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Towards a Standard Approach for Wear Testing of Wheel and Rail Materials

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ABSTRACT

Examination of the literature for wear testing methodologies for wheel and rail material reveals that while only a few different techniques have been used there is a wide variety in exactly how the tests have been conducted and the resulting data reported. This makes comparison of the data very difficult. This work, carried out as part of the International Collaborative Research Initiative (ICRI) which is aiming to bring together wheel/rail interface researchers from across the world to collate data and knowledge to try to solve some of the common problems that are faced, has examined the different approaches used and attempted to pull together all the good practice used into a test specification for future twin disc testing for wheel and rail materials. Adoption of the method will allow data to be compared reliably and eventually enable data to be compiled into wear maps to use as input, for example, to multi-body dynamics simulation wear prediction tools.

1 INTRODUCTION

The wheel/rail contact is a crucial aspect of the successful operation of railways. It is however, an extremely complex interface. A large variety of loading conditions and contact geometries exist due to the many different rail and wheel profiles, rail cant and curve radii, and railway vehicles running on a network. Contact mainly occurs at wheel tread/rail head in tangent track with wheel flange/rail gauge corner contact being seen in curves. The latter is usually more severe which leads to greater damage being seen. Friction forces and relative motion (creepage) in the contact are also highly variable. Natural lubricants such as humidity, precipitation, and leaves can negatively influence the friction in the wheel/rail contact, causing braking problems and wheel slip in traction. These problems can be overcome by applying man-made lubricants to reduce wear in curves and friction modifiers to increase adhesion. This, however, further adds to the wheel/rail interface complexity.

Effective management of the wheel/rail contact is an important aspect of rail infrastructure operations. In 2010 the cost of rail maintenance in Western Europe (excluding Spain) and Scandinavia was estimated to be 7.8 billion euro [1]. All the influencing factors have to be taken into account as they interact closely (as indicated in Figure 1) (adapted from [2]). For example, measures used to reduce wear, such as lubrication, can influence fatigue and adhesion [3], and the measures used to increase adhesion, such as sanding, can have a detrimental effect on wear [4]. A fine balance has to be found in determining maintenance schedules and lubrication regimes to keep railway networks running smoothly. This is becoming increasingly difficult as new specifications on wear and reliability are being imposed to increase the time between re-profiling, increase safety and reduce total life-cycle costs. In parallel with these requirements, vehicle missions are changing. More trains are running so track is seeing increased tonnage, and trains are running faster. These are leading to an increase in the severity of the wheel /rail contact conditions [5, 6].

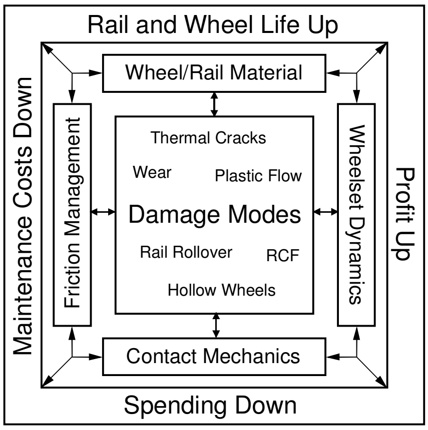


Figure 1. Systems Approach to Wheel/Rail Interface Management and Research (adapted from [2])

One very important aspect of the wheel/rail interface management is wear control. Before a strategy can be implemented, whether it is the introduction of a more durable wheel or rail material or a new lubrication approach, testing must be carried out to prove the performance benefit. There are a number of ways this can be achieved, as will be seen in the next section. However, at the moment there is no standard way to carry out a wear test or present wear data for the wheel/rail interface. This causes a problem in that it is impossible to compare output from different investigations or from different test rigs. With a more standardised approach data could be drawn together, for example, to form wear maps or be used more reliably as input to predictive models.

The aim of this paper was to critically review the different types of wear test and analysis methods available, discuss the advantages and disadvantages of each and then to recommend a standard approach to wear testing and provide guidance on good practice.

This is being carried out as part of the International Collaborative Research Initiative (ICRI), the aim of which is to bring together researchers from across the globe to work on common research problems.

2 CURRENT WEAR TEST METHODOLOGIES

2.1 Testing Across the Scales

A number of different techniques have been used for studying wear of railway wheel and rail steels. Field measurements have been used in the past to study the causes of wheel and rail wear [7-9]. A large amount of data has also been gathered from simulated field experiments carried out on specially built test tracks [10]. Laboratory methods used range from full-scale laboratory experiments [6, 11, 12, 13, 14] (see Figure 2a) and scaled-down tests [15] to bench tests using a twin-disc set-up [16-21] (see Figure 2b) and pin-on-disc tribometers [22-25] (see Figure 2c).

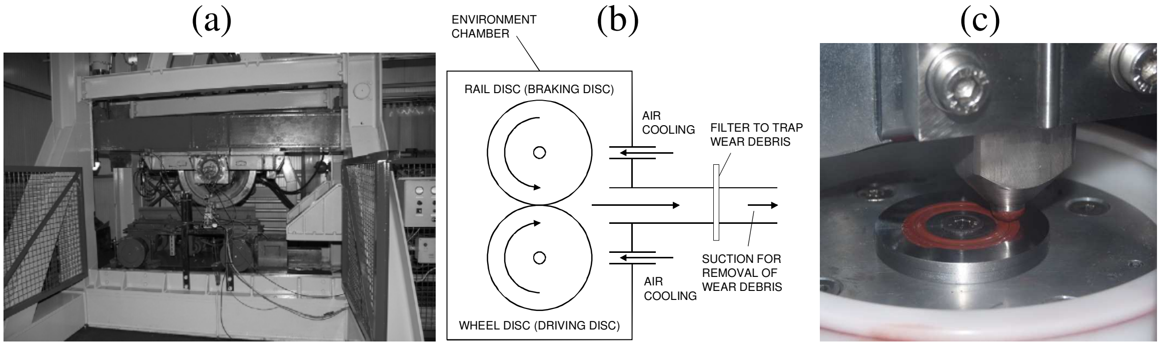


Figure 2. Laboratory Based Test Methods: (a) Full-Scale [13]; (b) Twin Disc Machine [21]; Pin-on-Disc [25]

Laboratory tests give high levels of control over test conditions and allow accurate measurement of friction and wear rate. Specimens are relatively easy to manufacture and the tests are cheap to run relative to a field test. It is straightforward, using this kind of approach, to generate reliable wear coefficients that can be integrated into wear models for predicting rail or wheel profile evolution. However, the contact geometry is simplified and representative environmental conditions are hard to achieve. This can lead to range of issues that could affect the applicability of the data generated. These issues will be discussed in detail in later sections. At the other end of the test spectrum, field testing provides real contact and environmental conditions. However, there is no control over the conditions and these will vary considerably. For a fixed point on a rail, for example, every wheel passage will be at a slightly different point and contact stress and slip will vary. This means that the data generated is hard to use. Full-scale laboratory tests provide a good compromise. They give real, but controllable contact conditions and allow useful wear data to be generated. However, there are few of these rigs and they are costly to run and there are some critical differences from field operation that must be used to put the results into context. The main issues, summarised by Stock et al. [26], were the lower rolling speed, the fact the same wheel was used and the contact was always at the same point on the rail, different longitudinal and lateral forces and different environmental conditions from those seen in the field. However, the new high-speed wheel-rail rig “Rolling Load Test Machine” at TTCI has been designed to run at more representative speeds; with random positioning of the wheel on the rail and multiple actuators it addresses many of these concerns [27].

2.4 Scaling-up Wear Data

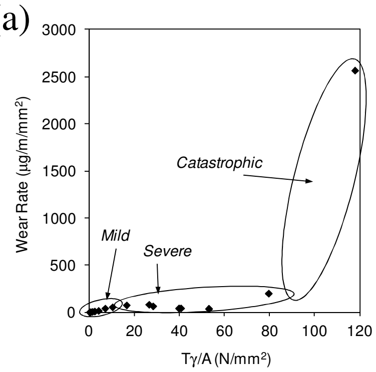
Two approaches are typically used for modelling wear processes, both of which are semi-empirical. The first utilises the Archard adhesive wear equation [28]:

$$V = k\frac{Pl}{h} \tag{1}$$

where, V is wear volume, P is load, l is sliding distance, and h is hardness. The second relates material loss to energy in the contact using the $T\gamma$ wear value, where T is tractive force (normal force × traction coefficient) and γ is creep in the contact (relative velocity of wheel and rail divided by velocity of vehicle) [21]. Both rely on establishing wear coefficients using the laboratory scale tests described previously (e.g., pin-on-disc or twin-disc).

An example of a $T\gamma$ wear curve derived from twin-disc testing of R8T wheel material against 900A rail material is shown in Figure 3a. $T\gamma$ is here divided by the contact area as an attempt to normalise data, to allow scaling of measurements from laboratory specimen contacts to the full-scale contacts. Three wear regimes are apparent, for each of which a wear coefficient can be defined. The wheel tread/rail head contact would be expected to fall in the mild regime, while the wheel flange/rail gauge corner would be in the severe and in some circumstances (for very tight curves) the catastrophic regime). Figure 3b shows an Archard wear coefficient (dimensionless) map derived from pin-on-disc and twin-disc tests [29], where regimes of wear are defined in terms of contact pressure and sliding velocity. The wear coefficients can be integrated with Multi-Body Dynamics (MBD) simulations for prediction of wheel or rail profile evolution [12, 30, 31, 32]. The typical process is illustrated in Figure 4. Global contact parameters from the MBD simulations are input to local contact analysis that defines $T\gamma$ or contact pressure and sliding velocity distribution within the contact from which material loss can be calculated using the wear coefficients.

(a)



(b)

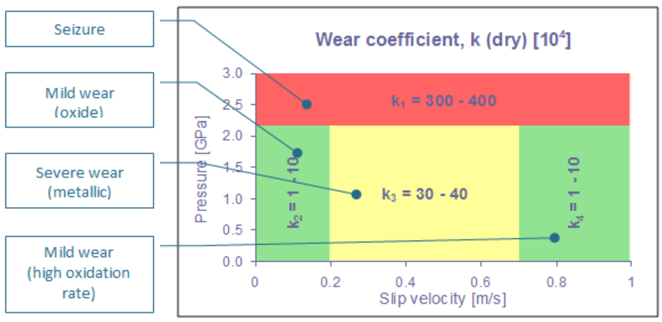


Figure 3. Wear Maps for (a) R8T Wear Rates (against 900A Rail) from Previous Twin-Disc Testing [21]; (b) 900A Rail (against R7 Wheel) [29] (dimensionless wear coefficient)

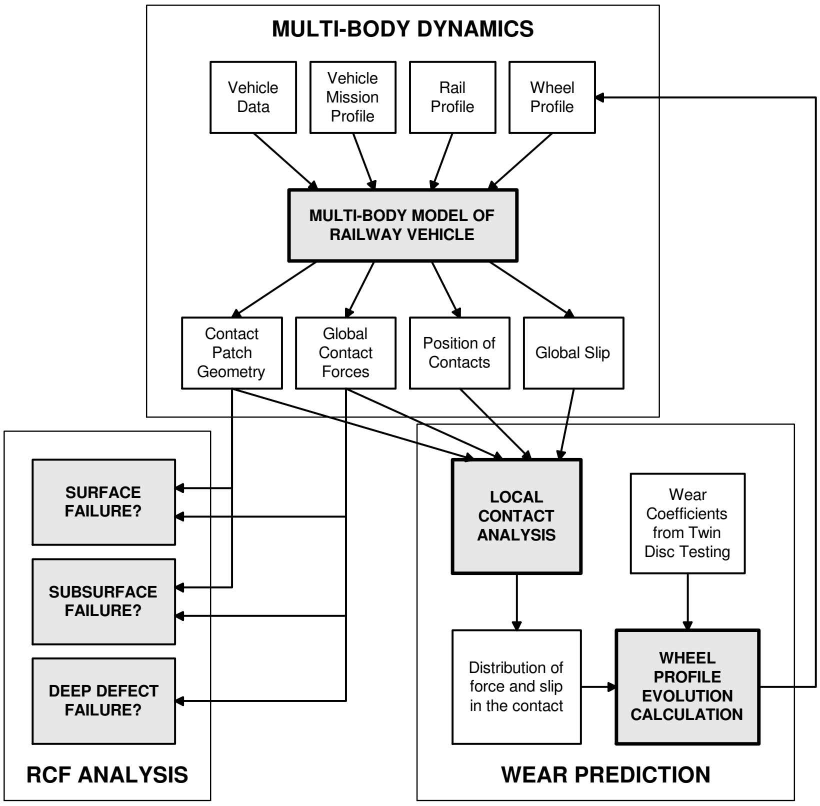


Figure 4. Typical Process in MBD based Damage Prediction [12]

Two approaches have been used for establishing the validity of applying wear coefficients from small-scale tests to predict wear in full-scale situations. The first is to validate models that utilise wear coefficients with data from full-scale tests. This was done for the wear coefficients defined from the wheel material wear data presented in Figure 3a. These were integrated into a MBD simulation of the Lucchini full-scale test rig and then used to model a test run on the rig [12]. The model output was compared with profile measurements taken on the wheels used in the test. Very good correlation was achieved.

The second approach is to compare wear from small-scale and full-scale tests using $T\gamma$ values. In principle the mass loss should be the same for the same $T\gamma$ value. This has been carried out for twin-disc data and full-scale data from the Voestalpine rig (details reported in Buckley-Johnstone et al. [33] for 260 grade rail run against R8T wheel material in dry conditions and with a top of rail friction modifier applied). The comparison is shown in Figure 5. The wide spread on the full-scale $T\gamma/A$ values comes about because friction and slip are not measured and are taken from separate measurements and a numerical simulation of the contact, respectively (see [34] for details). More full-scale data is needed, with reliable slip and friction data, to give full confidence in this approach, but it looks as if it will be a viable means to compare data from different scale tests. Further evidence of this is presented in Lewis & Olofsson [35]. Here the full-scale test results from Olofsson & Telliskivi [8] were compared with pin-on-disc and twin-disc wear data in the form of a wear coefficient.

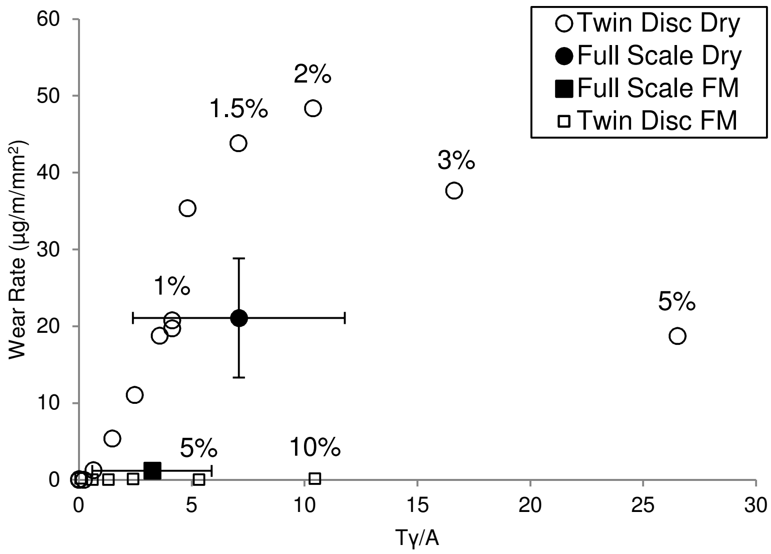


Figure 5. *Tγ/A* Wear Rate Data Twin-Disc / Full-Scale Comparison for Dry and Applied Friction Modifier Conditions [33] (% refers to slip value used for test)

2.3 Issues with Small Scale Tests

While small-scale tests are a good option in terms of the good levels of control and hence repeatability, providing scalable wear coefficients relatively quickly and affordably, there are a number of issues that can reduce confidence in their output. These are discussed in detail below and mitigation methods outlined.

Temperature Effects

The small contact size and constant reloading of the same material as the specimens rotate that occurs in both pin-on-disc and twin-disc testing can lead to high temperatures building up, particularly due to unrealistic heat transfer out of the contact. This can have two effects: the material could reach a point where it starts to soften, accelerating wear; or oxide generation within the contact will rise, leading to a thicker third-body layer within the contact than would be apparent in the field, as shown in Figure 6 [36]. The disc was encapsulated post-test in order to analyse the layer. In-situ measurement would be very difficult.

Typically air cooling is provided during twin-disc testing [21]. Some analysis of twin-disc contact temperature has been carried out and shows that temperatures can be kept to a level that would be apparent in an actual contact [37]. It is unclear what effect the oxide generation will have on wear, although it could act as a protective layer on the surface. It does lead to higher dry friction than would be measured in the field though [38]. In the past, brushes have been used to scrub discs to reduce the build-up [16], but this technique could not be used if another third-body material was being applied during the test.

(a) (b)

Figure 6. Third-Body Layers (a) On a the Surface of an R8T Wheel Specimen from a Twin-Disc Test Run against 900A Rail (1500 MPa and 0.3% slip) (3BL = third-body layer); (b) On an Actual Rail [36]

Specimens

Careful consideration of specimen design and manufacture is very important. Most researchers use actual wheel and rail materials. The specimen design for the SUROS twin disc machine was actually configured around the size of a rail head [39]. In the early work on this rig and for other twin disc machines, the discs were cut from sections of rail and wheel (as shown in Figure 7a). However, investigation of the micro-hardness across the disc revealed that there was significant variation in some cases. This led to discs being cut in the manner shown in Figure 7b, which has helped to resolve this. For pin-on-disc approaches it is harder to cut specimens from actual wheels and rails because often the discs required are too big. However, discs limited to the size of the rail head width can be handled and a similar cut to that presented in Figure 7b can be performed [22]. Specimens can be purpose-made from the same materials, but replicating the wheel or rail microstructure and hardness properties becomes harder.

(a) (b)

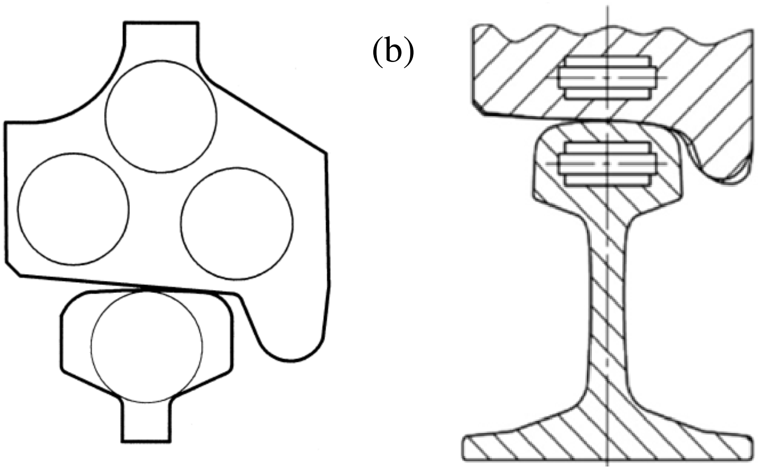


Figure 7. Specimen Cutting from Wheel and Rail (a) Original SUROS Approach and (b) Current Approach

The other important issue for specimen preparation is the final surface finish. It is actually quite hard to obtain data from the field. While measurements are relatively easy to take, they are rarely quoted in research reports or papers. For the SUROS discs the typical roughness value is about 1μm Ra which was believed to represent a worn-in rail or wheel roughness. In a wear test the roughness will evolve and reach a steady state value, but this value will vary depending on the contact pressure and slip conditions [40].

Contact Conditions and Test Length

Collaboration with the dynamic modelling community has led to good data-sets on wheel/rail contact conditions being generated. An example is shown in Figure 8, generated using GENSYS modelling of the Stockholm Local Railway network [29].

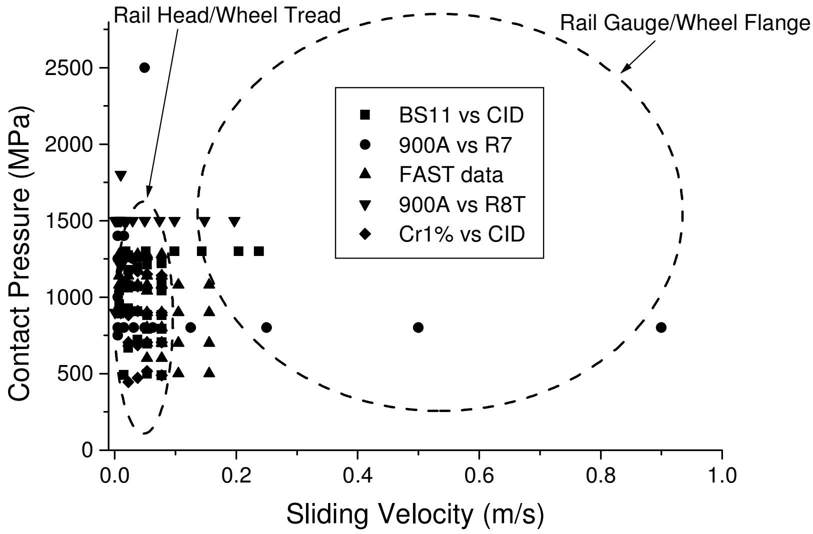


Figure 8. Wheel/Rail Contact Conditions on the Stockholm Local Railway Network derived from GENSYS Simulations with Available Wear Data Overlaid (from [35], contact information originally from GENSYS simulations of the Stockholm Local Railway Network from [29])

Comparing the results of dynamic modelling with conditions used in wear tests described in the literature [35] has helped to identify that there are gaps in the wear-test data. Most data is for the wheel tread/rail head contact for passenger vehicles. In the future, effort needs to be placed on looking at specific scenarios, for example freight, passenger, and light rail so that the wear tests needed to fill gaps can be undertaken. In assessing the performance of a material it is essential that it is tested across a range of conditions covering all possible operational scenarios. This is to ensure that data needed to feed into a wear model is acquired without any extrapolation, and that wear transitions are identified.

One problem with wear tests is that as they progress the contact pressure can change as the specimens become worn. This can, in the extreme case of a ball-on-flat test, lead to changes of several orders of magnitude as the contact develops from its initial "point" contact to a completely conformal contact as a wear groove is formed [41]. This could also be the same for a twin-disc test where one or both discs are initially crowned. Very often, due to test apparatus limitations, researchers modify the specimen geometry to achieve high contact pressures representative of those in the actual wheel/rail contact, but end up with something much lower as the test progresses. This problem can also affect tests with third-body materials, particularly lubricants, where a reduction in contact pressure can cause a shift in the lubrication regime.

Test length can also be an issue when assessing wear. Quite often it appears that researchers select the test length based on time and cost constraints rather than technical requirement. Also tests are rarely stopped to check how wear rate is varying with cycles, although this eliminates the problem of realigning the partially formed wear scar correctly into the interface after measurement. During a wear test the wear rate will vary as materials initially run-in and then potentially work harden. It would be expected for a rolling-sliding type test that the wear rate would rise from test start to reach a peak from which point it would fall before reaching a plateau value (see Figure 9). This means it is very important that a test is run until a steady state wear rate is reached. Blau [41] discussed this issue at length and has provided some very useful insights for researchers involved in designing tribological tests.

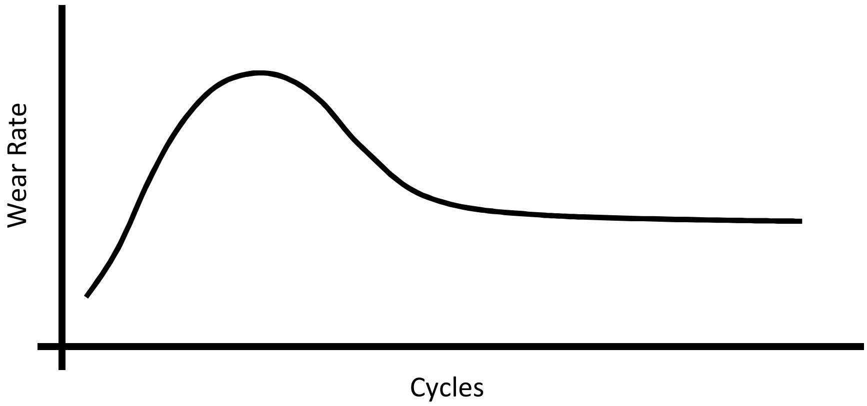


Figure 9. Typical Wear Curve for a Rolling-Sliding Wear Test

The only way to establish that a steady state condition has been reached is regularly to check the wear rate, which, while time consuming, can ensure that the correct assessment of a wear performance is made. Data are shown in Figure 10 from recent twin-disc tests on laser clad layers [42]. The materials all had different work hardening characteristics, which is evident as some have clearly reached a steady-state wear condition while others, even after 50,000 cycles, have an increasing wear rate. If shorter tests had been run a completely different ranking would have been obtained.

Statistical analysis of results is becoming a greater area of focus in tribological testing. In the past, when specimens have been expensive to produce and tests time consuming and costly, no or few repeats have been carried out. It is essential for wear tests of wheel and rail materials for repeats to be run even if this is at the expense of achieving tests over a wide range of contact parameters. It is important that we have full confidence in the wear coefficients being presented for use in wear modelling tools.

Consideration is needed here of material lost and material displaced due to plastic flow that is no longer supporting loads. This is mentioned later when wear measurement is discussed.

Environmental Conditions

Where the $T\gamma$ approach has been integrated with Multi-Body Dynamics (MBD) modelling packages the wear prediction is typically based on data acquired using twin-disc testing under dry conditions. For dry predictions these work well as evidenced by the successful modelling to establish the wheel profile evolution of a run on a full-scale wheel-set test rig run in dry conditions by Braghin et al. [12]. In reality a contact will rarely be dry and clean so it is essential that environmental conditions are replicated. This is very difficult for full-scale rigs, but has been achieved in pin-on-disc tribometer testing (it can be argued that a 100% sliding test simulates the sliding component of a partially slipping contact) where temperature and humidity have been varied to assess their effects on wear [24]. This is a relatively unique rig - in most other cases these cannot be controlled.

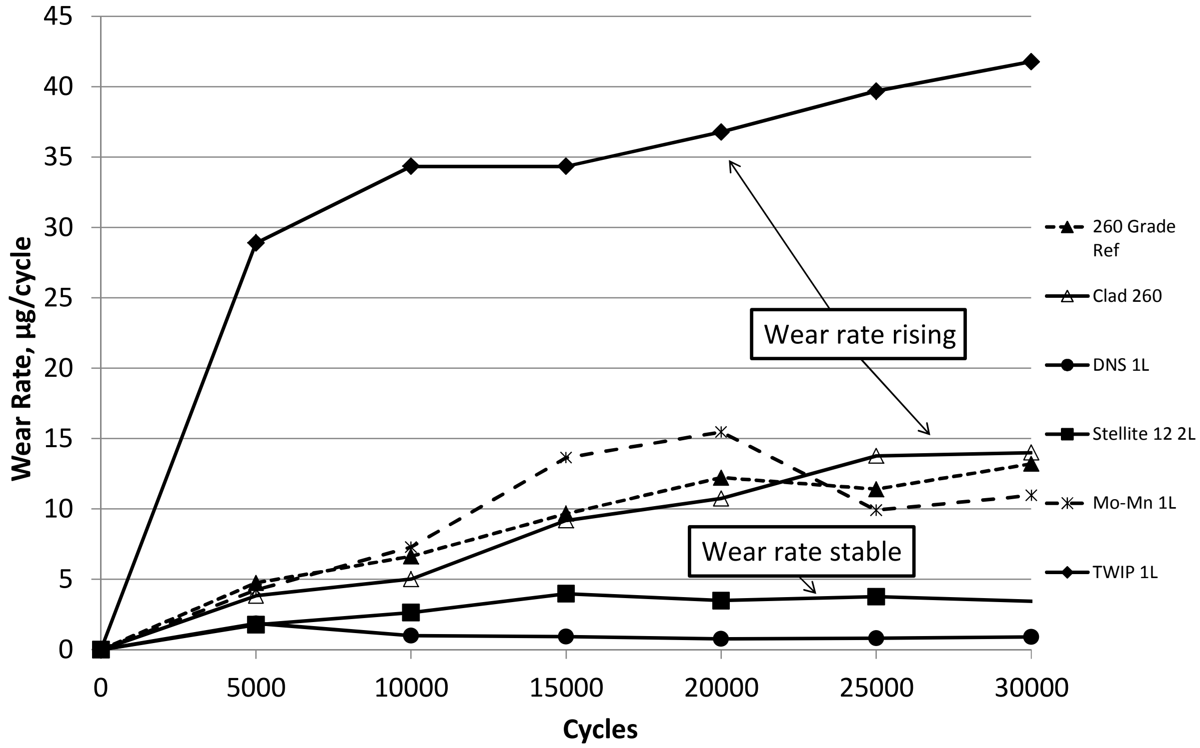


Figure 10. Wear Data from Twin-Disc Tests of different Clad Layers [42]

2.4 Third-Body Materials

Consideration of Third-Body Materials in Wear Tests and Wear Modelling

As mentioned above, a wheel/rail contact is rarely clean and dry. A number of third-body materials can be present in the wheel/rail interface, resulting from environmental conditions and those that are applied to control friction and wear. These have a large influence on wear [43] so it is important that they are considered when looking at wear tests and how they are carried out and how their effects are taken into account when modelling wear.

The conventional approach to account for a third-body material in the contact (including those resulting from environmental conditions such as water, or applied to the contact surface such as grease or sand) is to simply change the traction coefficient [31]. An example is shown in Figure 11, where dry and wet (i.e., water lubricated) wear data are illustrated from twin-disc testing using R8T wheel material run against 900A rail material (from [44]). Moving from a dry to wet condition may mean a change in traction coefficient from 0.5 to 0.2, reducing the $T\gamma$ value by 60%. Using the conventional approach, the reduced wear rate would be calculated using the dry wear data, with the change from dry to wet conditions indicated by Arrow 1 on Figure 11. However, in reality the change is from the dry wear curve to a wet wear curve (indicated by Arrow 2 in Figure 11) which gives a much lower wear rate. This is because when water is present the energy in the contact is being dissipated in a completely different way to how it is in a dry contact. The same applies for all third-body materials. Each will lead to a different wear curve. The same applies to the Archard approach. Different wear coefficients need to be defined for each third-body material.

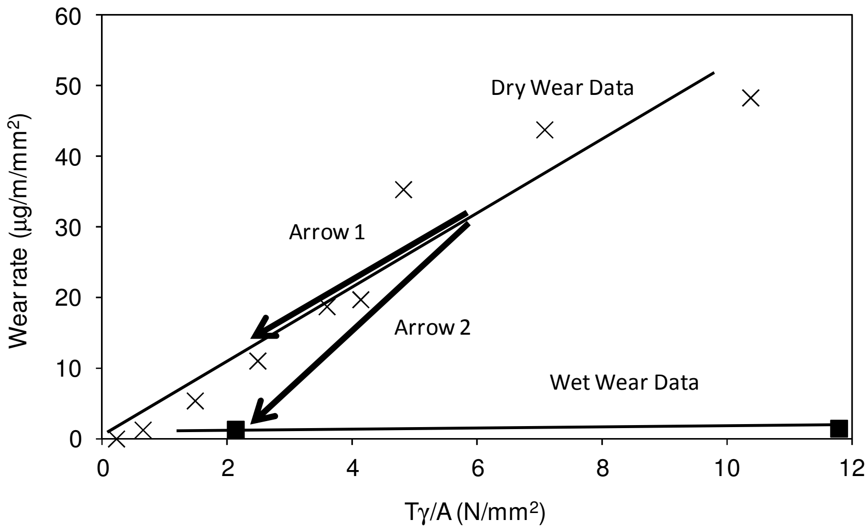


Figure 11. Dry and Wet Wear Curves from Twin-Disc Testing using R8T Wheel Material and 900A Rail (in mild wear regime according to data presented in Figure 3a) [44]

The whole issue of using traction coefficients as an input for MBD simulations raises another deficiency with current modelling practice. The traction coefficient is always a result of the contact conditions and varies continuously rather than being fixed as is assumed in MBD simulations. As relative slip in a wheel/rail contact changes, the traction coefficient changes, so a means to incorporate this (in the form of creep curves) is needed. Note that, as with wear coefficients, the creep curve associated with each third-body material will vary, as shown in Figure 12 [38].

Some simple contact modelling of a centred wheelset on a tangent track under high traction forces has been carried out using FASTSIM to assess the impact on the predicted tangential forces and wear when using fixed/varying traction coefficients and using dry wear curves with traction coefficient data to predict wear with water present [45]. This showed that if the surfaces were water contaminated, but assumed dry with a water traction coefficient, the difference in predicted forces were 144% and up to 250% for wear. Varying traction coefficients (using creep maps (slip versus traction coefficient versus contact pressure) derived from twin-disc data), rather than using a single value, also led to differences of up to 45%. This reinforces the need for developing a greater understanding of how traction coefficients and wear rates change when contact conditions are varied and third-body materials are introduced and shows that having reliable wear coefficients is not enough on its own to ensure accurate wear predictions. Until recently, however, not much testing has been carried out on different third-body materials. Some data now exists for both $T\gamma$ and Archard approaches for wet and greased contacts as shown in Figure 13 [44, 46], but much more is needed.

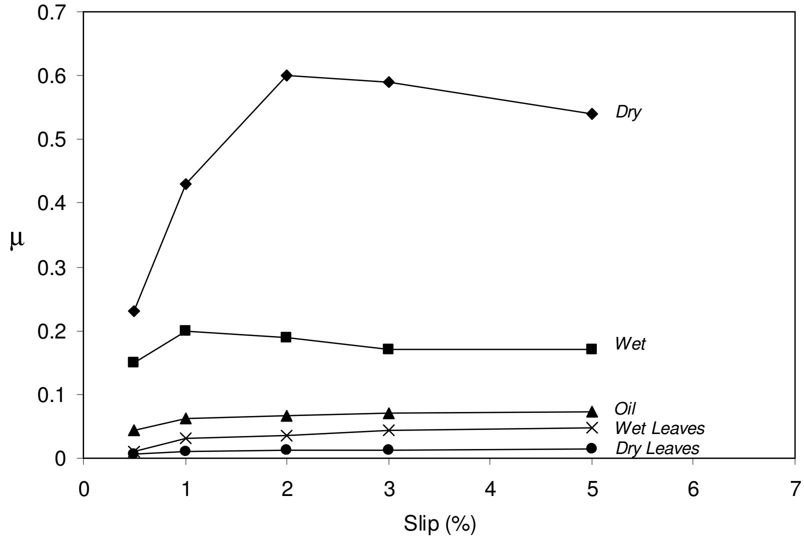


Figure 12. Creep Curves for a Range of Third-body Materials (R8T wheel material run against 900A rail at 1500MPa) [38]

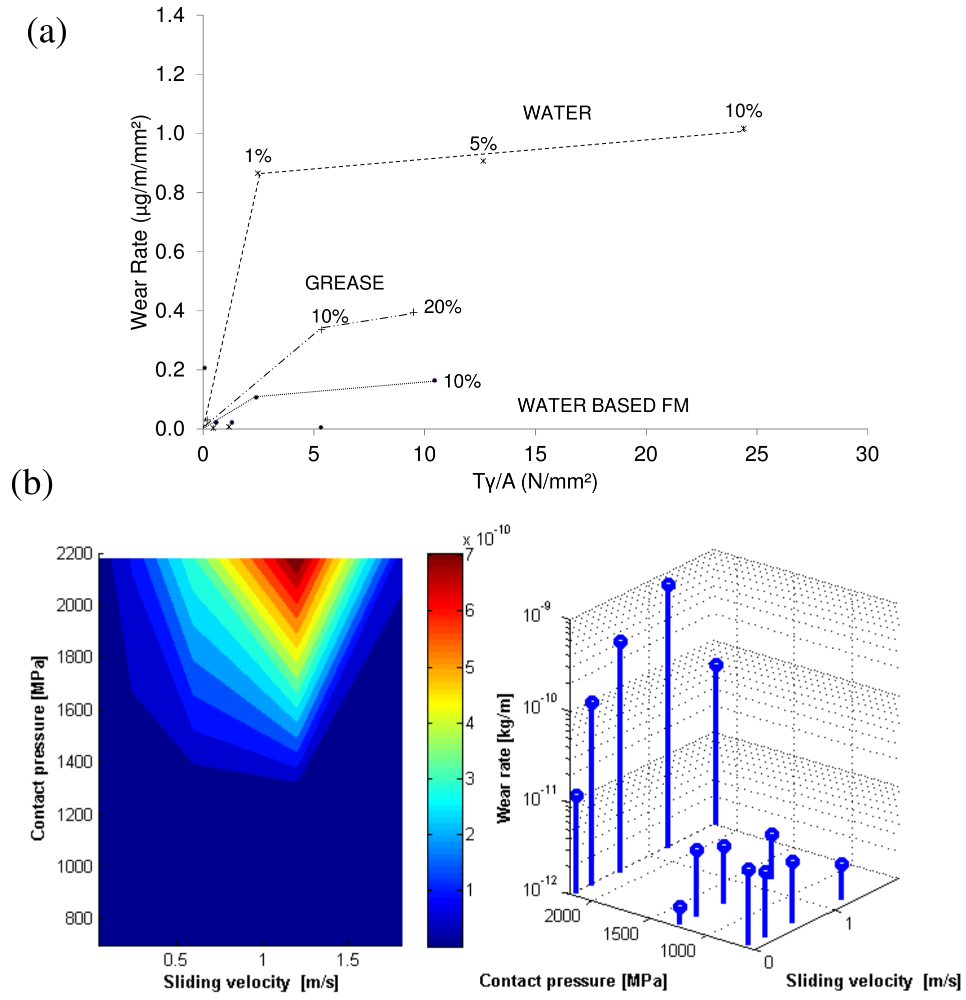


Figure 13. Wear Maps (a) *Tγ* Wear Map for Different Third-Body Materials [44] (% refers to slip value used for test); Archard Wear Coefficients for Grease [46]

Application of Third-Body Materials

It has been established (see above) that consideration of third-body materials is critical for accurate wear modelling. Furthermore, achieving representative application of the third-body materials is also very difficult.

Application of friction modifiers, lubricants and traction enhancers in field service is typically achieved via way-side applicators that rely on product pick up by the passing wheel either directly from a device located on the side of the railway track (for example a grease-dispensing unit) where it is carried down the track; or via devices that deposit product on the top of the rail head for the wheel to pass through (for example; liquid top-of-rail friction modifiers or traction enhancers). However, to try and reduce maintenance costs, many on-board applicators are now being used as this allows the product to be used in a more flexible manner. Solid stick friction modifiers or lubricants can be applied directly to the wheel tread or flange via spring-loaded devices, and sand or liquid lubricants can be sprayed directly into the wheel/rail interface.

On-board devices for application of lubricants can readily be been scaled for laboratory tests. Experimental set-ups for twin-disc testing have been attempted for: sand (see for example Arias-Cuevas et al. [47]); solid stick friction-modifier application [48]; and liquid friction modifiers [33]. Full-scale tests have been run for liquid friction modifiers [13], and sand [49]. In these studies the products have been continuously applied using scaled-down values from the field. Similarity between small-scale and full-scale wear tests with top-of-rail friction modifiers have been achieved as shown in Figure 5 (although much more data is needed to build confidence) [33].

In work to simulate way-side application, fixed amounts of various third-bodies, for example grease [3], top of rail friction modifier [25], and traction gels (including a leaf layer) [50] have been applied to one side of the contact (in either pin-on-disc or twin-disc tests). Here it is harder to know how much to apply, because although deposition rates can be established from the field, it is not known from this how much is actually transferred to the wheel and what proportion is wasted. More detailed analysis is required of field applications to determine how much product is active in the wheel/rail contact.

For some products, continuous application is not representative of actual operation. Work has been carried out on intermittent supply of product and this has led to different product ranking and behaviour [51, 52].

2.5 Measurement and Analysis

A search of the literature for wear tests carried out on wheel and rail materials reveals that not only have a wide variety of testing approaches been used, but also the range of measurements and analysis both before and after testing has varied considerably. Even though in all cases wear has been measured there have been many ways in which it has been presented. There are many metrics for defining wear rate. Blau [41] identified 23 different metrics for sliding (adhesive) wear alone! For rolling-sliding wear in twin-disc tests the most widely used metric is "µg of material loss/m rolled/mm^2 apparent contact area". In many cases, unfortunately, insufficient data has been included to allow the calculation of an Archard wear coefficient or define a $T\gamma$ value. This is usually because the material hardness has not been defined in the case of the Archard coefficient and for $T\gamma$ sometimes full-scale-test contact conditions are not defined or the friction coefficient has not been measured. The measurements and analysis carried out should focus on three areas:

1. Measurement of the material wear
2. Establishing the wear mechanisms
3. Definition of Archard wear coefficients or $T\gamma$ value

In the case of wear measurement, this can be carried out in a number of ways. The best approach is to measure mass loss. This requires careful initial cleaning of a specimen and mass measurement before and after a test. This is not always possible because of specimen size (especially for full-scale testing!), expected mass to be lost relative to total specimen mass, or because there is a need to preserve surface third-body layers for analysis. In this case a geometry change can be used. This could be simply a measurement of a diameter change for a twin-disc specimen. For a full-scale wheel or rail a profile change can be measured. There are bespoke tools available for such measurements [9]. Here an area of wear can be defined. For measurements involving a geometry measurement it is helpful if the material density is provided to allow a mass loss calculation. Another issue that needs to be considered here is how to deal with plastic deformation, i.e., material that has flowed, as a result of the contact conditions, to a point where it is no longer load-bearing (see Figure 14 for an example). In some cases researchers have included this as “lost” material, although it is still attached to the rail, albeit ‘displaced’ from the contact.

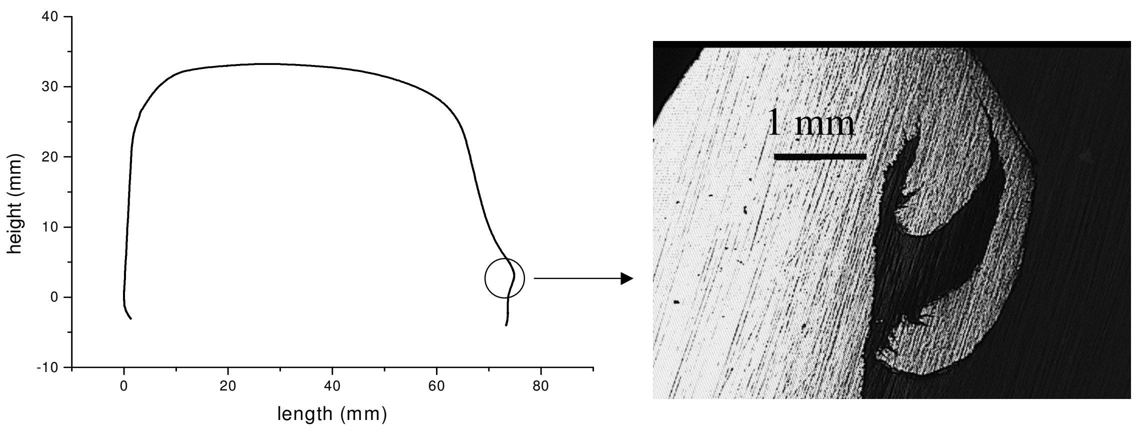


Figure 14. Rail Material Plastic Flow (from [8])

In order to study wear mechanisms a number of measurements can be taken and analyses carried out before, during, and after a test. Before a test, surface images can be taken (with optical microscopy or preferably using Scanning Electron Microscopy (SEM)), and roughness or surface morphology (using a stylus or non-contact profilometer) and surface hardness can be measured.

During a test, the friction coefficient should be recorded. This data can be used for a $T\gamma$ analysis and can also give very useful insights to wear transitions occurring during the test. Methods are under development that would allow surface monitoring and real time wear measurement (see for example Brunskill et al. [53]) that would give the information needed to establish how surfaces and wear rate evolve as a test progresses. When these methods are not possible, tests should be stopped periodically for specimen mass loss to be measured to check how wear rate is changing. Care must be taken if this is done as it can be difficult to realign specimens when restarting a test and stopping a test will disrupt any third-body layer present.

Temperature of the contact can also be measured. This can be achieved using a trailing thermocouple (only useful for disc surface temperature), a pyrometer that is focused at the contact (expected temperatures can be found in [37] where twin-disc contact temperatures for a range of conditions were measured and modelled), or a thermal camera. It is also beneficial if wear debris can be captured as the nature of the debris can be very useful in diagnosing the prevalent wear mechanism [54]. Figure 15 shows wear debris captured after a twin-disc wear test with sand. From this a low cycle fatigue mechanism was diagnosed [4].

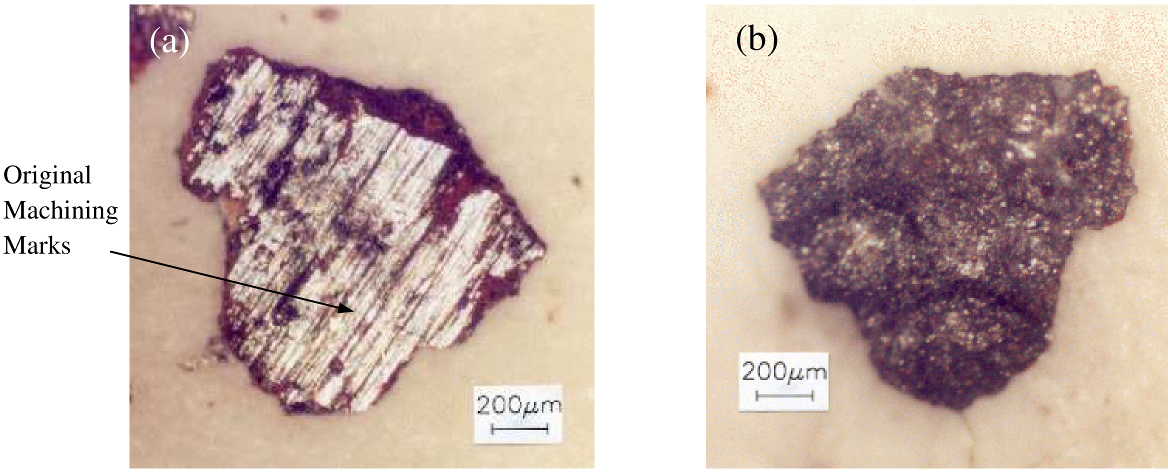


Figure 15. Wheel Steel Wear Debris Particle (from twin-disc test under dry conditions with sand application) (a) Top Surface; (b) Bottom Surface [4]

Post-test a lot more options are available. The pre-test surface analysis can be repeated to see how the surface characteristics have changed. If the specimen can be cut, sub-surface examination (after sectioning, polishing and etching to highlight the microstructure) can be carried out. This allows the deformation/plastic flow of the material to be defined, cracks to be visualised, and for hardness profiles to be measured to assess work hardening of the material. Optical microscopy or SEM can be used for the examination of the microstructure (see Figure 16) as well as methods like Electron Backscatter Diffraction (EBSD) [55] to give a very high resolution characterisation of deformation. Hardness measurements are best carried out using nano-indentation techniques, but if this is not available, micro-hardness measurements can provide a useful insight.

The chemical composition of the surface can also be analysed using techniques such as Glow Discharge Optical Emission Spectroscopy (GDOES) to establish if any modification of the material composition has occurred. This is especially useful when third-body materials have been added during a test (see for example Lewis et al. [25]).

Third-body layers generated on the surface of the disc can also be analysed. Meierhofer et al. [56] developed a method that involved encapsulating discs immediately after a test in Epoxy. This was then sectioned to expose the cross-section of the layer which was analysed using X-Ray Diffraction (XRD) to establish the oxide content. An example of a layer is shown in Figure 6a.

Figure 16. Sub-Surface Deformation in a R8T Wheel Disc Post Test (using optical microscopy) [21]

2.6 Summary

From the review of test approaches it is proposed that the best approach to studying the wear of wheel and rail materials is the twin-disc approach as this provides the most cost and time effective methodology and it provides close control of test parameters which leads to more reliable and usable data. However, the issues outlined above must be considered carefully and mitigation taken to ensure they have minimal effect on the test output. While here twin-disc testing is recommended, it is acknowledged that pin-on-disc can be very useful, especially for simulating cases where extreme sliding occurs such as very tight curves [22].

Full-scale testing has an important role to play in providing validation and evidence that the small scale test output is scalable and relevant [12].

3 A PROPOSED TEST APPROACH

A test approach is defined below that is based on all the information reviewed in compiling the previous sections. While it is recommended that a twin-disc method is used, many of the recommendations can equally apply to a pin-on-disc or full-scale test approach. For full-scale testing the critical element is definition of the contact conditions, particularly where friction and slip are not characterised on the rig itself. The specification is split into three areas: pre-test preparation and measurement; test execution and measurements during the test; and finally post-test measurement and analysis and data presentation. A quick guide to twin-disc testing is presented in Table 1.

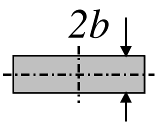
3.1 Pre-Test Preparation and Measurement

When testing a new wheel or rail material, tests should be planned to cover a sufficient range of $T\gamma$ values or contact pressure and slip values (for the Archard approach) to cover the full range of field operating scenarios (here MBD simulations may be required to specify the exact conditions needed, different conditions will be needed to look at wheel tread/rail head contacts versus wheel flange/rail gauge corner contacts). The range required should be informed by MBD simulations.

Sufficient repeats should be built into the plan to ensure statistical confidence in the data. Specific guidance on repeats in wear testing is not available. The ASTM G133 standard test method for ball-on-flat wear testing gives some useful data on coefficients of variability for wear and friction coefficients for both in-lab and between-lab testing [57]. This was a result of eight tests, but as pointed out the coefficients would vary for tests on different rigs and different material pairs. A decision on repeats should be part of an appropriate "Design of Experiments" approach to optimise the testing carried out. It should also be checked that baseline data is available for current wheel and rail grades. Whilst no specific guidance for wear testing exists, the ISO standard for statistical planning of fatigue test data [58] could be used instead. This recommends initially using eight specimens to estimate parameters of a 'known' curve, then "for reliability design purposes … test six specimens at each of five equally spaced stress levels".

If not already done, an uncertainty analysis of the wear and hardness measurements should be performed. If friction, temperature, or other parameters are utilized, then uncertainty analyses should be performed for these parameters also.

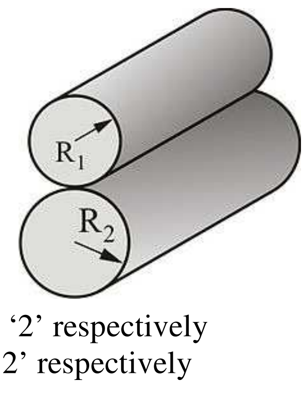
Test discs should be produced, where possible, from actual wheel and rail materials, cutting them parallel to the contact surface (as shown in Figure 7b). They need to be produced to a high accuracy, to ensure that the hole for mounting onto the test-rig shaft is central and that they are round. The discs should be ground to a surface-finish representative of a worn-in wheel or rail surface (Ra about 1 μm in the UK, but more data is needed to confirm this). It is recommended that flat, rather than crowned, surfaces are used to avoid an increase in contact area during a test and a subsequent reduction in the contact pressure. However, it is acknowledged that on some rigs, a crowned surface is required to reach high(er) contact pressures. Loads should be selected to achieve the desired contact pressures, calculated using the Hertzian line or point contact equations outlined in Figure 17 or 18 respectively.

Reduced radius	Contact area dimensions	Maximum contact pressure	Average contact pressure	Contact pressure distribution
$\frac{1}{R'} = \frac{1}{R_1} + \frac{1}{R_2}$	$b = \sqrt{\frac{4P'R'}{\pi E^*}}$ 	$p_0 = \frac{2P'}{b\pi}$	$p_{avg} = \frac{P'}{2b}$	$p(x) = p_0\sqrt{1-\frac{x^2}{b^2}}$

where:

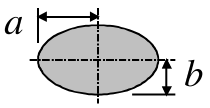
b is the half width of the contact strip (m)
P' is the load per unit length (N/m)
The reduced elastic modulus, E^*, is given by:

$$\frac{1}{E^*} = \left(\frac{1-\nu_1^{\,2}}{E_1} + \frac{1-\nu_2^{\,2}}{E_2}\right)$$



where:
ν_1 and ν_2 are the Poisson's ratios of the contacting bodies '1' and '2' respectively
E_1 and E_2 are the elastic moduli of the contacting bodies '1' and '2' respectively

Figure 17. Hertzian Line Contact Equations

Reduced radius	Contact area dimensions	Maximum contact pressure	Average contact pressure	Contact pressure distribution
$\frac{1}{R'}=\frac{1}{R_x}+\frac{1}{R_y}$ where: $\frac{1}{R_x}=\frac{1}{R_{1x}}+\frac{1}{R_{2x}}$ and $\frac{1}{R_y}=\frac{1}{R_{1y}}+\frac{1}{R_{2y}}$	$a=\sqrt[3]{\frac{3k^2\mathbf{E}(k)PR'}{\pi E^*}}$ $b=\sqrt[3]{\frac{3\mathbf{E}(k)PR'}{\pi k E^*}}$ 	$p_0=\frac{3P}{2\pi ab}$	$p_{\mathrm{avg}}=\frac{P}{\pi ab}$	$p(x,y)=p_0\left\{1-\frac{x^2}{a^2}-\frac{y^2}{b^2}\right\}^{1/2}$

where k is the ellipticity parameter ($k=a/b$) and $\mathbf{E}(k)$ is an elliptic integral of the second kind. The elliptic integral may be obtained from tables of mathematical data. Alternatively, an approximate solution is given by:

$$k=1.0339\left(\frac{R_y}{R_x}\right)^{0.6360} \quad \text{and} \quad \mathbf{E}=1.0003+\frac{0.5968R_x}{R_y}$$

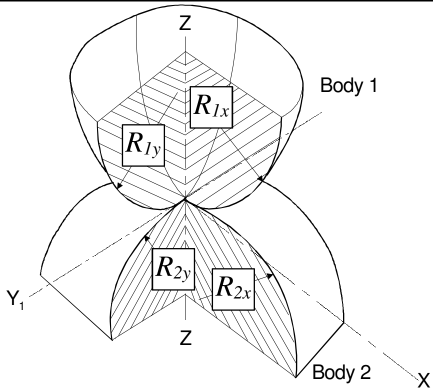


Figure 18. Hertzian Elliptical Contact Equations

Slip (or creepage) values should be stated as well as the method of calculation. The equations used for determining instantaneous slip, S_i, and cumulative slip, S_c, for the SUROS twin disc machine are [39]:

$$S_i(\%)=200\times\left(\frac{R_{TOP}V_{TOP}-R_{BOTTOM}V_{BOTTOM}}{R_{TOP}V_{TOP}+R_{BOTTOM}V_{BOTTOM}}\right) \quad (2)$$

$$S_c(\%)=200\times\left(\frac{R_{TOP}N_{TOP}-R_{BOTTOM}N_{BOTTOM}}{R_{TOP}N_{TOP}+R_{BOTTOM}N_{BOTTOM}}\right) \quad (3)$$

where R (mm) is the radius of the disk specimen indicated by the subscript, V is the speed of the disc in rpm and N is the number of cycles. On the SUROS twin disc machine the top disc is normally the rail disc and is acting as a brake, i.e. is going slower than the bottom, driving, wheel disc.

Within the field of tribology different versions of Equations 2 and 3 exist. It is important that the method used is defined so that data can be compared. However, it is recommended that the approach used for work performed using the SUROS twin disc machine be used.

When using the SUROS twin disc machine, the disc diameter can be measured to ±0.005 mm at the start of a test. During a dry test the diameter may reduce by up to 0.4 mm. This means that slip calculated using the initial values will become progressively less accurate during a test, although after an interruption, when the reduced diameter can be measured, account can be taken of this change.

Images of the test surfaces should be taken pre-test along with roughness and hardness measurements. Disc roundness should also be checked with an appropriate roundness measuring machine (e.g., a Taylor Hobson Talyrond device) which will also help confirm that the disc hole is central (typically ±0.005mm on the distance benteen disc centre and running surface). Prior to testing, discs should always be cleaned in an ultrasonic bath using, for example, acetone, and then weighed. Care is needed here as it can take a considerable time to clean specimens. They should then be mounted onto the rig and testing commence as soon as possible.

Where application of third-body materials is required, previous methods should be reviewed and one selected that is fully representative of field application. Discussions should ideally be held with suppliers/users to ensure that the best approach is chosen.

3.2 Test Execution and Measurements during the Test

Once mounted onto the rig, relative disc alignment should be checked before proceeding with the test. Disc air cooling should be applied during the test. Requirements in terms of flow rate and pressure will depend on the specimen size and rotational speed, slip and load being used and will need to be defined on a case-by-case basis.

During the test friction levels should be recorded for post-test analysis and for use in $T\gamma$ modelling. It would be good practice to collect slip, load, and temperature data as well to ensure that they were kept at the required levels during the test. Where possible, wear debris should also be collected (for example, by using a suction device and filter).

If a real time measurement of wear is not possible it would be appropriate for some of the tests to stop periodically (every 5000 cycles, as shown in Figure 10) to assess wear rate to ensure that the test length is long enough to achieve steady state wear. No criteria have been defined for when steady state wear is occurring, but a relatively constant wear rate would indicate this has been reached. Some materials take a long time to attain this state and it could be possible that some do not.

3.3 Post-Test Measurement and Analysis and Data Presentation

Post-test measurements should include final mass of the discs for wear-rate assessment. Final roughness of the surface should be determined and surface images taken, using optical microscopy and SEM.

To enable further investigation of wear mechanisms, discs should be sectioned and sub-surface deformation characterised as well as hardness. Sectioning could also be used to establish how much material has flowed out of the contact region to add to that lost completely. Where appropriate, techniques such as EBSD and XRD can be used to assess surface modification and third-body layers.

In reporting it is important that all information required for $T\gamma$ and Archard analysis is included:

- Disc mass losses
- Wear rate vs cycles
- Disc diameters

- Number of test cycles
- Disc rotational speeds
- Friction coefficients (average and a plot of friction coefficient against time or cycles)
- Load applied and Hertzian contact pressure
- Slip (%) and method of calculation
- Contact width
- Hertzian contact area
- Hardness, density and Young's Modulus for each disc material
- Application method and rate for third body materials added during the test

Further information for investigation of wear mechanisms etc. would include:

- Temperature vs cycles
- Surface images before and after test
- Post-test subsurface deformation
- Post-test subsurface hardness gradient

Test Planning	
• Obtain full range of contact conditions from MBD • Plan tests to cover full range (varying contact pressure and slip to give range of $T\gamma$ values) • Select loads to achieve contact pressures required (using Hertzian equations)	• Plan repeats • Manufacture discs from actual rail and wheel (set tolerances to achieve roundness and hole centrality required) • Cut parallel to contact surface • Grind to 1 μm Ra • Plan third-body product application where needed
Pre-Test Measurements	
• Mass (after cleaning) • Roughness • Surface hardness	• Surface images • Sub-surface hardness (dummy disc) • Sub-surface images before deformation (dummy disc)
Measurements during Test	
• Friction • Load and slip • Room temperature and humidity • Wear (either using appropriate technology or by stopping test periodically)	• Contact temperature • Any unusual behaviour of test specimens i.e. change in noise (frequency, amplitude), visible change in wear debris, visible change of test specimen surface
Post Test Analysis	
• Disc mass • Surface images • Roughness • Surface hardness	• Sub-surface deformation • Sub-surface hardness gradient • Third-body layer thickness and composition • Wear debris characteristics
Reporting	
• Mass loss and wear rate vs cycles with a statement of the measurement uncertainty • Material properties: density, hardness, Young's modulus, all parameters with a statement of the measurement uncertainty. • Disc details: geometry, roughness (start and finish), contact area (at test load)	• Test conditions: load, contact pressure, slip, disc rpm, duration (in cycles) • Friction vs cycles • Temperature vs cycles • Surface and sub-surface analysis • Wear debris analysis

Table 1. A Quick Guide to Twin-Disc Testing

4 CONCLUSIONS

In this paper different types of wear tests have been evaluated and the advantages and disadvantages of each discussed.

A large number of wear tests have been carried out, but there is inconsistency in the approaches used and in the presentation of data which means that in many cases the data cannot be compared across different studies. Some key issues that could lead to misleading interpretations are: failure to determine how wear rate varies across a test; and not running a test for enough cycles to reach a steady state wear rate. These could lead to inaccurate material rankings. It is important to understand how materials work-harden in order to assess wear resistance. This, however, is rarely characterised.

In an effort to try and ensure that data from future tests can be compared, good practice has been outlined in terms of the data that should be collected pre-test, during a test, and after the test, and how the mass loss from the wheel and rail materials should be presented. Wear testing is expensive and time consuming to carry out, so it is important that the maximum benefit can be gained from the results.

Approaches for scaling laboratory data from small-specimen tests to the full-scale have been outlined as this is of critical importance. Little work has been carried out on this so it is a key future research topic.

REFERENCES

[1] UIC, 2013, "Lasting Infrastructure Cost Benchmarking LICB, 15 years of benchmarking 1996-2010", © International Union of Railways (UIC), Paris, September 2013.

[2] Kalousek, J., Magel, E., 1997, "Optimising the Wheel/Rail System", *Railway Track and Structure*, January.

[3] Lewis, S.R., Lewis, R., Evans, G., Buckley-Johnstone, L.E., 2014, "Assessment of Railway Curve Lubricant Performance using a Twin-Disc Tester", *Wear*, Vol. 314, pp205-215.

[4] Lewis, R., Dwyer-Joyce, R.S., 2006, "Wear at the Wheel/Rail Interface when Sanding is used to Increase Adhesion", *Journal of Rail and Rapid Transit*, *Proceedings of the IMechE Part F*, Vol. 220, No. 1, pp29-41.

[5] Stanca, M., Stefanini, A., Gallo, R., 2001, "Development of an Integrated Design Methodology for a New Generation of High Performance Rail Wheelsets", *Proceedings of the 16th European MDI User Conference*, Berchtesgaden, Germany, 14-15 November.

[6] Heyder, R., Maedler, K., 2015, "The Influence of Wheel and Rail Material on the Wear of the Respective Contact Partner", *Proceedings of CM2015 10th International Conference on Contact Mechanics and Wear of Rail/Wheel Systems*, Colorado, USA, 30 August - 3 September 2015.

[7] Dearden, J., 1960, "The Wear of Steel Rails and Tyres in Railway Service", *Wear*, Vol. 3, pp43-49.

[8] Olofsson U., Telliskivi T., 2003, "Wear, Friction and Plastic Deformation of Two Rail Steels - Full Scale Test and Laboratory Study", *Wear*, Vol. 254, pp 80-93.

[9] Olofsson, U., Nilsson, R. 2002, "Surface Cracks and Wear of Rail: A Full-Scale Test on a Commuter Train Track", *Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit*, Vol. 216, No. 4, pp249-264.

[10] Steele, R.K., 1982, "Observations of In-Service Wear of Railroad Wheels and Rails under conditions of widely varying Lubrication", *ASLE Transactions*, Vol. 25, No. 3, pp400-409.

[11] McEwen, I.J., Harvey, R.F., 1985, "Full-scale Wheel-on-Rail Testing: Comparisons with Service Wear and a Developing Theoretical Predictive Model", *Lubrication Engineering*, Vol. 41, No. 2, pp80-88.

[12] Braghin, F., Lewis, R., Dwyer-Joyce, R.S., Bruni, S., 2006, "A Mathematical Model to Predict Railway Wheel Profile Evolution due to Wear", *Wear*, Vol. 261, pp1253-1264.

[13] Stock, R., Eadie, D.T., Elvidge, D., Oldknow, K., 2011, "Influencing Rolling Contact Fatigue through Top of Rail Friction Modifier Application – A Full Scale Wheel–Rail Test Rig Study", *Wear*, Vol. 271, pp134-142.

[14] Stock, R., Pippan, R., 2011, "RCF and Wear in Theory and Practice—The Influence of Rail Grade on Wear and RCF", *Wear*, Vol. 271, pp125-133.

[15] Kumar, S., Rao, D.L.P., 1984, "Wheel-Rail Contact Wear, Work, and Lateral Force for Zero Angle of Attack - A Laboratory Study", *Transactions of the ASME*, *Journal of Dynamic Systems, Measurement, and Control*, Vol. 106, pp319-326.

[16] Beagley, T.M., 1976, "Severe Wear of Rolling/Sliding Contacts", *Wear*, Vol. 36, pp317-335.

[17] Bolton, P.J., Clayton, P., McEwen, I.J., 1982, "Wear of Rail and Tyre Steels under Rolling/Sliding Conditions", *ASLE Transactions*, Vol. 25, No. 1, pp17-24.

[18] Bolton, P.J., Clayton, P., 1984, "Rolling-Sliding Wear Damage in Rail and Tyre Steels", *Wear*, Vol. 93, pp145-165.

[19] Krause, H., Poll, G., 1986, "Wear of Wheel-Rail Surfaces", *Wear*, Vol. 113, pp103-122.

[20] Garnham, J.E., Beynon, J.H., 1992, "Dry Rolling-Sliding Wear of Bainitic and Pearlitic Steels", *Wear*, Vol. 57, pp81-109.

[21] Lewis, R., Dwyer-Joyce, R.S., 2004, "Wear Mechanisms and Transitions in Railway Wheel Steels", *Journal of Engineering Tribology*, *Proceedings of the IMechE Part J*, Vol. 218, pp467-478.

[22] Sundh, J., Olofsson, O., Sundvall, K., 2008, "Seizure and Wear Rate Testing of Wheel-Rail Contacts under Lubricated Conditions using Pin-on-Disc Methodology", *Wear*, Vol. 265, pp 1425-1430.

[23] Sundh, J., Olofsson, U., 2011, "Relating Contact Temperature and Wear Transitions in a Wheel-Rail Contact", *Wear*, Vol. 271, No. 1-2, pp78-85.

[24] Lyu, Y., Zhu, Y., Olofsson U., 2015, "Wear between Wheel and Rail: A Pin-on-disc Study of Environmental Conditions and Iron Oxides", *Wear*, Vol. 328-329, pp 277-285.

[25] Lewis, S.R., Lewis, R., Olofsson, U., Eadie, D.T., Cotter, J., Lu, X., 2012, "Effect of Humidity, Temperature and Railhead Contamination on the Performance of Friction Modifiers: Pin-on-Disk Study", *Journal of Rail and Rapid Transit, Proceedings of the IMechE Part F*, Vol. 227, No. 2, pp115-127.

[26] Stock, R., Eadie, D.T, Oldknow, K., 2013, "Rail Grade Selection and Friction Management: A Combined Approach for Optimising Rail-Wheel Contact", *Ironmaking and Steelmaking*, Vol. 40, No. 2, pp108-114.

[27] Stabler, L., Irani, F., Li, D., Tajaddini, A., 2016, "Advancements in Testing Facilities for Railways", *Proceedings of WCRR 2016*, Milan, Italy, 29 May-2 June.

[28] Archard, J.F., 1953, "Contact and Rubbing of Flat Surfaces", *Journal of Applied Physics*, Vol. 24, pp981-988.

[29] Jendel, T., 2000, "Prediction of Wheel Profile Wear – Methodology and Verification", Licentiate Thesis, KTH, Stockholm.

[30] Enblom, R., 2009, "Deterioration Mechanisms in the Wheel-Rail Interface with Focus on Wear Prediction: A Literature Review", *Vehicle System Dynamics*, Vol. 47, pp661-700.

[31] Pombo, J., Ambrósio, J., Pereira, M., Lewis, R., Dwyer-Joyce, R.S., Ariaudo, C., Kuka, N., 2010, "A Railway Wheel Wear Prediction Tool Based on a Multibody Software", *Journal of Theoretical and Applied Mechanics*, Vol. 48, pp771-788.

[32] Pombo, J., Quost, X., Tassini, N., Lewis, R., Dwyer-Joyce, R.S., Ambrósio, J., Pereira, M., Ariaudo, C., Kuka, N., 2011, "A Wear Prediction Tool for Steel Railway Wheels using Three Different Wear Functions Wear", Vol. 271, pp238-245.

[33] Buckley-Johnstone, L.E., Harmon, M., Lewis, R., Hardwick, C, Stock, R., 2016, "Assessment of Friction Modifiers Performance using Two Different Laboratory Test-Rigs", *Proceedings of Railways 2016*, Sardinia, Italy, April 5-8 2016.

[34] Eadie, D.T., Elvidge, D., Oldknow, K., Stock, R., Pointner, P., Kalousek, J., Klauser, P., 2008, "The Effects of Top of Rail Friction Modifier on Wear and Rolling Contact Fatigue: Full-Scale Rail-Wheel Test Rig Evaluation, Analysis and Modelling", *Wear*, Vol. 265, pp1222-1230.

[35] Lewis, R., Olofsson, U., 2004, "Mapping Rail Wear Regimes and Transitions", *Wear*, Vol. 257, No. 7-9, pp721-729.

[36] Berthier, Y., Descartes, S., Busquet, M., Niccolini, E., Desrayaud, C., Baillet, L., Baietto-Dubourg, M.C., 2004, "The Role and Effects of the Third-Body in the Wheel-Rail Interaction", *Fatigue Fracture of Engineering Materials and Structures*, Vol. 27, pp423-436.

[37] Gallardo-Hernandez, E.A., Lewis, R., Dwyer-Joyce, R.S., 2006, "Temperature in a Twin-Disc Wheel/Rail Contact Simulation", *Tribology International*, Vol. 39, pp1653-1663.

[38] Gallardo-Hernandez, E.A., Lewis, R., 2008, "Twin Disc Assessment of Wheel/Rail Adhesion", *Wear*, Vol. 265, pp1309-1316.

[39] Fletcher, D.I. and Beynon, J.H., 2000, "Development of a Machine for Closely Controlled Rolling Contact Fatigue and Wear Testing", *Journal of Testing and Evaluation,* JTEVA, Vol. 28, No. 4, pp267-275.

[40] Ma, L., He, C.G., Zhao, X.J., Wang, H.Y., Wang, W.J., Zhu, Y., Liu, Q.Y., Jin, X.S., 2015, "Study on Wear and Rolling Contact Fatigue Behaviours of Wheel/Rail Materials under different Slip Ratio Conditions", *Proceedings of CM2015, 10th International Conference on Contact Mechanics and Wear of Rail/Wheel Systems*, Colorado, USA, August 30-September 3 2015.

[41] Blau, P.J., 2015, "How Common is the Steady-State? The Implications of Wear Transitions for Materials Selection and Design", *Wear*, Vol. 332-333, pp 1120–1128.

[42] Lewis, S.R., Lewis, R., Goodwin, P.S., Fretwell-Smith, S., Fletcher, D.I., Murray, K., 2015a, "Improving Rail Wear and RCF Performance using Laser Cladding", *Wear*, Vol. 332-333, pp581-591.

[43] Lewis, R., Dwyer-Joyce, R.S., Lewis, S.R., Hardwick, C., 2012, "Tribology of the Wheel/Rail Contact: The Effect of Third Body Materials", *International Journal of Railway Technology*, Vol. 1, No. 1, pp167-194.

[44] Hardwick, C., Lewis, R., Eadie, D.T., 2014, "Wheel and Rail Wear – Understanding the Effects of Water and Grease", *Wear*, Vol. 314, pp198-204.

[45] Rovira, A., Roda, A., Lewis, R., Marshall, M.B., 2012, "Application of FASTSIM with Variable Coefficient of Friction using Twin Disc Experimental Measurements", *Wear*, Vol. 274-275, pp109-126.

[46] Zhu, Y., Sundh, J., Olofsson, U., 2013, "A Tribological View of Wheel-Rail Wear Maps, *International Journal of Railway Technology*, Vol. 2, pp79-91.

[47] Arias-Cuevas, O., Li, Z., Lewis, R., 2011, "A Laboratory Investigation of the Influence of the Particle Size and Slip on Adhesion and Wear in the Wheel-Rail Contact", *Wear*, Vol. 271, pp14-24.

[48] Hardwick, C., Lewis, S.R., Lewis, R., 2014, "The Effect of Friction Modifiers on Wheel/Rail Isolation at Low Axle Loads", *Journal of Rail and Rapid Transit, Proceedings of the IMechE Part F*, Vol. 228, No. 7, pp768-783.

[49] Lewis, S.R., Riley, S., Fletcher, D.I., Lewis, R., 2015, "Optimisation of a Railway Sanding System, Part 1: Optimal Grain Entrainment into the Wheel/Rail Contact", in press, *Journal of Rail and Rapid Transit, Proceedings of the IMechE, Part F*.

[50] Lewis, S.R., Lewis, R., Cotter, J., Lu, X., Eadie, D.T., 2015, "A New Method for the Assessment of Traction Enhancers and the Generation of Organic Layers in a Twin-Disc Machine", in press, *Wear*.

[51] Fletcher, D.I., Beynon, J.H., 2000, "The Effect of Intermittent Lubrication on the Fatigue Life of Pearlitic Rail Steel in Rolling–Sliding Contact", *Journal of Rail Rapid Transit, Proceedings of the IMechE Part F*, Vol. 214, pp145-158.

[52] Evans, M.D., Lewis, R., 2015, "Alternative Twin-Disc Assessment of Solid Stick Flange Lubricants", *Proceedings of CM2015, 10th International Conference on Contact Mechanics and Wear of Rail/Wheel Systems*, Colorade, USA, 30 August-3 September 2015.

[53] Brunskill, H., Harper, P., Lewis, R., 2015, "The Real-Time Measurement of Wear using Ultrasonic Reflectometry", *Wear*, Vol. 332-333, pp1129-1133 .

[54] Roylance, B.J., Williams, J.A., Dwyer-Joyce, R.S., 2000, "Wear Debris and Associated Wear Phenomena - Fundamental Research and Practice", *Journal of Engineering Tribology, Proceedings of the IMechE, Part J*, Vol. 214, pp79-105.

[55] Linz, M., Cihak-Bayr, U., Trausmuth, A., Scheriau, S., Kunstner, D., Badisch, E., 2015, "EBSD Study of Early-Damage Phenomena in Wheel-Rail Model Test", *Wear*, Vol. 342-343, pp13-21.

[56] Meierhofer, A., Hardwick, C., Lewis, R., Six, K., Dietmaier, P., 2014, "Third Body Layer - Experimental Results and a Model Describing its Influence on the Traction Coefficient", *Wear*, Vol. 314, pp148-154.

[57] ASTM G133 - 05(2016), "Standard Test Method for Linearly Reciprocating Ball-on-Flat Sliding Wear", American Society for Testing and Materials, 2016.

[58] BS ISO 12107:2003, "Metallic materials, Fatigue testing, Statistical planning and analysis of data", British Standards Institution, 2003