Surface coatings for 3-piece freight bogie centre bearings

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Chapter 2: Literature review: Centre bearings

In this chapter, the literature relating to centre bearings is reviewed. The different materials and lubrication schemes currently used for the centre bearing surfaces of Australian rail freight bogies are listed. The effect of similar materials and lubrication schemes on frictional moment has been studied by Richmond [2] and Wu, Robeda and Guins [5] and is summarized. The effect of frictional moment on bogie dynamics during curve negotiation, bogie hunting along tangent track, and impact loads is discussed. Finally, the typical operating conditions of the centre bearing are described.

2.1 Three piece freight bogie

The most common freight bogie used worldwide is the two-axle three-piece freight bogie as shown in Figure 2. This bogie consists of two (2) wheelsets, two (2) sideframes, secondary suspension and a bolster. There is no primary suspension between the wheelsets and sideframes in these bogies. The secondary suspension consists of vertical spring nests in parallel with friction plate dampers located between the sideframes and the bolster. The secondary suspension forces tend to oppose lateral, vertical, and roll motions of the bolster relative to the sideframes. [1]. The wagon body rests directly on the centre bearing and its weight is transmitted downwards to the track through the bolster, secondary suspension, sideframes, wheelsets and wheel-rail interface. Conversely, forces arising from the track's geometry and irregularities are transmitted upwards through the same components [1]. The three-piece bogie can deform so that the wheelsets and sideframes can approximate the shape of a parallelogram. This type of deformation is known as "warping", "lozenging", or "parallelogramming" (Figure 3).
Figure 2 Schematic of 3-piece freight bogie showing location of centre bearing.

Figure 3 Warped bogie producing high wheel angle of attack [4].
2.2 Rails

The rails transmit the train loads to the roadbed and guide the moving train. The different track geometries—tangent, spiral, and curve are shown in Figure 4. The tangent section is straight, whilst the curve section has constant radius curvature. When trains are operated at other than low speeds, it is necessary to insert a spiral or transition curve between tangent and curve segments (Figure 4). The spiral is a curve of continuously-changing radius, decreasing from an infinite radius at the tangent end to a radius equal to that of the circular curve at the other end. In curves, the outer rail is raised, or superelevated, to balance the centrifugal forces which act away from the centre of the curve and tend to overturn the cars. [1].

Please see print copy for Figure 4

Figure 4 Track geometry. (a) Compound curve with spirals, (b) Reverse curves with intervening tangent. [1]
2.3 Side bearings

Side bearings are located at the sides of the bolster (Figure 2). There are essentially two different types of side bearings – gap and constant contact side bearings. For gap side bearings the wagon only contacts the bolster during curve negotiation. Constant contact side bearings (Figure 5) enable the wagon body to be in constant contact with the bolster at the side bearings and the centre bearing (Figure 6). During curve negotiation the wagon body rocks compressing the polymer element inside the constant contact side bearing until the wagon body wear plate comes into contact with the roller thus reducing bogie curving resistance. During motion along tangent track, the lightly compressed polymer prevents the wagon body from swaying which minimizes bogie hunting at high speeds.

Please see print copy for Figure 5

Figure 5 Stucki constant contact side bearing (CCSB) [7].
2.4 Centre bearings

The centre bearing connections consist of the cylindrical top centre attached to the wagon body that is located loosely inside the cylindrical centre bowl liner, i.e., a shallow dish fixed to the centre of the bolster of the freight bogie (Figure 7). The top centre is welded or bolted onto the wagon body, whilst the centre bowl liner is pressed into the bolster queen casting and then, in the case of steel liners, circumferentially stitch welded. Both the top centre and centre bowl liner have vertical rim wall and horizontal plate surfaces (Figure 7). A centre-pin or "kingpin" is placed through holes in the centres of the top centre and centre bowl liner [8].

The Australian Rail Operation Unit’s Draft Code of Practice for the Defined Interstate Rail Network, Volume 5 [10] provide recommendations for centre bearing diameter and material construction (Table 1). For a 50 ton bogie, the nominal centre bearing diameter is 305 mm. As the wagon mass is increased to 100 ton, the recommended centre bearing
diameter is increased to 405 mm [10].

However, there are two (2) other dimensions that are relevant to the centre bearing. The first is the diametral clearance between the top centre and centre bowl liner, which will be referred to herein as the centre bearing diametral clearance. Typically, for a newly installed centre bearing the nominal clearance is 5 mm. It is understood that this 5 mm clearance is required for the crane driver to lower the wagon top centre into the centre bowl liner without damaging either component. Typically at 10-15 mm clearance the bearing is condemned. The second dimension is the contact height between the vertical rim walls of the top centre and centre bowl liner, referred to herein as the rim wall contact height. 30, 34 and 55 mm rim wall contact heights have been observed in operation in Australia.

Pitch (or roll), sway (or rock), and yaw motions are highlighted in red in Figure 7. Pitch is rotation around the lateral direction, sway is rotation around the longitudinal or train travel direction, whilst yaw is rotation around the vertical direction, i.e. in the horizontal plane.

The materials recommended by the Australian Rail Operation Unit for centre bowl liners include 10-14% manganese steel, and asbestos free non-metallic materials [10]. Hadfield steel (12% Mn, 1.2% C) is a commonly used example of the former, whilst high density polyethylene (HDPE) is an example of the latter.
Eighteen (18) staff from the rail industry, and research centres within Australia were surveyed by telephone and email regarding existing bogie rotation systems for 3 piece freight bogies in Australia. The results are listed in Table 2. In general terms there are three main groupings of bogie rotation systems:
1. dry rubbing steel against steel centre bearing with constant contact side bearing, and

2. dry rubbing steel against steel centre bearing with gap side bearing, and

3. lower friction centre bearing with constant contact side bearing.

Various steels are used for centre bowl liners and top centres including: Hadfield steel, 1045 and 1053 medium carbon steels, 4330 nickel-chromium-molybdenum steel, Bisalloy molybdenum steel, and flame hardened 1030 medium carbon steel. The microstructures of these steels includes: austenite, pearlite-ferrite, and tempered martensite. Typically, all these steels have as supplied hardness values within the 200-300 HV range. However, in-service hardness can differ from as-supplied hardness as will be shown in the experimental results of Chapter 6.

The idea of greased centre bearings and polyethylene liners is to reduce the co-efficient of sliding friction, and thus the frictional moment, at the centre bearing compared to dry rubbing steel against steel centre bearings. In residential areas, these bearings have the added advantage of reduced noise [9].
Table 1. Australian Rail Operation Unit draft code of practice clauses relating to centre bearing dimensions and materials. [10].

| Clause 3.3.1.1 | “Three-piece bogies should incorporate the AAR standard flat cylindrical centre bowl applicable to the bogie classification. Nominal diameters to suit the top centre plate are as follows:

(a) 305 mm (12.0”) for 50 ton bogies.
(b) 350 mm (13.8”) for 70 ton (AAR 2E) bogies.
(c) 405 mm (15.9”) for 100 ton (AAR 2F) bogies.” [quoted from 10] |
| Clause 3.3.1 | “Centre bearings of the flat cylindrical type should be fitted with cup type wear liners of the following—

(a) 10-14% manganese steel;
(b) asbestos free non metallic material to AAR Standards S-305, S-306, S-307, S-308 and Recommended Practices RP-300 and RP-301; or
(c) other internationally recognized standards applicable to the design of centre plate used.” [quoted from 10] |
Table 2 Existing bogie rotation systems

<table>
<thead>
<tr>
<th>Generalised groupings</th>
<th>Centre bowl liner material</th>
<th>Top centre material</th>
<th>Lubrication</th>
<th>Side bearing type</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. dry rubbing steel against steel centre bearing with CCSB</td>
<td>Hadfield steel</td>
<td>Cast 1053 medium carbon steel</td>
<td>None</td>
<td>Constant contact</td>
</tr>
<tr>
<td>2. dry rubbing steel against steel centre bearing with gap side bearing</td>
<td>Hadfield steel</td>
<td>Cast 1053 medium carbon steel</td>
<td>None</td>
<td>Gap</td>
</tr>
<tr>
<td></td>
<td>1045 medium carbon steel</td>
<td>Unspecified steel</td>
<td>None</td>
<td>Gap</td>
</tr>
<tr>
<td></td>
<td>4330 nickel-chromium-molybdenum steel</td>
<td>Unspecified steel</td>
<td>None</td>
<td>Gap</td>
</tr>
<tr>
<td></td>
<td>Bisalloy molybdenum steel</td>
<td>Unspecified steel</td>
<td>None</td>
<td>Gap</td>
</tr>
<tr>
<td>3. lower friction centre bearing with CCSB.</td>
<td>Hadfield steel</td>
<td>Flame hardened 1030 medium carbon steel</td>
<td>Lithium grease</td>
<td>Constant contact</td>
</tr>
<tr>
<td></td>
<td>Bisalloy Bisplate 400 molybdenum steel</td>
<td>Unspecified steel</td>
<td>Lithium grease</td>
<td>Constant contact</td>
</tr>
<tr>
<td></td>
<td>HDPE plate liner</td>
<td>Unspecified steel</td>
<td>None</td>
<td>Constant contact</td>
</tr>
<tr>
<td></td>
<td>HDPE bowl liner</td>
<td>Unspecified steel</td>
<td>None</td>
<td>Constant contact</td>
</tr>
</tbody>
</table>
2.5 Centre bearing friction

2.5.1 Effect of centre bearing friction on curve negotiation

For proper curve negotiation, the bogies of a vehicle must rotate underneath and relative to the carbody at the centre bearing, as Figure 8 illustrates. This rotation is resisted by the friction force generated at the contacting surfaces of the centre bearing and side bearings [1].

Figure 8 Car body has a relative rotation to the bogie on curves [4].

It is possible that the surfaces of the plate and rim wall use different materials, thus the respective co-efficients of friction, $\mu_{\text{plate}}$ and $\mu_{\text{rim wall}}$, can be different. In general terms, there could be 3 conditions as listed in Table 3. Friction conditions 1. and 2. are existing combinations, whilst the author is not aware of any rail industry examples of Friction condition 3. The effect of $\mu_{\text{plate}}$ and $\mu_{\text{rim wall}}$ on the frictional moment for a 125 ton wagon load is shown in Figure 9 [2].
Table 3 Possible centre bearing friction conditions.

<table>
<thead>
<tr>
<th>Friction conditions</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. $\mu_{\text{rim wall}} = \mu_{\text{plate}} = \mu_{\text{bearing}}$</td>
<td>e.g. $\mu$ centre bearing = 0.2 (polymer liner) or 0.5 (unlubricated steel-against-steel)</td>
</tr>
<tr>
<td>2. $\mu_{\text{rim wall}} &gt; \mu_{\text{plate}}$</td>
<td>e.g. $\mu$ rim wall = 0.5, $\mu$ plate = 0.2.</td>
</tr>
<tr>
<td>3. $\mu_{\text{rim wall}} &lt; \mu_{\text{plate}}$</td>
<td>e.g. $\mu$ rim wall = 0.2, $\mu$ plate = 0.5.</td>
</tr>
</tbody>
</table>

The effect of different materials and lubrication schemes on frictional moment has been studied by Richmond in 1993 [2] and Wu, Robeda and Guins in 2004 [5] utilizing full-scale laboratory test rigs and/or test track data. The laboratory test rig used in both studies to measure the frictional moment is shown in Figure 10. The vertical load applied at the centre bearing plate surface simulated a fully loaded 125 ton wagon. No load is applied at the rim wall. Both yaw motions and rock and/or roll motions were applied to the loaded test rig. The test conditions are summarized in Table 4. 30,000 to 50,000 yaw cycles represents 240,000 to 400,000 service km’s [2], or around 5 years continuous service. If during the test, the frictional moment exceeded around 56.6 kN.m ($\mu=0.4$), then the test was terminated, presumably to prevent damage to the test rig itself. The rock/roll motion was paused when the frictional moment was measured.

These yaw and rock/roll motions, consistent with those that occur during curve negotiation, can produce at least two (2) different types of loading conditions as shown in Figures 11 to 12. A further two loading conditions that, although not simulated by the test rigs, can occur during curve negotiation in actual rail operations and are shown in Figures 13 to 14.
Figure 9 Theoretical frictional moment for 125 ton wagon load [adapted from 2].
Figure 10 Schematic of Thrall Car Manufacturing Co. frictional moment test rig [2].

Table 4 Frictional moment testing conditions.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical load (kN)</td>
<td>686</td>
</tr>
<tr>
<td>Centre bearing diameter (mm)</td>
<td>405</td>
</tr>
<tr>
<td>Average plate contact pressure (MPa)</td>
<td>5.6</td>
</tr>
<tr>
<td>Yaw motion</td>
<td>+/- 2.5 to 3.0º @ 7 to 10 cycles/min</td>
</tr>
<tr>
<td>Rock/roll motion</td>
<td>+/- 19.1 mm @ 14 to 20 cycles/min</td>
</tr>
<tr>
<td>Angular speed (mrad/s)</td>
<td>10 to 17</td>
</tr>
<tr>
<td><strong>Sliding speed</strong> at the rim wall (mm/s)</td>
<td>2.1 to 3.5</td>
</tr>
<tr>
<td>Total number of yaw cycles</td>
<td>30,000 to 50,000</td>
</tr>
<tr>
<td><strong>Sliding distance</strong> at the rim wall due to yaw motion (m)</td>
<td>884 to 1,060</td>
</tr>
</tbody>
</table>

Please see print copy for Figure 10
The first loading condition, as shown in Figure 11, includes yaw motion only and provides even load distribution without any rim wall contact. The test rig measurement of frictional moment is made under this type of loading condition.

The second loading condition, as shown in Figure 12, has yaw and roll moments and provides an uneven load distribution due to the addition of the roll moment. The wagon body may roll/sway in the centre bowl liner with little restraint from either the centre-pin or the rim wall of the centre bowl liner [1].

The third loading condition, as shown in Figure 13, provides a point load due to very high roll/sway moment. The centre plate has lifted off on one edge, and the whole load is placed on the point of the other edge. This can occur due to sway moment through track spirals [3]. For this loading condition, since the moment arm is further from the centre of rotation than in the first loading condition, the frictional moment is about 1.5 times that of the first loading condition [4].

The fourth loading condition, as shown in Figure 14, includes both centre plate and rim wall contact [5]. The centre plate contact has an uneven ring contact load distribution which is likely due to bolster elastic bending. This type of uneven load distribution has been observed in actual centre bearings through the use of force sensitive paper [5]. Due to the centre bearing diametral clearance of around 5 mm, the top centre may slide along the centre bearing plate surface until it contacts the rim wall. [1]. Contact at the rim wall adds a further component to the frictional moment as shown in Figure 9 [2].
Figure 11 Even load distribution - no rim wall contact [4].

Please see print copy for Figure 11

Figure 12 Uneven load distribution due to roll moment / bolster pitching [4].

Please see print copy for Figure 12
Figure 13 Point load due to roll moment / bolster pitching [4].

Figure 14 Uneven ring contact load distribution with rim wall contact [5].
A selection of results from Richmond [2] and Wu, Robeda and Guins [5] full-scale laboratory testing and test track studies of the effect of different materials and lubrication schemes on the co-efficient of friction are presented in Figure 15. In all cases, the top centre is steel (unspecified grade and hardness) and the centre bowl liner is plain carbon steel. For the unmodified polymer liner and fabric re-inforced thermosetting resin samples dirt contamination consisting of 0.5 cubic inch of 50% sharp sand:50% brake shoe dust was applied evenly to centre bearing at start of test [2]. For the polymer 1 - test rig sample, 1.0 cubic inch of this dust was applied evenly to centre bearing surface after 30,000 cycles [5].

For 125 ton wagon loads, Wu, Robeda and Guins recommend that polymer plate liners with a co-efficient of friction between 0.12 and 0.17 would perform well where curve negotiation is a concern, whilst a co-efficient of friction between 0.27 and 0.32 would accommodate both curve negotiation and bogie hunting along tangent track [5]. These recommended plate surface co-efficient of friction ranges are highlighted in Figure 15. The effect of friction on bogie hunting is discussed in the following section. In conjunction with a polymer plate liner, Wu, Robeda and Guins recommend the use of a steel rim wall liner because of the problems of rim damage in polymer bowl liners [5].

The un lubricated steel top centre against plain carbon steel centre bowl liner experienced excessive frictional moment of 31.7 kN.m and rising after 3,100 cycles and the test had to be terminated. At this stage the calculated co-efficient of friction was 0.27 and rising [2]. Wu reported a co-efficient of friction of 0.5 for an un lubricated steel against steel centre bearing tested using a similar frictional moment test [4]. For the molybdenum based grease bearing, the initial co-efficient of friction was 0.10, but the test had to be terminated after 5,500 cycles when the co-efficient reached 0.48 [2].
Richmond observed that most of the molybdenum based grease had been squeezed out from the bearing providing an essentially unlubricated condition [2]. The test track data also indicated that the molybdenum disulphide lubricant would only last around 6 to 8 months in service [5].

High viscosity lithium based grease applied to the steel top centre against plain carbon steel centre bowl liner provided, by Wu, Robeda and Guins friction range recommendations, too low a co-efficient of friction of 0.1 [2]. Dirt and water contamination caused higher, but inconsistent frictional moments that are undesirable to bogie dynamics. Observation of the contaminated centre bearing at the end of the test revealed the entire bearing was rusty with no sign of lubrication [2].

The modified UHMW polyolefin (polyethylene or polypropylene) polymer liner, fabric re-inforced thermosetting resin, and polymer 1 all provided acceptable and consistent co-efficient of friction values over the test durations that were within a range from 0.18 to 0.29 (Figure 15), whilst the co-efficient of friction of the unmodified polymer liner increased from 0.3 at the start of the test to 0.48 after 29,000 cycles which was inconsistent and out of the recommended range. It is clear that polymer liners are best suited to provide the consistent co-efficient of friction levels required to limit the frictional moment to below 28.3 kN.m in bogies loaded with 125 ton wagons.

The average contact pressure at the centre bearing plate surface for an unloaded 125 ton wagon would be 1.1 MPa, whilst under the simulated testing conditions discussed above it was 5.6 MPa. At this stage, there are no published papers on the effect of centre bearing plate friction on frictional moment for unloaded wagons, or lighter loaded wagons, such as 50 ton freight wagons.
Figure 15  A selection of results from Richmond [2] and Wu, Robeda and Guins [5] studies of the effect of different materials and lubrication schemes on co-efficient of friction utilizing full-scale laboratory test rigs and/or test track data.
2.5.2 Effect of centre bearing friction on bogie hunting

Bogie hunting is a yaw and lateral displacement oscillation of the wheelsets and bogie frame which is initiated by track irregularities such as misalignments. If the track is good quality, then bogie hunting is not a problem [1]. Bogie hunting is damped below a certain critical value of forward speed. Above the critical speed, the amplitude of oscillation increases until the bogie is banging from rail to rail, limited only by the wheel flanges. [1]. Bogie hunting occurs on tangent track and it may occur with empty or light cars at speeds as low as 56 km/hr. The yaw motion of the bogie causes it to trace a cyclic path along the track as shown in Figure 16 [1].

Bogie hunting causes accelerated wheel and rail wear, fatigue and wear damage to car components such as centerplates, bolsters and sideframes. In extreme cases, it can cause derailments. [1]. Some railways use unlubricated steel-against-steel centre bearings to reduce the risk of bogie hunting [11]. This is the case in Australia for lightly loaded (e.g. 50 ton wagon mass) freight cars running on largely tangent track with only limited curves. The wear of such unlubricated centre bearings is examined in this thesis. Constant contact side bearings also provide another mechanism to control bogie yaw motion, which in turn controls vehicle hunting.

Please see print copy for Figure 16

Figure 16 Bogie hunting. [1].
2.5.3 Effect of centre bearing friction on impact loads

Simson has investigated the effect of centre bearing friction on impact loads using a computer simulation model of longitudinal dynamics in a 73 ton coal wagon [12]. The model included constant contact side bearings and a 15 mm centre bearing diameter clearance [12].

The results show that point loading due to bolster pitching from longitudinal impact loads of around 175 kN, as shown in Figure 13, occurs routinely in empty wagons with centre bearing co-efficient of friction below 0.5 at a dynamic acceleration of 21 m/s\(^2\). For the fully loaded wagons, due to the higher friction force, rim contact only occurred for low friction (0.1 to 0.3) and extreme longitudinal dynamics (dynamic acceleration of 28 m/s\(^2\)) [12].

Some members of the Australian rail industry have reported failures of polymer centre bowl liners due to plastic flow caused by extreme impact loading events [12]. Wu, Robeda and Guins recommend the use of horizontal plate polymer liner, with steel rim wall liner because of the problems of rim damage in polymer bowl liners [5]. These failures could be due to:

- small rim wall contact height [3]. The rim wall contact height that is commonly used throughout the Australian rail freight industry is around 30-35 mm. A polymer centre bowl liner with a 55 mm rim wall contact height in service for around 8 years on a 100 tonne coal wagon has been observed without any plastic flow or cracking. The greater the rim wall contact height, the greater the rim wall contact area, the lower the peak contact stresses associated with longitudinal impact loads [3].

- large diametral clearance. Good design of the centre bearing diameter clearance also
has potential to reduce failures in polymer centre bowl liners. Typically the new centre bearing diameter clearance is at least 5 mm, and the bearing is condemned at 10-15 mm. The advantages of a tighter clearance are twofold:

1. The smaller distance traversed before impact, thus less time for the wagon to accelerate, means a lower impulse/impact velocity and subsequently lower peak contact stresses.

2. The smaller the difference in diameter between the top centre and centre bowl liner the greater the rim wall contact area. Again, this reduces peak contact stresses.

2.5.3.1 Alternative mechanical design for impact load situations

Over the years, the University of Ghent and research partners have studied the small- and large-scale friction and wear testing, and mechanical testing of polymer and polymer composite materials [13, 14, 15, 16, 17, 18, 19, 20]. The polymer composite materials and mechanical designs included:

1. 100 mm diameter discs of glass-fibre reinforced thermoplastic with PTFE friction-reducing modifiers, initially lubricated, against chromium plated steel. The discs are used in large spherical bearings, such as shown in the schematic in Figure 17, for segment gates and heavy steel constructions [13]. The discs can be reinforced by a steel core to minimize creep [13].

2. 175 mm diameter discs of ultra-high molecular weight polyethylene (UHMWPE), a material that behaves similarly under friction and wear testing to high density polyethylene but with a higher compressive yield strength and melting point. The disc-shaped UHMWPE pads have been reinforced around their
perimeter with a carbon-fibre reinforced epoxy ring (Figure 18). A lip of UHMWPE material protects against direct contact between the steel counterface and carbon fibre reinforced epoxy ring. Friction and wear testing at contact pressures up to 150 MPa provided acceptable results.

Such designs could be applied in a smaller scale to the rim wall surface to address the problem of yielding or creep in high loading situations.

Please see print copy for Figure 17

Figure 17 Schematic of dry rubbing bearing design with polymer circular pads as bearing surface [13].
Please see print copy for Figure 18

Figure 18 Hybrid UHMWPE-discs with a carbon/epoxy composite reinforcing ring [17].
2.6 Typical operating conditions

The typical operating conditions at the centre bearing plate, and rim wall under curve negotiation for a 96 tonne wagon load traversing the Goonyella Line, Central Queensland are listed in Table 5 and Table 6 [6].

Table 5 Centre bearing plate loading conditions.

<table>
<thead>
<tr>
<th></th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical load (kN)</td>
<td>470</td>
</tr>
<tr>
<td>Centre bearing diameter (mm)</td>
<td>355</td>
</tr>
<tr>
<td>Plate contact area (mm$^2$)</td>
<td>72,805</td>
</tr>
<tr>
<td>Average <strong>plate contact pressure</strong> (MPa)</td>
<td>6.5</td>
</tr>
</tbody>
</table>

Table 6 Centre bearing rim wall sliding and load conditions.

<table>
<thead>
<tr>
<th></th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Transition curve length (m)</td>
<td>40</td>
</tr>
<tr>
<td>Train speed in transition curve (km/hr)</td>
<td>40</td>
</tr>
<tr>
<td>Angular speed (mrad/s)</td>
<td>8</td>
</tr>
<tr>
<td>Yaw motion / transition (º/transition)</td>
<td>1.7</td>
</tr>
<tr>
<td>Sliding distance / transition (mm/transition)</td>
<td>5.8</td>
</tr>
<tr>
<td>Number of transition curves / per return trip</td>
<td>170</td>
</tr>
<tr>
<td>Number of return trips per year</td>
<td>320</td>
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<tr>
<td>Number of transition curves traversed in 5 years</td>
<td>54,400</td>
</tr>
<tr>
<td><strong>Rim wall load</strong> in transition curve (kN)</td>
<td>60</td>
</tr>
<tr>
<td><strong>Sliding speed</strong> at the rim wall (mm/s)</td>
<td>1.6</td>
</tr>
<tr>
<td><strong>Sliding distance</strong> at the rim wall due to yaw motion over 5 years (m)</td>
<td>1,586</td>
</tr>
</tbody>
</table>
2.6.1 Sliding speed

The sliding speed of 1.6 mm/s (Table 6) is in close agreement with the calculated sliding speed of 2 mm/s at the rim wall for the TTCI test rig [5].

2.6.2 Sliding distance

The sliding distance over a five year period of 1,586 m is 1.8 times the calculated sliding distance of 873 m at the rim wall for the TTCI test rig over 50,000 yaw cycles required to pass the standard test [5].

2.6.3 Geometry

Initially the geometry is non-conforming due to the different diameters of the new top centre and centre bowl liner, or diametral clearance. Over time the worn top centre is likely to conform to the worn centre bowl liner, as illustrated in Figure 72 and Figure 73 in Chapter 6. However, through the life of the centre bearing a wagon top centre does not necessarily remain on the same bogie centre bowl liner. If it is moved it is quite possible that the new centre bowl liner doesn’t have the same worn geometry as the previous one, so the contact will be non-conforming again.

2.6.4 Curve negotiation rim wall loading conditions

According to dynamic modeling work by Simson, the load on the rim wall surfaces through transition curves is 60 kN [6]. Based on this load, the calculated average and maximum contact pressures for different material pairs are listed in Table 7. The formulas used are presented in Appendix A. The low elastic modulus of HDPE of 0.8 GPa, spreads the load over a large contact area (13,133 mm2) providing a relatively low average contact pressure of around 5 MPa. This compares to around 50 MPa for the new AISI 1053 steel top centre against Hadfield steel centre bowl liner.
Table 7 Centre bearing rim wall average and maximum contact pressures of different new and worn centre bearing material pairs based upon loading conditions listed in Table 6.

<table>
<thead>
<tr>
<th>Top centre material</th>
<th>Centre bowl liner material</th>
<th>Condition of bearing</th>
<th>Centre bearing diametral clearance (mm)</th>
<th>Rim wall contact height (mm)</th>
<th>Elastic contact length (mm)</th>
<th>Contact area (mm²)</th>
<th>Average contact pressure (MPa)</th>
<th>Maximum Hertzian contact pressure (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>High density polyethylene</td>
<td>New or worn with non-conforming geometry</td>
<td>5</td>
<td>35</td>
<td>328.6</td>
<td>13,133</td>
<td>5.2</td>
<td>10.0</td>
<td></td>
</tr>
<tr>
<td>Hadfield steel</td>
<td>New with non-conforming geometry</td>
<td>3</td>
<td>35</td>
<td>425.4</td>
<td>14,891</td>
<td>4.0</td>
<td>7.7</td>
<td></td>
</tr>
<tr>
<td>Hadfield steel</td>
<td>Worn with conforming geometry</td>
<td>5</td>
<td>55</td>
<td>262.1</td>
<td>14,418</td>
<td>4.2</td>
<td>7.9</td>
<td></td>
</tr>
<tr>
<td>AISI 1053 medium carbon steel</td>
<td>High density polyethylene</td>
<td>-</td>
<td>35</td>
<td>-</td>
<td>3,253 (a)</td>
<td>18.4</td>
<td>-</td>
<td></td>
</tr>
</tbody>
</table>

(a) Based on worn subtended contact angle of +/- 15º about longitudinal direction and conforming worn geometry.
2.6.5 Impact loads

In the case of the AISI 1053 steel top centre and Hadfield steel centre bowl liner components observed in Chapter 6, they were used in the unlubricated condition. These steel centre bearing components were from vehicles comprising a 50 ton wagon load. This load is less than the 73 ton load used by Simson [12] to model the impact loads as mentioned in section 2.5.3. It will be shown in Chapter 6 that there were no obvious signs of impact wear in the observed worn components. As Simson has indicated, these impact loads would certainly be an important consideration when polymer plate and polymer centre bowl liners are used [12].

2.6.6 Environment

Rail freight trains operating in the Pilbarra region of Western Australia commonly experience ambient temperatures of at least 45 ºC during the summer months. High density polyethylene has a maximum useful service temperature of 80 ºC [22]. Iron ore and coal dusts, sand and water are foreign substances that can contaminate the centre bearing. Foam dust seals help protect against dust and sand. Water has no adverse affect on the mechanical properties of high density polyethylene [22] which is the most common polymer used for centre bearing surfaces.

2.7 Centre bearing rim wall wear

The literature search didn’t reveal any previous reports or papers regarding centre bearing rim wall wear. However, it is understood that Dr Stephen Marich initiated the introduction of polymer centre bearing liners into Australia to address the problem of rim wall wear in the 1980’s [21]. There have been no reports, from industry, of failures of polymer liners due to wear.
Hypothetically, one, or a combination, of the following four loading conditions could cause rim wall wear of the centre bearing in the longitudinal direction:

1. Sliding of the rim wall surfaces during bogie rotation through transition curves, and/or
2. Combined impact and sliding of the rim wall surfaces during entry to transition curves, and/or
3. Impact only between the rim wall surfaces due to extreme longitudinal train dynamics that can occur particularly in empty wagons [12], and/or
4. Sliding between the corner of the top centre and the centre bowl liner due to pitching of the bolster caused by longitudinal impacts [12]. The relative contributions of these loading conditions to rim wall wear is not known.

As mentioned in section 2.6.5 there were no obvious signs of impact wear in the observed worn components, thus loading conditions 3. and 4. could be precluded. However, for an impact load less than the normal rim wall curve negotiation load as described in section 2.6.4 combined with sliding (loading condition 2.) the resultant worn microstructure may be similar to pure sliding as in loading condition 1, which would mean the first two loading conditions could contribute to wear.

Chapter 6 includes a study of a worn high density polyethylene centre bowl liner, and worn AISI 1053 medium carbon and Hadfield steel centre bearing surfaces. The worn AISI 1053 medium carbon and Hadfield steel centre bearing components were from vehicles comprising a 50 ton wagon load. This load is less than the 96 ton load used in calculating the typical operating conditions [6] discussed in section 2.6 above. Subsequently, inferences made about the wear observed at the rim wall are based on a 31 kN rim wall load, instead of the 60 kN load for a 96 ton wagon load.