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Development of a new running gear for the Spectrum intermodal vehicle

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ABSTRACT: The European Union (EU) Seventh Framework Programme (FP7) project Spectrum [12] set out to develop a freight vehicle which would facilitate the exploitation of the low density, high value (LDHV) goods market. Key to the performance criteria for the vehicle were: increased speed to enable mixed running with passenger services; improved ride quality to avoid damage to the LDHV goods; and reduced track damage for longevity and sustainability on increasingly stressed infrastructure. This paper presents aspects of the development of a novel running gear arrangement for the Spectrum vehicle, focussing on the dynamic performance of a Vampire vehicle model and the steps to realising stable running. Finally, the estimated performance of the Spectrum vehicle concept is compared against calculations for a conventional freight wagon with respect to curving, vertical track forces and potential savings in track access charges through implementation of Network Rail's Variable Track Access Charge Calculator. It was found that the novel Spectrum concept could offer savings in Variable Usage Charges of between 8% and 16% compared to the conventional equivalent.

1. INTRODUCTION

The European Union (EU) Seventh Framework Programme (FP7) project Spectrum [12] set out to develop a freight vehicle which would facilitate the exploitation of the low density, high value (LDHV) goods market. Key to the performance criteria for the vehicle were: increased speed to enable mixed running with passenger services; improved ride quality to avoid damage to the LDHV goods; and reduced track damage for longevity and sustainability on increasingly stressed infrastructure.

This paper presents aspects of the development of a novel running gear arrangement for the Spectrum vehicle, focussing on the dynamic performance of a Vampire vehicle model and the steps to realising stable running.

2. BACKGROUND

The concept of choice for the Spectrum vehicle was to be a six-axle articulated vehicle (comprising three two-axle bogies and two decks) capable of carrying up to four 25 foot refrigerated swap-bodies (or containers) or two 45 foot refrigerated swap-bodies or combinations thereof. The vehicle would feature non-uniform axle loads, having a maximum 22.5 t axle load at the centre bogie and 17 t axle load at the outer bogie, and be capable of integrating with passenger services at speeds of up to 160 km/h.

To achieve the requisite vehicle performance a bogie concept was envisaged which sought inspiration from the higher performance of passenger bogies, and the simple, cost effective solutions implemented on freight bogies. The base concept was to feature viscous damped, coil sprung trailing arms for the primary suspension and a standard UIC centre bowl and side-bearer arrangement at the secondary. The primary suspension arrangement would allow optimisation of the plan view suspension for curving, while maintaining good vertical ride quality. Utilising the standard UIC secondary suspension hoped to: minimise cost (while still providing adequate dynamic characteristics); allow for potential retrofitting of the bogie to existing stock in the future; and to decouple the counter dependency between the bogie and wagon design processes which ran concurrently within a demanding project time frame.

3. SUSPENSION CHARACTERISTICS

Initial values for the suspension parameters were calculated based on engineering theories, such as those for railway vehicle dynamics [6], spring characteristics [5] and rubber theory [10].

For example, to determine appropriate values for the trailing arm bush characteristics a set of equations was derived based on simple geometry. From these the contribution of the stiffnesses of the trailing arm bush and primary coil springs in the three principal plan view directions (longitudinal – 'X'; lateral – 'Y'; and yaw – 'W') to the overall primary yaw stiffness (PYS) could be determined.

For example, the influence of the trailing arm bush on the primary yaw stiffness (KY) was found to be as shown in Equations (1) and (2), where the suffix TB indicates a parameter of the Trailing arm Bush, suffix X or Y indicates the longitudinal and lateral directions respectively, Y is the lateral semi-spacing of the component (TB), K indicates the stiffness of the given component in a given direction and θ represents the wheelset yaw angle.

$$KY_{TBX} = Y_{TB}^2 . K_{TBX} \tag{1}$$

$$KY_{TBY} = \frac{(Y_{TB}\theta)^2}{2} . K_{TBY} \tag{2}$$

Similarly the influence for the primary coil springs (PC) was found as in Equations (3) and (4).

$$KY_{PCX} = dx. K_{PCX}. Y_{PC} \tag{3}$$

$$KY_{PCY} = dy. K_{PCY}. X_{TA} \tag{4}$$

Here, the terms dx and dy represent the longitudinal and lateral deflection across the coil spring, resulting from a given wheelset yaw displacement (θ), as defined in Equations (5) and (6), where X_{TA} indicates the length of the trailing arm.

$$dx = X_{TA}\left(1 - \frac{\theta^2}{2}\right) - \mathrm{Y_{PC}}\theta - \mathrm{X_{TA}} \tag{5}$$

$$dy = X_{TA}\theta + Y_{PC}\left(1 - \frac{\theta^2}{2}\right) - Y_{PC} \tag{6}$$

The total effective primary yaw stiffness is then simply the sum of the contributions of the primary coil spring and trailing arm bush stiffnesses in the longitudinal (X), lateral (Y) and yaw (W) directions (Equation 7).

$$KY = \sum\left(KY_{TB(X,Y,W)}\right) + \sum\left(KY_{PC(X,Y,W)}\right) \tag{7}$$

From studies on the relationship between track damage and primary yaw stiffness [3] it is understood that a primary yaw stiffness of around 12 MNm/rad represented an initial value to design to, which would lead to comparably good curving performance and low levels of wheel and rail damage. From the derived equations and target PYS suitable trailing arm bush parameters were determined.

4. STABILITY ANALYSIS

A mathematical model of the concept vehicle and running gear was constructed in the Vampire rail vehicle dynamics simulation package, in order to optimise the suspension parameters for stability, ride performance and derailment resistance.
A parametric approach was employed to optimise parameters for stability. Parameters which were included in the variations were: trailing arm bush stiffness, damping and dynamic stiffening; secondary side-bearer friction coefficient; side-bearer longitudinal stiffness and clearance; nominal geometry (bogie wheelbase and trailing arm length); the addition of anti-roll bars; the inclusion of viscous yaw dampers; and the addition of novel controls and constraints such as the control arm of the Infra-Radial project [8] (researched in the EU FP7 Sustrail project [7]).
The vehicle stability was assessed against both speed and conicity – with the conicity being implemented via the Vampire square root creep law, as opposed to using real wheel and rail profiles. Linear analyses, such as an Eigen value analysis, were of limited validity for the Spectrum concept due to the friction elements used to represent the UIC secondary suspension. A transient analysis was therefore used to identify conditions of instability by exciting the vehicle with an initial period of sharp irregularities followed by perfectly smooth track. The decay (or lack of)

of wheelset lateral motion following transition onto the smooth track provides an indication of the vehicle's stability [13].
Initially two modes of instability were noted for the articulated vehicle, instability at low conicity (around γ = 0.025) and at combined high speed (greater than approximately 35 m/s) and high conicity (around γ = 0.2). These observations are in line with current understanding of the different instability or hunting modes (low speed carbody instability and high speed wheelset instability).
The parametric study (varying parameters as mentioned above) was unable to achieve stable running for the requisite range of conicities, speeds and load conditions. The investigations found that there was a coupling of the bogie lateral modes and the yaw motion of the vehicle semi-body via the high lateral stiffness of the UIC secondary suspension (a metallic stiffness, as there is no designed compliance). Decoupling these lateral modes by the introduction of a much lower artificial stiffness at the centre bowl was found to significantly improve stability. This observation is supported by Carter's 1928 understanding of the instability of a locomotive [4] (discussed more recently by Knothe and Böhm [9]), where the speed limit for stable running was shown to be dependent upon the lateral bogie-body stiffness.
Figure 1 below shows a measure of the stability/instability for every conicity and speed combination assessed (20 cases) for each case of varying lateral stiffness (13 cases). The instability measure here takes the RMS of the leading wheelset lateral displacement for each conicity and speed combination, and finally the mean of those RMS values is taken for each lateral stiffness. The figure shows that a stiffness value below approximately 0.2 MN/m is desirable to optimise the secondary lateral stiffness for stability, as below this value there is little further reduction in response and above this value the wheelset response increases significantly.

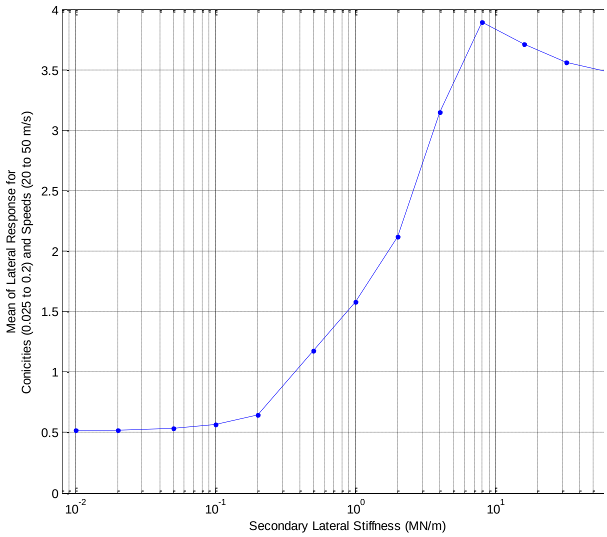

Figure 1 Mean lateral response for a range of secondary suspension lateral stiffness values

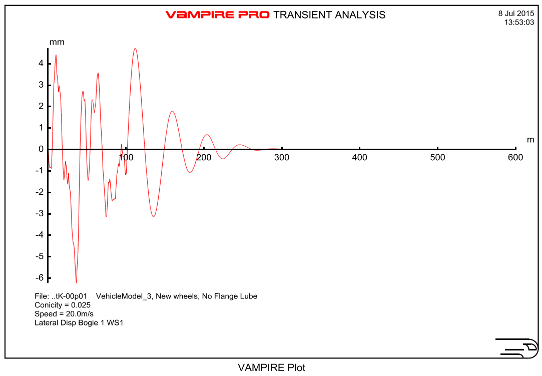

Figure 2 Example of stable response to sharp lateral irregularities (0 to 100 m). RMS of decaying response from 100 m to 600 m is 0.9 mm

A practical implementation of the low stiffness was devised, whereby the centre bowl (normally resting on the bogie transom) is suspended by swing links from the transom. The swing links provide a lateral stiffness of constant frequency (irrespective of payload), and the pivot points of the links provide load dependent damping. The notion of swing links to provide low lateral bogie-body stiffness is a tested, cost effective solution, and has been used on legacy passenger bogies and can be seen in various guises on leaf sprung freight vehicles and on some locomotives. The detail of the proposed swing link arrangement can be seen in the cross-section of the development CAD model in Figure 3, while the finalised bogie concept is shown in Figure 4.
The swing links were modelled in detail in the Vampire vehicle model, by using a combination of Pinlink, Friction and Bumpstop elements. The properties of the swing links (length, pin diameter, friction etc.) were subsequently optimised to achieve the requisite performance for a range of linear conicities, speeds and also for representative wheel and rail profiles. The side-bearer arrangement was also slightly modified to include lateral stops to prevent damage from the lateral forces on the side-bearer, which do not arise (at least to the same extent) in the normal configuration. The lateral friction and clearances associated with the side-bearers were also included in the vehicle model.

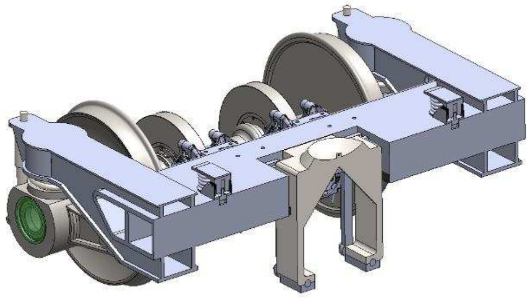

Figure 3 Cross section of the proposed swing link arrangement in the developing CAD model

Figure 4 The final CAD model of the Spectrum bogie concept

5. PERFORMANCE EVALUATION

The dynamic performance of the final vehicle concept was assessed by undertaking computer simulations using the Vampire rail vehicle dynamics package, in the following areas: virtual conformance to the assessment requirements of EN 14363[2]; evaluation of the curving and vertical performance, with respect to the degree of rail and track damage which could be expected; and estimation of the cost benefits resulting from improved curving and vertical dynamic performance over a conventional freight vehicle. These analyses are presented in the following sub-sections.

5.1. EN 14363 Assessments

The Spectrum vehicle was simulated through an artificial route developed to satisfy the conditions of the EN 14363 On Track Tests for a maximum running speed of 160 km/h – which actually necessitates accounting for 10% over-speed and assessment up to 176 km/h. Three different load cases were simulated: tare, part laden and fully laden. For each vehicle load case the performance of the vehicle model was assessed against the requirements of the standard for: track shifting forces (ΣY_{max}); flange climb derailment risk (Y/Q); dynamic vertical wheel load (Q_{dyn}); quasi-static vertical wheel load (Q_{qst}); quasi-static lateral wheel load (Y_{qst}); maximum body lateral acceleration (y^*_{max}); maximum body vertical acceleration (z^*_{max}); RMS of body lateral acceleration (y^*_{rms}); RMS of body vertical acceleration (z^*_{rms}).

For some of the measures the limit value is dependent upon vehicle variables, such as axle load, so to ease the evaluation of the results each metric was normalised by the appropriate limit value. The performance of the vehicle may then be expressed as a percentage of the limit value for each metric. The data for each wheelset or semi-wagon body end (i.e. leading end of the leading semi-body, trailing end of leading semi-body, leading end of trailing semi-body etc.) was calculated as appropriate, and the results were segregated by the EN 14363 test zone, where Zone 1 is straight track and very large radius curves, Zone 2 is large radius curves, Zone 3 is small radius curves (400 m to 600 m) and Zone 4 is very small radius curves (250 m to 400 m).

Figure 5, below shows an example of the output produced for each measure of the Standard assessed – in this case the sum of the lateral wheel-rail forces for each axle (track shifting forces).

In addition to simulating the on-track tests from EN 14363, the static tests for safety against derailment were also simulated.

For example, the vehicle model was simulated through a virtual laboratory wheel unloading test in tare, part, and fully laden states. The upper subplot of Figure 6 shows the wheel unloading for the left and right contacts of the leading and trailing wheelsets of the leading bogie. The lower subplot shows the corresponding wheel lift applied. The leading bogie is tested during the first 60 seconds of the simulation and the centre bogie (wheel lift not shown) during the last 60 seconds. The wheel unloading limit imposed by EN 14363 is 0.6 (that is a reduction of up to 60% of the static wheel load). The Spectrum vehicle was found to conform to the wheel unloading criteria in the tare, part and fully laden conditions – in all cases, the maximum wheel unloading experienced was at the leading wheelset and was never more than 40%.

Similar analyses were also undertaken to prove conformance with the Standard for the bogie rotational resistance (X-factor) test and the flange climb resistance (Y/Q) test.

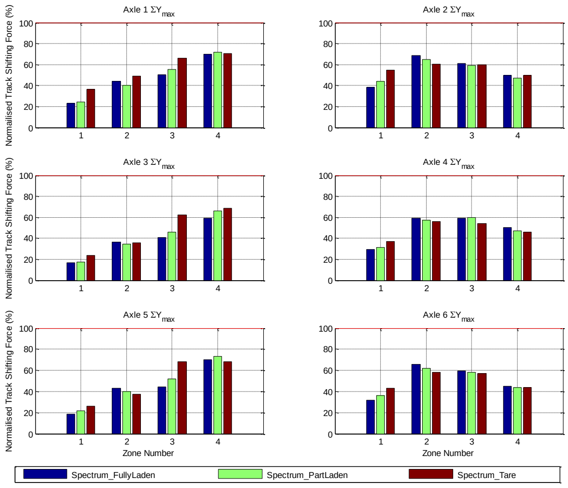

Figure 5 Sum of lateral forces (track shifting forces) calculated according to EN 14363

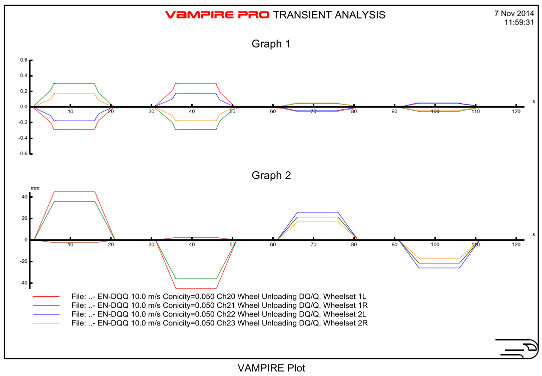

Figure 6 Wheel unloading in tare condition

5.2. Curving Performance

The curving performance of the vehicle is assessed to a limited extent by the EN 14363 on track tests, where the sum of the lateral wheel-rail forces (track shifting forces) is limited. The purpose of this limit is to ensure vehicles do not drive excessive track damage. An alternative measure of curving performance is the evaluation of the energy dissipated through the wheel-rail contact. This gives an indication of the curving efficiency of the vehicle and the level of rail damage (rolling contact fatigue (RCF) and wear) which the vehicle can be expected to generate. An assessment based on this energy dissipation forms part of the track access charging scheme used in the UK [1].

In idealised conditions an unconstrained wheelset will achieve a radial position when curving, that is the axle centre-line points to the centre of the curve. At a radial position the rolling radius difference between the left and right contacts does not lead to any longitudinal force at the wheel-rail contact; this scenario may be referred to as perfect curving. In contrast, a wheelset which is constrained within a vehicle or bogie is normally unable to achieve a radial position. Consequently the rolling radius difference, coupled with an angle of attack (the angle between a line normal to the axle centreline and the tangent of the curve), leads to the generation of potentially high wheel-rail contact forces. The magnitude of the contact forces is influenced by the plan view compliance of the primary suspension system (often characterised by the primary yaw stiffness (PYS)) and by the radius of the curve and the installed cant – or more precisely the degree of cant deficiency.

The energy dissipated at the wheel-rail contact may be calculated as the product of the tangential force 'T' and the creep (or micro-slip) 'γ' that has occurred over a given distance. This product (force x distance) gives the work done per unit distance of track, or in SI units J/m. Studies [3] & [11] have successfully correlated the magnitude of the measure Tγ with wear and with the propagation of RCF damage. The relationship between

Tγ, wear and RCF is not linear but for the general case lower Tγ values are favourable for maximising asset life of wheels and rails and minimising energy consumption.

The Tγ values were calculated for the leading bogie of the Spectrum wagon in tare, part laden and fully laden conditions, for a range of curve radii (from straight track down to 250 m radius) and for a range of cant deficiencies (from 100 mm cant excess to 200 mm cant deficiency), totalling 247 cant/curve track cases for each vehicle.

To aid the comparison of the data the average Tγ was taken for all the cant deficiency cases of each curve radius, thus giving a more generalised indication of the performance for a range of conditions. The mean Tγ values are plotted against curve radius for the wheel-rail contacts of the leading wheelsets of the outer and centre bogies of the Spectrum vehicle in Figures 9 to 8. For comparison the Tγ values are also plotted for a generic Y-Series intermodal freight wagon. The colours denote similar load conditions and allow comparison of the Spectrum vehicle's performance, i.e. the black solid line (for the outer bogie of the Spectrum vehicle in laden condition with 143 kN axle load) and the black dot-dashed line (for the centre bogie of the Spectrum vehicle in laden condition with 197 kN axle load) are both below the red dotted line (for the Y-Series vehicle in laden condition with 218 kN axle load). The key to the Tγ plots is shown in Figure 7, where the load in brackets is the axle load. The differing axle loads are a consequence of configuring the articulated and conventional vehicle models with similar payload cases, and it is acknowledged that they complicate comparisons between the two vehicles.

With some inspection it may be concluded from these results that, in the general case, the Spectrum vehicle is expected to generate less wheel and rail damage as a result of curving forces than a Y-Series vehicle.

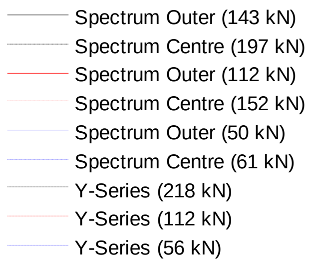

Figure 7 Key to Figures 8 to 10

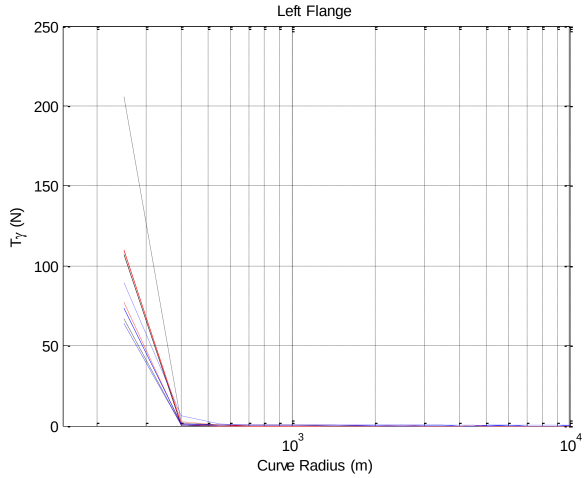

Figure 9 Mean Tγ for left flange contacts

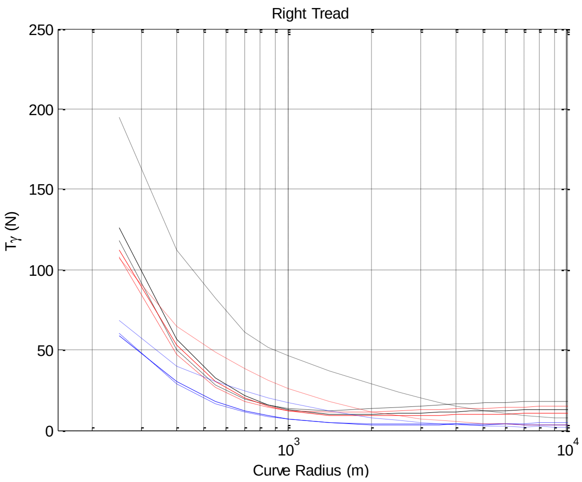

Figure 8 Mean Tγ for right tread contacts

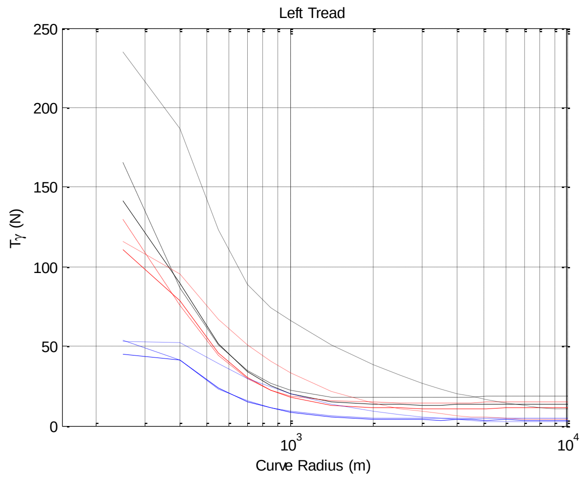

Figure 10 Mean Tγ for left tread contacts

5.3. Vertical Performance

The EN 14363 on track tests include an assessment of the dynamic vertical forces (Q_{dyn}) to ensure that vehicles do not generate excessive vertical track forces which would lead to disproportionate levels of damage. In the UK a more refined assessment of dynamic vertical forces forms part of the track access charging calculation for freight vehicles. The assessment compares the standard deviation (SD) of the vertical wheel-rail contact forces against the vertical SD of the track. A linear trend line through a data set of $Q_{dyn\text{-}SD}$ against track vertical SD provides a ride force coefficient (from the gradient of the trend line) and a ride force constant (from the y-axis intercept of the trend line).

A process similar to the UK access charging assessment was undertaken for the Spectrum vehicle in tare, part laden and fully laden load conditions. The track file used for this assessment was the same as that used for the EN 14363 'On track tests' and is therefore representative of the range of track qualities expected when running at 160 km/h. For comparison a conventional Y-Series vehicle was also assessed on the same track but at the normal operating speed of 120 km/h.

The SD of the vertical wheel-rail contact forces were calculated for 200 m track sections. The vertical SD of the track was calculated for the same 200 m sections. To ensure a fair comparison of performance the vertical force SD was normalised by the vehicle's axle load. The subplots in Figure 11 show the normalised vertical track forces plotted against the SD of the vertical track alignment (blue circles) for the Spectrum vehicle model (left column) and the Y-Series vehicle model (right column). The linear trend line is shown in solid red, and an additional line at 3σ from the trend line is shown in dashed red. In all cases the vertical track forces per kN of axle load are generally lower for the Spectrum vehicle than for the Y-Series. For the fully laden cases the difference is less obvious as the static axle load represents a more significant portion of the dynamic wheel load.

The data is shown in a different manner in Figure 12, where the normalised vertical force SD is plotted against the track section number. The mean of the force SDs is shown by the red dashed line and the magnitude is indicated by the text in each subplot. The differences between the vertical performance of the two vehicles can be clearly seen, especially in the tare and part laden cases – again in the fully laden case the difference a less apparent due to the increasing influence of the static axle load. In the tare case the mean of the vertical force SDs is nearly 40% less for the Spectrum vehicle than those for the Y-Series vehicle; in the part laden case the Spectrum values are over 60% less than the Y-Series values; and for the laden case the difference reduced to be nearly 20% less. The reader is reminded that in this analysis the Spectrum vehicle was run at 160 km/h while the Y-Series vehicle was run at 120 km/h.

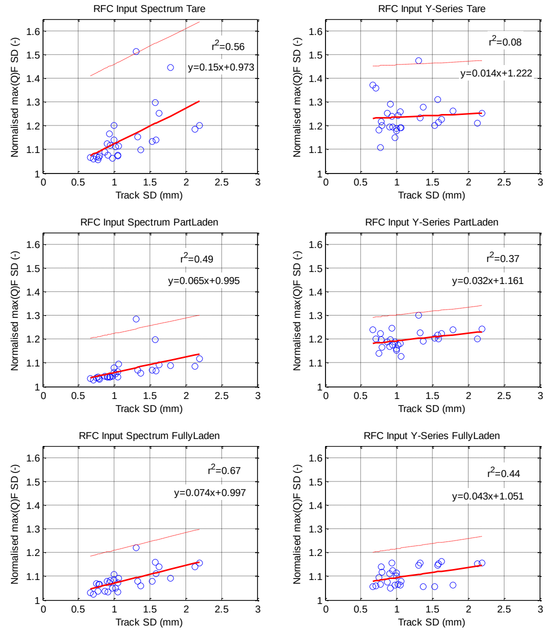

Figure 11 Normalised vertical force SD versus vertical track alignment SD

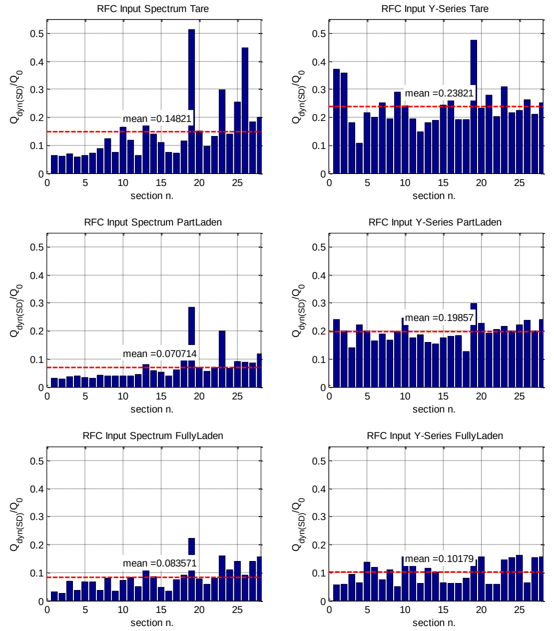

Figure 12 Normalised vertical force SD versus track section number

From this analysis of the vertical track forces it is apparent that the Spectrum vehicle should generate significantly lower vertical track forces compared to a conventional Y-Series vehicle. Consequently vertical track damage should also be less.

5.4. VUC Calculations

The Spectrum vehicle has been designed to provide higher dynamic performance compared to conventional freight vehicles. Primarily this was to achieve high speed running and improved ride quality for the LDHV goods. The running gear components necessary for this improved performance have a penalty of being generally more expensive than conventional freight bogie components. It is therefore likely that incentives would be required to maximise the uptake of such novel technology and improve the business case for investment in the rolling stock.
As well as benefiting the customers and cargo, the improved performance of the Spectrum vehicle also has the potential to benefit the infrastructure: in terms of lower vertical track forces and reduced curving forces, which will reduce track degradation and rail damage and associated maintenance costs. The full benefits of the novel technology can therefore only be seen from a holistic view point.
In continental Europe no differentiation is made between freight vehicles of differing dynamic performance when it comes to calculating track usage charges, despite the fact that some vehicle types will contribute to rail and track damage more than others. The track usage charges therefore provide a potential route to incentivising higher performing freight, by more fairly apportioning the maintenance costs based on a vehicle's contribution to rail and track damage.
In the UK a Variable Usage Charge (VUC) has been developed [1], which is based on a vehicle's vertical dynamic performance and curving behaviour (amongst other factors).
To demonstrate how a VUC might benefit higher performing freight in Europe a case-study was completed using vehicle dynamics simulation output and Network Rail's VUC calculators[1]. The case-study compared the discount

[1] The Network Rail calculators and documentation used in this case-study are in the public domain and may be freely downloaded from Network Rail's website:
http://www.networkrail.co.uk/using-our-network/cp5-access-charges/

factors which would be applied to the Spectrum vehicle and to an equivalent generic Y-Series vehicle. The process is described in the following paragraphs.

The calculation of the discount factor is based upon the vehicle's dynamic vertical ride forces obtained from vehicle dynamics simulations. The ride forces are processed to provide a ride force metric (RFC). The RFC is calculated separately for tare and laden conditions, and is then converted to a level of suspension Discount Factor. A spreadsheet calculator developed by Network Rail is used to determine the discount factor for a vehicle. For this process, vehicles in their tare and laden conditions are run over a specific track named Track for Banding (TfB) and respective Ride Force Constants and Coefficients (RFCC) are determined for each vehicle. The RFCC are based on the correlation between the standard deviations of the vertical ride forces and the track vertical irregularities (as was presented in Section 5.3).

For the case-study the VUC calculation process was carried out for the Spectrum vehicle and for a generic Y-Series vehicle which represented a 60' container flat. Table 1 below summarises the key vehicle parameters for the analysis.

Table 1 Vehicle Data

Vehicle	**Spectrum**	**Y-Series**
Type	Freight wagon	Freight wagon
Number of Axle	6	4
Tare Weight (in tonnes)	31.9	22.9
Laden Weight (in tonnes)	94.8	89.0
Unsprung Mass (in kg)	1402	1402
Freight commodity	Domestic intermodal	Domestic intermodal
No. 20' containers (TEUs)	4	3

The $T\gamma$ tables and suspension factors were input into the VTAC calculator for both vehicles in tare and laden conditions to calculate the VUC in £/kGTM (pounds per thousand gross-ton miles).
To allow comparison of the two vehicle types, which have different capacities, two costs were calculated:

- For the tare condition, the cost to transport a Twenty-foot Equivalent Unit (TEU) of capacity 1000 miles was calculated in £/kTEU.M (pounds per thousand twenty-foot equivalent unit miles)
  Cost per TEU = (VUC x Tare Weight) / vehicle TEU capacity

- For the laden condition the cost to transport one ton of payload one thousand miles (£/kNT.M [pounds per thousand net ton miles])
  Cost of carrying payload = (VUC x Laden Weight) / (Laden Weight - Tare Weight)

The calculated costs are compared in Table 2 below and are summarised as percentage cost saving in Table 3.

Table 2: VUC and Cost calculations

Vehicles	**Spectrum**		**Y-series**	
Loading Condition	Tare	Laden	Tare	Laden
VUC (in £/kGT.M)	1.3585	2.3847	1.5242	3.1745
Tare Cost (£/kTEU.M)	13.827	--	14.988	--
Laden Cost (£/kNT.M)	--	3.595	--	4.275

Table 3: Cost Benefit for Spectrum vehicle

Cost savings for Spectrum versus Y-series (%)	
Tare VUC	10.9
Laden VUC	24.9
Tare vehicle per container (£/kTEU.M)	7.8
Carrying payload (£/kNT.M)	15.9

From Table 3 it can be seen that, in tare condition the Spectrum vehicle would cost approximately 8% less per twenty foot equivalent unit transported, and in laden condition the cost would be approximately 16% less per ton of payload carried.

6. CONCLUSIONS

A new suspension arrangement has been developed for the articulated Spectrum intermodal vehicle. The arrangement combines viscous damped, coil sprung trailing arms and a UIC centre bowl and side-bearer arrangement with the addition of lateral swing links at the centre bowl. The components of the suspension system, while not being new in design, are applied in a novel arrangement and application – providing dynamic performance closer to that of passenger stock, while minimising increased cost and complexity.

The research found that the high lateral stiffness present in the standard UIC secondary suspension arrangement imposes limitations to vehicle stability, which were exacerbated for the articulated Spectrum vehicle. Decoupling the lateral bogie modes via a reduced secondary lateral stiffness was found to significantly improve stability.

The novel bogie design was assessed against the requirements of EN 14363 (on-track tests and for safety against derailment). Additionally the curving performance and vertical track forces were assessed through evaluation of $T\gamma$ values and the ride force constants and coefficients respectively. The Spectrum concept was found to have performance benefits when compared to a model of a conventional freight wagon with a Y-Series type bogie.

The benefits were quantified by applying Network Rail's Variable Access Charge calculator for determining discounts applicable for the Variable Usage Charge. The analysis found that in tare condition the Spectrum vehicle would cost approximately 8% less per twenty foot equivalent unit transported and in laden condition the cost would be approximately 16% less per ton of payload carried.

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