A PARAMETRIC STUDY OF BOGIE ROTATION FRICTION MANAGEMENT UTILISING VEHICLE DYNAMIC SIMULATION

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ABSTRACT

In 1999 QR introduced a fleet of 2500 new coal wagons with nylon bogie centre bowl wear liners into service in Central Queensland. Within months wheel wear on these wagons and other wagons were exceeding 5mm per 100,000km. Initial investigations focused on traditional causes of increases in wheel and rail wear such as badly profiled rail and poor rail/wheel lubrication. Wagon bogie rotational resistance was not considered until later following concerns raised regarding the possible additional rotational resistance created by the constant contact side bearers. Rotational resistance testing was undertaken by QR's Rollingstock Engineering Division in conjunction with Coal & Freight Services Group and some interesting results were found. The constant contact side bearings were not the source of major rotational resistance under the loaded wagon as first thought. Also the rotational resistance of the polymer liners was found to be excessive and concluded to be the root cause of the high wheel wear. A centre bowl lubrication programme was initiated and wheel wear was under control within two measurement periods with wheel life wear improved on all wagons, old and new. Since that time QR has undertaken extensive trials of different centre bowl types and found several more suitable materials which are now in use.

In keeping with the above thrust, the Rail CRC Australia's Bogie Rotation Friction Management project (Project 82) is investigating a range of factors that affect bogie rotation friction, including centre bowl friction, curve transition design and side bearer type. An extensive simulation-based parametric study was performed to determine the effects of varying side bearer type, centre bowl friction level, wagon loading conditions and speed on the wear characteristics of the wheelsets of a rail vehicle that runs on a selected track system. A 62 degree-of-freedom vehicle model was used for the simulations; the model was developed using the VAMPIRE® simulation package. New wheel and new rail profiles were used for the analysis to give an indication of the initial wear characteristics of the wheelsets. In this paper, an analysis of the results obtained is presented and conclusions are drawn based on the findings.

1 INTRODUCTION

In previous papers by the co-author [1][2], the vehicle model used in simulation studies (a QR National VSA wagon) had a roller-assisted constant-contact side bearer (CCSB) model. Newer and more detailed side bearer characteristics were obtained from Gemco Rail (in Western Australia), a local distributor of side bearers and related products [3]. The following side bearer types could now be modelled with improved accuracy: roller-assisted CCSB, standard-travel CCSB, extended-travel CCSB and the gap type side bearer. The significant difference between the earlier models and the current model is the specific modeling of side bearer configurations. Also, since manufacturers' test data was used, there was more confidence in the model parameters. With the availability of newer vehicle models, it became instructive to investigate the relative influence of the different side bearers on the wheelset wear characteristics as different operational conditions are varied. Other parameters of interest (beyond side bearer type) were the centre bowl friction level, the wagon loading condition and the vehicle speed. This led to an extensive parametric study to investigate these influences.

2 MODELLING

2.1 Case Study Track and Vehicle

An earlier case study had been performed using a QR National VSA wagon – a 106 ton gross, 20.1 ton tare narrow gauge hopper wagon with a light weight design 4.1 ton standard three piece bogie with CCSB [4][5]. The centre bowl was originally fitted with a nylon liner. The nylon liners caused excessive wheel wear due to the development of very high centre bearing frictions. The centre bearings of VSA wagons are now routinely lubricated with the use of self lubricating polymer plate type liners due to wheel wear benefits of reducing bogie rotation friction.

The case study wear analysis was performed on the Goonyella system of QR National in Central Queensland. Some of the results from that study are presented here. Table 1 gives the total wear energy impacts across the system assuming no longitudinal train loads. Wear rates increase dramatically as the centre bearing rotation friction increases. Figure 1 shows the wear rate increase for the nine test case curves used in the case study. The case study also included track sections for balloon loops and turnouts but those wear losses were insignificant with 86% of wheel/rail wear losses occurring in just three most prevalent curve types they being 304 m, 500 m and 808 m curves.

Table 1 Total wheel/rail wear energy in case study [5]

Center Bearing Friction Coefficient	New Wheel [MJ/ wagon]	Worn Wheel [MJ/ wagon]
0.10	84.8	63.5
0.30	96.1	64.2
0.50	104.8	82.1
0.99	127.7	86.5

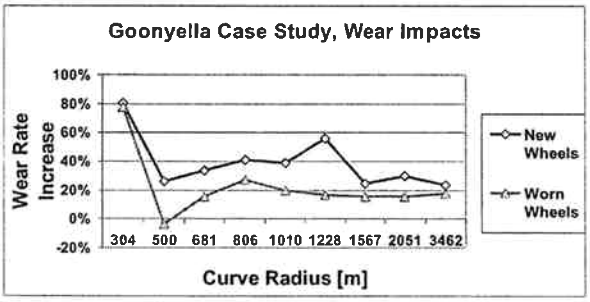


Figure 1 Wear impacts of bogie rotation friction in case study curves [5]

2.2 Parametric Study Track and Vehicle

The track model developed for the study represents the tight curves in the Central Queensland coal haulage operation of QR National. The wheels rail profile used for the parametric study was a typical new wheel new rail combination as typical of the Central Queensland coal haulage operations. Figure 2 shows the contact profile of the new wheel new rail combination used in the parametric study.

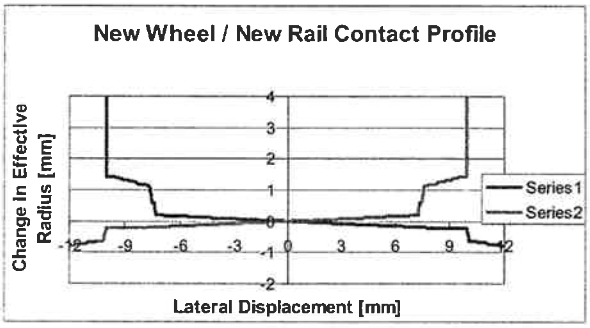


Figure 2 Contact profile for new wheel new rail profiles

A typical tight curve has a radius of 304 m with a nominal 50 m transition curve, along with a 65 mm cross-elevation. The full curve is 242 m in length before exiting with another 50 m transition and is as used in the case study [4][5]. For the parametric study, the track model was modified by adding an identical left-curving track after a 200 m straight track in-between the two curves. This was to check the consistency of right- and left-curving results and to obtain a result independent of curve direction. A smooth track was modelled. The track characteristics are shown in Figure 3.

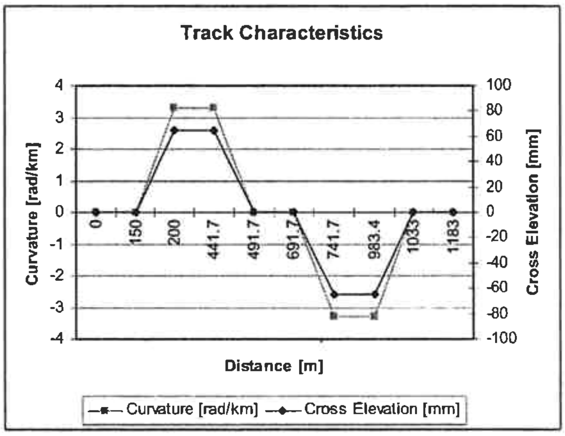


Figure 3: Track characteristics used for simulations

2.3 Side bearer models

Side bearers improve vehicle wagon stability during curving and in the case of constant-contact side bearers (CCSB), help to reduce or eliminate hunting instability on straight track. The spring stiffness characteristics of the roller-assisted CCSB (CCRS-R), the standard-travel CCSB (CCSB-ST) and the extended-travel CCSB (CCSB-XT) side bearers are shown in Figure 4 as plots of vertical force against vertical deflection. The load deflection data is for existing CCSB designs [3] suitable for use on a study wagon. The gap type side bearer (not shown) is modelled as a piecewise linear curve with a value of zero until the side bearer clearance is reached, after which it has a very steep gradient, symbolising a very stiff contact. For the roller-assisted CCSB, the actual modelled situation is a combination of a stiffness characteristic curve and the piecewise linear curve of the gap type side bearer, which represents the hard contact between the roller and the body wear plate.

All of the models have a side bearer travel limit of 5/16 inches except the extended-travel CCSB (CCSB-XT) which has a travel limit of 5/8 inches. The CCSB-R and the CCSB-XT have preloads of 20.066 kN, while the CCSB-ST has a preload at setup height of 24.08 KN.

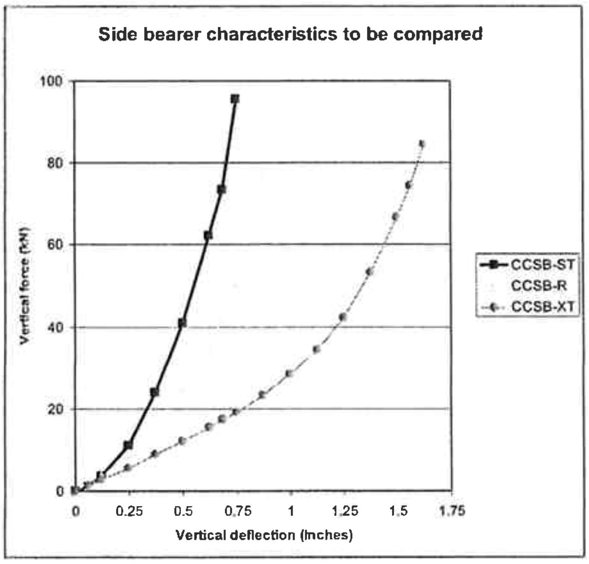


Figure 4: Side bearer spring/pad stiffness characteristics [3]

2.4 Previous simulation study

A previous analysis of the wear characteristics of the same VSA wagon on different track systems was conducted in [2]. The main focus of that investigation was to study the effect of cant deficiency/excess on the wear performance of wagons and individual wheelsets. The current analysis however differs from the previous one in the following important ways: firstly, this analysis uses new wheel and rail profiles, whereas the previous analysis used new wheels but a worn asymmetric rail profile which had been recorded by QR in a 306 m curve [4][5]. The worn profile has reduced wheel conicity; this analysis is comprehensive with regards to side bearer types, which have been modelled using reliable technical data, and; a wide range of centre bowl friction values and vehicle speeds has been investigated.

2.5 Simulation conditions

The track speed is limited to 60 km/h. The coal wagon consists usually transport coal when fully loaded at speeds of about 40 kph and return empty at speeds of up to 60 kph. Hence the investigation was designed to cover this speed range. Simulation runs were done for loaded and empty wagons with vehicle speeds of 40, 50 and 60 kph.

A new wheel and new rail profile were used with the chosen track, in order to be able to assess the initial wear behaviour of the wheelsets.

A total of six centre bowl friction levels were considered: 0.1, 0.3, 0.4, 0.5, 0.7 and 1.0. A rail friction coefficient of 0.3 was used. The contact between the side bearer pad and the body plate/wedge was assumed to have a friction coefficient of 0.35. Also, the coefficient of friction at the contacts between the axle box and the bogie side frames was assumed to be 0.5.

Simulations have been done with a 62 degree of freedom vehicle model using the VAMPIRE® vehicle simulation package. The output data of interest were the wear indices of the left and right wheel treads and the flange (which were combined in a single wheelset to form a total average wear index), and the wheelset angles of attack. For a given speed and wagon loading, 24 simulations were performed (4 side bearers and 6 centre bowl friction levels), resulting in a total of 144 simulations and a wide range of output data.

The simulations were run at a step size of 0.1 ms and for a total distance of 1180 m, including both curves. A set of simulations were run with a loaded wagon, and then repeated for an empty wagon.. The outcomes of the loaded and empty wagon cases are investigated separately and also compared.

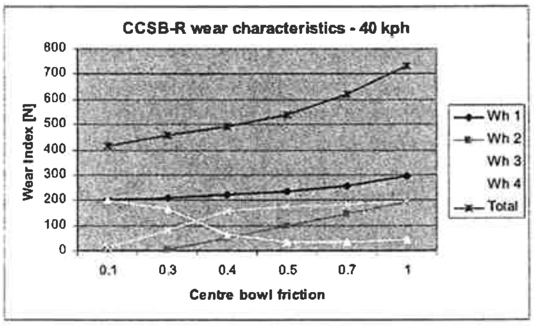
3 SIMULATION RESULTS / OBSERVATIONS

3.1 Instantaneous and Average Wear and AoA values

The instantaneous and average values of the wheelset wear and angles of attack were used quite extensively in this study to interpret the outcomes of the simulations and to draw comparisons between simulation runs with different conditions. The wear value or index used is the product of the creepage and the creep force at the contact patch. This is equivalent to the wear energy expended per unit length. The average values were calculated over the entire section of track modelled, which includes both the curved and straight parts of the track. In the case of the wheelset angles of attack (AoA), the average absolute value is determined; this avoids cancelling out of positive and negative values as the right-curve becomes a left-curve. This average value can then be correlated against the average wear index for the same run. The instantaneous values are important help interpret the average values, especially in cases where most of the wear is being experienced at particular locations along the curve.

3.2 Correlation between Wear Rate and AoA

A correlation study between the average wheelset angle of attack and the average wear rate over the entire curve was conducted. Figure 5 represents a loaded wagon case while Figure 6 is for an empty wagon. Figure 5a shows the average wear across the four wheelsets as the centre bowl friction is varied, and Figure 5b shows average wheelset angles of attack for the same centre bowl friction variation, using a roller-assisted CCSB on a loaded wagon at a speed of 40 kph. Figure 6a and Figure 6b are similar plots, this time for an empty wagon. As stated earlier, the average wear index value for each wheelset is calculated over the entire curve.



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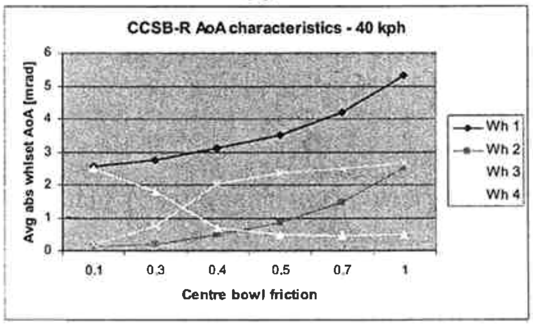

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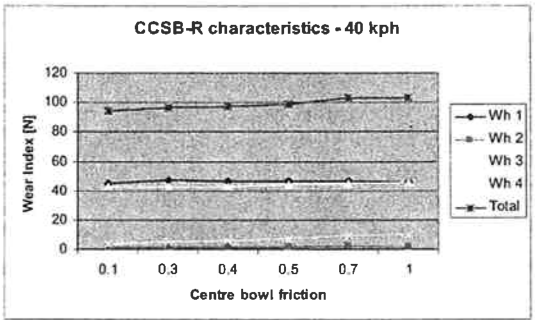
Figure 5: Plots of (a) average wear across wheelsets and (b) average wheelset angles of attack, versus centre bowl friction, for loaded wagon running at 40 kph



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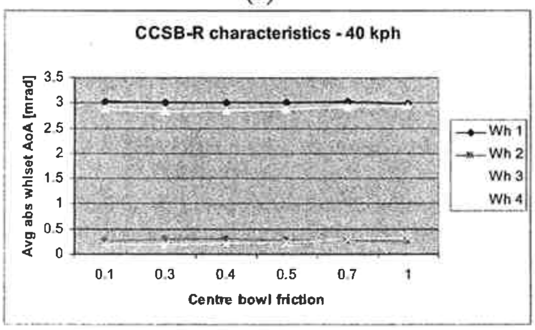


(b)

Figure 6: Plots of (a) average wear across wheelsets and (b) average wheelset angles of attack, versus centre bowl friction, for empty wagon running at 40 kph

A comparison of the two sets of figures shows very strong correlation between the wear number and the wheelset angle of attack. For example, the correlation coefficient, R, calculated for the wheelsets 1 to 4 on a loaded wagon at 40 kph were 99.92, 97.29, 99.30 and 99.41 %, respectively. With all the wheelsets combined, the correlation coefficient was 97.10 %. Similar analyses were performed at 50 and 60 kph, and for both loaded and empty wagon cases. Table 2 displays these results. The correlation of wheelset 2 is particularly low for the empty wagon at high speed (60 kph). At 60 kph, the actual wear and angle of attack values for wheelset 2 are quite low in comparison to the other wheelset values and does not vary much over the different centre bowl friction values.

Table 2: Correlation coefficients (AoA vs Wear No.) for loaded and empty wagons

	Wh 1	Wh 2	Wh 3	Wh 4	Overall
Loaded wagon:					
40 km/h	0.9992	0.9729	0.9930	0.9941	0.9710
50 km/h	0.9734	0.9866	0.9908	0.9959	0.9647
60 km/h	0.9881	0.9322	0.9847	0.9709	0.9362
Empty wagon:					
40 km/h	0.9852	0.8959	0.9479	0.9178	0.9881
50 km/h	0.9002	0.6748	0.9566	0.9335	0.9969
60 km/h	0.6527	0.1616	0.9626	0.9261	0.9991

3.3 Effect of centre bowl friction on wheelset wear rate

Regardless of the choice of side bearer or the speed, there is a general increase in the total wear index as the centre bowl friction is increased. For the loaded wagon case, the total wear increases at a much higher rate than for an empty wagon. The total wear values in the loaded wagon case are also 4 to 7 times higher in the empty case, the ratio increasing with centre bowl friction.

Among the different wheelsets, the greatest wear occurs consistently on wheelset 1. In the loaded case, it can be seen from Figure 5 that wheelset 1 has the highest wear, which increases steadily with centre bowl friction. Wheelset 4 has the lowest wear at low centre bowl friction, but its wear rate also increases rapidly with friction, eventually exceeding the wear on wheelsets 3. Wheelsets 3 and 4 themselves appear to have wear profiles that vary in such a manner that their combined total wear remains approximately constant; as the centre bowl friction increases, the wear rate of wheelset 3 drops while that of wheelset 4 increases with friction. These trends are also observed in the 50 and 60 kph simulation runs for the loaded vehicle case.

The above interesting behaviour observed in the loaded case was investigated further, especially since it was not consistent with previous studies using worn rail profiles. Plots of wheelset lateral deflection and wheelset angles of attack, against centre bowl friction, showed that with low friction (0.1), wheelsets 1 and 3 were flanging on the high rail while wheelsets 2 and 4 remained well within the lateral travel limit. Wheelsets 1 and 3 also averaged around -5 mrad AoA for the right-curving track (+5 mrad for the following left-curving track).

As the centre bowl friction increases, wheelset 1 remains in flange contact against the high rail with increasing angle of attack. As the centre bowl friction increases wheelset 2 approaches the high rail with an increase in its angle of attack (-1.7 mrad at 0.5 centre bowl friction) from approximately zero. On the rear bogie, the increasing yaw moment at the centre bowl as its friction is increased appears to cause the bogie to warp, thus removing the wheelset 3 from flange contact and driving wheelset 4 into the low rail. The new wheel on new rail profiles used for this simulation contribute to this effect with a higher contact angle on the rail head as the contact point approaches the flange than in previous studies, which increases the lateral component of the rail contact forces; the resultant wheel-rail contacts drive the wheelset tracking towards the inner rail.

Table 3 shows the steady state values of the lateral displacement and the angle of attack obtained for the loaded wagon during the first (right-hand) curve at 40 kph, at 0.1 and 0.5 centre bowl friction. The values obtained in the left-hand curve are similar but with opposite signs. Flange contact begins at ±10.3 mm of lateral travel.

Table 3: Steady state lateral displacement and angle-of-attack values in curve at 40 kph.

	CB Friction	Wh 1	Wh 2	Wh 3	Wh 4
Lat. displ. (mm)	0.1	-11.15	-7.72	-11.15	-3.11
	0.5	-11.16	-10.2	-10.92	7.77
Angle of attack (mrad)	0.1	-5.17	-0.02	-4.61	-0.16
	0.5	-5.58	-0.08	-3.82	0.11

For the empty wagon case (Figure 6), the centre bowl friction has a limited influence on the wear characteristics. Wheelsets 1 and 3 both have much larger wear values when compared to wheelsets 2 and 4. This is similar to the 'moderate curve' behaviour described in [2]. It should also be noted that the wear values are much lower in the empty case than in the loaded case.

3.4 Effect vehicle speed on wear rate

The effect of running speed on the wheelset wear rate is not consistent across the wheelsets and seems to change depending on the centre bowl friction. If a crude average is taken across all side bearer types, then in most cases, the wear on wheelsets 1 and 3 tend to increase slightly with speed while the wear on wheelsets 2 and 4 decrease with speed. Overall, there tends to be a slight increase in the total wear with speed in both the loaded and the empty wagon cases. Figure 7 shows the percentage change in the total average wear from the 40 to 50 kph, and from 50 to 60 kph, as a function of the centre bowl friction. The change from 50 to 60 kph seems to be more consistent than from 40 to 50 kph. The bulk of the changes seem to be within ±6 %. However, at higher friction, greater percentage changes are encountered at the higher speed range.

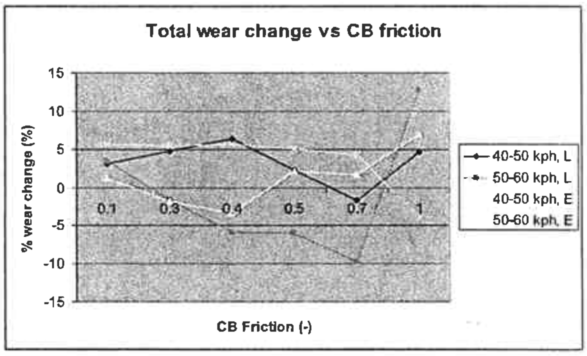


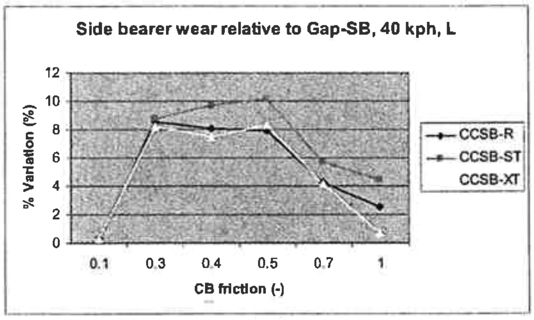
Figure 7: Percentage change in Total wear from the 40 to 50 kph, and from 50 to 60 kph, for loaded and empty wagons

Flange contact on wheelset 2 is sensitive to the speed or cant deficiency of the curve. Using the new rail profile flange contact for wheelset 2 is generally restricted to the curve entrance transition but the strength of entrance transition contact on wheelset 2 impacts on the front bogie warp and resulting AoA.

3.5 Effect of choice of side bearer

There was some variation with the use of different side bearers. It was observed that in most cases, the highest total wear occurred with the standard-travel CCSB, while the least wear occurred with the gap-type side bearer. In the loaded cases, the wear values are high compared with the empty wagon cases. As a result, the percentage variation in the loaded cases appears to be lower. There is some variation in the individual wheelset wear values, especially in the loaded case, where wheelsets 3 and 4 vary across the side bearers in such a manner as to have a constant combined wear value. Figure 8 shows the percentage variation of the CCSB-R, CCSB-ST and CCSB-XT relative to the Gap SB, plotted against the CB friction, for simulations at 40 kph. The first plot (a) is for the loaded condition while the next one (b) is for the empty wagon case. In both cases, the CCSB-ST varies by the highest percentage while the CCSB-XT varies by the least amount relative to the gap SB. This trend is consistent over most of the speeds and wagon loading conditions.

It must however be noted that as stated earlier, the CCSB-ST has a higher preload at setup height than the CCSB-R and CCSB-XT. This is likely to be a significant contributor to the higher wear experienced with this side bearer.



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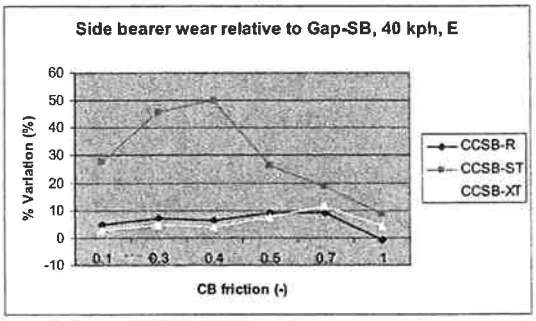

(b)

Figure 8: Percentage variation of the CCSB-R, CCSB-ST and CCSB-XT wear values relative to the Gap SB, for loaded and empty wagon at 40 kph

3.6 Difference between loaded and empty wagon behaviour

The major difference in the loaded and empty wagon characteristics arise from the relative importance of the influence of centre bowl and side bearer frictions on the bogie rotation characteristics in each scenario. With an empty wagon, the effects of varying the centre bowl friction are less prominent than with a loaded wagon, and lower forces and wear occur. However, a change in the side bearer preload will be more noticeable in an empty wagon due to the much lower centre bowl frictions. This explains the much higher percentage variations in the CCSB-ST wear value in the empty wagon case as compared with the loaded case (Figure 8b).

4 CONCLUSIONS

The observations and analyses from this simulation study suggest that using different wheel-rail contact profiles can lead to quite different wear characteristics across the wheelsets. With a new wheel profile and worn rails, previous studies have shown flanging on wheelsets 1, 2 and 3 on a 304 m curve. The current study shows that with new wheel and rail profiles, flanging occur on different wheelset depending on the centre bowl friction level. At low friction, wheelsets 1 and 3 flanged, whereas at higher centre bowl friction, flanging occured on wheelsets 1 and 4 with Wheelset 4 flanging on the low rail. At higher speeds or cant deficiency and high centre bowl friction flange contact increased on wheelset 2.

The following conclusions were drawn from the observations made from this simulation study:

1. There is a very strong correlation between the wear index and the average absolute wheelset angle of attack.
2. The sum of the wheelset wear numbers, i.e., the total wear, increases with centre bowl friction.
3. The greatest wear occurs consistently on Wheelset 1.
4. The wear characteristics across the wheelsets depend on whether the wagon is loaded or empty.
5. For the loaded wagon, at low centre bowl friction, the least wear occurs on wheelset 2. However, as the centre bowl friction increases, the wear on wheelset 2 tends to increase. At higher speeds wheelset 2 will flange and the wear increses beyond that of wheelsets 3 and 4.
6. For the loaded wagon, wheelsets 3 and 4 seem to 'share' their wear in such a way that their combined wear for a given set of parameters and conditions, remains approximately constant. This tendency can be observed as the centre bowl friction is varied and also as the choice of side bearer is varied. For instance, as the centre bowl friction increases, the wear on wheelset 3 tends to reduce then increase or level out; on the other hand, the wear on wheelset 4 tends to increase then later reduce or level out.
7. With empty wagons, there is very little variation in the individual wheelset wear values as the centre bowl friction is varied. Wheelsets 1 and 3 have much higher wear values than wheelsets 2 and 4. This condition was referred to as 'moderate curving' in [2]. In addition, for the empty wagon cases, the wear values are lower than the equivalent loaded wagon cases.
8. With loaded wagons, there is no clear trend with the total wear on the wheelsets as the speed is increased. With empty wagons, the total wear appears to increase slightly with speed.
9. The overall wheelset wear may increase or decrease with speed, depending on the variations in the tracking of wheelset 2 and 4. The wear on wheelsets 1 and 3 generally tends to increase with speed while that on wheelsets 2 and 4 tends to decrease.
10. In most cases, the least total wear occurs with the Gap SB and the greatest with the CCSB-ST. The CCSB-R and the CCSB-XT usually fall in-between these two. These results match the constant contact preloads of the chosen CCSB products and suggest the preload is more significant than the side bearer design.

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REFERENCES

[1] S. Simson, M. Pearce: *Longitudinal Impact forces at 3 piece bogie centre bearings,* ASME/IEEE Joint Rail Conference, RTD, 2005, v 29, pp 45-50

[2] S. Simson, M. Pearce: *Centre Bearing Rotation forces during curve transitions,* Proc. Conference on Railway Engineering 2006, April 30 – May 3, RTSA, Melbourne, pp71-77, 2006

[3] GEMCO Rail, "Stucki Product Information." Perth, Received 21 Nov. 2005.

[4] Simson S. A., Pearce M., Project research report of the Rail CRC titled "Project 82 milestone 10 report", Rail CRC, Central Queensland University, Rockhampton QLD 4702, Australia.

[5] S. Simson, M. Pearce: *Wheel Wear Losses from bogie rotation resistance, effects of Cant and Speed,* ASME/IEEE Joint Rail Conference, RTD, 2006, vol. 31, pp. 109-114