Sliding Wear of Rail and Wheel Steels: Effect of Hardness Ratio, Normal Load and Lubrication

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Keywords: Wheel-rail system, Sliding wear, Pin on disc test, Hardness ratio, Normal load, Lubrication

ABSTRACT

The pin-on-disc test model system was used to simulate contact wear between wheel flange and rail gauge corner. Pins were extracted from a class C cast steel wheel and discs were extracted from premium and intermediate steel grades rail, both hypereutectoid and with hardness between 321 and 392 HB. Rails are pearlitic and the bainitic wheel is in its first life. Dry and lubricated tests were performed with normal load variation (40, 80 and 120 N). Under dry condition, there was no influence of the pair hardness to mass loss of the tribosystem studied, contrary to what is stated by literature. For the lubricated tests, mass loss was significantly lower and regardless of both, hardness ratio and normal load, than in the other tests. This shows the technological importance of: (i) implementing wheel flange lubrication (locomotive embedded process); (ii) improving rail lubrication techniques (mobile or way side process) and/or friction coefficient measurement techniques and (iii) need for revision of wheel standards and/or specifications.

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1. INTRODUCTION

Railway engineers work daily with the problem of high wear rate on rails and wheels, so efforts to solve this problem by increasing wheel and rail hardness (up to about 400 HB) have been made by researchers for many years [1]. This decision has caused controversy and discussion around the world, and it has not been resolved or ended [1].

Optimization of rail making technology has been carried out along the last four decades. However, wheel standards had remained unchanged since 1969 [2]. Considering this, studies on wheels have been made aiming to increase hardness to the 340 - 363 HB range, striving for reduction of wear rate of this component due to evolution of rail properties [2]. Yet, new studies have been carried out showing that material hardness only is not a sufficiently reliable criterion for achieving wear resistance [2].

One specific study [3] suggests a relationship between wheel hardness and rail hardness to reduce the (wheel-rail) system’s wear rate. The proposed ratio between rail/wheel hardnesses should be within 0.6 and 0.8 (wheel harder than rail).
rail). In contrast, another study [4] show that ratio between rail/wheel hardneces should be equal or greater than 1.0, while [1] show this ratio should be within 1.0 and 1.1. Table 1 depicts maximum hardneces used for rails and wheels in some countries, as well as the rail/wheel hardneces ratio “R” [1,2].

Table 1. Ratio “R” for rail/wheel hardneces in some countries [1,2].

| Country     | HBrail max | HBwheel max | R  |
|-------------|------------|-------------|----|
| Russia      | 401        | 363         | 1.1|
| Japan       | 400        | 360         | 1.1|
| USA/Canada  | 390        | 390         | 1.0|
| Brazil*     | 405        | 340         | 1.2|
| Europe      | 360        | 310         | 1.2|
| China       | 390        | 390         | 1.0|

*Adapted by the author

The hardneces ratio for wheel-rail pairs shows a relationship with wear rate for these components [2-4]. However, there are several other factors related to rail wear, such as track geometry, axle load, and speed, among others. In curves with sharp radius (R<350 M), wear is greater on the high rails and on wheels that run on these curves [5]. On these sites (goal of this study), on the contact between wheel flange and rail gauge corner, pure sliding occurs (Fig. 1), which causes greater wear rates than those seen on tangent tracks or moderate and/or large radius curves, where mechanisms of rolling contact fatigue are predominant.

![Fig. 1. (a) Gauge corner and lateral wear of rail head and (b) wheel flange wear.](image)

According to literature [6], studies report some field wear tests, which were used to propose wear trends for the wheel, the rail and the wheel-rail system [4]. Field wear tests (track) carried out at TTCI (Transportation Technology Center Incorporation), using different grades of rail steels with hardneces between 314 and 405 HB and wheels with hardneces between 249 and 317 HB, showed that the optimum rail/wheel hardneces ratio (Hrail/Hwheel) should be equal to or greater than 1.0 [4]. These tests were conducted in a closed-circuit railway inside TTCI, where the train runs on non-lubricated rail curves with radius of 350 m.

The relative wheel-rail wear from field tests at TTCI were combined and summarized (Fig. 2) [4], and some conclusions may be drawn in regard to the wheel-rail system, highlighting that when hardneces ratio is above 1.0, wheel wear stabilizes and rail wear is reduced. For railway companies, it is expected that rail wear should be lower than wheel wear, because rail replacing needs disruption of normal train traffic.

![Fig. 2. Rail/wheel hardneces optimal ratio [4,6].](image)

In the study carried out by [1], experiments were made using a model named "test rig" with a vertical disc (wheel material C74GW-T-B) against a horizontal bar (rail material M76TT), in a dry test. Hardneces of samples taken from the rail was within 321 and 401 HB. Hardneces of samples taken from wheels were within 293 and 363 HB. Based on experimental results from wear tests for the wheel-rail system, a mathematical model was developed to show the system’s behavior to wear, with recommendations that optimal hardneces ratio (Hrail/Hwheel) should be between 1.0 (Hrail 362 HB / Hwheel 361 HB) and 1.1 (Hrail 401 / Hwheel 363 HB), so that a lower wear rate could be achieved for the wheel-rail system.

In the same study [1], for R = 1.0, the lowest wear values for the wheel-rail system were reached when hardneces was 360 HB for both, rails and wheels (Fig. 3). The use of steel pairs with hardneces ratio R = 1.0, but from hardness
categories lower or higher than 360 HB, showed greater mass loss for the wheel-rail system.

Fig. 3. Dependences of wear of (1) wheel and (2) rail on ratio of their hardness [1].

A Brazilian study [7] was conducted by means of a pin (extracted from pearlitic rails with 387 HV) on disc (extracted from class C AAR 335 HV forged and 341 HV cast steel wheels) wear test model. The tests were carried out using round tip pins, constant normal load, under dry conditions and varying sliding speed (0.1 to 0.9 m/s). Wear rate for pin and disc was two to three times greater in the tribosystem with forged steel disc. The pair with forged disc showed hardness ratio R = 1.2, while the cast steel disc pair showed R = 1.1.

Fig. 4. “Magic pentagon” of wheel-rail interface. The arrows highlight the hardness ratio and lubrication parameters.

Friction management (lubrication and/or friction modification) as an isolated solution provides a certain level of benefits to railway operators. However, in order to maximize the benefits of friction management, it is necessary to consider this maintenance practice as just one of the factors that contribute to maintaining the wheel-rail system in good operating conditions. Friction management strongly interacts with three other factors such as rail and wheel metallurgy, rail and wheel profile (grinding/milling) and track geometry (superelevation of curves) as a function of current train speed [8]. At the Brazilian company VLI Logística, the railway engineering department adopted a fifth factor called infrastructure (of track and rolling stock), generating Fig. 4 named “magic pentagon” of the wheel-rail interface. Also, it is noticeable in this illustration two specific factors associated to the present study: ratio between pair hardresses and lubrication.

The objective of this study is to evaluate the impact of the ratio between hardness of the pair in contact, the normal load and lubrication in sliding wear tests with a pin-on-disc configuration using rail and wheel steels.

2. MATERIALS AND EXPERIMENTAL METHODS

2.1 MATERIALS

The pin-on-disc sliding wear tests were performed using two classes of rail steels for the discs (premium and intermediate) and a steel class for the pins (C). Table 2 shows chemical composition (mass percentage) of premium and intermediate class rails, specified by rail standard [9], and of wheel’s class C steel, specified by wheel standard [10].

Table 2. Chemical composition [9,10].

| Material/ [%]                  | C   | Mn  | P   | S   | Si  |
|-------------------------------|-----|-----|-----|-----|-----|
| Premium and intermediate rail | 0.74-0.86 | 0.75-1.25 | 0.020-0.030 | 0.020-0.030 | 0.10-0.15 |
| Class C wheel                  | 0.67-0.77 | 0.60-0.90 | 0.030-0.040 | 0.005-0.040 | 0.15-1.00 |

The rail standard [9] specifies that the rail head microstructure must be completely formed by pearlite, both for premium rails and intermediate rails. The wheel standard [10] specifies that the microstructure should not contain martensite, not being specified what should be the wheel microstructure.
Consistently with pearlitic microstructure, rail standard [9] sets Brinell hardness (HB) ranges for rails, and wheel standard [10] sets the Brinell hardness for wheels, as shown in Table 3. The strong correlation between the increase on rail wear resistance and refinement of pearlite interlamellar spacing (reduction of distance between cementite lamella) and its hardness is due to the rail’s manufacturing process and addition of alloy elements, aiming to produce refined pearlite. In an effective approach to the rail’s manufacturing process, refinement of interlamellar spacing and high surface hardness happen because of the rail head hardening thermal processing. This thermal processing is achieved by accelerated cooling of the rail head (normalization), which can be done with forced air spray cooling, water spray, oil spray or immersion in liquid bath with polymers. Cooling can be “online”, while the steel is still austenitic immediately after hot rolling, or via "offline" processes by reheating laminated rails [11]. After wheel forging or casting steps, the wheel standard specifies that these must be heat treated by means of accelerated cooling with water spray cooling (normalization) on the wheel’s rolling surface, and then receives a second heat treatment (tempering) for stress relief [12,13]. This heat treatment provides reduction in interlamellar spacing of the pearlite and increase in surface hardness. It is interesting to note that the wheel standard has the maximum and minimum acceptable hardness, but does not specify the homogeneity of this value on its surface. The same wheel can have hardness values with variation above 50 Brinell points and still be accepted for use. However this may cause surface defects and/or early wear [13].

### Table 3. Surface hardness [9,10].

| Steel class (Material) | Hardness [HB] |
|-----------------------|---------------|
|                       | Minimum       | Maximum |
| Premium rail          | 370           | 410*     |
| Intermediate rail     | 321           | 360**    |
| Class C wheel         | 321           | 363      |

**Notes:**
* See Note 3 in Table 4-2-1-4-2b inside AREMA: there is no maximum hardness specified for premium rails in the AREMA standard, but a note: "If 410 HB is exceeded, the microstructure through the head shall be examined at 100X or higher for confirmation of a fully pearlitic microstructure in the head".
** There is no maximum hardness specified for intermediate rails in the AREMA standard. But for the author’s expertise with these rail types, it does not go beyond 360 HB.

### 2.2 EXPERIMENTAL METHODS

Microstructures of materials (pin and discs) and wear surfaces after testing were analyzed in a scanning electron microscope (SEM) Jeol JSM - 6010LA. For analyzes, a 20 kV voltage and magnification between 20x and 10,000x were used.

Pin and disc hardness measurements were made along the longitudinal axis, starting at the contact surface; that is, the pin tip (wheel surface) that will contact the disc top face (rail head surface). A hardness test was also performed on the pin tip (cross section) and on the upper face of discs.

For identification of oxides present on wear surfaces, a Raman spectrometer by Horiba was used. All Raman spectra were obtained by using a 1,000X magnifying lens and laser beams with 532 nm wavelength. For identification of oxides, resulting spectra were compared to those from other studies [14,15].

The methodology for extracting pin and discs, respectively from the wheel and the rails, is similar to the methodology adopted in other studies [7,16]. However, in the present study, the pin was extracted from the wheel and the disc from the rail head.

In order to ensure disc parallelism, as well as to maintain an initial roughness pattern, the discs were ground. The initial roughness of discs was: $S_a = 0.24 \, \mu m \pm 0.05 \, \mu m$ and $S_q = 0.36 \, \mu m \pm 0.06 \, \mu m$, where $S_a$ is the arithmetic mean of the surface roughness or average roughness, and $S_q$ is the quadratic mean roughness (standard deviation of the height distribution curve).

Sliding wear tests were performed at LFS (Laboratory of Surface Phenomena)/USP Polytechnic School – University of São Paulo) using a pin-on-disc tribometer in a Plint TE67 machine (Fig. 5). The test consists of constant contact of a pin, due to the application of a normal load against a horizontal disc with constant counterclockwise rotation. Pins were 3 mm in diameter, 15 mm long and had flat face. Discs were 60 mm in diameter and 5 mm thick. To ensure parallelism between body contact surfaces, before all tests, pins were submitted to a sanding process, passing through sandpaper...
with granulometry of # 220, # 400 and # 600, for approximately 30 seconds on each sandpaper. The disc, with constant rotation of 70 rpm, received the pin contact at a distance of 25 mm from the disc center, resulting in tangential speed of 0.18 m/s. Dry and lubricated tests were carried out for 3,600 seconds. Normal load was varied by 40, 80 and 120 N. For each condition, at least three repetitions were made with different pins and discs. The frictional force (tangential force) was monitored during the test using a load cell. The coefficient of friction was determined by the ratio between frictional force and normal load. Mass loss was obtained by the difference in mass before and after testing, using a digital scale with accuracy of 0.0001 g. Before weighing, the samples were cleaned in an ultrasound bath with ethyl alcohol for 10 min, and then remained for 10 min in an oven to ensure drying.

For lubricated tests, a rail lubricant commonly used in railways was applied. Before starting the test, 1 g of lubricant was applied on the disc. As can be seen on Fig. 5b, a device to direct grease to the wear track at each disc rotation was used, in order to keep lubricant on the contact surface between disc and pin.

It is observed that normal loads (40, 80 and 120 N) adopted in the present study are not usual in tests for assessing wheel flange wear versus rail head corner. Literature shows normal loads such as 24.6 N [7]; 10, 30 and 50 N [17]; 220 N [18]; 420 to 1980 N [1]; and 0 to 2,000 N [19].

The average coefficient of friction (COF) is the mean value of three tests performed for each condition. For each test, an average coefficient of friction was obtained.

3. RESULTS AND DISCUSSION
3.1 MICROSTRUCTURE

The resulting microstructure for discs (premium and intermediate rails) is completely pearlitic (Fig. 6). Qualitatively, premium discs premium showed a more refined microstructure than intermediate discs, because the former showed smaller colony sizes and smaller interlamellar spacing. Colony refining and interlamellar spacing result from a more severe normalization thermal processing on the rail head. A smaller rail head interlamellar spacing in comparison to another is probably related to differences in the thermal cycle, since compositions are very similar (Table 2). It is known that higher cooling rates result in smaller interlamellar spacing [20]. The same effect was verified for rail [21] and wheel [13] steels.
3.2 HARDNESS

Figure 8 shows hardness values on pin and disc testing surfaces (Fig. 8a), and also longitudinal hardness values (Fig. 8b). They are in agreement to those specified by the respective standards for these components, with hardness of 321 HB being at the lower limit of the specification for hardness on the wheel's rolling surface by the AAR standard, a value 10 % distant from the value of 350 HB considered usual/expected for wheels of this class.

At 4 mm, pearlitic microstructure starts. Wheel standards do not specify wheel microstructures, but its hardness. However, in contrast to what is expected from field practice, it is assumed that microstructure should be pearlitic, as well as the rail microstructure. This unexpected change in microstructure along the wheel's life has not received attention in technical literature, and may constitute a technological opportunity for improvements, since bainitic and pearlitic microstructures have different tribologic behaviors [17] even while having the same hardness.

On the pin area to be tested (wheel surface), a bainitic microstructure was found, as shown on Fig. 7. Microstructural analysis of the wheel where the pin was extracted from showed that in the first 4 mm of its first life microstructure is completely bainitic, i.e., 25 % of the wheel's first life showed bainitic microstructure. Note that:

- Wheel's 1st life = 16 mm.
- New flange = 35 mm.
- New flange may be worn, depending on the railway, up to 19 mm.
- This difference from 35 to 19 mm is called the wheel's 1st life (16 mm).
- When it reaches 19 mm, flange thickness must be reprofiled, starting the 2nd life and later coming to the 3rd and last life of the wheel.

This lower hardness value was probably due to tempering of the bainite formed on the wheel surface during thermal processing that follows cooling, as well as evidenced in a sample of
forged wheel that received thermal treatment with austenitizing temperature of 890 °C and tempering temperature of 590 °C [12]. The same value of 321 HB was measured on the wheel surface, a value 10 % distant from the value of 350 HB considered usual/expected for wheels of this class. From microstructural and hardness characterizations, the expected results for the wheel (pearlite/350 HB) were not obtained. On the contrary, the material found had another microstructure and lower hardness (bainite/321 HB) for the first life of the wheel.

Figure 8b shows hardness values on the longitudinal section of pin and discs. From the pin surface up to 4 mm, hardness ranges around 320 HB, reaching approximately 350 HB.

Table 4. Ratio between hardnesses of materials studied in function of the wheel's first and second lives.

| Parameter           | Steel                          | 1st wheel's life (and pin on disc test) | 2nd wheel's life |
|---------------------|--------------------------------|----------------------------------------|------------------|
| Hardness            | Pin wheel class C              | 321                                     | 350              |
|                     | Disc premium rail              | 392                                     | 384              |
|                     | Disc intermediate rail         | 347                                     | 351              |
| Hardness ratio      | Disc premium rail/pin wheel C  | 1.2                                     | 1.1              |
|                     | Disc intermediate rail/pin wheel C | 1.1                                 | 1.0              |

The first 4 mm of pin (from the wheel surface or pin tip) correspond to 25 % of what is known as the wheel's first life. At the end of a period corresponding to an approximately 16-mm wear of the wheel tread, the common practice at wagon repair shops requires that the wheel shall be removed from service and re-profiled. That is, in the course of its first life, the wheel comes into contact with the rail with hardness around 321 HB, and with bainitic microstructure. After re-profiling, the same wheel comes into contact with the rail with hardness of 350 HB and pearlitic microstructure. During wear testing in this present study, the whole test took place in the region of bainitic microstructure and with hardness of 321 HB. Therefore, hardness ratio in the trial on intermediate rail was 1.1 and in the trial on premium rail disc it was 1.2. However, in the case of an intermediate rail, the first life would run with hardness ratio of 1.1 and in the second life with hardness ratio of 1.0. These values would be 1.2 and 1.1, respectively for a wheel working against a premium rail, as can be seen on Table 4. That is, for sliding wear tests, premium rail hardness is 20 % higher than that observed on class C cast wheel pin \((R = 1.2)\). Intermediate rail disc hardness is 10 % higher than that observed on class C cast wheel pin \((R = 1.1)\).

3.3 FRICTION

Figure 9 shows the average coefficient of friction (COF) considering a test period between 1,100 and 3,600 seconds. This interval was adopted in order to ensure that all values would be achieved in a permanent regimen, thus excluding values at the running-in initial period.

![Coefficient of friction (COF) as function of the normal load for all conditions.](image-url)

Fig. 9. Coefficient of friction (COF) as function of the normal load for all conditions.
In dry tests using premium discs, a COF reduction from 0.6 to 0.4 is observed with an increase in normal load from 40 to 120 N. For dry conditions with intermediate discs, there is no significant variation in COF with increasing normal load. For conditions with higher values of normal load (80 and 120 N), lower values of COF were found with premium discs.

During contact of moving surfaces, part of the energy dissipated by friction is transformed into heat, causing a significant increase in temperature of the surfaces in contact [22]. Reductions in coefficient of friction with an increase of the normal load, as shown for the conditions with premium disc, may be associated to formation of an oxide film due to the increase of surface temperature and consequently greater reactivity of the same with atmosphere. It is known that the presence of oxide films on surfaces in contact can result in a reduction of the coefficient of friction, as it will inhibit metallic contact between the surface roughnesses, thus reducing formation of adhesion joints [23-26]. On the other hand, the surface must be sufficiently hard to support this oxide layer. Higher hardness surfaces, as they present less plastic deformations, are able to promote greater support to the oxide layer, thus preventing it from being broken and removed during sliding [24-26]. This justifies the lower coefficient of friction at higher normal loads for the higher hardness disc (premium). However, in this study, no temperature measurements were made on pin or disc during the tests, and it is understood that these measurements must be carried out in future studies in order to support justifications.

For lubricated tests, the coefficient of friction was relatively stable, showing no statistically significant variations, but with a minor trend to increase with normal load increment from 80 to 120 N. Values found for COF were between 0.05 and 0.10, and were significantly lower compared to those in dry tests (without lubrication). COF results found in this study are consistent with those found in literature for similar conditions [27,28].

One of the main goals of lubrication in railways is to reduce friction between wheels and rails, aiming to increase locomotive energetic efficiency (reducing diesel consumption) and working life of rails and wheels (reducing sliding wear). Under experimental conditions adopted in the present study, the used grease provided COF reduction from 0.6 to 0.05, thus showing its efficiency for reducing friction.

3.4 WEAR

Results from dry testing with premium (Fig. 10) and intermediate (Fig. 11) discs demonstrate mass loss behavior in function of the normal load.
In dry tests, with normal load of 40 N, wheel pins had similar mass loss (wear) when slid against discs of premium and intermediate rails. In the case of the intermediate disc, pin mass loss was equal to that of the disc; and in the case of the premium disc, pin mass loss was greater than that of the disc. The results found for mass loss of premium disc steel versus class C pin are consistent with literature [4] for the normal strength of 40 N. That is, for hardness ratio equal to 1.2 (premium rail disc versus class C wheel pin), mass loss of the wheel is greater than that of the rail. However, the results found for mass loss of intermediate disc steel versus class C pin are not in accordance with literature [4], as they show that for hardness ratio equal to 1.1, mass loss of both bodies are the same.

With an increase of normal load from 40 to 80 N, mass loss of class C wheel pins increased, both for the test on the premium disc and the dry test on the intermediate disc. Pin mass loss was statistically equal to mass loss of premium and intermediate discs. Pin mass loss had similar results, when tested on intermediate and premium discs. Mass loss of the intermediate disc was similar to that of the premium disc.

With an increase of normal load from 80 to 120 N, mass loss class C wheel pins increased for the tests.
on premium and intermediate discs. Pin mass loss was statistically equal to mass loss of the premium disc and greater than the intermediate disc. Pin mass loss was similar when tested on premium and intermediate discs. Mass loss of the intermediate disc was similar to that of the premium disc. Mass loss results for intermediate disc steel versus class C pin are consistent with literature [4] for normal load of 120 N. That is, for hardness ratio equal to 1.1 (intermediate rail disc versus class C wheel pin), mass loss of the wheel is greater than that of the rail. However, the results found for mass loss of premium disc steel versus class C pin are not in accordance with literature [4], as they show that for hardness ratio equal to 1.2, mass loss of both bodies are the same.

Upon analysis of mass loss results for the pin-disc system (Fig. 12), that is, the sum of pair's mass loss, it is observed that these results are similar for both conditions, with premium and intermediate discs. This evidence shows that variation of hardness ratio studied in the present work did not have significant influence on the wear of the pin/disc (wheel/rail) set, as shown individually for each component.

For the lubricated tests with premium disc (Fig. 13) and intermediate disc (Fig. 14), results showed that adding a lubricant lead to mass loss statistically equal for pin and both discs, regardless the contact being permanent for the disc or intermittent for the pin. Using the rail lubricant, the pin's mass loss was reduced in more than two orders of magnitude when compared to dry tests. For the discs, mass loss reduction was also around two orders of magnitude when compared to results from dry tests.

![Fig. 13. Mass loss results as a function of normal load applied to premium disc and class C pin under lubricated condition.](image1)

![Fig. 14. Mass loss results as a function of normal load applied to intermediate disc and a class C pin under lubricated condition.](image2)
As previously mentioned, lubrication on the wheel-rail contact aims to increase energy efficiency of locomotives and to increase service life of rails and wheels. In the case of this study, results showed that system lubrication was able to significantly reduce component wearing (pin and disc).

Figure 15 shows microstructures of class C cast pin wear subsurface for the highest normal load (120N): (a) dry/versus premium disc and (b) lubricated/versus premium disc. Note: the pin’s worn surfaces are at the bottom of the images, as indicated by the arrows.

After testing (both conditions at 120 N), microstructures of disc and pin subsurfaces were analyzed in SEM. The microstructure of pin wear subsurface after dry tests is shown in Fig. 15. In all samples, it is possible to notice a layer immediately above the pin wear surface, which presents high plastic deformation (hardening). Therefore, hardening existing in pins for the dry condition is more severe than that existing in lubricated pins. Right after this hardened layer, a bainitic microstructure is observed on all pins. This result shows that throughout the wear test, the microstructure tested was bainitic, i.e., tests occurred in the wheel’s first life. The hardened layer was always thicker in dry tests than in lubricated tests, as expected due to higher coefficient of friction and, therefore, greater tangential force acting in these tests.

Figure 16 shows disc microstructures. In all samples, as it happened with pins, it is possible to notice a region immediately under the wear surface showing intense plastic deformation (Fig. 7).

Figure 16. Microstructure of the worn subsurface of the disc for the highest normal load (120N): (a) dry/intermediate disc and (b) lubricated/intermediate disc. Note: the disc’s worn surfaces are at the bottom of the images, as indicated by the arrows.

After testing, pin and disc wear surfaces were analyzed in SEM and Raman spectrometer. Figure 17 shows disc wear surfaces for conditions with premium rail and normal load of 40 N and 120 N. In the present study, it was decided to present only images of the disc wear surface for conditions with premium rail, since the observed phenomena were the same for conditions with intermediate rail, both for pins and discs. In all
samples (pins and discs), mechanisms of adhesion, abrasion and oxidation were observed. Figures 17a and 17c show pin wear surfaces after testing with 40 N and 120 N, respectively, observed in secondary electrons. On both surfaces, it is possible to notice adhesive joints, as well as regions with material removal. These regions with material removal visibly increase with an increase of test normal load, corroborating mass loss increase as a function of normal load (Figs. 17 and 18). Abrasion scratches in the sliding direction (horizontal) are possibly caused by wear particles, whether metallic or oxidized, which have higher hardness than the surface. The oxidative mechanism is confirmed by images obtained with backscattered electrons shown on Figs. 17b and 17d for normal load of 40 N and 120 N, respectively. The dark regions of the image are oxides and the light ones are metallic regions. These mechanisms are typical of sliding wear and have already been reported by other authors who also used pin-on-disc tests [17,7,29,16].

Fig. 17. Wear surface of discs, under premium rail conditions, after testing with normal loads at: (a) and (b) 40 N under secondary electrons and backscattered electrons, respectively; (c) and (d) 120 N under secondary electrons and backscattered electrons, respectively.
Figure 18 shows the results obtained by Raman spectroscopy also for the discs under conditions with premium rail and normal loads of 40 N and 120 N. Figures 18a and 2c show the analyzed regions, characterized by the presence of oxides. Figure 18b and Figure 18d show spectra characteristic of these regions. For all conditions tested, Hematite iron oxides were found on the wear surfaces. Previous results for reduced coefficient of friction indicated an increase in surface temperature that possibly resulted in increase of oxide formation rate. However, the results of Raman spectrometry showed that the type of oxide formed was not altered by this temperature increase.

4. CONCLUSIONS

Under all dry test conditions (40, 80 and 120N), the system’s mass loss was equal for both hardness ratios used (R = H_Rail / H_Wheel = 1.1 or R = 1.2). That is to say, there was no impact of the correlation between pair hardesses on the tribo-system here studied, contrary to what is stated in literature.

However, for the present study, body mass loss singly show that:

- Under dry test conditions with normal load of 40 N, the material of premium rail discs wears less than the material of class C wheel pins (R = 1.2).
- For normal load of 80 N, mass loss of the three bodies are statistically the same (R = 1.1 and R = 1.2).
- For normal load of 120 N, the material of intermediate rail discs wears less than the material of class C wheel pins (R = 1.1).

Results of mass loss in the lubricated test, for all normal loads studied and hardness ratios R = 1.1 and R = 1.2, showed that mass loss was the same for pin and disc. Disc and pin steels showed similar mass loss and the influence of the pair hardness ratio was not evidenced.

Moreover, as contributions to technological actions in railway industry, the following could be listed:

i. Application of lubricants between the wheel flange and the corner/lateral of the rail head forms an interface film that may reduce the difference in wear between the wheel-rail pair, regardless of their hardness. As shown in this study, under dry
conditions, wheel materials wear more than those of the rails. This shows the need for having a lubrication process embedded in locomotives (wheel flange lubrication).

ii. For more severe wear conditions (small radius curves), lubrication is even more important because the interface film will cause component wear to equalize, and be significantly reduced when compared to dry conditions (regardless of normal load or hardness ratio). For this, in addition to a lubrication process embedded in locomotives, it is also important to adopt lubrication on high rails in sharp curves using a mobile process (road-rail vehicle) or way side process (fixed lubricator positioned at the edge of the track).

iii. The coefficient of friction (COF) in lubricated laboratory tests was 0.05 in average, while for field tests this value is 0.20 in average [29]. The discrepancy between coefficients of friction measured on field and in the laboratory (0.05 and 0.20, respectively), suggests that there is room for improvements in field lubrication techniques, seeking greater reductions for the coefficient of friction in practical terms; or measurement techniques, reducing measurement differences on field and in the laboratory.

iv. The standards and/or technical specifications of wheels should be revised, in order to make the microstructure completely pearlitic and hardness on the surface between 350 to 363 HB. This aims to avoid tempered bainitic microstructures with very low surface hardness as found in this study (320 HB), which would lead to premature wear of the wheel and reduction of its working life.

v. Aiming reduction of the system's wear, it is recommended to adopt the ratio between rail/wheel hardnesses R ≥ 1.0, proposed in the field study by Steele and Reiff (1982), and/or 1.0 ≤ R ≤ 1.1 proposed by the mathematical model set by Razhkovskiy et al. (2015).

Acknowledgement

The authors would like to express their gratitude to all staff at USP (University of São Paulo), ITV (Vale Technological Institute) and VLI (Valor da Logística Integrada) who supported this study.

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