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GAUGE FACE CONTACT IMPLICATIONS OF BOGIE ROTATION FRICTION IN CURVING

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1 ABSTRACT

QR National's experience of wheel wear in the operation of its central Queensland coal fleet has shown a high sensitivity to the proportion of the wagon fleet with high bogie rotational resistance. In periods where significant parts of the fleet exhibited high centre bearing rotational resistance rail lubrication conditions also were deteriorated and wheel life fleet wide was noticed to shorten. QR National believe that this increase in wheel rail wear was the result of reduce rail lubrication effectiveness caused by unfavorable contact mechanics at the gauge face of the rail.

The Rail CRC Australia Bogie Rotation Friction Management project has conducted a case study of the QR National VSA wagon. The case study analysis shows wheel rail wear on the Goonyella system operations of VSA wagons are very sensitive to bogie rotation friction especially for the tightest curve (304 m) where wheelsets 1, 2 and 3 all flange. The wheel rail contact mechanics are dependant on the bogie rotation friction specifically the angle of attack generated on axle 1 and 2 increases due to bogie rotation friction. The flange contact forces and resulting contact stresses change only slightly with increases of bogie rotation friction, under 10%. The significant change in the gauge face contact mechanics is the creepage angle. The creepage angle on the gauge face of the 304 m curve changes from 47 to 65 degrees to the rail axis for axle 1 and 18 to 57 degrees on axle 2. With this change in the flange contact creepage angle lubricant is increasingly swept off the rail gauge face increasing the friction coefficients for following wheelsets.

Previous studies on bogie rotation friction has shown that wear losses from high bogie rotation forces are due to warp of the bogie. Bogie warp occurs in the track transition in order to rotate the bogie however in constant curving there is no need to retain the warped bogie shape and transition design can be used to relieve bogie warp[1]. An alternative curve transition design is suggested for the 304 m curve. This alternative design termed a "curvature overrun transition", has small section of tighter curve radius in the transition section whilst the curve body is a slightly larger radius giving a minimal change to the curve alignment. The curvature over run design gives significant improvements to the angle of attack and contact mechanics in the curve body. However wheel wear savings only occur for vehicles having high bogie rotation friction and rail wear rates are effectively higher due to higher peak rail wear.

2 INTRODUCTION

The ability of bogies to rotate freely and negotiate curves efficiently, significantly reduces the contact forces on the gauge face on the high rail and top of rail and tread contact patch on the low rail at the rail/wheel interface. This is critical to the wheel and rail wear performance. If the rotation of the bogie is limited and these forces are too high, the angle of attack of the wheelset will prevent lubricant distribution also removing other lubricant throughout the curve and beyond. High bogie rotational forces result in higher forces at the wheels and also require higher torque to overcome the resistance to slip as the wheelset attempts to negotiate the rolling radii difference. This will result in high flange and gauge face wear and rolling contact fatigue on the top of rail and in the tread of these wheelsets. If the majority of wagons in the system have high bogie rotational resistance and are not transferring lubricant successfully from lubricators to rail gauge face, then all rollingstock will exhibit high rates of wheel wear.

2.1 QR National

In 1999 QR introduced a fleet of 2500 new coal wagons with nylon bogie centre bowl wear liners and constant contact side bearers into service in Central Queensland. Within months wheel flange wear on these wagons and other wagons were exceeding 5mm per 100,000km. Rail lubrication during this period was also noted to deteriorate and was linked to changed flange contact mechanics of the new rollingstock. The experience of QR had following this time of high flange wear on all wheelsets reducing to wheelsets 1 and 3 then until wheel flange wear was under control is reflected in the simulations undertaken by CRC Project 82.

QR Network Access operate track side gauge face lubricators to cover all their tighter curves in central

Queensland coal haulage operation. The effectiveness of such track side lubricators is dependent on the flange contact load and creepage at the flange contact.

Improving the design of the transition of curves to assist bogie steering shows promise for better rail wheel interaction in curves, reducing friction, wear and energy consumption. This has significant advantages with the current high cost of energy and there fore the future improvement of haulage tonnages for QR National.

2.2 Rail CRC Case Study

The Rail CRC Australia Bogie Rotation Friction Management project has been investigated the cost benefit of managing friction levels of centre bearings using vehicle simulation. Initial simulation studies[2] suggested that benefits in reduced wheel rail wear from lower centre bowl rotation resistance in three piece bogie wagons become significant only in medium to gentle curves. These results match test track findings of the Transport Technology Centre[3,4]. Such curves exhibit lower wheel rail wear thus the overall wheel rail wear impact is moderate. These findings did not match all the experiences of QR National. Consequently case study of bogie rotation friction management has been conducted on a QR National 106 ton gross, narrow gauge coal hopper wagon[5,6,7]. Further QR National reported that periods of operation where the fleet had high centre bearing friction coincided with periods of poor gauge face lubrication leading to extreme wear problems.

The case study analysis shows wheel rail wear on the case system operations of VSA wagons are very sensitive to bogie rotation friction[5,6,7]. The tightest curve is particularly adversely affected by centre bearing friction condition. The results showing a reduction in the total vehicle operation are given in Table 1. Bogie rotation friction was found to increase the angle of attack on the leading bogie producing wear rates and higher lateral creepage at the high rail gauge face. Lateral creepage

Table 1 Case study curving wear energy with centre bearing case

Centre Bearing Case	New Wheel [MJ/wagon]	Worn Wheel [MJ/wagon]
-010	85.9	64.4
-030	97.2	65.5
-050	106.0	83.1
-090	129.1	87.5

In the tightest curve wear increases where greater, wheel wear for empty and loaded operations for self lubricating polymer centre bearing liners giving 40% reductions compared to Mn steel and 80% over a damaged wear liner surface. Nearly all the wear rate reductions come from the loaded wagon case. It was found the sensitivity and behavior of vehicle curving due to bogie rotation friction was dependent on the type of curve as defined by the number of wheelsets in flange contact. The high sensitivity of the tightest curve from the VSA case study is due to multiple factors: the curve type being near tight, three wheelsets in flange contact[4,6]; and the load wagon operation being at near cant equilibrium.

2.3 Transition Curvature Over-run

Previous Rail CRC studies[1] of bogie rotation friction have identified the negative effects form high bogie rotation friction in curving are due to bogie warp on the lead bogie. The warp occurs primarily from the side frame yawing relative to the bolster. Warp due to bogie rotation on the rear bogie is favorably orientated and reduces wheelset angles of attack on the rear bogie in constant curving. Notably the warp deflections reverse during the exit transition and the rear bogie becomes unfavorably warped with the lead bogie favorably aligned. The bogie warp generated in the entry transition that is retained in to the constant curve has a dependency on lateral loads[1] which can be either from cant deficiency or lateral coupler forces. The most severe case of bogie rotation friction sensitivity is identified as a vehicle at cant equilibrium with no additional lateral coupler loads.

The primary author has previously suggested that a transition over run curve design be used to allow the bogie to unwarp prior to constant curving. The concept assumes that bogie rotations occurs only after warp deflection of the bogie and that friction in the bogie connections keeps the bogies warped shape after rotation is complete. The transition over run design thus rotates the bogie to a higher curvature and then relives that curvature allowing the warp deflection of the bogie to relieve.

The previous study[1] had been conducted on a medium radius curve, (wheelsets 1 and 3 in flange contact) and gentle radius curve (wheelsets 1 only in flange contact). These studies suggested significant advantages from curvature overrun transitions in both wheel rail wear and wheelset angle of attack of the leading wheel. The case study analysis[5,6] of the VSA operations has identified near tight curves as the most concerning for bogie rotation friction due to the high sensitivity of these curves to rotation resistance and the higher wear rates in these curves. The flanging contact forces of axle 2 in a near tight curve being directly related to the bogie rotation friction and resulting bogie warp.

3 SIMULATION STUDY

The simulation modeling has been performed using VAMPIRE®. The vehicle model is as previously described a fully non-linear model of three piece bogie dynamics with planar friction modeled at the centre bearing the friction wedge dampers and the bearing adaptors[1,4,6]. The contact mechanics modeling in VAMPIRE® uses a Kalker model[8] in a pre-calculated lookup table based on lateral position with corrections for high angles of attack. The creep coefficients in the simulation study are set at 0.45 at the rail top surfaces and at 0.15 for the gauge face surface to represent dry rail conditions with track mounted flange lubricators. This simulation method is fast but is subject to errors especially with sudden changes in wheel rail contact angles as experienced with worn wheel and rail profiles at higher angles of attack. For this reason the study has been conducted with new wheel profiles. Only the loaded wagon case has been investigated here as centre bearing friction has little impact on the empty wagon behavior.

3.1 304.5 m Near Tight Curve, Design A

The curve used in the case study is an assumed average curve from 27 curves in the selected VSA coal wagon operation of QR National. The Curve radius is 304.5 m and the length is 291.7 m between transition mid points. Rail profiles used are recorded profiles from one of the curves in QR VSA wagon operations, a worn 306 m radius curve. For a new wheel profile the combination of wheel rail profile and curve radius and the VSA wagon causes near tight curving having three wheels in flange contact.

A standard of 50 m long curve transitions is used as specific transition curve lengths where not available. For most of the 27 curves from the case study classifier as a type 304.5 m short transitions are likely. Most of 27 curves are located on the one large hill where the track grade is 1-50. The loaded trains descend this grade at reduced speed of 40 kph under dynamic braking compared to the empty wagons ascending at a speed limit of 60 kph. The cant deficiency is thus near equilibrium for the loaded wagon which is unfavorable for high bogie rotation resistance[1].

The presence of lateral coupler forces in the load train effectively adds cant deficiency. Considering then the coupler forces for the front of train case improves the bogie behavior for high bogie rotation frictions. This work has considered the rear of train case where there are no coupler forces.

3.2 Alternate Curve Design

An alternate curve design featuring transition curvature over run has been formulated (Figure 1), called design B. The curve B design has been constrained to minimize the lateral displacements made to the track and give the same change in direction. The design also needs to be symmetric for traffic to run in both directions. As the rotation required of a bogie is based on the arc of track between the bogie centers the tighter curvature has to be held for a distance greater than the bogie centre distance. The chosen length of the curvature hold is 13 m a smaller 5 m curve reversal is used at the tangent ends of the transition. The chosen maximum curve radius of is 277 m whilst the body of the curve maintains a curve radius of 310.5 m.

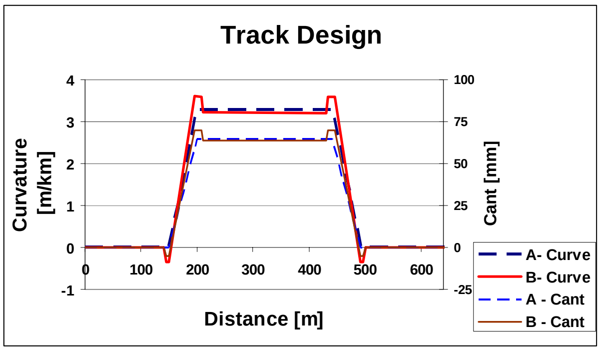

Figure 1 Track curvature design A and B

The alignment change from the 304.5 m radius curve to the design B is shown in Figure 2. The alignment shift for the two curve designs has been kept under 330 mm or less than a third of the gauge width. This would allow the design to be implemented at most sites without significant formation work.

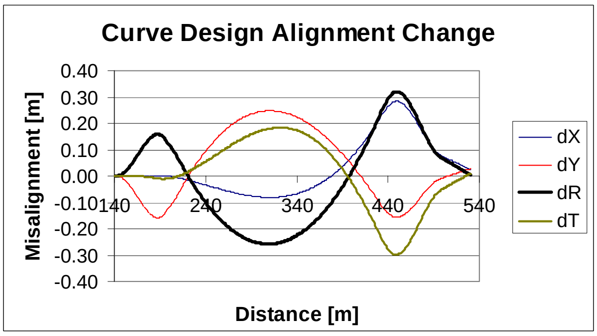

Figure 2 Track design alignment change

4 RESULTS

4.1 Curve A Performance

The curving performance of the VSA wagon is generated by the warp angle of the bogie and the resulting wheelset angles of attack. These are given in Figure 3. In Figure 3 the wheelset angles of attack of wheelset 3 and 4 do not match the rear bogie warp because wheelset 4 is not in flange contact and the tracking position of wheelset 4 changes with bogie warp. On the front bogie wheelsets 1 and 2 are in flange contact and thus the angles of attack of these wheelsets increase as a direct result of bogie warp.

The rear bogie warp is not a mirror reverse of the front bogie due to flange contact forces which cause a counter warping moment to the bogie rotation friction. Figure 4 shows the warping behaviour of front and rear bogies during curving against the track curvature. The rear bogie warp deflection stops with the on set of wheelset 3 flanging about 38 m in to the transition. The rear bogie unwarps on the transition exit as the track twist is relieved freeing up the friction resistance to warp movement at the friction wedges.

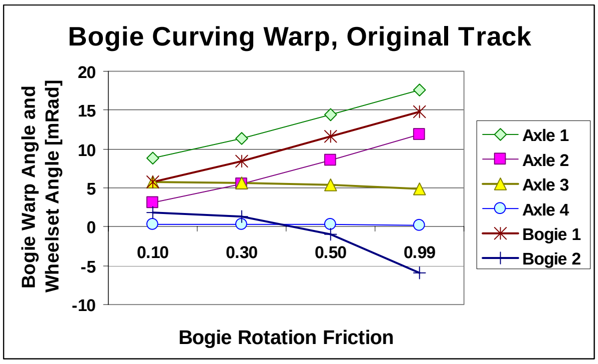

Figure 3 Bogie Warp on Design A Track

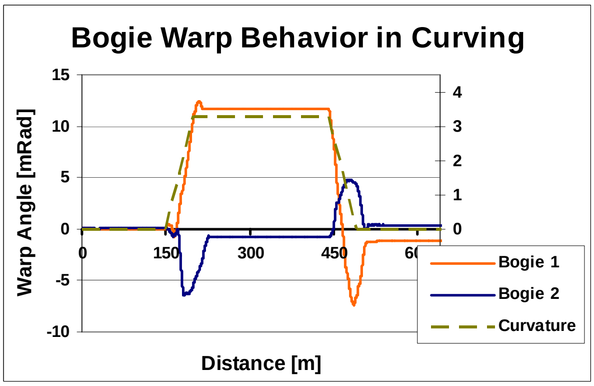

Figure 4 Bogie warp during curving, design A, centre bearing friction of 0.5

At the gauge face the creepage becomes increasingly angled laterally down the gauge face of the high rail. Figure 5 shows the total creepage in meters made over the gauge face of the entire curve. As the bogie rotation friction increases from a well lubricating polymer liner to a steel liner [frictions of 0.1 – 0.5] there is a 325 increase in the lateral creepage on the gauge face this increases for to 42% for a damaged liner [friction 0.99]. All of this increase occurs due to the front bogie warp.

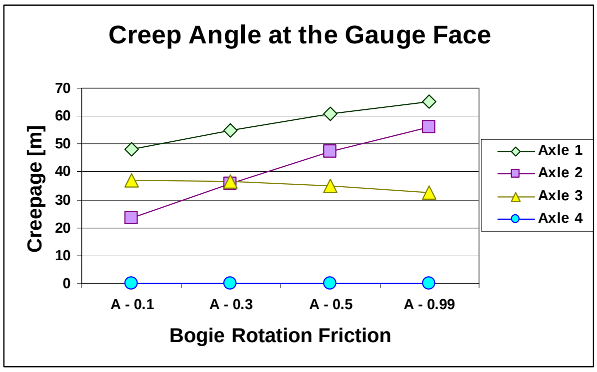

Figure 5 Creep angle design A

4.2 Curve B Performance

The curvature over run transition design is intended to reduce the retained bogie warp in the constant curve alleviating the wheelset angles of attack. Figure 6 shows the obtained warp and wheelset angles seen on the constant 310.5 m curve radius section of design B. There are significant and increasing warp angle reductions on the front bogie achieved by the transition design. The warp and angle of attack angles on the front bogie are reduced by ~1.4 mRad at centre bearing frictions of 0.1 and ~5.0 mRad for the damaged centre bearing condition of 0.99. However the rear bogie becomes unfavorably angled towards the high rail particularly for low centre bearing frictions. This is caused by the flange contact on axle 3 and not axle 4 for the ~300 m radius curve. With the bogie unfavourably warped towards the high rail after curve transition we now get flange contact on axle 4 reducing or eliminating the rail wear benefits from the lead bogies improved warp angle.

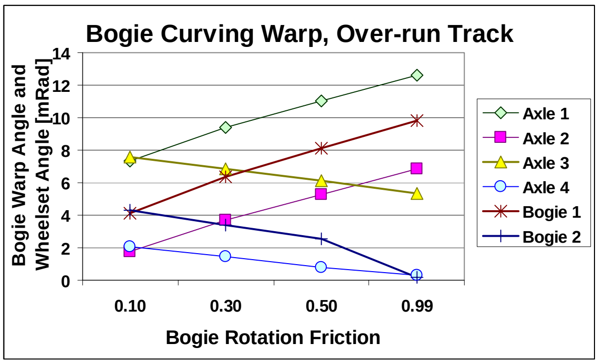

Figure 6 Bogie warp on design B Track

The bogie warp behavior is shown in Figure 7 for centre bearing friction of 0.5. In Figure 7 we see the rear bogie warp negatively (away from the high rail) at first until flange contact on axle 3 occurs at ~180 m. The bogie unwarps as the track twist under the wagon reduces. Then as the curvature reduces bogie 2 becomes warped positively (towards the high rail) and axle 4 contacts the high rail. At higher centre bearing friction axle 4 avoids flange contact during the 310 m radius section of the design B curve.

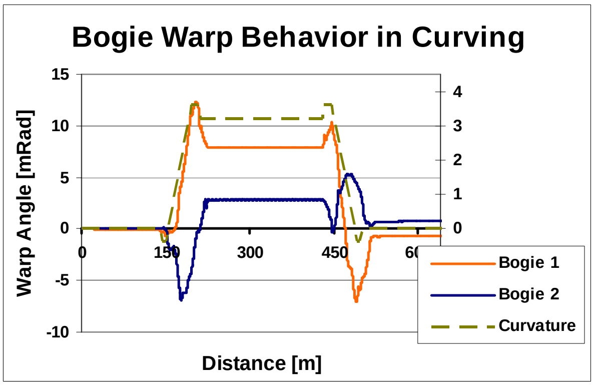

Figure 7 Bogie warp during curving, design B, centre bearing friction of 0.5

The lateral creepage on the gauge face is shown in Figure 8 is much reduced except for the additional creepage generated by axle 4 flange contact. The net result is to reduce the influence of centre bearing friction on the lateral creepage rate on the gauge face. For low bogie rotation friction levels the gauge face lateral creepage is increased (9.1 % at 0.1) and at high bogie rotation friction the lateral creepage reduces (-13.1% at 0.99).

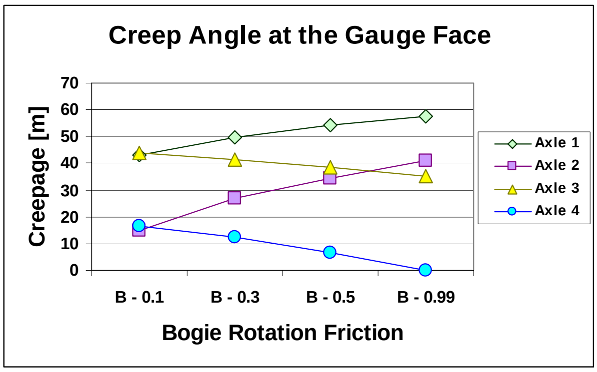

Figure 8 Creep angle design B

4.3 Wear and Derailment Performance

As discussed the bogie rotation resistance produces a direct increase in the angle of attack on the leading bogie for design A curves of a near tight radius such as the 304.5 m curve with the VSA wagon. The intention of curve B design is to reduce the unfavorable effects of bogie rotation resistance by rotating the bogie bolster to nearer the required angle and allowing the bogie warp to be reduced for the bulk of the curve. Figure 9 shows the curving wear energy over the track length for both the A and B curve designs for variable centre bearing friction conditions. The curve B design effectively reduces wheel wear at high centre bearing frictions but for lubricated center bearings the wear energy is slightly increased. The distribution of wheelset wear between the front bogie and the rear bogie is made more even for all bogie rotation frictions with the curve B design.

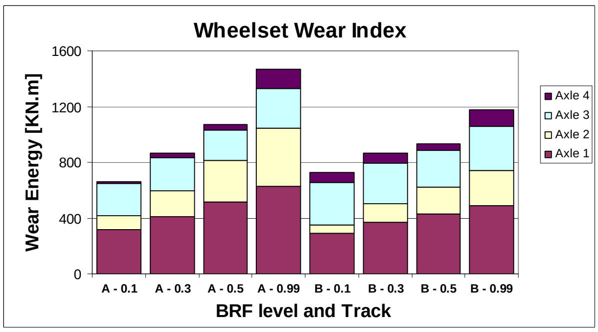

Figure 9 Wheel rail wear energy over track length

The rail wear from a rail replacement perspective is determined by peak wear at any given rail location in the curve. For the curvature over run curves, design B the peak rail wear is always going to be higher than the tradition transition design. The peak rail wear also remains sensitive to the bogie rotation friction no matter which curve design is used.

Derailment index of L/V had a very similar response to the AoA and bogie warp responses so whilst there was sum reduction of the average derailment index the peak derailment index values increased. No attempt has been made here to determine an overall derailment risk indicator.

4.4 Tight Curving

Tight curves are defined as curves where all four wheelsets are in flange contact. As such the bogie un-warping behavior identified in this study where the rear bogie unwarps at the end of the curve entrance transition (Figure 4 and Figure 7) is not likely to occur in tight curves. It should also be noted that the difference between near tight and tight curving in some cases may only be case of cant deficiency. Wheel squeal which occurs in part due to high angles of attack may be mitigated by transition over run design. Further investigation is required to determine the likely behavior in tight curves.

5 CONCLUSIONS

A near tight curve has been defined as any curve where the combination of curve radius and the wheel rail profiles generate flange contact on three of the four wheelsets. Near tight curves require the bogie to exhibit warp deflection such that rear wheelset in the trailing bogie will not flange whilst all other wheelset are in flange contact. Where a rail vehicle experiences near tight curving rail wear and the lateral creepage on the gauge face increase rapidly with bogie rotational friction.

Use of curvature over run transitions to force greater rotation from a bogie bolster and allow un-warping of the bogie frame before the main body of the curve has been shown to produce higher wheel wear for low rotation friction bogies on near tight curves. This has been shown to be due to increased wear occurring on the rear bogie that becomes unfavorable warp with wheelset angles of attack directed towards the high rail.

Lateral creepage on the gauge face of the high rail has like wise been shown to rapidly increase with bogie rotation friction and is assumed to indicate the increased difficulty of maintaining gauge face lubrication. The use of the design B transition over run curve has not significantly improved the total lateral creepage on the rail gauge face with a mix of results depending on bogie rotation friction.

These results are based on a starting curve type of near tight and do not apply to medium and gentle curves. Behavior in tight curves where wheel squeal behavior might be altered by overrun transitions is not predicted from these results.

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3 BIOGRAPHICAL NOTES

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