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Centre Bearing Rotation Forces During Curve Transitions

Scott A. Simson
BE, Grad Dip IT, ME, RPEQ
Centre for Railway Engineering, CQU

Michelle E. Pearce
BE
Centre for Railway Engineering, CQU

SUMMARY

Centre bearing forces and wear losses in three piece bogies have been simulated using VAMPIRE®. Bogie rotational resistance has been modelled at centre bearings and constant contact side bearers accounting for uneven loading during track transitions. The detailed model handles cylindrical centre bearings accounting for rim wall contact as well as the centre plate planar friction connections and uneven loading. The simulations have been performed for a range of curve radii assuming gauge face lubricated track.

Wagons of moderate or high friction resistance to bogie rotation at centre bearings and side bearers have shown altered wheel wear, reducing maintenance cycle times in operations for rollingstock. In this paper, simulation studies have examined the effects of cant deficiency / excess and transition curvature on the wear performance of wagons and individual wheelsets, in particular the high wearing leading wheelsets. Wheel wear is shown to be altered due to changes in bogie rotation friction moments retained through the curve when cant deficiency of the curve is altered.

The simulation results also show that transition curvature design can be used to reduce high wheel/rail wear in 3 piece bogies generated by high bogie rotation resistance. In the same way, transition curvature design can be used to alter the wheelset angle of attack during constant curving on 3 piece bogies.

INTRODUCTION

Rail CRC Project 82 has been investigating bogie rotation friction management in 3 piece freight bogies. The cost benefits of managing bogie rotation friction levels come through limiting bogie hunting and poor vehicle curving performance, [1]. In both curving and hunting, 3 piece bogie performance is not only dependent on bogie rotation friction but is heavily dependent on wheel rail contact profiles and effects of the bogie suspension in warping (lozenge) and steering.

Related research within Rail CRC Project 82 [2,3] and by TTCI [1] has found that wear losses in gentle curves for 3 piece bogies increase significantly with centre bearing rotation friction. In very tight curves there is almost no change in the total wheel rail wear. The friction moment of bogie rotation however is dependent on rotational movement of the bogie which occurs only in the curve transition. Hence a closer examination of the bogie rotation movements and forces during curve transitions has been carried out to determine if transition design can be used to relieve or remove the effects of high bogie rotation resistance on curve wear.

BACKGROUND: BENEFITS OF BOGIE ROTATION FRICTION MANAGEMENT

Rail CRC Project 82 has found that high centre bearing friction affects the wear performance of bogies in curves in differing manners depending on the tightness of the curve [2,3]. Five curve types are identifiable by there curving behaviour.

- Tight Curves: when the curvature exceeds the total conicity offered by the wheel profile, all wheelsets in the vehicle are forced into flanging.
- Near Tight Curves: Three wheels, postions 1, 2 and 3 flange against the high rail. Such curves are only seen in bogies with low warp stiffness. The vehicle model in this paper has a higher warp stiffness.
- Moderate curves: when leading wheelsets in each bogie flange and trailing wheels are free to track between the rails.
- Gentle curves: Wheelset 3 flanging force / tracking becomes dependent on bogie rotation friction, and wheelset 1 is flanging.
- Tangent curves: The total conicity offered by the wheel profile allows the wagon steer around this curve without flanging.

Depending on the track curvatures wagons operate over, wear losses may be heavily affected or relatively unaffected by bogie rotation friction levels.

- Tight Curves: Wear rates increase on wheelsets 1 and 4 but decrease on wheelset 3 for little or no overall change.

- Near Tight Curves: Wear rates increase rabidly on wheelsets 1 & 2 and decrease on wheelset 3 for a large change in overall wear. Exceptionally high wear increases can occur with cant excessive curve due to wheelset 4 tracking near the low rail increasing wear rates dramatically on axles 1, 2 and 3 [3].
- Moderate curves: Wear rates increase on wheelset 1 but decrease on wheelset 3 for little overall change. Cant deficiency and the tracking position of trailing wheels effect the sensitivity to bogie rotation friction in moderate curves [2,3].
- Gentle curves: Large increase in the total wheel wear occurs due to both flanging on wheelset 1 and tread wear increases on bogie 1 due to increased angles of attack.
- Tangent curves: High percentage increase in wear occurs due to tread wear angle of attack and wheelset flanging in the curve transition. The actual wear energy though is very small as to be insignificant.

Wear results for varying curvature are given in Figure 1. Figure 2 shows the distribution of wear over the four axles for low and high bogie rotation friction levels. The percentage increase in total curving wheel/rail wear seen in Figure 1 shows a steady increase with a peak result at 1250 m radius. The peak in percentage wear at the 1250 m radius is due to the transition from a gentle curve type to a tangent curve type for the wheel profile being tested. At 1250 m radius the occurrence of wheelset 1 flanging wear in curving for the chosen wheel rail profile is dependant on the bogie rotation friction level. The percentage increase in wear total between the low rotation friction case and high rotation friction case represents the on set of flange contact during the steady radius part of the curve.

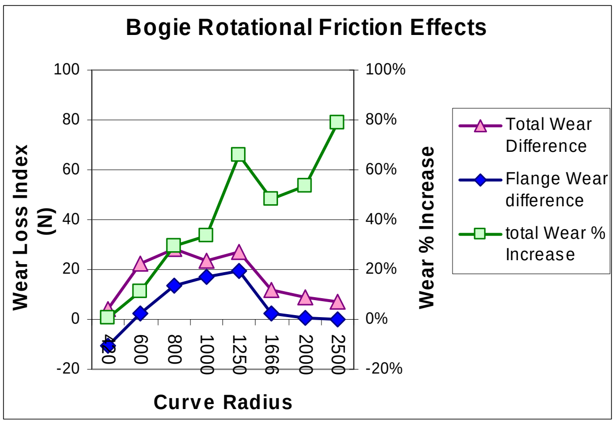

Figure 1 Wear Losses from High Bogie Rotation Friction

In Figure 2 the high wear rates for axle 1 on high bogie rotation friction wagons in tangent curve radiuses such as 1666 m radius and larger is due to the flanging contact that occurs in the inward curve transition on axle 1 with the high bogie rotation friction.

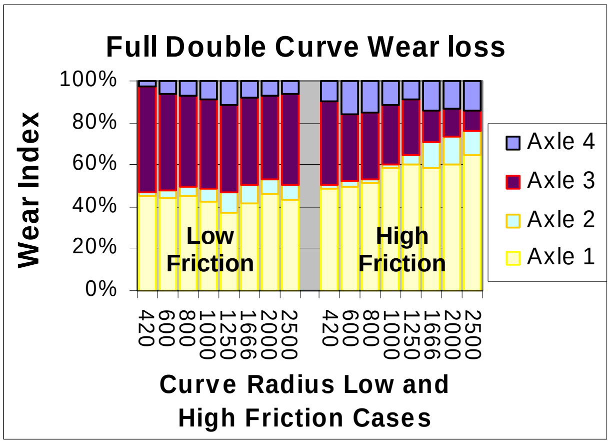

Figure 2 Percentage of Wheel Wear by Axle for Low and High Bogie Rotation Friction

SIMULATION MODEL

Simulation has been done with a 62 degree of freedom vehicle model using the VAMPIRE® vehicle simulation software. The simulation model uses a default 73 ton hopper wagon with 3.6 ton bogies on narrow gauge (1065 mm) track. The wagon in the loaded condition offers a high centre of gravity for body. The centre bowl for this model is 300 mm diameter top centre in a 315 mm diameter bowl liner. The model includes CCSB (Constant Contact Side Bearers) with either 6 or 15 KN preload on a wear pad, high friction coefficients 0.5 and a 6.5 mm gap to rollers located 570 mm from the wagon centre line. The 15 KN CCSB preload represents a large part of the empty wagon static load mass of 8.1 tonne. The simulation calculations in VAMPIRE® have been set at 20 kHz with the output data being analysed using just 1 kHz. Simulation runs are performed over double reverse curves of the specified radius with 50 m transition curves and default 2/3 cant deficiency.

Bolster Wagon Body Connection Model For Centre Bearing Loads During Transition

The centre bowl connection model has been developed from a model used for centre bowl impacts, [4] and is a more detailed full vehicle model than used by Wu [1] or Katta and Thomas [5]. The bolster to wagon body connection model includes nineteen non linear spring elements and thirty four friction elements (sixteen linked pairs for plate friction and two line friction elements). The model assumes a cut out chord on either side of the top centre to prevent point loading of the centre bowl during wagon sway movements. The cut out chord is for an eighty-one degree arc as seen in narrow gauge rollingstock operations. Figure 3 shows the location of spring and friction elements over the centre plate.

Six non linear springs and matching linked friction element pairs are used for the plate of the centre bowl as pictured in Figure 3. Eight non linear springs are used for the centre bowl rim with matching plate friction. Each rim spring element has 7.5 mm clearance. Each CCSB has a non linear spring and plate friction pair for the wear plate and a non linear spring and line friction element for the roller. The use of plate friction modelling is important to rotational friction modelling as it accounts for longitudinal, lateral and pitch movement effects on the rotational resistance of the centre bearing [4].

As shown in Figure 3 six spring connections are used to represent vertical loads at the centre bowl. The six spring connections are spaced at edge locations front and rear for bolster pitching point loading cases and corner locations for sway edge load of the chord cut outs. The six friction element pairs to match have been positioned to give the correct rotational friction to resistance and translational frictions. Rotational friction remains a little conservative for edge or point loads seen during curve transitions.

Note Well: Assumed in this model is that the king pin has no contact with the bolster except for vertical lifting forces.

Other Model connections

Sideframe bolster and the sideframe bearing adaptor connections are also modelled with non linear spring and friction damper connections. Bearing adaptors have 2 mm of longitudinal and lateral clearance from the sideframe and plate friction interaction. The tight lateral restraint was used to maintain the model's numerical stability but limits the model's ability in angle of attack modelling. The secondary suspension spring nest has stiffnesses in 6 axes. Additional lateral non linear stiffnesses are included for the bolster stops. Plate friction is used for the friction wedges affecting the warp movements in the bogie. The friction wedge model uses an altered friction coefficient for up and down strokes to account for the friction wedge angle.

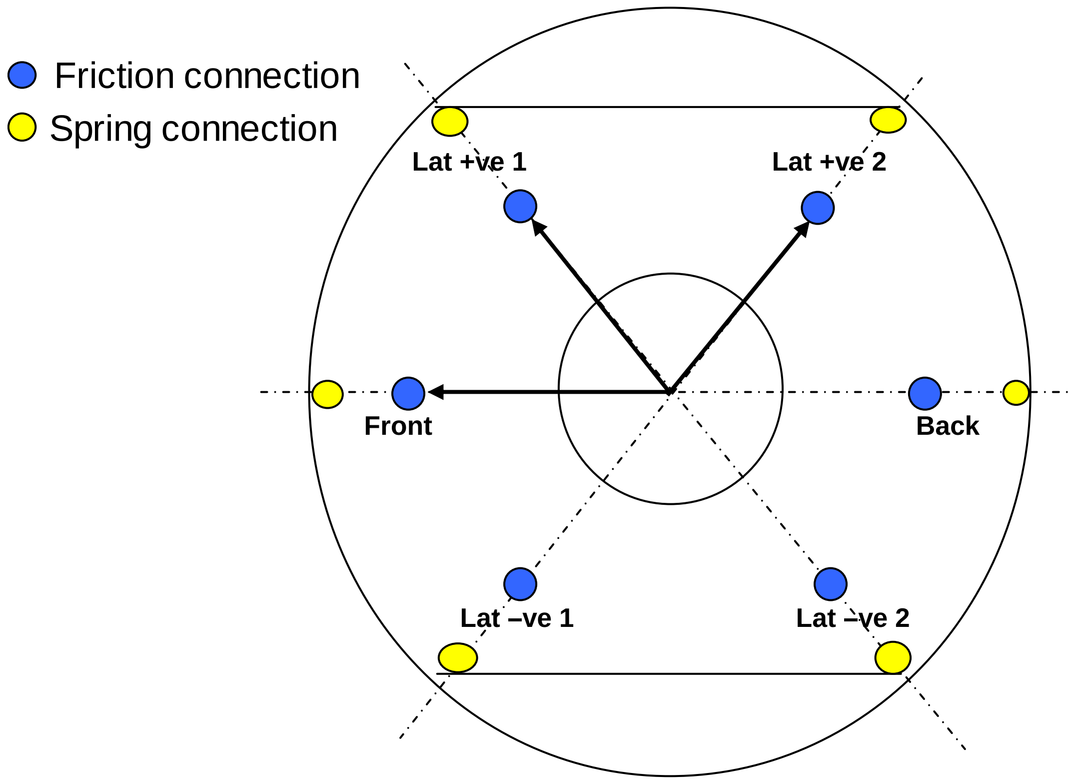

Figure 3 Centre Bowl Plate Spring and Friction Connections

RESULTS: TRANSITION CURVE FORCES IN CENTRE BOWL ROTATION

Vertical Forces

During the track transition curve the wagon experiences changing track curvature at the same time as changing track cant. This requires the wagon to twist at the same time as the bolster rotates. At a cylindrical centre bearing, point or line loads are experienced depending on top centre shape and the stiffness of the centre bearing contact. Figure 4 shows simulation results on the average vertical load for variable cant deficiencies during the approach transition at each of six model connections depicted in Figure 3. From Figure 4 it is evident uneven line loading of centre bearings occurs in transition curves on 420 m radius narrow gauge curve. Line loading is achieved on both front and rear top centre plates in any canted curve.

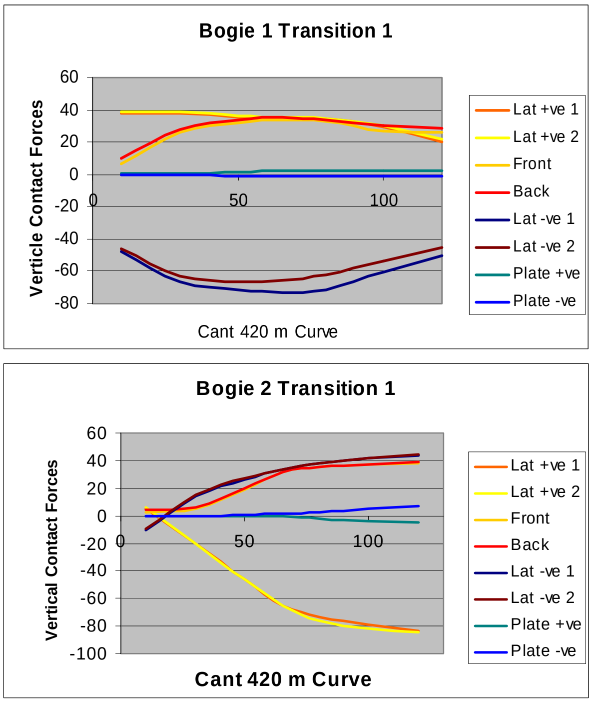

Figure 4 Average vertical loads at the centre bearing during track entry transitions

During constant curving, vertical loads at the centre bearing are dependent on the cant deficiency of the curve. Vertical loads at the centre bearing for the cants of the track at 60 kph on a 420m narrow gauge curve of the vehicle are shown in Figure 5.

In comparing Figure 4 and Figure 5 it is seen that for a cant deficient curve, the lead bogie is tilted in the same direction as experienced during the lead in transition and the rear bogie is rocked back to an opposite tilt as it exits transition in a cant deficient curve. As the cant is increased (reducing cant deficiency) the lead bogie is made to rock back from its curve transition position and the rear bogie stays in the same transition tilted line contact loading. Rocking of the vertical loads from side to side of the centre bearing unlocks the friction wedges of the secondary suspension, freeing the bogie to warp movement.

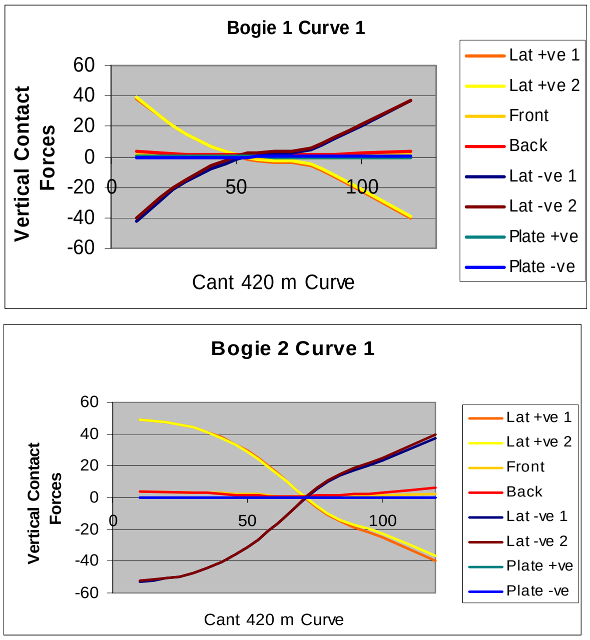

Figure 5 Average vertical loads at the centre bearing during track curves

Rotational Forces

Rotation friction moments have to be retained from the curve transition to the constant radius curve to have an impact on wheel and rail wear as the majority of wheel rail wear occurs during constant curving. Figure 6 shows how the vast majority of wheel and rail wear occurs in the constant radius of the curve and not in transition curves.

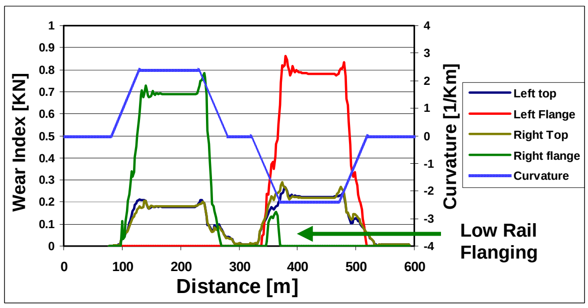

Figure 6 Rail wear vs distance 420 m radius curve 65 mm cant flange lubricated

Track twist over the wagon length in transitions generates uneven vertical loads as shown in Figure 7. Associated with the track transition is the rotational movement on the plate of the centre bowl. The uneven vertical load generates the line contact forces giving a longitudinal friction component to the bogie rotation friction.

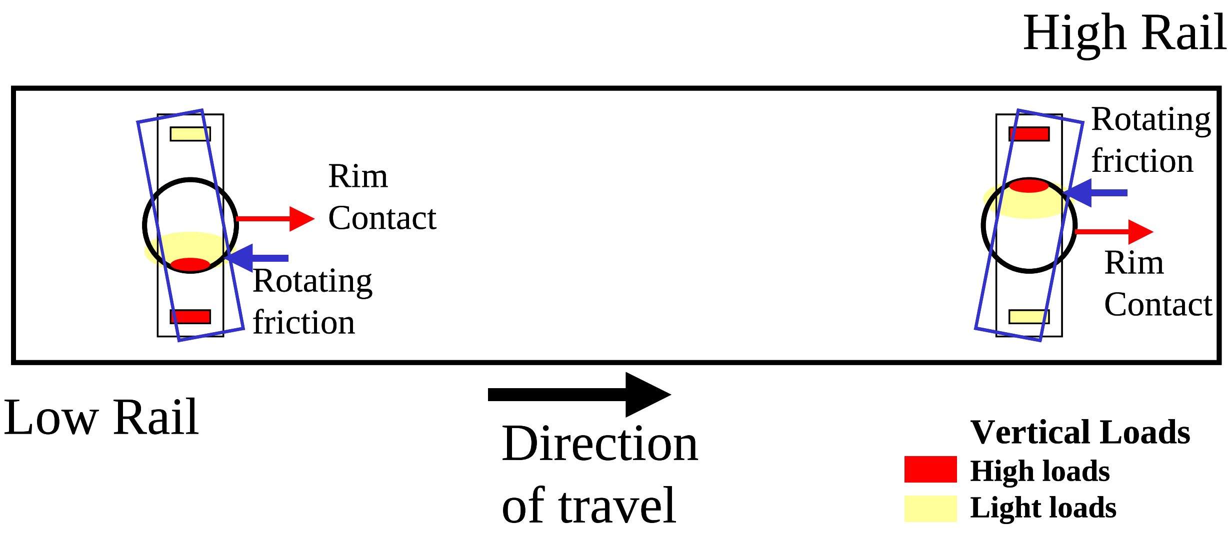

Figure 7 Transition Twist Generating Rim Contact Forces

As shown in Figure 7 the track twist and the rotation direction of both bogies is such that both bolsters are pushed backwards by the plate friction such that the rim contact between top centre and the rim of centre bearing liner occurs at the front or leading face of the top centre. The rim contact friction is then dependent on the friction levels on the plate of the centre bearing and the wagon's twisting stiffness to generate the contact force against the rim wall.

Simulation results show the retention of bogie rotation friction forces into the constant curve is dependent on the bogie warp deflection. In the transition curve the bogie will first warp at the spring nest then rotate at the centre bearing to accommodate the track curvature. The secondary suspension spring nest provides the elastic spring deflection to bogie warp movements. During the transition curve the spring nest yaws to allow the bogie to warp. When the yaw deflection reaches the friction limit for centre bearing rotation the bogie rotates. It is the warp deflection of the bogie that is retained into the curve and maintains the friction moment at the centre bearing. Any combined movement of the spring nest and the centre bearing will tend to relieve the warp deflection of the bogie and the friction at the centre bearing as there is no continued curvature change. The friction wedges also oppose yaw rotation of the spring nest. Thus relieving bogie warp is made easier when the wagon sways or bounces, to break the friction forces at the friction wedges.

CANT DEFICIENCY SIMULATIONS

Simulations where carried out to investigate the effect of cant deficiency on the wear results from high centre bearing friction. Rocking the leading wagon bogie on the centre bearing and bogie suspension was found to be a significant factor in the wear results. Figure 8 shows how rocking movements of the front bogie relieves rotational friction by plotting average rotational friction against the curve cant. Figure 9 shows the wheel wear effects of centre bearing rotational friction being nullified as cant is increased to cant excess.

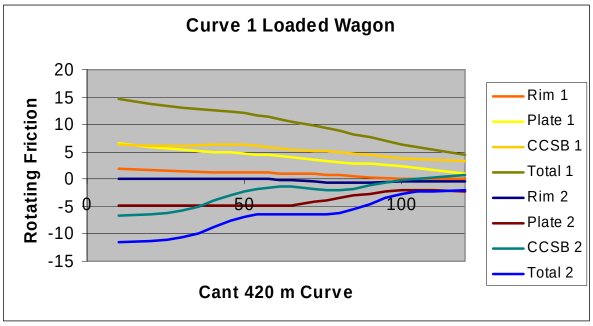

Figure 8 Averaged Rotation Friction During Curving as Affected by Cant Deficiency

The improvement in wheel rail wear from cant excess is not reproduced in reduced wheelset angle of attack. Figure 10 shows how the high cants allow for increases in the leading wheel angle of attack in low friction centre bowls. The uneven wheel loads in curve transitions seen with higher cants allows for greater movement at the bearing adaptors. This result is dependent on the dynamic modelling of the primary and secondary suspension and has not been fully investigated in this research.

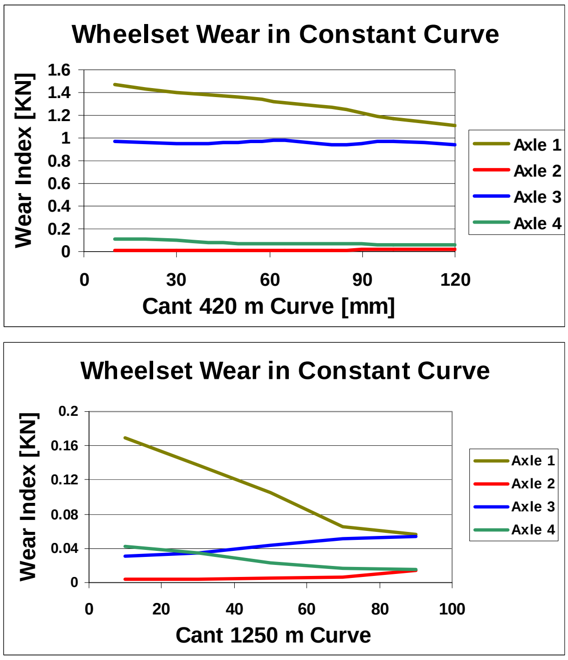

Figure 9 Wear effects of Cant on each axle due to Cant variation on 420 and 1250 m curves

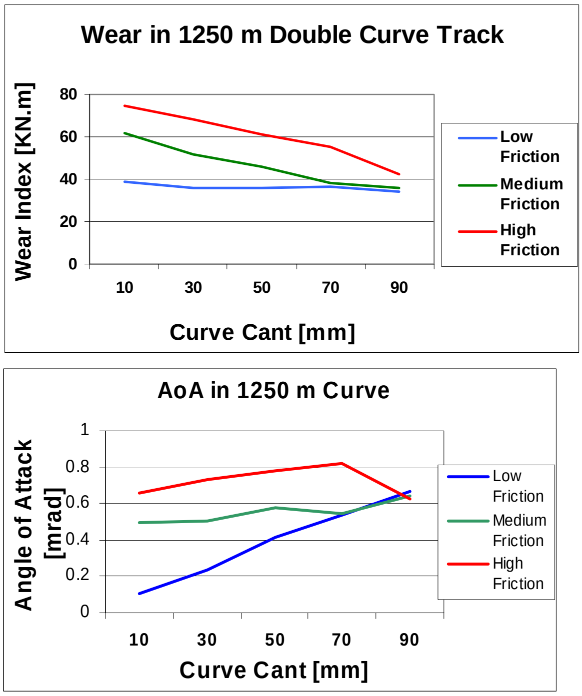

Figure 10 Wear index and angle of attack in 1250 m curves with cant variations

TRANSITION DESIGN SIMULATIONS

Transition curve design was investigated for its effect on the retained bogie rotation friction forces in the curve body and in post curve tangent sections. For this investigation 1250m curve was chosen to use with the vehicle model with as new wheel profiles. In these conditions the vehicle model has a curving wear performance that is the most sensitive to centre bearing friction. The most beneficial design change came from curvature over runs as depicted in Figure 11. Other alternate designs considered applied the curvature or cant in advance of one another. The curvature over run design was the only truly beneficial design. This design warps and then rotates the bogie further than required before easing the curvature to allowing the bogie warping to be relieved during constant curving.

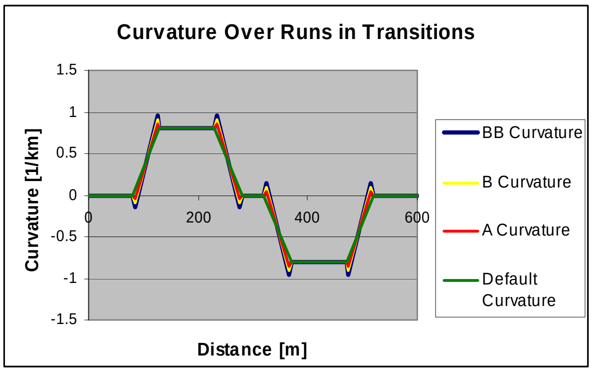

Figure 11 Curvature Over Run Designs

A transition curve with curvature over run, by relieving bogie warp, relieves the friction moment that is retained for the constant curvature of the track curve. This reduces the effects of high centre bowl friction in the bogie thus reducing the high flange contact forces on wheelset 1. Figure 12 shows the decrease in wheel rail wear and angle of attack on the leading wheelset achieved on the 1250m radius curve for the transition over run track designs.

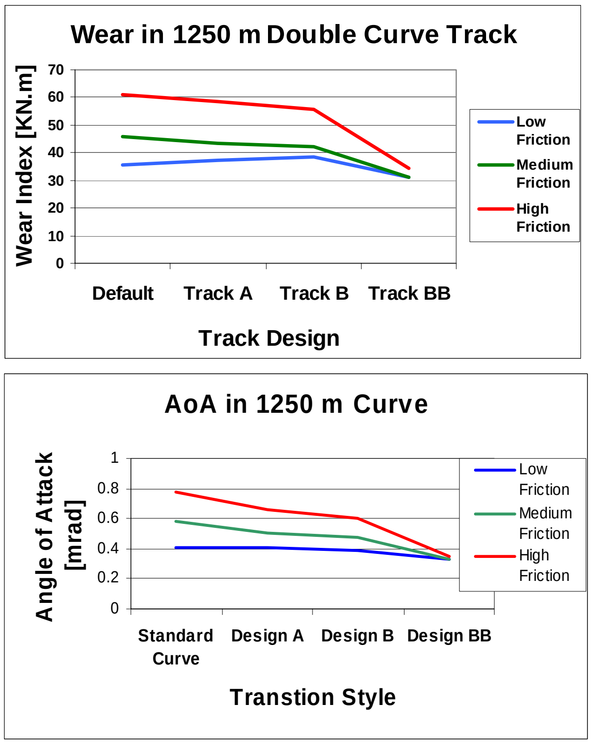

Figure 12 Wear and Angle of Attack wheelset 1 Results for Transition Designs

In track designs A and B the design cant limit of 50 mm for 80 kph train speed was not exceeded. Transition design BB had a higher than design cant (58 mm) applied with the curvature over run. The extra cant and associated suspension rocking is shown to be significant in relieving the friction moment and the bogie warp, reducing both wear losses and angle of attack values on the lead wheelset.

CONCLUSIONS

A detailed non linear vehicle model has been established for modelling bogie centre bowl rotational forces during curving. The model has shown how elastic warp distortion of the bogie maintains the bogie rotation moment through the constant part of the curve. In turn the elastic warp generates the increased wheel rail wear on the lead bogie increasing overall wear in medium to gentle curves as reported in literature [1].

The effects of high centre bearing rotational friction can be reduced if the bogie warp distortion is relieved by the track transition design. Cant excessive curves relieve bogie warp due to the wagon rocking on its suspension after completing the curve transition. However the leading wheelset angle of attack is not necessarily improved by this design.

A curve transition that over runs to a tighter than required curvature before returning to the design curvature works to improve both the wear performance and angle of attack on the wagons leading wheelset. The transition curvature over run provides the improved curving performance by relieving the elastic warp deflection in the bogie and the bogie rotation friction moment. The effects on the wheel/rail wear from high bogie rotation friction can be reduced by an appropriate curvature over run curve transition design.

REFERENCES:

1. Wu, H., 2002, Effect of center plate lubrication on vehicle curving and lateral stability, Research Report R-959, Association of American Railroads / Transport Technology Center, Inc, Pueblo Colorado.
2. Simson S. A., Pearce M., Sun Y. Q., 2005, Project 82 milestone 4-5 and 7 summary, Cooperative Research Centre for Railway Engineering and Technologies, Rockhampton
3. Simson S. A., Pearce M., 2005, Project 82 milestone 10 report, Cooperative Research Centre for Railway Engineering and Technologies, Rockhampton
4. Simson S. A., Pearce M., 2004, Longitudinal impact forces at 3 piece bogie center bearings Proc. 2005 ASME/IEEE Joint Rail Conference, ASME, Pueblo Colorado, pp. 45-50
5. Katta, R. R., Thomas C. F., 2004, A dynamic stiffness and damping model for rail car center plate polymer liners *Proc. 2004* ASME/IEEE Joint Rail Conference, ASME, Baltimore Maryland, pp. 63-73