Surface coatings for 3-piece freight bogie centre bearings

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Chapter 3: Dry sliding wear of metals

3.1 Mechanical wear processes

Mechanical wear processes, as distinct from corrosive wear processes, can be classified according to 5 groupings – abrasion, erosion, adhesion, surface fatigue and impact. The first 4 types of wear processes or mechanisms are shown in Figure 19 in order of increasing severity [23]. More than one process may contribute to wear. The most severe wear process is addressed during design for wear reduction.

Given the operating conditions at the rim wall as described in Chapter 2, the processes that could be of interest in this research are those underlined in Figure 19 – polishing, adhesion, galling, pitting, delamination, ratcheting wear, plus impact wear. However, with regards to the higher wearing AISI 1053 steel top centre, it will be shown in Chapter 6 that the likely wear mechanism is plastic strain accumulation combined with low-cycle fatigue [24], or delamination [25]. Ratchetting wear which is described briefly in this chapter could also be a secondary wear mechanism.

Please see print copy for Figure 19

Figure 19  The different mechanical wear processes [23]. The processes of interest in this thesis are underlined.
3.2 Contact between real surfaces

The surface roughness of most engineering surfaces limits the contact between solid bodies to a very small portion of the apparent contact area [26]. The sum of the individual asperity contact areas ($\Sigma A_i$) is known as the real or true contact area ($A_r$) (Figure 20) [26].

Figure 20 Contact of rough surfaces. The real or true contact area of rough surfaces, $A_r$, is the sum of the asperity contact areas, $A_i$. [adapted from 26].

3.2.1 Chemical aspects

Even though surfaces are nominally clean and dry, in reality “metals will usually be covered with a film of oxide, which is covered by a second film of adsorbed gases and hydrocarbons (oils)” [quoted from 27, p22]. The thickness of the adsorbed layers is of the order of 10 nm [27]. These films interfere with the bonding or adhesion between the sliding substrate materials [27]. At light loads during sliding, there will be some squeezing out of the adsorbed films producing some contact between solid (e.g. oxide, sulfide) layers [27]. At high loads during sliding, the solid layers of the mating substrate materials may fracture, “particularly if the substrate beneath the solid layer deforms plastically” [quoted from 27, p23], permitting direct contact between the substrate
materials [27].

3.3 Friction and wear

According to Meng there are over 300 equations for friction and wear [28]. It is not the purpose of this thesis to describe any model in detail, but rather to determine which theory and model could apply to the worn components observed in Chapter 6. The basic laws of friction and Archard’s wear law are briefly described. The rigid-perfectly plastic, plane strain slipline field theory and model of asperity deformation introduced by Challen and Oxley [29] can be used to explain the microstructures of the worn AISI 1053 medium carbon steel top centre observed in Chapter 6, so is described in more detail.

3.3.1 Friction

The co-efficient of friction is a dimensionless quantity. It evolved from the work of many philosophers, scientists and engineers; in particular, da Vinci (1452-1519), Amontons (1663-1705), and Coulomb (1736-1806) [30]. The basic laws of friction (usually attributed to Amontons) are:

1. “the frictional force is directly proportional to the force acting normal to the contacting surfaces” [quoted from 31, p145], and

2. the frictional force is independent of the apparent area of contact of the surfaces [31].

The first law can be expressed as:

\[ F = \mu N, \]  

(1)
where \( F \) is the friction force, \( \mu \) is the co-efficient of friction, \( N \) is the normal force [30].

There are two types of friction co-efficients:

- **static co-efficient of friction** \((\mu_s)\) – “represents the friction opposing the onset of relative motion (impending motion)” [quoted from 30, p586], and

- **kinetic co-efficient of friction** \((\mu_k)\) – “represents the friction force opposing the continuance of relative motion once that motion has started” [quoted from 30, p586].

It is also referred to as the dynamic co-efficient of friction and by the symbol \( \mu_d \).

The static and kinetic co-efficients of friction are both material- and system-dependant [30].

Amontons thought the cause of friction was due to the collision of surface irregularities, and Coulomb said friction was due to the interlocking of asperities [32]. Hardy observed that by applying one monolayer of lubricant between the contacting surfaces the friction was reduced, thus casting doubts about the idea that friction is due to interlocking asperities. Bowden and Tabor researched friction and lubrication from the 1930’s, and proposed the adhesion theory of friction in 1959. “The frictional force is explained as the force needed to shear the welded junctions formed by adhesion at the tips of contacting asperities. The welded surfaces are assumed to be parallel to the sliding direction and the normal stress, \( \sigma \), and shear stress, \( \tau \), acting on them are taken to be independent of each other and to be related respectively to the indentation yield stress (hardness) and shear flow stress of the contacting metals.” [quoted from 31, p145].

If the real contact area equals \( A_r \), then the normal force \( N = A_r \sigma \) and the frictional force \( F = A_r \tau \). “The co-efficient of friction is therefore given by \( \mu = F/N = A_r \tau / A_r \sigma = \tau / \sigma \) and
the basic laws of friction are satisfied.” [quoted from 31, p145]. However, the ratio of the shear flow stress to its indentation yield stress for ductile metals is between 0.17 and 0.2 [32] and this is much less than the values for μ normally obtained for un lubricated metals [31].

In an effort to take account of the interdependence σ and τ, the model was modified by Tabor by using the von Mises yield criterion [32, 31]. The schematic in Figure 21 “shows a slab of material loaded against a rigid plane surface, representing in very idealised form an asperity contact” [quoted from 33]. In Figure 21(a) the material is subjected to compression by a normal stress, p₀, and is on the point of yielding. Now, when a tangential stress applied (does not imply sliding) to the asperity junction as in Figure 21(b), the material experiences an additional shear stress, τ. “According to the von Mises yield criterion, for the material to remain at the point of yielding, the normal stress, p₁, must be reduced compared to p₀.” [quoted from 33]. If the normal load remains constant, then the area of contact must grow known as junction growth. However this model does not limit junction growth, so “in theory it could continue until the whole area of specimen was actually in contact, and the μ would reach a very high value” [quoted from 33]. In most practical cases junction growth is limited by the ductility of the material, and by the presence of weak interfacial films such as adsorbed gases, and thin oil films [33].
Figure 21 Schematic of a slab of material loaded against a rigid plane surface: (a) with no tangential load applied, (b) with tangential friction force applied [33].

\[ p_1^2 + 3\tau^2 = p_0^2 \]  \hspace{1cm} (2)

where \( p_1 = N/A \) (normal stress), \( \tau = F/A \) (shear stress), \( p_0 = \sigma_y \) (yield strength).

This concept of an interfacial film separating the surfaces was incorporated by Tabor into the adhesion model [31]. In this model, the shear stress at an asperity junction “resulting from an applied tangential force (equal to the frictional force when sliding occurs) will only cause sliding if it is equal to the shear strength of the interfacial film” [quoted from 31]. If the shear stress is less than the shear strength of the interfacial film, then “deformation of the asperities will occur under the combined action of the normal and shear stresses and the area of the junction contact will increase (junction growth); this process will continue until the tangential force is sufficient to shear the interfacial film” [quoted from 31]. Using this model Tabor derived the relation [31]

\[ \mu = 1/\sqrt{\delta(f^2 - 1)} \]  \hspace{1cm} (3)

where \( \delta \) = empirical factor which Tabor took to be 9, \( f = \tau /k \), with \( \tau \) = shear strength of
the film, \( k = \) shear flow stress of the deforming material.

If \( f = 1 \) such as in clean surfaces [32], then \( \mu \) approaches \( \infty \). “However, if \( f \) is reduced by 5% to 0.95, then \( \mu \approx 1 \)” [quoted from 31]. This is consistent with experimental results which show \( \mu \) is in excess of 100 for chemically clean surfaces, while the introduction of even a small amount of contamination reduces \( \mu \) to the order of 1 [31].

A ploughing component of friction, described as the deformation force needed to plough the softer surface by the hard asperities of the counterface [31], is sometimes added to the adhesive component in an attempt to account for surface roughness effects, that is:

\[
\mu = \mu_a + \mu_p
\]  

(4)

where \( \mu_a \) and \( \mu_p \) are the adhesion and ploughing components of the co-efficient of friction [31]. “Bowden and Tabor argued that for most situations with metal surfaces \( \mu_p \ll \mu_a \) and \( \mu_p \) can therefore be neglected” [quoted from 31].

3.3.2 Wear

3.3.2.1 Archard wear law

In 1953 Archard proposed that the wear of metals based on the original asperity deformation model of Bowden and Tabor could be described by a simple empirical law [34, 31]. The assumptions are:

- Contact occurs at the asperities, and
- The asperity deformation is plastic, and
- The true area of contact equals the sum of all asperity contact areas, and
- The true area of contact is closely proportional to the normal load [35].
Figure 22 is a schematic showing the evolution of a single contact area as two asperities move over each other [35]. At (c) the asperities are co-incident and the asperity contact has reached maximum size [35]. It is assumed that the radius of this contact equals a (Figure 22). “The fraction of the normal load, δN, supported by this area is then given by” [quoted from 35]:

\[ \delta N = \sigma_y \pi a^2 \]  \hspace{1cm} (5)

where \( \sigma_y \) = yield pressure (approximately equal to the indentation hardness).

**Figure 22 Schematic showing the evolution of a single contact area as two asperities move over each other [35]. Asperity radius = a.**

During further sliding, the load is transferred to other asperity contacts, which in turn plastically deform. “Wear is associated with the detachment of material from the asperity contacts and the volume of each wear debris particle will relate to the size of the individual asperity contact” [quoted from 35]. Assuming that this volume, \( \delta V \), is proportional to the cube of the contact radius, \( a \), and the detached particle is hemispherical in shape then [35]:

\[ \delta V = \frac{2}{3} \pi a^3 \]  \hspace{1cm} (6)
“It is assumed that a proportion, $\varphi$, of asperity contacts generate wear debris particles”

[[quoted from 35]. For a sliding distance of $2a$, the wear rate in terms of worn volume loss per unit sliding distance, $\delta Q$, can then be expressed as:

$$\delta Q = \varphi \frac{\delta V}{2a} = \frac{1}{3} \varphi \pi a^2$$

(7)

The total wear rate, $Q$, is the sum of the contributions over the real contact area:

$$Q = \Sigma \delta Q = \frac{\varphi}{3} \Sigma \pi a^2$$

(8)

The total normal load is:

$$N = \Sigma \delta N = \sigma_y \Sigma \pi a^2$$

(9)

Substituting $\Sigma \pi a^2 = N/\sigma_y$ (from equation (9)) into equation (8) gives:

$$Q = \varphi \frac{N}{(3 \sigma_y)}$$

(10)

If $\varphi/3 = K$ (dimensionless wear co-efficient), and assuming $\sigma_y$ = indentation hardness, $H$, then equation (10) becomes the Archard wear law:

$$Q = K \frac{N}{H}$$

(11)

Equation (11) is normally expressed in terms of the total worn volume, $V$:

$$V = K \frac{NL}{H}$$

(12)

The wear co-efficient, $K$, “allows for the fact that while all asperity interactions contribute to friction only some contribute to wear” [quoted from 31]. Alternatively, it could also “represent the number of cycles of deformation required by each asperity before a wear debris particle is ejected” [quoted from 35]. In any case, experimental results have shown that the worn volume is proportional to the normal load applied and
sliding distance, and inversely proportional to the hardness of the wearing surface as per equation (12) [31]. Wear co-efficient data for different tribological systems collected over time has proved very useful in design [31]. The dimensional normalized or specific wear rate \( \text{(m}^3/\text{N.m}) \) is also used in design. It is defined by the equation:

\[
\text{Normalised or specific wear rate} = \frac{V}{NL}
\]

The limitations of Archard’s wear law as described by others are listed below:

- Doesn’t consider the physics and physical metallurgy of metal deformation [25]. “The involvement of fracture in the process appears far too severe, particularly for the sliding of smooth well-lubricated surfaces over each other.” [quoted from 31], and

- “It indicates that wear particles will be welded to asperities on the harder surface” [quoted from 31], but it is not clear how they become separated as debris [31], and

- “The theory doesn’t provide any insight to the wear of metals under different sliding conditions” [quoted from 25], and

- It assumes that both contacting asperities deform plastically. “This will not be the case where one surface is significantly harder than the other.” [quoted from 31].
3.3.2.2 Plasticity Index

“The plasticity index, $\Psi$, helps to predict the onset of plastic flow at asperities” [quoted from 35, p14] and incorporates surface roughness. Several models for plasticity index exist: Greenwood and Williamson, Whitehouse and Archard, Bower and Johnson, to name a few [26]. They use statistical methods “to describe the complex nature of the contact between two rough surfaces” [quoted from 26, p463]. The Greenwood and Williamson model defines the plasticity index, $\Psi$, by the equation:

$$\Psi = \frac{E_r}{H} \times \left(\frac{\sigma^*}{r}\right)^{0.5} \quad (14)$$

Where $E_r$ = the reduced or composite elastic modulus (Pa), $H$=indentation hardness (Pa), $\sigma^*$ = standard deviation of the surface peak height distribution (approximated by RMS surface roughness [36]) (m), $r$ = asperity radius ($1/r$ can be approximated by the RMS curvature [36]) (m).

The composite elastic modulus, $E_r$, is given by the equation:

$$\frac{1}{E_r} = \left(1-\nu_1^2\right)/E_1 + \left(1-\nu_2^2\right)/E_2 \quad (15)$$

Where $\nu_1$ and $E_1$, and $\nu_2$ and $E_2$ are the Poisson’s ratio and Young’s elastic modulus for contacting materials 1 and 2, respectively.

The Greenwood and Williamson model predicts whether the deformation is elastic or plastic for static contacts prior to the onset of yielding. For “$\Psi < 0.6$ elastic deformation dominates and if $\Psi > 1.0$ a large portion of contact will involve plastic deformation” [quoted from 26, p465]. The Bower and Johnson model provides a plasticity index, $\Psi_s$, for repeated sliding however it is beyond the scope of this thesis to discuss it further.
3.3.2.3 Stresses in sliding contact

The surface stresses due to frictional traction are shown in Figure 23 [36]. As shown there is a maximum compressive and tensile stress, ahead and trailing, respectively, the contact point of sliding motion.

Please see print copy for Figure 23

Figure 23 Surface stresses due to frictional traction for a cylinder sliding perpendicular to it’s axis (x-axis represents sliding direction) [36].

The different forms of component response to repeated loading during sliding conditions are shown in Figure 24 [37]. At sufficiently small loads, no material reaches the yield point and the load is carried entirely elastically and so is fully reversible (Figure 24a.) [37]. At a higher load, the component may plastically deform on the first sliding pass, but due to the generation of compressive residual stresses from the sliding pass and possibly strain hardening of the material, the new elastic limit known as the elastic shakedown limit (Figure 24b.) is higher than the original elastic limit, thus the load is again carried entirely elastically [37].

At loads above the elastic shakedown limit, plastic deformation will occur on each sliding pass [37]. In this case, the response can either be cyclic plasticity shown in
3.3.2.4 Plastic strain accumulation

In cyclic plasticity, the plastic strain is fully reversing, whilst during ratchetting, the strain cycle includes both a reversing and uni-directional component [38]. “In the case of a linear kinematically hardening material (i.e. one with a hardening modulus independent of the mean stress in the cycle) the steady cyclic state will consist of a closed loop of reversed plastic strain” [quoted from 38, p39]. This is the case when face-centred cubic metals with low stacking fault energy, remain in Stage II of the resolved shear stress – resolved shear strain flow curve, as is described for Stellite 6 in Chapter 5, section 5.3.4. However, this is the exception rather than the rule, so “non-linear hardening (i.e. a decreasing hardening modulus with increasing peak stress) ratchetting is expected, leading to accumulating shear strain in the near-surface layer” [quoted from 38, p39].

Ratchetting wear manifests itself as “very thin slivers or lamellae of material, typically less than a micron in thickness” [quoted from 37], which “are extruded from the contact patches on the softer surface” [quoted from 37]. This is shown schematically in Figure 25. In cross-section, the “filmy” wear particle may look similar to an ironed out wave as described by the Challen and Oxley wave removal model [29].

Figure 26 shows the effect of co-efficient of friction on plastic flow. “The magnitude and location of maximum shear stress are affected by tangential forces with the point of maximum shear stress moving towards the surface as friction increases and reaches the surface when $\mu$ approaches 0.33” [quoted from 35, p6]. At a co-efficient of friction above 0.33 the maximum contact pressure, $p_0$, required to cause plastic flow is
significantly lower than required to cause sub-surface plastic flow when $\mu<0.25$. At $\mu>0.33$, “plastic flow wear mechanisms dominate such as asperity adhesion-shear and fatigue crack propagation” [quoted from 35, p6].

Figure 24 The different forms of component response to cyclic loading. a) perfectly elastic, b) elastic shakedown, c) plastic shakedown or cyclic plasticity, d) incremental collapse or ratchetting failure [37].

Figure 25 Schematic showing wear particles formed by ratchetting wear [37].
Figure 26 Shakedown map for line contact showing Hertz pressure (normalized by yield stress in shear) plotted against friction co-efficient. [39].

3.3.2.4.1 Challen and Oxley wave formation model

In an attempt to overcome some of the limitations of Archard’s wear law, “Challen and Oxley introduced the rigid-perfectly plastic plane strain slipline field model of asperity deformation” [quoted from 31, p146] as shown in Figure 27. This model and associated low cycle fatigue wear model [40] can be used to describe some of the aspects of the observed wear behaviour in Chapter 6.
Figure 27 Wave model of asperity deformation: (a) slipline field, (b) hodograph [31].

In Challen and Oxley’s wave model, “the frictional force, which opposes the sliding of a hard metal surface over a soft one, is assumed to result from the pushing of waves of plastically deformed material in the soft surface ahead of asperities on the hard one” [quoted from 31]. Thus, “a frictional force can be accounted for without fracture necessarily occurring” [quoted from 31]. The model’s independent variables are defined as: “the slope of the hard asperity, α, and the normalized strength, f, of the interfacial film” [quoted from 31]. The co-efficient of friction is defined by the equation [31]:

\[ \mu = \frac{(\sin \alpha + \cos (\arccos f - \alpha))}{(\cos \alpha + \sin (\arccos f - \alpha))} \]  \hspace{1cm} (16)

where \( A = 1 + \pi/2 + \arccos f - 2\alpha - 2 \arcsin [(1-f)^{0.5} \sin \alpha] \)

Theoretical and experimental values for \( \mu \) are shown in Figure 28. There is good
correlation between experimental and theoretical results [31]. From Figure 28, it is shown that with boundary lubrication providing a low interfacial film strength ($f = 0.14$) the co-efficient of friction, $\mu$, only approaches 1.0 at high asperity angles of around 40º, whereas for an dry sliding ($f = 0.9$), $\mu$ approaches 1.0 at asperity angles of around 13º. For this current research, it would be reasonable to assume that the interfacial film strength, $f$, and co-efficient of friction, $\mu$, of an unlubricated AISI 1053 steel top centre against Hadfield steel centre bowl liner would be around 0.9 and 0.5, respectively. Based on these assumptions, from Figure 28 the asperity angle of the Hadfield steel against the softer AISI 1053 steel would be around 7º. “A relatively crude calculation [40] suggests that when a deforming material is steel then plastic waves are produced for $\alpha > 0.1^\circ$” [quoted from 31, p153]. Figure 29 shows the plastic flow for one single pass when $\alpha = 20^\circ$. Figure 30 shows schematically the effect of repeated sliding cycles on surface and sub-surface plastic flow. A photograph showing a wave without macroscopic cracks, and the plastic deformation that occurs as the wave moves across the surface is shown in Figure 31.
Figure 28 Theoretical (lines) and experimental (symbols) results for $\mu$. The open symbols used for tests in which a wave was formed and solid symbols for tests in which cracking occurred. Chain-dotted lines are for $f$ values one standard deviation on either side of mean $f$ values. The specimen material is 5086-H32 aluminium-magnesium alloy [31].

Figure 29 Experimental plastic flow field: grid 0.1 mm sides, $\alpha = 20^\circ$ [24].
3.3.2.4 Low cycle fatigue

The Challen and Oxley wave model describes the frictional force without involving fracture [31]. Using this model, the wear generation due to low cycle fatigue [40] and wave removal and chip removal [29] have been proposed. The former is of particular interest to this research. Kragelsky takes the view that it is cyclic stresses below the elastic limit that are responsible for the low cycle fatigue [40]. Challen, Oxley and Hockenhull (1986) suggest it is related to the plastic strain amplitude, with the number
of cycles to failure given by a Manson-Coffin type relation where [40]:

\[ n = \left( \frac{C}{\varepsilon_a} \right)^D \]  \hspace{1cm} (17)

where \( n \) = number of cycles to failure, \( C \) = constant related to the monotonic fracture strain for the process and material considered, \( \varepsilon_a \) = strain amplitude, and \( D \) = constant approximately equal to 2 for a range of materials. [40]

More recently (1999), Williams also comments that it seems likely that in many cases near-surface failures are of a ductile nature and result from an accumulation of plastic strain very close to the component surface rather than being driven by elastic stress intensity factors [37]. Challen, Oxley and Hockenhull assume that the strain amplitude is equal to the shear strain increment, \( \gamma_p \), from their model. From here they derive that the Archard wear co-efficient equals:

\[
K = 9 \times \sqrt{3} \times r \mu / (C^D \Delta \gamma_p^{1-D})
\]  \hspace{1cm} (18)

Where \( r \) = ratio of plastic work to total work, that the hard wedge in their model imparts on the softer surface, in the generation of a wear particle, \( \Delta \gamma_p \) = incremental shear strain per cycle, and:

\[
r \mu = \left( (A \sin \alpha + \cos (\arccos f - \alpha) - Bf) / (A \cos \alpha + \sin (\arccos f - \alpha) \right)
\]  \hspace{1cm} (19)

and:

\[
B = \left( \sin \left( \frac{1}{2} \arccos f - \alpha \right) \right) / \left( \sin \left( \frac{1}{2} \arccos f \right) \right)
\]  \hspace{1cm} (20)

Suh’s delamination theory of wear describes the detachment of a particle due to low cycle fatigue caused by subsurface void and crack formation [25]. The case of fatigue wear initiated by subsurface processes is considered. The rate of void formation may be
increased when there are hard particles present in the metal, since the motion of dislocations is blocked by them [25]. When these hard particles are stronger than the cohesive strength of the matrix, cracks and voids can be nucleated under the stress of dislocation pile-ups [25]. Small cracks can also form when these hard particles break up under the dislocation pile-up stress [25]. Voids can also form during plastic flow by the decohesion of the matrix-particle interface and by plastic flow of the matrix around a hard particle [25]. These processes are shown schematically in Figure 32. The cracks and voids may link together by three different mechanisms: growth of voids, crack propagation, and the plastic shear deformation of the metal between the voids. In Chapter 6, it will be shown to be the latter.

Please see print copy for Figure 32

Figure 32 Illustration of a process of subsurface crack formation by growth and link up of voids [26, adapted from 25].

The mechanism of surface crack initiated fatigue wear is illustrated schematically in Figure 33 [26]. A primary crack originates at a weak point on the surface and propagates downwards along weak planes such as slip planes, or dislocation cell boundaries [26], the latter occur in heavily strain-hardened metals [42]. A secondary crack can develop from the primary crack or the primary crack can connect with an
existing subsurface crack [26]. To generate a wear particle the crack must re-connect with the surface [26]. The tensile stresses that occur during the dry sliding contact of metals would assist opening of these cracks [35], and the final tearing of the wear particle. It has been found that during unlubricated sliding, in particular reciprocal sliding, wear particles can form due to the growth of surface initiated cracks [26].

Wear scars typical of plastic strain accumulation in conjunction with low cycle fatigue are shown in Figure 34, and Figure 35.

Please see print copy for Figure 33

Figure 33 Schematic illustration of the process of surface crack initiation and propagation [from 26].
Figure 34 Micrograph of micropits and cracks on the deformed aluminium surface [34]. Micron bar is shown by length of dotted line and represents 30 µm.

Figure 35 Fatigue wear particle formation on cast iron [26].
3.3.2.6 Wear mechanism maps

Wear mechanism maps provide useful information about the type of wear for a given set of conditions. A load-speed wear mechanism map for medium carbon steels based largely on pin-on-disc data is shown in Figure 36 [37]. The normalized sliding velocity, $V$, is defined:

$$V = V r_0 / a$$ \hspace{1cm} (21)

Where $V =$ actual sliding velocity, $r_0 =$ radius of the apparent area of contact, $a =$ thermal diffusivity.

The normalized pressure, $p$, is defined:

$$p = p / H$$ \hspace{1cm} (22)

Where $p =$ nominal average contact pressure, $H =$ indentation hardness.

The normalized sliding speed and pressure conditions for new and worn centre bearing (AISI 1053 steel top centre against Hadfield steel centre bowl liner), and the start and end of wear test (Hadfield steel pin against AISI 1053 steel plate) are also plotted on Figure 36. At the contact pressures relevant to the centre bearing operating rim wall and wear test conditions steel will begin to oxidize at sliding speeds greater than 1 m/s as shown in Figure 36. The reduced adhesion due the oxide layer helps provide a mild wear situation. Since, the average sliding speed for the rim wall and wear test are approximately 1.6 and 20 mm/s, respectively, a mild oxidational wear regime will not occur in these instances. The centre bearing conditions as plotted on Figure 36 would suggest a severe wear situation, whilst the wear test conditions are bordering on a severe wear situation. Mild wear typically produces fine wear debris (0.01-1 µm), predominantly of oxides, while severe wear produces particles of 20-200 µm size [35].
Figure 36 Load-speed wear mechanism map for medium carbon steels based largely on pin-on-disc data. Thick lines delineate different wear mechanisms and thin lines are contours of equal wear rates. Chain lines represent constant values of $pV$ factor. [37]. The normalized sliding speed and pressure conditions for new and worn centre bearing (AISI 1053 steel top centre against Hadfield steel centre bowl liner), and start and end of wear test (Hadfield steel pin against AISI 1053 steel plate) are shown on Figure 36 by symbols listed below.

<table>
<thead>
<tr>
<th>Symbol on Figure 36</th>
<th>Conditions</th>
<th>Normalised velocity</th>
<th>Normalised pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>◆</td>
<td>New centre bearing</td>
<td>0.60</td>
<td>0.095</td>
</tr>
<tr>
<td>□</td>
<td>Worn centre bearing</td>
<td>1.01</td>
<td>0.033</td>
</tr>
<tr>
<td>△</td>
<td>Start of wear test (1053 steel plate)</td>
<td>0.14</td>
<td>0.118</td>
</tr>
<tr>
<td>○</td>
<td>End of wear test (1053 steel plate)</td>
<td>0.31</td>
<td>0.023</td>
</tr>
</tbody>
</table>

Please see print copy for Figure 36
Figure 37 illustrates a wear mechanism map based on the sliding of a hard, rough material against a softer counterface [37]. At very low asperity attack angles and interfacial shear stress that you could find in the wear of polished surfaces with lubrication the load is carried mostly elastically and fatigue is the dominant wear mechanism. At higher asperity attack angles and low interfacial shear stress, a mild wear regime is encountered. A transition from mild to severe wear occurs at low-medium interfacial shear stress and at an asperity attack angle of approximately 0.9°.

Figure 37 Wear mechanism map for soft surface sliding against a harder rougher counterface. The interfacial shear stress, $\tau/k$, is analogous to the interfacial film strength, $f$. $\theta$ is the asperity attack angle. Thicker lines delineate different wear mechanisms and thin lines are contours of equal wear co-efficient [37].

Please see print copy for Figure 37
3.4 Alternative materials and processes selection

The materials and process selection is based upon the operating conditions/tribological parameters at the rim wall and the observed wear mechanism. Hadfield steel is maintained as the mating material.

3.4.1 Lubrication

Excellent lubrication of sliding surfaces can reduce the adhesive wear co-efficient by several orders of magnitude compared to no lubrication as shown in Figure 38 [43]. Lubrication is often the most cost effective method to reduce adhesive wear. Solid lubricants such as molybdenum disulphide and molybdenum- and lithium-based greases have been used previously to lubricate primarily the horizontal plate section of the centre bearing. However, due to the large diametral clearance and non-conforming geometry between the top centre and centre bowl liner the lubricant would likely be squeezed away from the longitudinal rim wall region where it is most needed to reduce wear. Thus lubrication of rim wall surfaces is not considered a practical option to reduce wear.

3.4.2 Materials and processes selection

The materials selection chart for bearings as shown Figure 39 provides the normalized dry sliding wear rate of materials for a given bearing pressure. Existing materials: high density polyethylene (HDPE), and medium carbon steel are highlighted in boldtype. HDPE can have normalized wear rates two orders of magnitude less than medium carbon steels which would be consistent with the limited results presented in this chapter. During discussions with industry there have been no reports of polymer centre bowl liners failing due to excessive wear of the rim wall. It is beyond the scope of this
project to investigate HDPE or polymer composites as possible alternative materials in any detail.

Figure 38 Wear co-efficients for various sliding conditions [43].

For all bearings, the selection of a material for one of the bearing surfaces guides the selection of the material for the mating surface. Hadfield steel is currently used as a centre bearing surface. It has good un lubricated sliding wear resistance and excellent impact toughness [44]. As demonstrated in this chapter, Hadfield steel forms a relatively smooth work hardened layer in service. For all these reasons, Hadfield steel has been selected as one of the candidate bearing surfaces.

The wear mechanism of the AISI 1053 steel top centre was incremental plastic strain accumulation in conjunction with low cycle fatigue. The design criteria for the material(s) to replace AISI 1053 steel and mate against the smooth work hardened Hadfield steel are:
• Similar hardness, thus shear yield strength, to Hadfield steel (550 HV). The idea is for the load to be carried elastically, and thus provide an adhesive or spalling/fatigue wear condition. Need to avoid a 2-body abrasive wear condition that can occur between materials of significantly different hardness. 2-body abrasive wear generally leads to a higher wear rate than adhesive or spalling/fatigue wear [45].

• Improved low cycle fatigue characteristics.

Nitrided steels, as highlighted on the chart (Figure 39), can have normalized wear rates an order of magnitude less than medium carbon steels. Stellite 6 has good unlubricated wear resistance [46]. Berg et. al. reported that plasma nitriding of similar steels provides a surface hardness of around 500 HV [50], whilst a reported hardness value for Stellite 6 laser clad layers was around 550 HV [51]. The hardness profiles and wear resistance, especially with respect to fatigue initiated wear, of plasma nitrided steels and Stellite 6 will be discussed further in Chapters 5 and 6, respectively.

The tempered martensite microstructure of AISI 4016 molybdenum steel also provides good impact toughness so will be used as a base material for plasma nitriding and laser cladding experiments. Mild steel is also used in laser cladding as a comparison.

The case depth of the plasma nitrided steel wear tested in this research was 375 µm. This is an order of magnitude lower than the wear depth observed in Chapter 6. Further testing beyond the work in this thesis would be required to determine the required plasma nitrided case depth. Laser clad thicknesses up to 2.0 mm are possible using the powder blown technique [52].
Please see print copy for Figure 39

Figure 39 Normalised wear rate versus maximum bearing pressure for dry sliding against a steel counterface [53].