therefore a “homogeneous operation” traffic is generated. Dedicated lines are nevertheless not a very common option for transportation of generic goods. In this case, freight trains have to run on the same lines of conventional passenger trains, generating a mixed and “heterogeneous operations” traffic.

To allow freight trains running in between passenger trains, it is therefore requested a higher maximum speed and a better track friendliness. The current design of freight wagons is not able to guarantee a track friendliness comparable to modern passenger vehicles, whose running dynamics is in continuous improvement, noticeably reducing track maintenance costs.

It is commonly agreed that it is nearly impossible to increase the freight transport efficiency with the current solutions used for freight bogies [1]. The most common bogie for freight wagon in Europe, the Y25 bogie that was released in 1967, does not allow a safe running at speeds over 120 km/h. The maximum axle load is currently bound to increase from 22.5 t/axle to 25 t/axle but the original Y25 concept was never changed and the limitations of this bogie are now evident.

In this paper, an innovative bogie architecture for freight wagons is introduced. The new architecture is designed to allow a straightforward replacement of the Y25 bogie, while the main drawbacks of the current design are avoided introducing new concepts for the primary suspension, the friction damping and the bogie frame. The bogie is designed with the aim to be stable in tangent track at least at a speed of 140 km/h and to increase the track friendliness in curved track. Comparison with Y25 and other freight bogies are presented in terms of $T\gamma$ (or *Tgamma*, i.e. “wear number”) and vertical forces at the maximum speed. Finally, an evaluation of the Variable Usage Charge is shown, in order to estimate the track access charge according to UK specifications.

2. Description of an innovative freight bogie

While developing a new bogie for freight vehicles the designer has to take into account a number of demanding and often contradictory constraints, from the very high ratio between empty and laden axle load to running safety on twisted track and to the structural strength of the bogie frame.

However, recent innovations in bogie frames for passenger vehicles may be implemented also in freight bogie technology without increasing the production and maintenance costs, e.g. inboard bearings bogies (or inside frame bogies). Even if an inboard bearings bogie for freight wagons has been already proposed in the past, today this technology is strongly consolidated and it can be used without major problems. The main advantage of this kind of bogie is the lower mass and the reduction of the yaw moment of inertia. When applied to passenger bogies, this brings in many cases to a lower wheelbase increasing the track friendliness of the vehicle.

The 1.8 m wheelbase of Y25 is sufficiently low, but the mass is mainly concentrated far from the bogie centre (i.e. in the axleboxes), raising up the moment of inertia.

Figure 1 shows how the proposed innovative bogie, with inside frame architecture, can reduce the overall dimensions with respect to a conventional Y25 bogie introducing interesting new features like:

- the *AIR Wheelset* technology [2] or conventional inboard bearings wheelsets;
- wheel web mounted brake discs;

- horizontal coil springs with progressive stiffness (two springs *per bogie*).

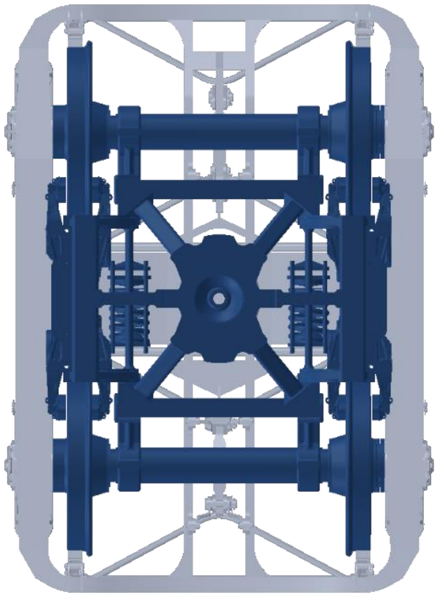

Figure 1 *Schematic representation of the overall dimensions of an Y25Ls(s) bogie (grey), compared with the innovative inside frame bogie (blue).*

The bogie is designed with a conventional centre bowl and two side bearers, in order to guarantee a straightforward replacement on standard wagons. As shown in Figure 2, the main difference is the primary suspension arrangement, where one progressive coil spring is used to replace the eight springs used on each side of an Y25 bogie.

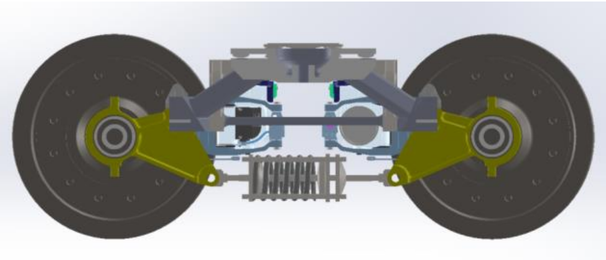

Figure 2 *Primary suspension arrangement of the modified bogie from an inside view. One progressive coil spring is used on each side.*

The primary suspension acts in longitudinal direction as already done in other kinds of bogies, with the difference that each spring connects the two arms on one side and therefore the two wheelsets. Load dependent friction damping is provided by the cylindrical pin connection between the arm and the frame.

The final characteristics of the bogie are described in Table 1 and compared with a conventional Y25 bogie. It is worth to highlight how the mass is reduced even if brake disks are introduced.

Table 1 *Main characteristics of the two bogies.*

	New bogie	Y25	Difference
M [kg]	4058	4750	-15%
Ixx [kgm^2]	3001	4667	-36%
Iyy [kgm^2]	3853	5610	-31%
Izz [kgm^2]	4564	6631	-31%

3. Vehicle and track dynamic models

A multibody model was developed with the VI-Rail software package [3] to investigate the dynamic behaviour of the new bogie. Both empty and laden wagon conditions were evaluated, resulting in the vehicle characteristics described in Table 2. The model represents a Sgnss 60' intermodal wagon, generally equipped with Y25Ls(s) bogies.

Table 2 *Main data used for the models*

	Empty	Laden
Axle load [t]	4.25	22.5
Total mass [kg]	17000	90000
Wheelbase 2a$^+$ [m]	1.8	
Vehicle pitch 2a* [m]	14.2	

The bogie model includes a flexible body, representing the bogie frame, whose flexibility is central to guarantee the needed vertical force equalization over twisted track. In fact, due to the primary suspension arrangement and the low static load when empty, the spring flexibility is not sufficient to obtain $\Delta Q/Q$ values lower than 0.6. The model was tuned through *FEM* simulations, resulting in a total wheel unloading of $\Delta Q/Q$=0.51 when considering both bogie and carbody twists.

The friction damping on the centre bowl and the side bearers was set to comply with the requirements of [4], where the yaw resistance of the bogie (*X factor*) shall be lower than 0.16 for the empty wagon and 0.1 for the laden wagon. The *X factor* is calculated as:

$$X = \frac{M_z}{2a^+ * P_0}$$

where P_0 is the nominal axle load in kN.

The total value of the yaw moment resistance M_z is shown in Figure 3 for both empty and laden condition, resulting in an *X factor* equal to 0.13 and 0.06 respectively.

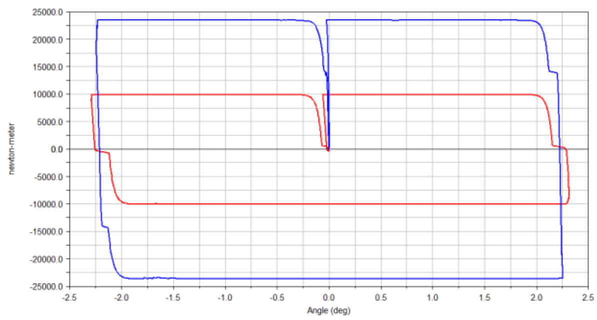

Figure 3 *Yaw resistance vs. the yaw angle of the bogie for an empty wagon (red) and for a laden wagon (blue).*

All running dynamics simulations were performed with S1002 wheel profile, coupled with 60E1 rail profile inclined of 1:40, in order to avoid double contact and to generate a medium to high value of equivalent conicity (γ_{eq}=0.17 at 3 mm of lateral displacement). The wheel-rail friction coefficient was set to *f*=0.4 in all simulations.

Track flexibility was considered according to [5] and lateral and vertical irregularities (i.e. alignment and longitudinal level defects) were added to the track geometry. Two different kind of irregularities are used:

- irregularities sampled from measurements on an Italian regional line, where running dynamics homologations tests are usually performed;
- *ERRI High level* irregularities generated according to [11].

The PSD function of these irregularities are plotted in Figure 4. The first kind is used for running dynamic simulations (i.e. according to the requirements of [4]), while the second kind is used to investigate the response of the bogie in terms of railhead damage.

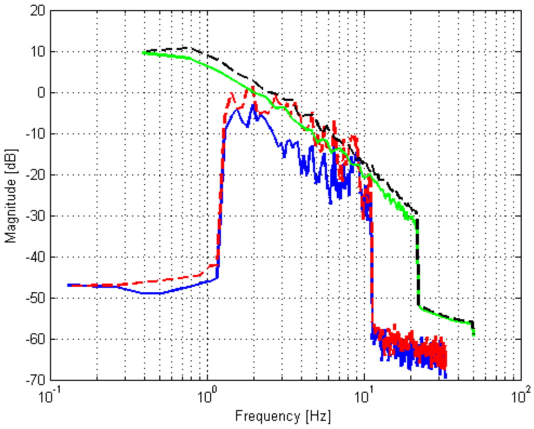

***Figure 4** PSD distribution of lateral (alignment, solid blue) and vertical (longitudinal level, dashed red) for measured defects and the corresponding ERRI "High Level" irregularities lateral (solid green) and vertical (dashed black).*

4. Dynamic behaviour of the bogie

4.1. Lateral dynamics

The frequency response of the vehicle was evaluated by simulations on straight track at a speed of 148 km/h. The frequency of lateral wheel forces has then been calculated for both empty and laden conditions (Figure 5).

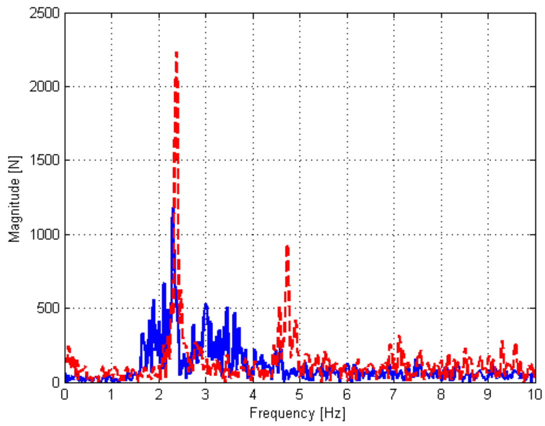

***Figure 5** Lateral frequency response for empty (dashed red) and laden (solid blue) conditions*

The main lateral frequency of the vehicle is about f_0= 2.3 Hz. This value is used to investigate the stability of the vehicle according to [4], where the maximum *RMS* value of the sum of guiding forces ΣY on left and right wheel (filtered in the range of the critical frequency $f_0 \pm 2$ Hz) is compared with half of the Prud'Homme limit, i.e.

$$\Sigma Y_{lim} = k_1(10 + \frac{P_0}{3})/2$$

where k_1=0.85 for freight vehicles and P_0 [kN] is the axle load. The stability assessment values are therefore calculated and shown in Table 3.

***Table 3** Assessment values for stability evaluation*

	Empty	Laden
Axle load [kN]	41.69	220.73
Limit value [kN]	10.16	35.52

The maximum *RMS* value over runs of 300 m of the last wheelset of the wagon model is plotted in Figure 6 as the percentage of the limit value vs. the running speed for both loading conditions.

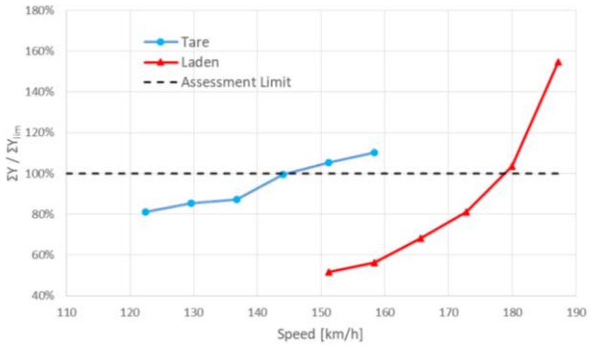

***Figure 6** Stability assessment in tare condition and laden condition.*

The limit value is reached for a speed of about 145 km/h for the empty wagon and for a speed of 180 km/h for the laden wagon. However, for the laden wagon, a maximum speed of 160 km/h is considered sufficient to perform the evaluations that are the object of this paper.

4.2. Vertical dynamics

The frequency spectrum of vertical wheel force is plotted in Figure 7. A bad vertical dynamics of the bogie is one of the main causes of railhead damage accumulation.

Without considering singular track defects like joints or welding, which are responsible of high frequency impact forces where unsprung masses and track stiffness are dominant, the characteristic of the primary suspension is a relevant factor for random distributed vertical irregularities [6].

One of the main advantages introduced by the new suspension arrangements is a better response of the bogie to random vertical irregularities. This is particularly true for such irregularities with wavelengths lower than the bogie wheelbase, for example short pitch corrugation, due to the lower stiffness (half of the nominal) "seen" during the passage over the defect. However, this kind of irregularities have to be considered separately from the longer irregularities which are taken into account in this paper. Therefore, a future paper will specifically describe in detail the response of the suspension "short-circuit effect" on short irregularities compared to a conventional suspension.

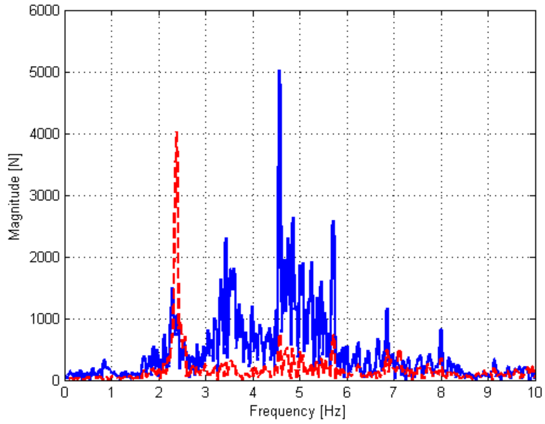

Figure 7 *Vertical frequency response for empty (dashed red) and laden (solid blue) conditions*

In this paper a more general evaluation of the vertical dynamics is described. A part of the *Track Access Charge* (*TAC*) for freight vehicle in *UK* is due to the vertical wheel dynamic force produced by the wagon. In particular, the parameter used to evaluate the performances of a bogie is the *Ride Force Count* (*RFC*). The *RFC* is a metric of the vertical dynamic force which assesses the track friendliness of freight bogies and their suspension system. From this index a *Suspension Discount Factor* is then calculated which scales the final *Variable Usage Charge* (*VUC*) paid by the operator. The *Discount Factor* is defined as a function of *RFC* by a curve (or discrete bands).

RFC values are calculated by means of dynamic simulations on specific track, named *Track for Banding* (*TfB*) and evaluating the correlation between the standard deviation of the vertical track irregularities and the standard deviation of the dynamic vertical force, calculated for 200 m-long track sections. By means of a linear interpolation of the data, it is possible to derive the *Ride Force Coefficient* (gradient of the trend line in kN/mm) and the *Ride Force Constant* (intersection of the trend line with the *y-axis*). Then, these parameters supply the *Suspension Discount Factor* (SDF) evaluating the distribution of the standard deviation of the vertical irregularities. An explanation of this process can be found in [7].

This method was therefore chosen to evaluate the vertical performance of the new primary suspension arrangement. To compare the results with those from other freight wagons, data available from [8] and [9] were used, in which a four axle *Y-series* flat 60' wagon represents the reference for another innovative freight bogie developed for the six-axle *Spectrum* vehicle project. To permit a fair comparison, the vertical forces (low pass filtered at 20 Hz) are initially normalized by the axle load. The simulation speed is set to 120 km/h, 145 km/h, 160 km/h. The axle load of the reference vehicles is summarized in Table 4.

Table 4 *Axle load of reference vehicles*

Vehicle	Tare axle load [t]	Laden axle load [t]
Y-series reference	5.75	22.25
Spectrum reference	5.31	15.8

It is worth to highlight that the *TfB* used for the *RFC* calculation has a specific track irregularities distribution, which was not available to the authors. However, the ERRI defects, which are normally used to model track irregularities in multibody simulations, show higher values of standard deviations compared to the measured ones. This can be explained (see Figure 4) by considering that the measured values are filtered in the *D1* wavelength range (3÷25 m) according to [4] and [10], while the others are numerically synthesized for a broader wavelength range (2÷100 m). Filtering for example the *ERRI High Level* defect in the *D1* range, it is possible to see how the distribution shifts to lower values of standard deviation. This is shown in Figure 9, where a track divided in 100 m long sections was simulated. In these conditions the original high level defects distribution is very close to *TfB* and therefore it can be used for the *RFC* calculations.

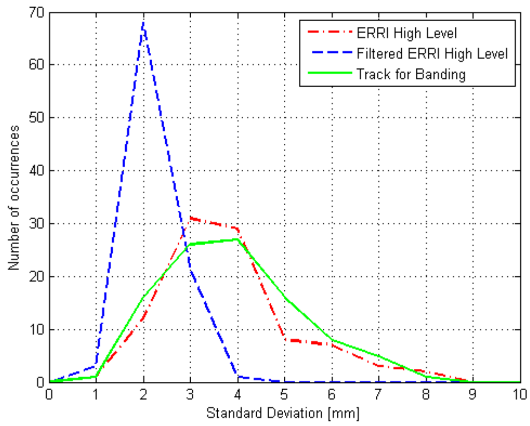

Figure 8 *Number of occurrences of the standard deviation of vertical ERRI High Level defects as generated (red), filtered in the D1 wavelength (blue) and the Track for Banding distribution (green).*

The evaluation is shown in Figure 10. While the *Ride Force Coefficient* and the *Ride Force Constant* will be used with the aim to perform *VUC* calculation, the most relevant and directly usable parameter is the mean value over the track sections. Table 5 shows the comparison between the vehicles.

Table 5 *Dynamic vertical force comparison between bogies*

Vehicle		Mean value (tare)	Mean value (laden)
Y-series reference (120 km/h)		0.238	0.102
New bogie	**120 km/h**	0.127	0.052
	145 km/h	0.162	0.072
	160 km/h	-	0.085
Spectrum reference (160 km/h)		0.148	0.083

The results show that the mean dynamic vertical force *per unit of axle load* are low for the new bogie even at high speeds. At the maximum speed, the values are only slightly higher than those of the Spectrum vehicle.

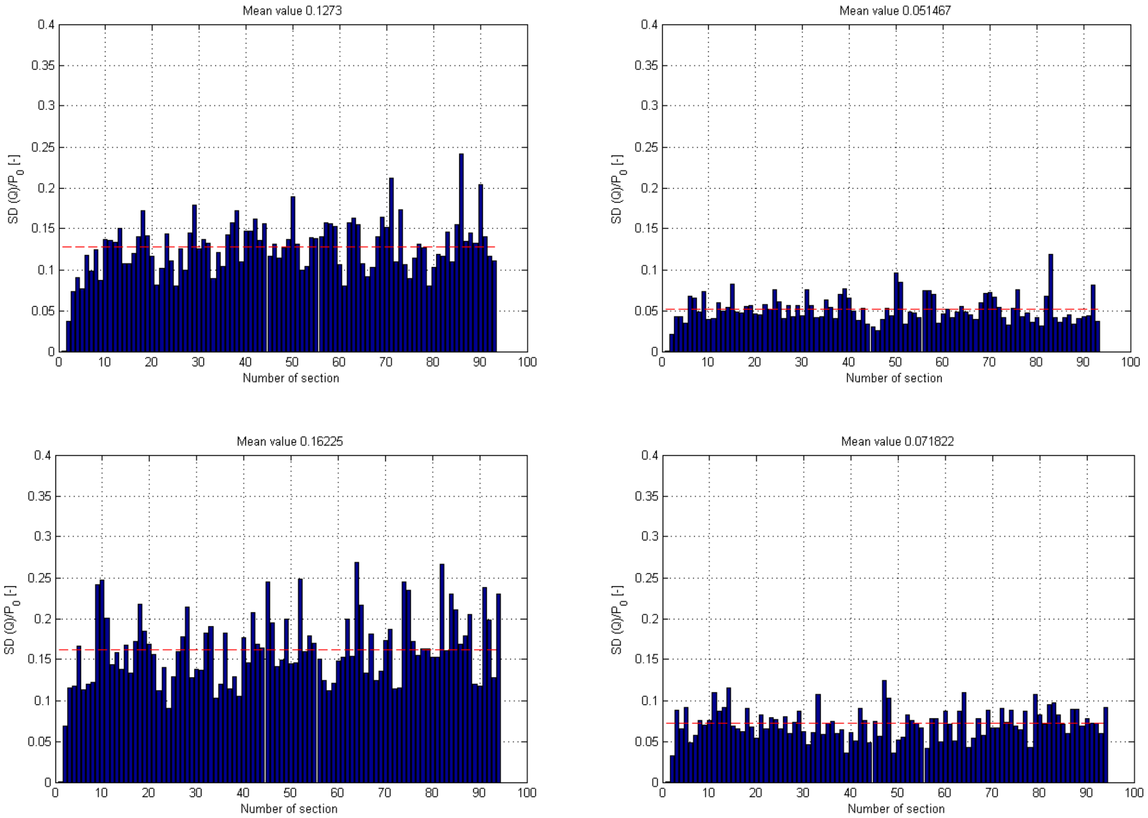

Figure 9 *Standard deviation of the vertical dynamic force over 100 m long track sections for the empty wagon (left) and for the laden wagon (right) at the speed of 120 km/h (above) and of 145 km/h (below). Dashed red line represent mean values.*

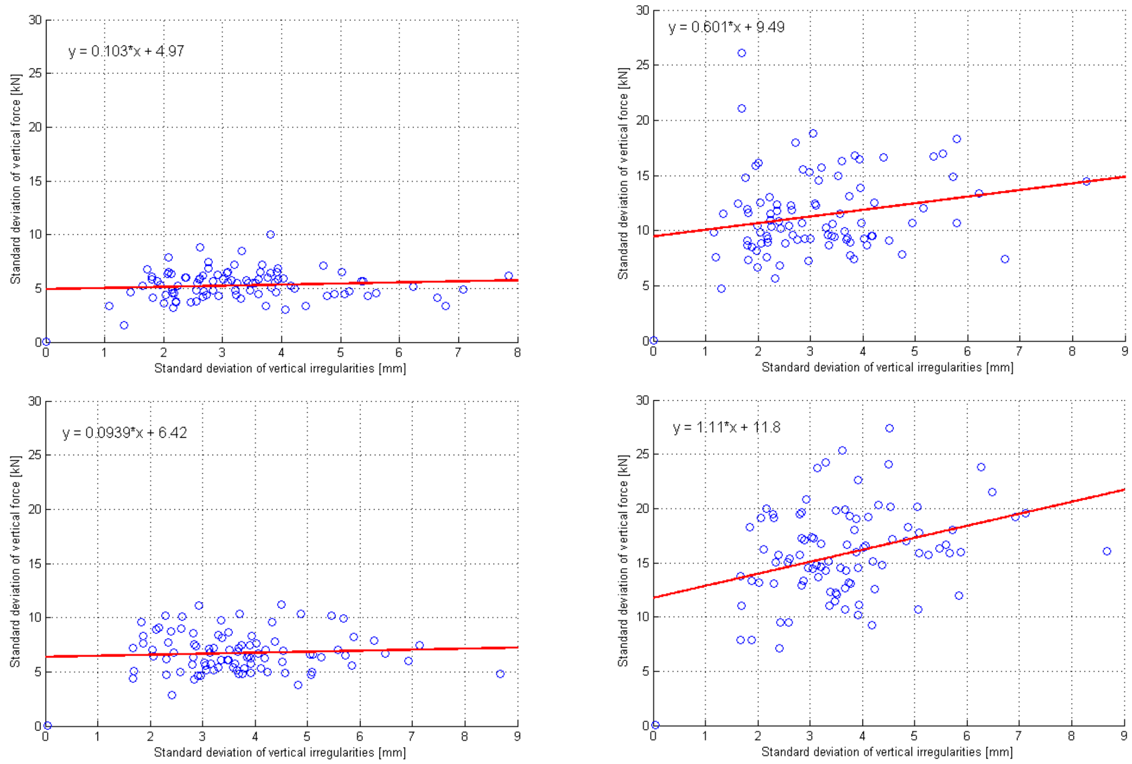

Figure 10 *Standard deviation of the vertical dynamic force over 100 m long track sections for the empty wagon (left) and for the laden wagon (right) plotted against the standard deviation of vertical irregularities at the speed of 120 km/h (above) and of 145 km/h (below). Red lines are the best fit linear regressions.*

With the aim to perform a *VUC* calculation for the freight wagons, the suspension discount factor was evaluated. This value is obtained separately for the tare and laden conditions from the RFC value, obtained as

$$RFC = \sum_{i=1}^{n} (SD(i) * P1 + P2) * N(i)$$

where *SD* is the standard deviation class (1 mm each) of vertical irregularities, *P*1 is the Ride Force Coefficient [kN/mm], *P*2 is the Ride Force Constant [kN] and *N* is the number of occurrences for the *i-th* class of standard deviation.

The results in terms of *Suspension Discount Factor* are shown in Table 6 and Table 7. The values are derived from seven bands (from 0.858 to 1.098), where each band corresponds to a particular range of *RFC*, different for tare and laden conditions.

Table 6 *Suspension Discount Factor comparison in empty condition*

Vehicle		*RFC*	Band	Discount factor
Y-series reference		680-715	2	1.058
New bogie	120 km/h	489	6	0.898
	145 km/h	631	4	0.978
Spectrum reference		465-564	6	0.898

Table 7 *Suspension Discount Factor comparison in laden condition*

Vehicle		*RFC*	Band	Discount factor
Y-series reference		>1650	1	1.098
New bogie	120 km/h	1029	6	0.898
	145 km/h	1438	4	0.978
	160 km/h	1689	1	1.098
Spectrum reference		>1650	1	1.098

Even at *v*=145 km/h the new bogie remains in the bands for which the discount factor is *SDF*<1.

5. Steady state curving behaviour

5.1. T-gamma and Rail Surface Damage

The track friendliness of a vehicle, and therefore of a bogie, is often evaluated by using the *Tγ* value. The product of the tangential force *T* and the tangential creepage *γ* is a measure (in J/m) of the energy dissipated at the contact of each wheel and it can be used to assess both track friendliness and the steering ability of a bogie. The lower the *Tγ* is, the lower the damage introduced by the wheels to the rails is. In [15], this parameter was used to compare different kind of freight bogies considering their effect on rolling contact fatigue.

The importance of *Tγ* as a parameter to evaluate the track access charge of a vehicle is confirmed by its introduction in the track deterioration models ([12], [13]).

Figure 11 describes the relationship between *Tγ* and the damage introduced by an axle of the vehicle. In the evaluation, only *Tγ* from high rail is taken into account. The total rail surface damage (and the related maintenance cost) is therefore the result of the combination of the two damage rates, as explained in Table 8.

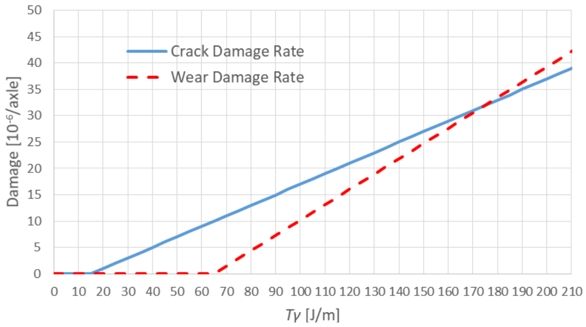

Figure 11 *Assumed damage rate for wear and crack growth as function of Tγ in the Rail Surface Damage (RSD) model.*

Table 8 *Range of Tγ, effects on rail damage and fixing actions*

Tγ [J/m]	Kind of damage	Action
0-15	No damage	-
16-65	Crack growth	Rail grinding
66-175	Crack growth > Wear	Rail grinding (wear contributes to remove cracks)
>175	Crack growth < Wear	Rail renewal

In the *Variable Usage Charge*, 85% of the charge is assigned to track maintenance and renewal, 70% of which linked to vertical rail forces and 30% to horizontal rail forces. While the first is only a function of axle load, operating speed and unsprung mass, the second depends on the steady state curving behaviour of the vehicle and therefore on *Tγ*. The model prescribes a number of simulations in curves of different radius to generate a *Tγ* table used to calculate the charge for freight or passenger operators.

An automatic tool was developed and included in VI-Rail that generates the fifteen curves (from 200 m to 10000 m) with the conditions described in [16]. The curving behaviour is evaluated with the following values of cant deficiency:

- h_d=40 mm for passenger vehicles;
- h_d=0 for freight vehicles with v_{max}= 100 km/h.

Therefore, freight vehicles simulations are performed at the equilibrium speed. Contact conditions shall be modelled considering the lubrication of the gauge face of the high rail (*f*=0.2).

A comparison between *Tγ* values from the external wheels (high rail) of the new bogie and the Spectrum bogie is shown in Figure 12. Values are related to the leading bogie whose *Tγ* values are greater than the trailing bogie.

Unfortunately, due to the lack of information, a direct comparison in laden conditions or with a wagon equipped with Y25 is not possible. An indirect comparison will be given in the next paragraph with the *VUC* calculations.

It is worth to highlight that at the moment of writing the tool was in the finalization stage and the wheel and rail profiles used for the new bogie are not the ones (measured semi-worn profiles) required in [16]. However, for low radius curves it is assumed that conicity has only a slight effect on the curving behaviour of a bogie.

The aforementioned assumption of h_d=0 is seen as particularly conservative as running faster is one of the main goals of freight operation. Higher cant deficiency values are therefore investigated.

Due to the roll angle of the wagon and the increase of the load of the external wheels, the suspension arrangement of the new bogie generates a better steering performance as it allows the wheelset to align radially when the vehicle runs with h_d>0 (a_{nc}>0). In such case the external suspension arms rotate more than the internal ones, generating an outer wheelbase greater than the internal one.

As the effect of cant deficiency was found not relevant for the empty vehicle, $T\gamma$ values and angle-of-attack for the laden condition only are shown in Figure 13 for h_d=0, 40, 92, 153 mm. These values correspond to a_{nc}=0, 0.26, 0.6 (typical for freight traffic) and 1.0 (typical for passenger traffic) m/s^2.

While for very low radius curves (200 m) the steering ability is quite poor in all the investigated situation, the effect of increasing cant deficiency is clearly beneficial for mild curve radii (R=600÷800 m) where at the equilibrium speed the transition between wheel and rail flange contact to tread contact occurs. This transition shifts towards lower curve radii when the cant deficiency increases, as the angle of attack tends to zero more rapidly.

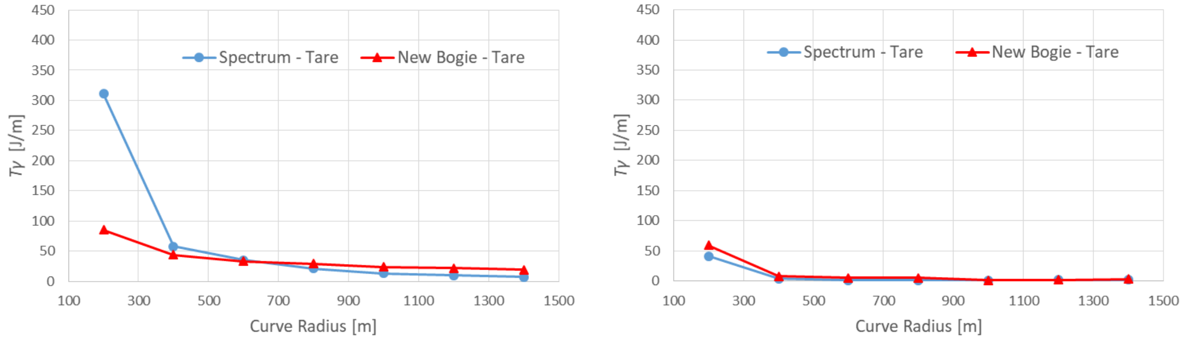

Figure 12 *Front wheelset (left) and rear wheelset (right) $T\gamma$ comparison of leading bogie of the spectrum vehicle (from [9]) with the new bogie (tare condition, zero cant deficiency).*

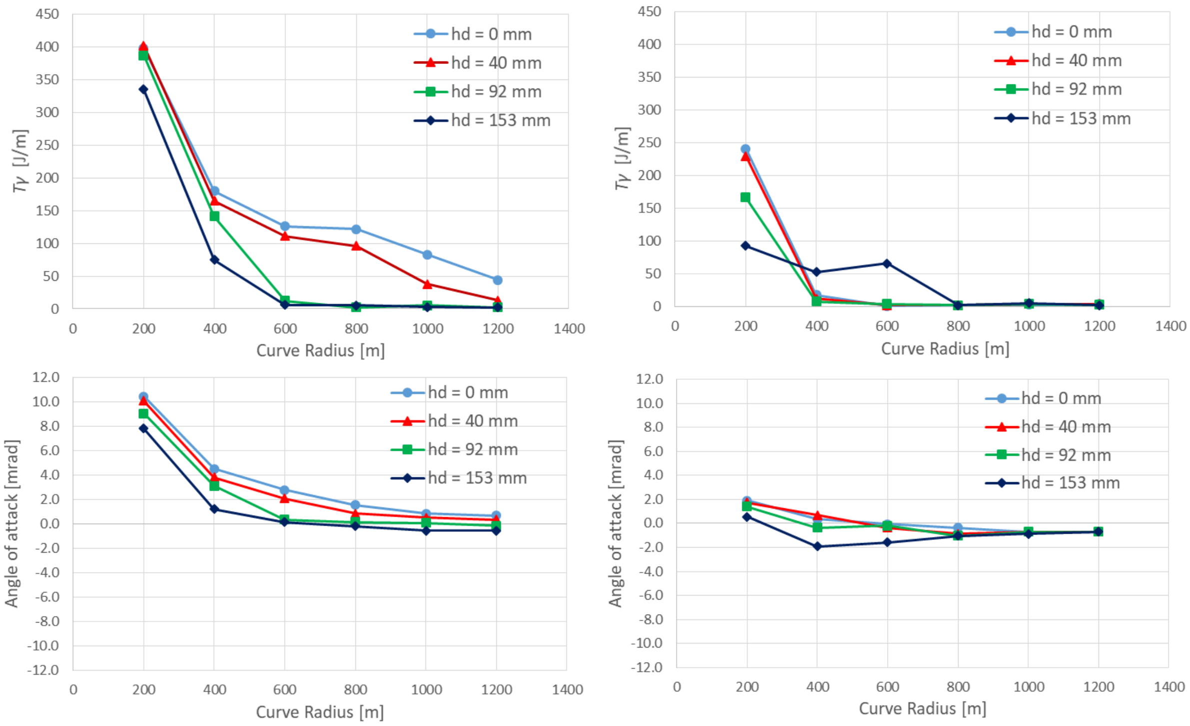

Figure 13 *$T\gamma$ (top) and angle of attack (bottom) of front axle (left) and rear axle (right) of the leading bogie for different curve radii and cant deficiency values for the new bogie in laden condition.*

5.2. Variable Usage Charge calculation

Wear damage (*WD*) and *crack damage* (*CD*) according to [14] were calculated from the steady state calculated $T\gamma$ values. Track maintenance and renewal costs (in £/km*axle) are then obtained by weighing *WD* and *CD* with the distribution of curves over the UK network as follows:

$C = 14500 * CD - 2000 * WD$ if $T\gamma < 175$ J/m
$C = 12500 * WD$ if $T\gamma > 175$ J/m

in which multipliers are derived through the parameters listed in Table 9. A spreadsheet that allows to perform the calculation procedure is freely available on the Network Rail website [16]. The other costs that generate the final charge, such as signalling costs (5%) and civil costs (10%), are not relevant for the scope of this paper.

Table 9 *Parameters used to correlate RSD to maintenance and renewal costs [14]*

Grinding cost	2000 £/km
Renewal cost	250000 £/km
Depth ground	0.5 mm/grinding cycle
Side wear limit	10 mm

Taking into account that for freight vehicle the previously calculated *Suspension Discount Factor* is applied, Table 10 shows the final charge in £/kGTM (kGTM=million gross tonne miles) for the empty case. Table 11 shows both £/kGTM and the charge of carrying payload (i.e. the cost for kNTM= million net tonne miles) for the laden case compared with the two reference wagons.

As shown in previous paragraph, $h_d>0$ has a considerable effect on the curving behaviour for the laden case. Therefore, the comparison is made for both the cases of 0 mm, 40 mm and 92 mm of cant deficiency.

$$VUC_{net} = VUC * \frac{Laden\ weigth}{Laden\ weigth - Tare\ weigth}$$

Table 10 *VUC comparison in empty condition*

Vehicle	*VUC* [£/kGTM]
Y-series reference	1.5242
New bogie	1.1046
Spectrum reference	1.3585

Table 11 *VUC comparison in laden condition*

Vehicle		*VUC* [£/kGTM]	VUC_{net} [£/kNTM]
Y-series reference		3.1745	4.275
New bogie	h_d=0	3.3072	4.077
	h_d=40 mm	3.0788	3.796
	h_d=92 mm	2.7366	3.374
Spectrum reference		2.3847	3.595

Charges at the equilibrium speed for the new bogie are comparable with Y25 bogies, lower in empty conditions (-27%) and slightly higher in laden condition (+4%).

While the reference vehicle had a tare of 23 t, the wagon equipped with new bogies has a tare of 17 t resulting in a lower charge *per unit of payload*.

The comparison with the Spectrum reference is nevertheless quite unfavourable especially for the laden wagon, although laden axle loads are very different (Spectrum = 15.8 t/axle, new bogie = 22.5 t/axle). This gap reduces when running at h_d=92 mm, resulting in a 20% *VUC* decrease, thanks to the radial aligning of the wheelsets.

6. Conclusions

In the last years, the necessity of improvements related to bogie technology for freight wagons is becoming a crucial topic inside railway transportation system. Greater operating speed and better track friendliness are needed to increase freight efficiency and the conventional architecture (Y25 for example) are not suitable for different services respect to the current ones.

In this paper a novel bogie architecture with inside frame, brake discs and progressive coil springs is presented and applied to a freight wagon. Multibody simulations were performed to investigate the dynamic behaviour of the vehicle. Critical speed over defective track were calculated showing that maximum speed over 140 km/h is reachable. At the maximum speed, vertical dynamic forces over defective track are still lower than the ones generated by a wagon equipped with Y25 bogie, generating a lower impact on railhead damages. This is due to the innovative arrangement of the primary suspension, which at same time allows to reduce the number on components (two springs per bogie) and to improve the vertical response to distributed irregularities. The horizontal spring arrangement, together with the lower yaw moment of inertia, has shown good results also in curving behaviour resulting in lower $T\gamma$ values especially for mild radius curves (R=600÷800 m).

The comparison with other freight bogies in terms of *VUC* calculations according to Network Rail specifications has shown comparable charge both in empty and laden conditions, with advantages due to the lower weight of the bogie and therefore of the tare weight of the simulated wagon. However, the effect of the cant deficiency is not included in the current analysis methods, resulting, in the authors opinion, in a too conservative costs evaluation.

Even if further investigations are scheduled for future papers, the innovative design has shown favourable features in terms of track friendliness.

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