RAILWAY TRANSITION CURVE STEERING WITH YAW ACTUATED VARIABLE STEERING BOGIES

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Abstract

Curve transition steering has not been previously considered in idealised steering analysis and active steering bogies have been developed with considations focused on the constant curving difficulties. A new bogie design, called an AY-VS bogie, is presented allowing yaw actuation of the bogie with variably controlled steering angles. This design is an advance on the previously published AY-FS bogie. Yaw angle and steering angle calculations have been performed for a linear transition are used to identify the steering angle correction required for AY-VS in a curve transition. The bogie is shown to provide improved stability performance over AY-FS. Curving improvements have also been achieved.

1   Background

The previous studies of railway steering ideals, [1], [2], [3], have only treated curves where the radius and cross level and cant are constant. Railway tracks also feature transition sections between constant radius curves and tangent track which are sometimes referred to as spirals as both the cross level (or cant) changes as the curvature changes. Most railway systems use linear transition curves where both the cant and curvature change linearly through the transition.

Past studies by the authors [3], [4], [5] have shown: That the steering task for hauling locomotives is more difficult due to the saturation of wheel rail creep forces with traction loads; The steering performance of traction bogies is sensitive to the lateral loads; Steering of traction bogies is best achieved when the bogie yaw angle and bogie steering angle are achieved independently from the wheel rail creep forces. The use of a yaw actuated forced steered bogie has been shown to give superior traction steering in constant radius curving and passive forced steered bogies out perform other passive steering bogie designs.

Forced steering bogies refer to a design type where the yaw angles of the end axles are linked to the yaw angle of the bogie frame relative to the vehicle body. Wickens [6] reports that Liechty was successfully in developing forced steering bogies in the 1930's. A modern application of forced steering has been used by Japan Railways Hokkaido on narrow gauge railway for 140 kph express passenger trains [7]. A variation of forced steering occurs with articulated vehicles where the angle between adjacent vehicle bodies is used to provide a set steering angle to the bogie wheelsets. Articulated vehicle bogie steering is in common usage amongst tramway vehicles where very tight curvatures are experienced. Design of forced steering bogies is typically made to achieve a radial alignment of the axles with yaw alignment of the bogie frame to the rails. Recent studies of force steered bogies have identified improvements in lateral wheel forces from increasing the steering angle produced from bogie yaw rotation to exceed radial steering, [7]. However such adjustments to the linkage arrangement has a direct negative effect on the bogies stability performance.

Force steered and articulated steering bogies in particular will result in comparatively higher levels of track degradation during the curve transition. For forced steering in transition curves, the bogie yaw angle and the steering angle of bogies is no longer proportional, as the steering angle required by bogie wheelsets depends on curvature under the bogie itself and the bogie yaw angle on the curvature under the whole vehicle. The alignment error for forced steering becomes greater as the rate of curvature change increases and as a result is worse for shorter transitions.

2   Transition Steering Task

The steering task for constant radius curves [1], [2], [3], has been reported previously. The objective with an idealised transition steering is to minimise the wheel rail creep forces and minimise the track shifting forces. In a

curve transition there are additional factors not present in constant radius curves. Throughout the curve transition the vehicle body and the bogies under go yaw acceleration and decelerations as angular velocity changes with the curvature. There is also a change in the lateral forces as the centrifugal acceleration changes with increasing curvature. There is a requirement to change the lateral tracking position of the wheelsets.

There are also other potential complications from changing wheel loads. The wheel loads are effected by track twist of the transition and by vehicle sway motions that occur from the introduction of lateral rail forces. The wheel load complications make the concept of perfect steering [1], [2], where lateral forces are balanced by creep forces, much more challenging again showing the advantage of achieving ideal traction steering [3] (where yaw moments from longitudinal creeps are permitted). In any case a yaw moment is required in order to provide yaw acceleration to the bogies.
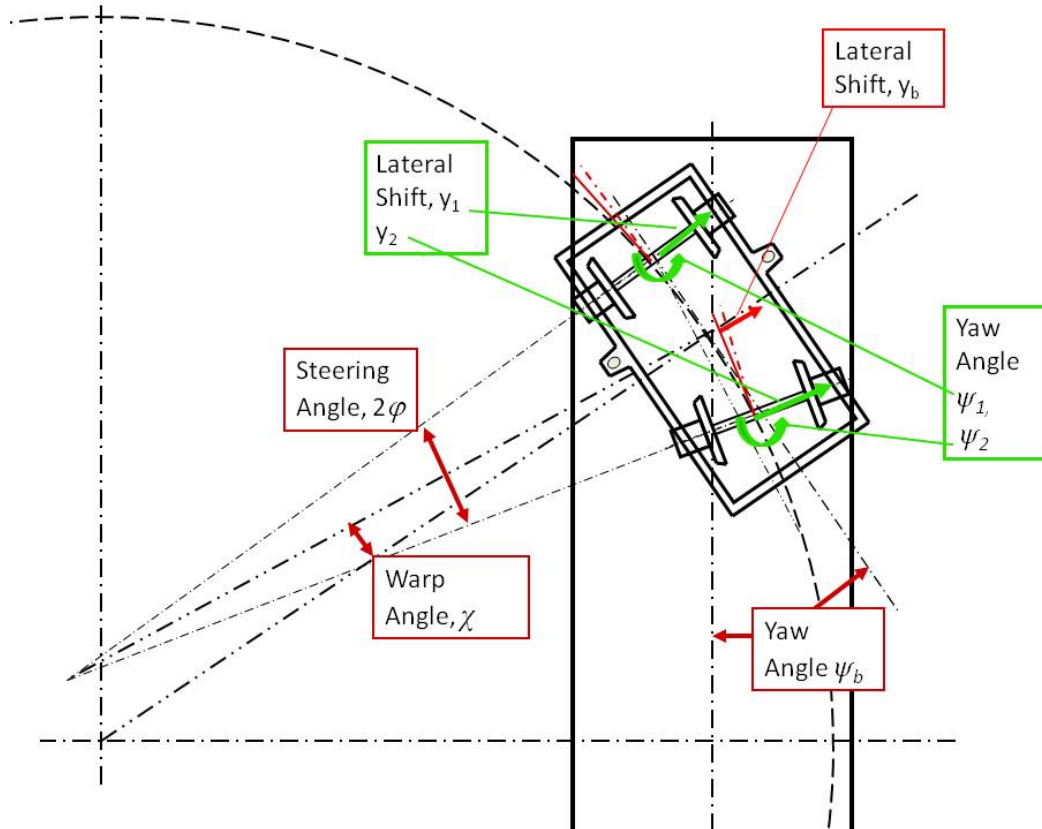

Figure 1 Wheelset steering angles and degrees of freedom

The authors then have used their concept of ideal traction steering from constant curving for transition steering. The ideal traction steering concept uses the bogie yaw and steering angles to minimise the lateral creep forces. The maintaining of the vehicles lateral force and vehicle yaw moment balance is assumed to be achieved with wheel rail lateral contact loads from gravitational stiffness. Lateral loads on each wheelset in a bogie are to be evenly distributed thus producing minimal track shifting forces. To balance the wheelset lateral loads and minimise lateral creep forces we must determine the appropriate bogie yaw angle and steering angle during curve transition whilst the warp angle of the wheelsets is best kept rigid and near zero for stability.

Under this concept the ideal traction transition steering requires a bogie design permitting independent control of the wheelset steering angle and the bogie yaw angle. The wheelsets will need to travel small lateral distances on the track gauge but the angle is small due if the transition curve length is several times longer than the bogie spacing. The presence of track alignment irregularities or changing rail profiles is ignored here.

2.1   Transition Steering Angles

In order to actively steer to the ideal traction steering concept the required bogie yaw and bogie steering angles need to be determined. To calculate the bogie yaw angle and the steering angle required in linear curve

transitions an equation for track curvature of the transition is given in Equation 1. Then the steering angle for the bogies can be calculated by using a small angle simplification and integrating Equation 1, (Equation 2). Assuming the vehicle centre is distance zero, then integrating for the half bogie centre distance in each direction determines the bogie yaw angle. That results in Equation 3, the required yaw angles. Similarly integrating for the wheelset centre distance of each bogie gives the steering angles as given in Equation 4. Using Equation 3 and Equation 4 the variation then in the steering angle needed to correct a forced steering wheelset steering angle in a transition curve is Equation 5.

Equation 1 Transition curvature with distance

$$\mathrm{K}(x) = M \cdot x + C$$

Where: K(x) is the curvature at distance x;
   M is the change in curvature;
   C is the curvature at distance zero;
   x is the distance along the track.

Equation 2 bogie yaw angle

$$\psi_i = \int \left(\mathrm{K}(x) \cdot x \right) dx$$

Equation 3 Required bogie yaw angles

$$\psi_i = C \cdot A \pm 1/3 \cdot M \cdot A^2$$

Where: ψ_i is the bogie yaw angle of the bogie i;

Equation 4 Bogie steering angle

$$\varphi_i = h \cdot \left(C \pm M \cdot A \right)$$

Where: φ_i is the half steering angle of the bogie i;
   h is the half wheel base between end wheelsets of the bogie;
   A is the bogie centre semi spacing

Equation 5 Variation in the half steering angle from forced steering

$$\Delta\varphi_i = \pm 2/3 \cdot M \cdot A \cdot h$$

3 Yaw Actuated Variable Steering Bogie Design

Simulations of constant curvature traction steering capable bogie designs such as forced steered and yaw actuated forced steered bogies show increased wheel wear rates in transition curves, [4], [5]. These bogies achieve steering angle alignment by making the steering angle a match to the bogie yaw angle. For a curve transition though the steering angle needed at each bogie varies from the constant proportion of the yaw angle due to the change in curvature, Equation 5. A new design of bogie is proposed by the authors allowing yaw control of the bogie with variable adjustment to the steering angle position, [8]. Figure 1 gives a general layout of the bogie in a transition curving position. Yaw actuators (70) control the bogies yaw angle and the steering actuator (71) controls the variation of the steering angle which is required for transition curves. The steering actuator moves the end pivot (72) on the body linkage (67).

The yaw actuated variable steering bogie is described thought the paper as an AY-VS bogie. The steering linkage arrangement for AY-VS bogie as depicted in Figure 2 is but one possible arrangement. All arrangements feature a body linkage (67) offset from the rotation centre of the bogie and the end pivot (72) is actuated to effect variations in the wheelset steering angle. The AY-VS bogie can be designed in either two or three axle arrangements. The appendix A shows the three axle arrange for tangent and constant curves together with a two axle arrangement in a transition position.
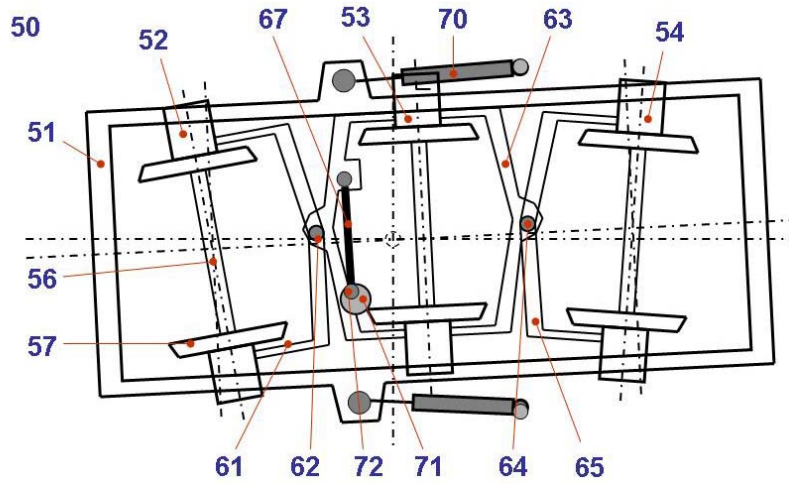
Figure 2 AY-VS Bogie, three axle arrange in transition curving [8]

3.1 AY-VS Bogie Control Methods

The AY-VS bogie gives separate and direct control of both the bogie yaw angle and the bogie steering angle. In active control the stability of the bogies improve with the application of a control that includes the negative derivative of the yaw misalignment [5] [9] and the proportional control of steering angle misalignment, [9]. The limitation is that the negative derivative yaw control impairs the transition curve steering of the bogie. Furthermore, the steering angle control potentially excites oscillation of the steering angle. Control input and output delays are also significant in impeding the bogie instability.

The control task for steering around curves is complicated by the transition. As discussed in section 2.1 in a curve transition the yaw angles for each bogie are no longer directly proportional to the current track curvature and force steering gives an incorrect steering angle. In a linear curve transition the yaw angle for ideal steering is related to both the curvature and the change in curvature Equation 3. In constant curving the change in curvature *M* is equal to zero the equation simplifies. The steering task for an AY-VS bogie requires the variable steering actuator, (item 71 in Figure 2) to correct the steering angle from the force steered position. The steering angle correction is shown in Equation 5.
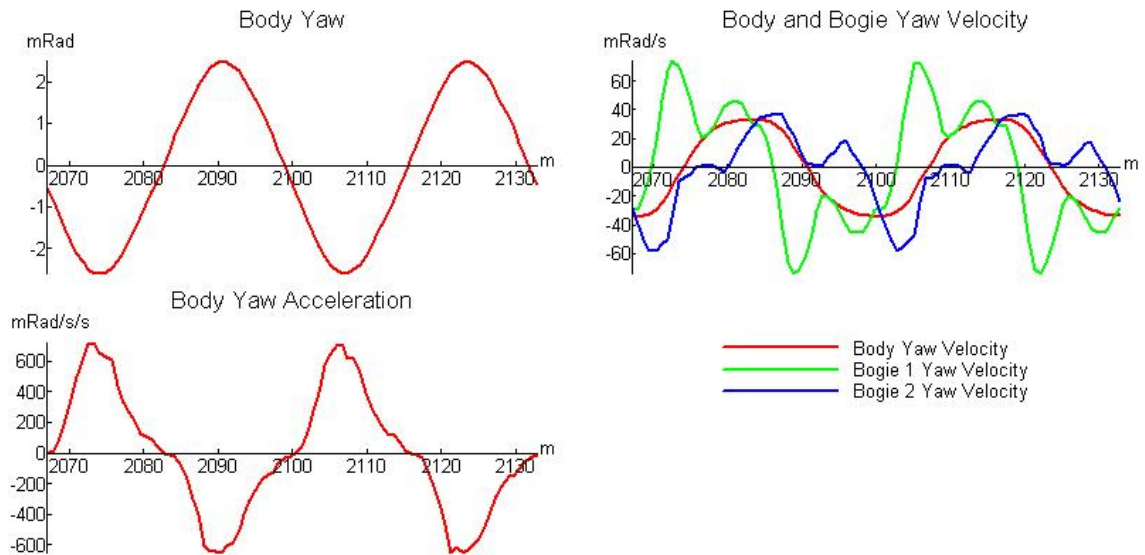

Figure 3 Force steered bogie hunting instability

As previously found when using curvature estimates for control [5], actuator movements for the change in curvature can act to excite the instability modes, particularly the front bogie hunting mode. Thus it is important to isolate the change in curvature estimates from frequencies that could excite bogie instability mode. The best

method is to use the vehicle body yaw accelerations as the oscillations in body yaw during hunting are smaller compared bogie motions and have less of the high frequency components of flange contact, see Figure 3. However the body yaw motion still experiences high frequency noise from flange contact impacts that need to be filtered to maintain stability and the body yaw acceleration needs to be filtered.

Steering angle actuation can excite a second instability mode in the bogie as found in AWY bogies [9]. The steering angle actuation induced instability (second mode) in an AY-VS bogie is similar to the rear bogie hunting mode in forced steering. Figure 4 shows that the hunting motion of the yaw and steering angles of both bogies in forced steered vehicle have distinct differences. The steering mechanism should force the steering angle of the front bogie in phase with the yaw angle where as the rear bogie yaw and steering angles should be in opposing phase. In Figure 4 it can be seen that the steering angle on the front bogie is nearly 90 degrees out of phase (Steering angle peak before the yaw angle peak) and is driving the instability. The hunting has a wave length of 30 m but steering angle has additional motions at roughly 3 times higher frequency which is a wavelength much closer to the kinematic wave length for the wheel conicity. On the second or rear bogie the steering angle has been phase shifted also to drive the yaw motion of the bogie. In rear bogie case the steering angle phase shift is in the opposite phase to the front bogie.
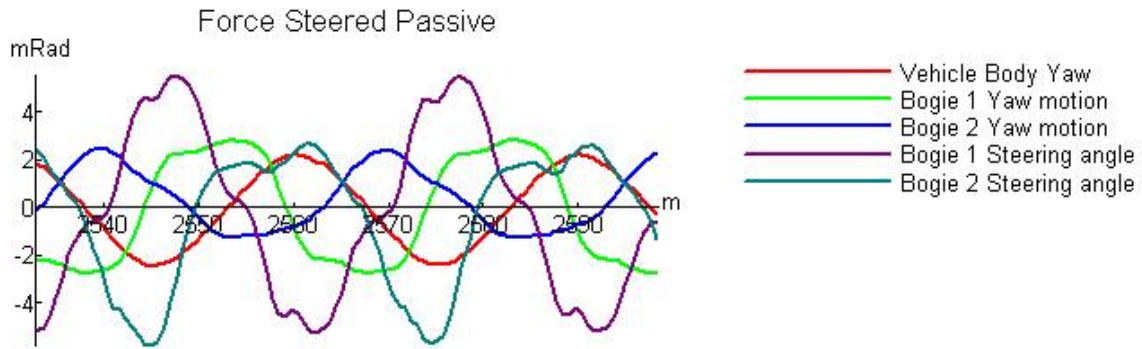

Figure 4 Hunting motion steering and yaw angles for a passive forced steered bogie

The steering angle control responding to rear bogie then needs to promote the opposite phase shift to the motion which for the rear bogie will now be opposite to steering. This makes steering actuation in AY-VS bogies problematic to the rear bogie stability. The best performance for a generic AY-VS controller is achieved by low pass filtering of the steering angle actuations to isolate the control from instability and not excite the rear bogie hunting. This approach relies then on the yaw actuation to give stability to AY-VS bogie designs in response to bogie-to-body relative yaw velocity. An alternative idea is to add steering angle actuation to oppose the yaw angle velocity of the bogie. This control addition, however, will impair the transition steering performance of the bogie.

The following simulations use a controller with yaw actuation to damping hunting instabilities by directly opposing yaw angle velocities. The alternative control using steering angle actuation to oppose motion in the steering actuator has also been modelled. Note that the phase shift in bogie steering angle during hunting is due to the flexibility of the steering linkages and suspension elements. Active control to compensate for the flexibility could further improve stability.

4   Simulation

The simulations in this study are performed using VAMPIRE® software version 4.32. The choice of VAMPIRE® software for use in this study was made based on its availability and ease of use. This is the same modelling package as used by the authors previously [4], [5], [9]. The limitations of the VAMPIRE® software are given in these references and are not repeated here.

The vehicle models used are the similar as those previously used by the authors [4], [5], [9], considering locomotives with three axial bogies. For each bogie design the model uses the same bogie frame and wheelset dimensions and masses. Two vehicle models have been used in this paper with designs having the three axles steered and with either AY-FS or AY-VS bogies. Six controllers have been used in stability testing two of which are only applicable to AY-VS bogies. Together with a passive operation this makes a total of 11 vehicle

configurations. A further idealised controller is used in the curving analysis. The active control has been implemented as either a precedence control having prior knowledge of the track alignment or as a curvature estimating controller sensing the yaw velocity of the vehicle body.

All the locomotives simulated are narrow gauge railway bogies with the gauge distance of 1065 mm. The basic parameters of the bogie suspensions are as given in Table 1. Primary yaw damping is provided by longitudinal elements at the axle boxes. The bogies have secondary yaw friction damping with friction co-efficient of 0.05 at each of four suspension mount points with x and y positions as are given in Table 1. The suspension setting for the AY-FS, AY-VS and the force steered bogies are identical excepting for the presence of a locking spring in the AY-FS bogie to restrain the steering actuator. All vehicles include a mass for the steering actuators. The steering actuated is modelled as separate mass connected to the vehicle body with free movement only in yaw. When simulating constant coupler force, as done in all the traction simulations of this paper, the VAMPIRE® simulation has a sudden longitudinal jerk on the vehicle body on initiation. If the steering cam mass is too small this causes the vehicle simulation to exceed acceleration limits in the software. As a consequence the AY-VS bogie simulations have been performed with slightly heavier locomotive. The locomotive has the additional mass of two 525 Kg steering actuators making the locomotive 115.636 Tonne reducing the adhesion level required in the simulation of 186 KN to 16.4 % adhesion. Four rail frictions have been simulated for curving as done previously the friction levels being: 0.175; 0.20; 0.35; and 0.50.

Table 1 Common Vehicle Parameters

Parameter	Value	Units
Vehicle Body mass	75.62	Mg
Body vertical centre mass	1.68	m
Bogie centres	12.77	m
Bogie mass	9.66	Mg
Bogie Vertical centre mass	0.603	m
Wheelset centres	1.88	m
Wheelset mass	3.26	Mg
Wheelset radius	0.515	m
Steering Actuator mass	0.525	Mg
Primary suspension stiffness z axis	2.00	MN/m
Primary damper z axis	0.30	MNs/m
Primary stiffness end axles y axis	12.4	MN/m
Primary stiffness middle axles y axis	0.04	MN/m
Primary stiffness end axles x axis	0.04	MN/m
Primary stiffness middle axles x axis	24.0	MN/m
Secondary suspension stiffness z axis	1.41	MN/m
Secondary suspension damper z axis	0.20	MNs/m
Secondary suspension damper x axis	1.0	MNs/m
Secondary suspension x position	1.06	m
Secondary suspension y position	0.74	m

All the bogie designs feature steering arms that connect the end axles to a steering frame that is centred over the middle axle as shown in Figure 2. The extra locking between the steering arm (see item 67, Figure 2) and the vehicle is only in the forced steered and AY-FS bogies.

Control of the AY-VS bogie actuation is achieved with estimated track curvature or known track curvature data. Curvature estimates are made from the yaw velocity and yaw acceleration of the vehicle body. Simulations have been performed on the curving performance under traction curving conditions in tight curves for a precedence yaw control, a curve estimating yaw control and an idealized controller. Both the precedence and the curve estimating control have been implemented with two levels actuation and sensing speeds. The low speed control is limited by a 16 Hz low pass filter of the controller inputs and outputs. The faster control called the version 2 controller has 26 Hz filters on controller input output. The version three controllers are only for AY-FS bogies and use the variable steering actuation to add a steering angle correction to nullify the effects of the yaw angle derivative. The variable steering controllers add additional steering angle during curving to compensate for the deflection of the steering links due to friction in the primary suspension.

4.1 Simulation Results Stabiltiy

Stability tests of the AY-VS bogies are shown in Figure 5 and Table 2. The results show the stability of the AY-VS bogie is similar to the AY-FS bogie when similar controllers are used. The additional dampening in the version three controllers improves the AY-VS bogie stability. (The critical speed increases by approximately 10 kph.) The stability of the curve estimation controller is poor with the 16 Hz response of the version one controllers. However with the faster response of the version two controllers using 26 Hz filters the precedence yaw control and the curve estimation control have near matching critical hunting speeds and better result than the version 1 controllers. An idealised (instantaneous) control gives stability until the primary suspension stiffness permits warp angle oscillation (critical speed of ~380kph –result not shown).

Table 2 Critical speed from stability testing, AY-VS bogie comparisons

Bogie Type	AY-FS-3 Precedence	AY-VS-3 Precedence	AY-FS-3 CE	AY-VS-3 CE
FS - Passive	159	159	159	159
Ver 1	192	197	161	160
Ver 2	217	209	207	211
Ver 3	n/a	220	n/a	225

The hunting instability motions for the passive force steered bogie the AY-FS bogie version 2 control and the AY-VS bogie version 3 control are shown in Figure 6 for a speed of approximately 250 kph. It is noticeable the addition of yaw control (AY-FS and AY-VS) has reduced oscillation of the more stable rear bogie. The wave length of the oscillation increases from 30 m for the passive force steered bogie to 33 m for the AY-FS then 36 m for the AY-VS bogie designs. The longer wave length oscillation decreases the angle amplitudes consequence as the rail gauge restraint limits the actual horizontal movement. Note the actuation control uses the same gain settings for both the front and rear bogies. Hence the control gain settings that have been adjusted though simulation testing to improve performance are a compromise between the lead and rear bogies. Further work on an optimised control strategy needs to use individual control gain settings for front and rear bogies. Whilst such an implementation then requires addition input to the controller to identify the direction of travel, the stability behaviour of the two bogies is sufficiently varied to suggest that performance gains are likely.
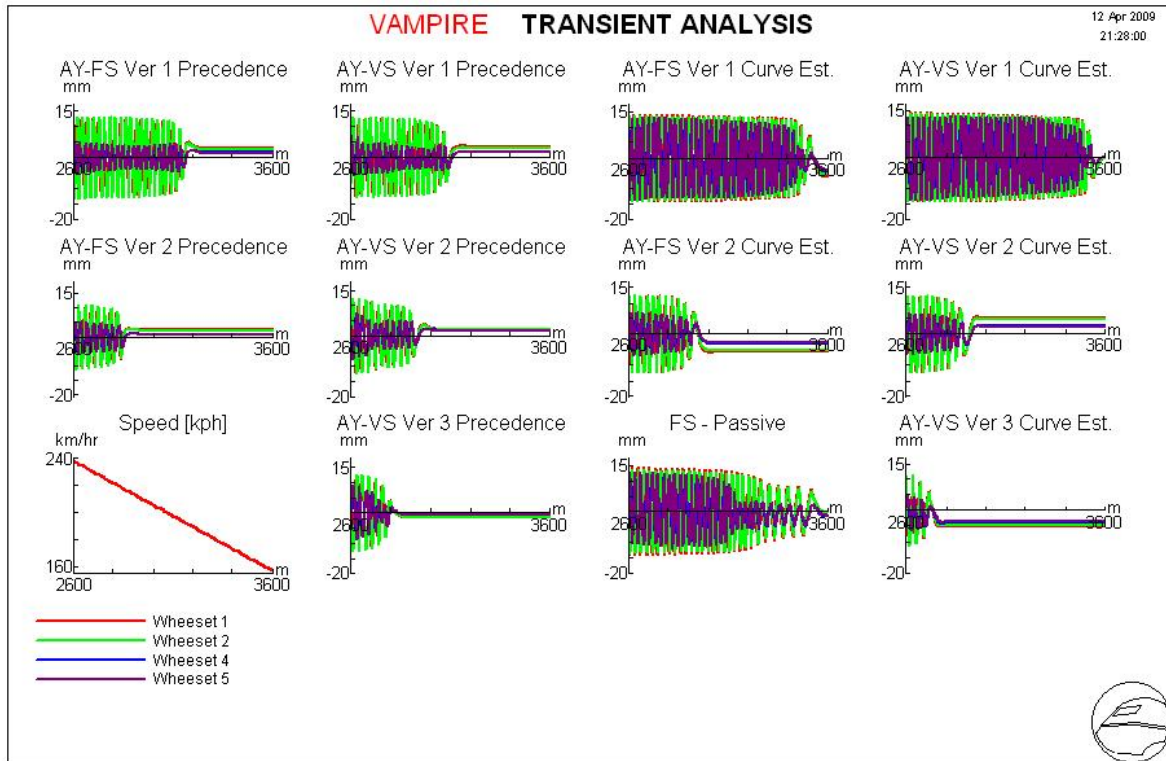

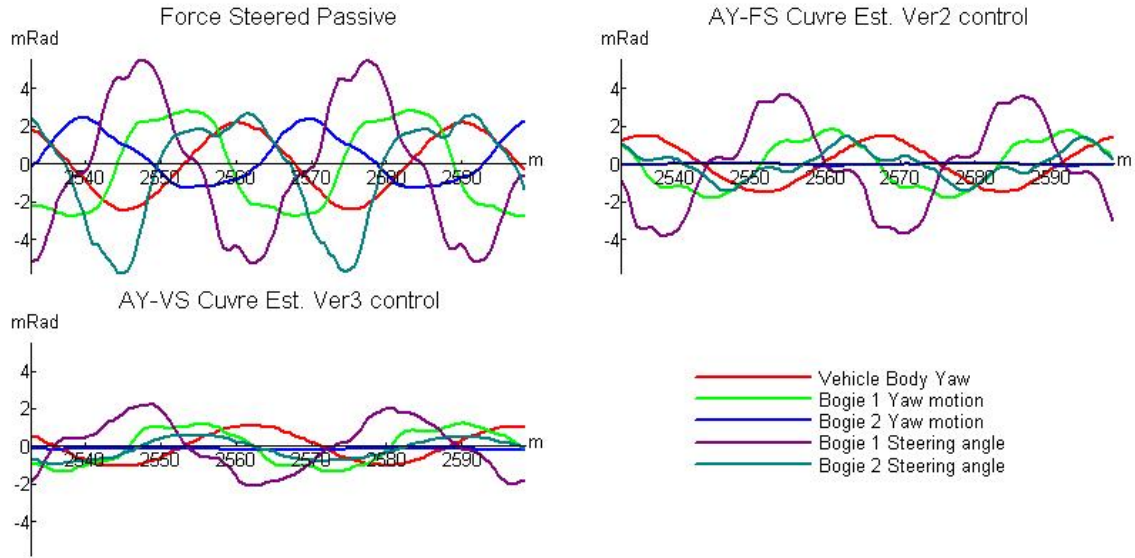*Figure 5 Stability test results for AY-VS, AY-FS and FS bogies.*

Figure 6 Hunting Instability motions with active control.

4.2 Simulation Results Curving

The traction curving performance is shown in Figure 7 as summed wheelset creep energy. Figure 7 presents the version 2 or fast response control results for precedence and curve estimating control together with the ideal control. Note for simulation the ideal control is effectively a precedence control with a response delay of almost zero (i.e ~1 kHz low pass filtering). The faster response delay of the ideal control allows for much stronger control signal gains to be used in the ideal controller resulting in better alignment against the modelled suspension friction. This is particularly true of the transition curves where both the faster response and the higher gains improve the yaw alignment to the changing curvature.
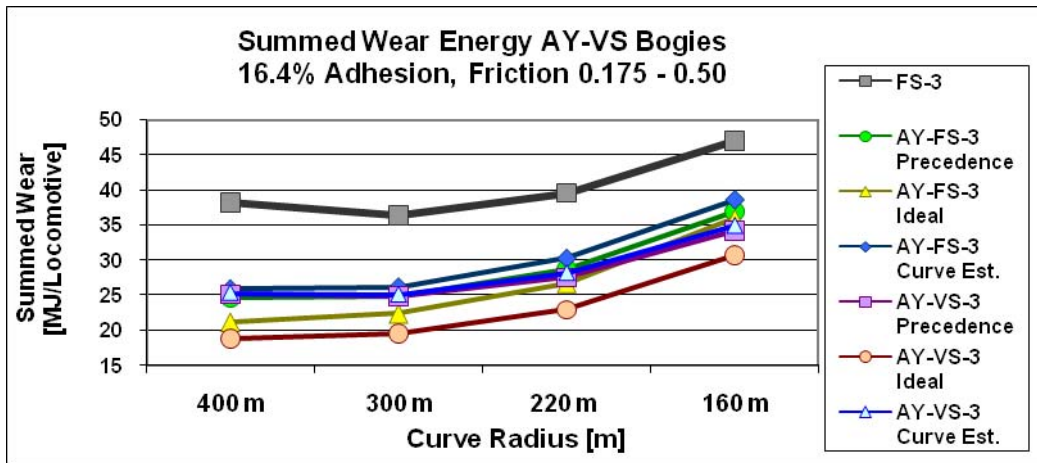

Figure 7   Traction curving performance, wheelset creep energy, for AY-FS and AY-VS bogies, version 2 and ideal controllers

The performance difference in curve estimation and precedence control is small. The curve estimation control has sensing delay in determining the start and end of the curve transitions and thus has slightly impaired curving performance. Wheelset alignment results during curving are shown in Figure 8 for curve estimating control for the AY-FS and AY-VS bogies. The wheelset misalignments that result are small and greater performance gains are associated with the bogie yaw alignment in the transition where wheelset flanging occurs. Hence in the ideal controllers have in Figure 7 much lower wear energy results for 400 m and 300 m curve radii where the stronger gain settings enable flange contact to be prevented. The tight 220 m and 160 m curve radii are too tight for the modelled wheel profile which has an effective concity at flange contact that is only sufficient to negotiate 356 m curves radius without longitudinal creepage differences.

8

The variable steering provides significant curving improvements for the idealised control. The variable steering improvements increase with change in curvature per unit distance, *M*, as a sharper curvature increase, results in greater error of the wheelset steering positions. The curve estimation and precedence yaw controllers used with variable steering require much smaller gains due to the slow application of the steering angle variation. The performance improvements are also dependent on the bogie semi spacing and wheelset semi spacing of the bogie as indicated in Equation 5. The results show that for the variable steering, worthwhile improvements will need to have the steering variation control performed with minimal delay as seen in the idealised control. It can be also noticed in Figure 8 that whilst the wheelset alignment gains AY-VS bogie on the front bogie are dramatically better particularly near the maximum curvature. On the rear bogie the wheelset 6 angle of attack remains high.
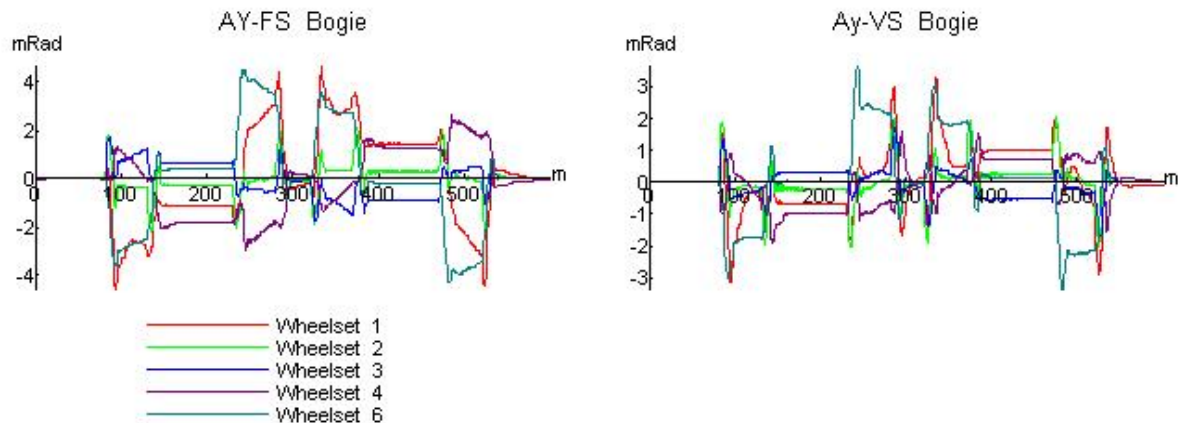

Figure 8 Wheelset angle of attack for AY-FS and AY-VS bogies, curve estimate control 160 m curves

5   Conclusions

Control of the AY-VS bogie has been achieved with estimated track curvature sensing the vehicle body yaw velocity or with precedence control knowing the track curvature data. Simulations have been performed on the curving performance under traction curving conditions in tight curves and the stability of the bogie has been examined using non linear simulation.

The stability testing found that using convention steering angle actuation in the AY-VS bogie contributed to the rear bogie instability due to the presence of the forced steering linkage which changes the normal phase shift of the bogie yaw angle and steering angle. Improved stability of the AY-VS bogie was achieved by actuating the steering angle to oppose the bogie yaw motion. The stability of the curve estimation control was found to be lower if the controller actuation response is slow but for faster response control an equivalent stability performance was achieved. The flexibility of the steering connections is important to the bogie stability considerations. Control gains used in testing were maintained as the same for front and rear bogies and further work should consider optimisation of independent controls for each bogie.

The result of the curving analysis are summarized in Figure 7. Performance improvements are dependent on a rate of track curvature change and the vehicle dimensions as indicated by Equation 5. Improvements from variable steering are also dependent on the yaw actuation responsiveness and accuracy. The performance of the yaw actuation in the curve transition being more important to the amount and severity of flanging contact during transition curving. The AY-VS bogie simulations showed improved angle of attack alignments for the front bogie more successfully than the rear bogie alignment.

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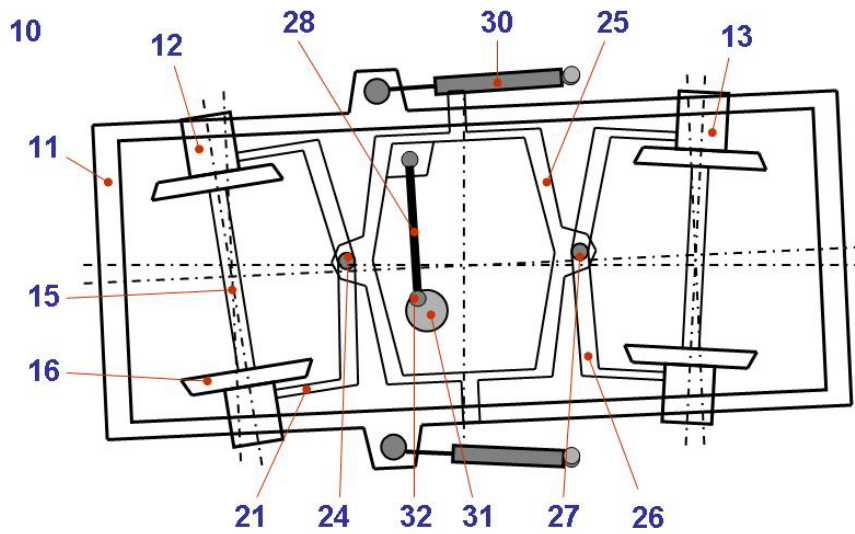
Appendix A

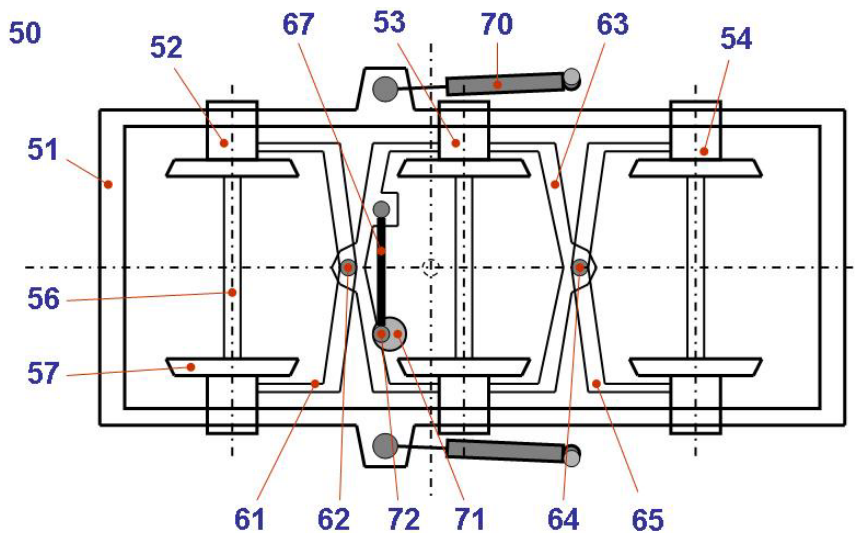*Figure 9 AY-VS Bogie, two axle arrange in transition curving [6]*

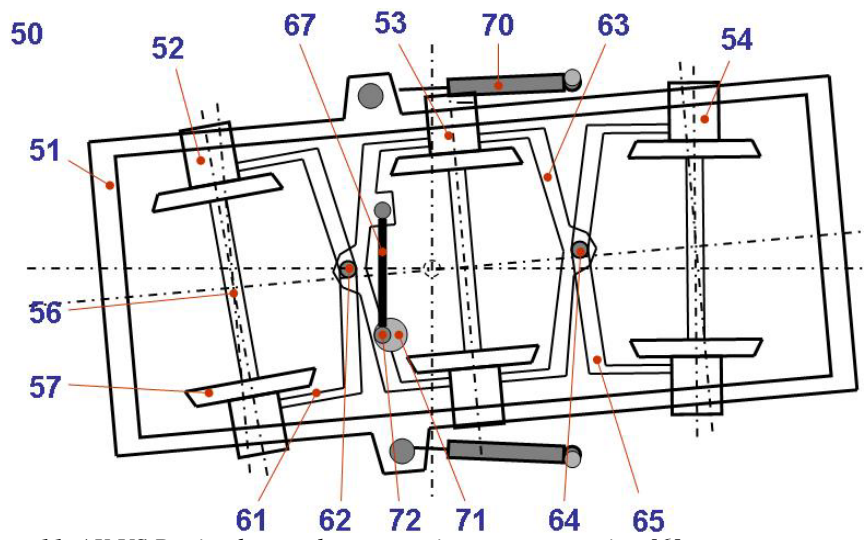*Figure 10 AY-VS Bogie, three axle arrange in tangent track [6]*

Figure 11 AY-VS Bogie, three axle arrange in constant curving [6]