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A Comparison of Friction Modifier Performance using Two Laboratory Test Scales

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Abstract

This paper describes two methods for assessing friction modifier performance carried out at two different test scales. Study A used wear data from a full-scale rig test at voestalpine Schienen GmbH [1] and compared it to wear data from twin disc tests using the SUROS test machine at The University of Sheffield. Study B compared 'retentivity' data from a full-scale rig at The University of Sheffield and SUROS tests. Study A concluded that a good correlation existed between the two scales although assumptions made in the full-scale contact calculation introduce large spread into the results. There was a greater correlation between the two data sets at more severe contact conditions. Study B showed a different baseline coefficient of traction between the two scales and that a longer test length is required to fully evaluate the 'retention' of the friction modifier on the full-scale rig. The article expands on a previous conference presentation [2] on the same subject. Additional information on the test procedure and test rigs is included here. Surface and sub-surface analysis of SUROS test samples has also been added. The analysis has shown that applying friction modifier leads to a similar wear mechanism as for dry contact, but wear is less severe and there is less subsurface deformation. A discussion describing the differences in test scales and comparing lab tests to field operation is also included.

Keywords: wheel rail contact, friction modifier, scaling, twin disc testing, wear rate

1 Introduction

Friction modifiers (FM) are used to provide an intermediate coefficient of friction in the wheel-rail interface, (usually between 0.3-0.4) thereby improving energy efficiency of the railways by ensuring friction is not too high. The intermediate friction level will also ensure safe train operation by not compromising traction and braking of the train. Friction modifiers also produce a positive gradient on creep curves. A positive gradient on creep curves prevents roll-slip oscillations which can lead to damage [3].

The ability to perform controlled testing of wheel-rail interaction is vital to improve the understanding of the wheel-rail interface. Under most circumstances, it is uneconomical to perform testing under fully representative conditions. Access to track and instrumented rolling stock is limited, expensive and difficult to control, which leads to the need for representative laboratory tests. There are many different scales and styles of test facility that exist to allow for representative contact conditions within controllable environments. These can range from simple table top tribometers through to full-scale component tests. It is often the case that small-scale test rigs give results quickly, cheaply, and with more control over parameters than larger, more complex test rigs. Reducing the complexity of the test rig to gain control over different parameters is at the expense of accurately portraying the system, which can lead to differences between results from laboratory and in-service observations. Understanding the fundamental operating and tribological principles of the system to be tested is key to designing representative small-scale tests.

The aim of this work was to compare the performance of a water based friction modifier when subjected to two different scales of laboratory experiments. The two scales were: 1) twin disc using 47 mm diameter discs; 2) full-scale linear test rig using a full size wheel (diameter approximately 900 mm). These separate, but comparable, test regimes have looked at the performance of the FM with respect to wear amounts and coefficient of traction levels. Study A compared wear and $T\gamma/A$ data for dry and FM interfacial conditions. Study B compared coefficients of traction in terms of evolution, retention and baseline levels where a single application of FM was applied initially.

2 Background

Top of rail (TOR) FMs are widely used in the North American heavy haul environment as well as in passenger/transit systems all over the world. There are a number of different material concepts with regard to materials for TOR application, which has led to confusion. However, a paper has recently been published [4] which has clearly defined FMs according to their "drying behaviour" and how to differentiate them from TOR lubricant materials. FMs are particles suspended in water, which quickly evaporates in the wheel-rail contact leaving behind solid particles to mix with the existing third body layer to provide the optimised friction level. Non-drying materials provide the optimised friction level through a mixed lubrication mechanism (TOR lubricants and sub-classes). In addition to these two classes, solid stick FM's are also available which are applied to the wheel, and provide intermediate friction levels though similar mechanisms.

The benefits of friction modifiers are well documented. They reduce rolling contact fatigue (RCF) and wear by reducing lateral forces in curves[3-4], and also lead to a reduction in noise [5–10]. There are also reductions in low frequency vibrations [10] (which leads to reduced corrugations and improved ride comfort) and reduced fuel consumption [11] (via reduced rolling and curve resistance). Additionally there is no impact of FM's on track isolation circuits [12] or braking capabilities, which are important safety aspects of any product to be applied to the rail.

A recent field test using a TOR lubricant (a hybrid material containing water and oil) [13] showed that the friction coefficient was highly dependent on the amount of TOR lubricant applied. If too much is applied then the friction coefficient is too low for safe operation of the train. Additionally, if the amount of TOR lubricant applied is too little then the friction coefficient is above the desired, intermediate, levels. This supports the statement that TOR lubricants work in the mixed mode lubrication regime and that a very close control of application rates is necessary to obtain a desired friction level [4].

Recent research has focussed on the optimisation of the application of FM's, i.e. how much to apply and when, how far down the track the effect lasts and how it interacts with oxides on the rail [11, 15-16]. Most of the current research has been either field studies or full-scale rig studies, both of which are costly in terms of time and money. Therefore, if twin disc test results are shown to provide scalable results, then research can be carried out at a faster rate and lower cost. This is because small-scale twin disc rigs can be used to carry out large test programs quickly, meaning many variables can be tested in a relatively short timeframe. A small number of the most promising results can be tested on full-scale rigs, and field trials used to verify the small-scale results.

To be able to compare wear data from different test rigs a *Tγ/A* approach is used. *T* refers to the tractive force, *γ* is the amount of slip in the contact and A is the contact area. Relating wear to *Tγ* is an approach widely used to predict the wheel profile evolution within multi-body dynamic simulations. Originally, it was used as an empirical wear index as wear is related to the energy lost due to creepage in the wheel-rail interface [16]. *Tγ* is divided by the contact area in this work to allow scaling between small-scale specimens and full-scale test rigs. Whilst using the *Tγ/A* approach allows comparisons of the test rigs to be made, how each individual parameter affects performance cannot be analysed. This approach was first used by Bolton and Clayton [17] in twin disc tests and has since been used in full-scale tests [18].

3 Test Methodology

Both studies used the SUROS test rig [15] for the twin disc tests, a schematic of the rig is shown in Figure 1. The discs are machined from rail and wheel steel with the dimensions shown in Figure 2. Both studies include results from full-scale rigs. Study A used data from tests run on the full-scale rolling rig at voestalpine Schienen GmbH [1] and Study B the full-scale wheel-rail rig at The University of Sheffield shown in Figure 3.

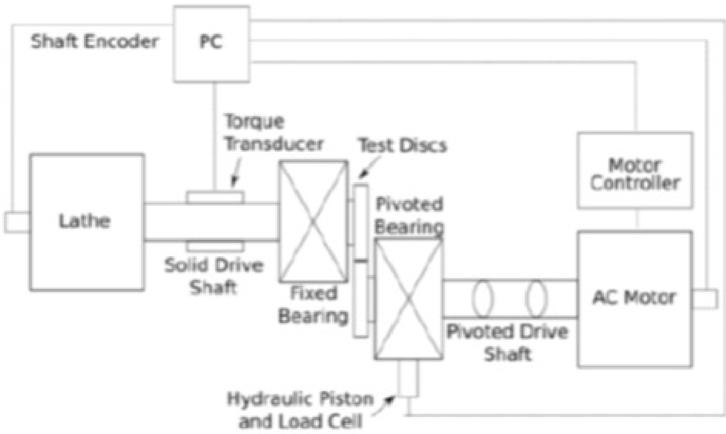

Figure 1- Schematic of the SUROS twin disk tester

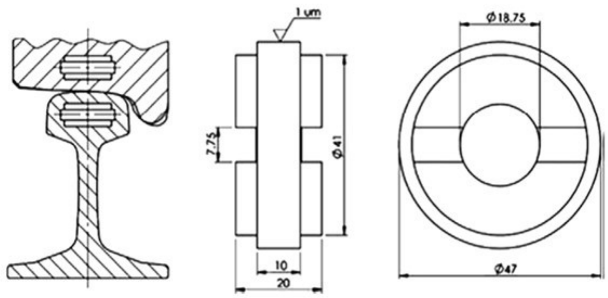

Figure 2- Dimensions of SUROS test specimen

Figure 3- Full-scale test rigs at: left) voestalpine Schienen GmbH [19], right)

University of Sheffield

3.1 Study A

Full-scale tests on the voestalpine rig used vertical and lateral loads of 23 tonnes and 4 tonnes, respectively. Full details of the rig's operation have been previously outlined [1, 16]. Dry tests were run as well as tests with FM sprayed on to the railhead every 250 wheel passes for a duration of 10,000 wheel passes. A wheel pass is one movement of the wheel through the test area. The wheel and rail are separated whilst the rail returns to its starting position so that the wheel is always passing over the test section in the same direction. Wheel and rail profile measurements were performed both pre and post-testing using a Greenwood Engineering MiniProf. This allows wear to be calculated, the difference between the post-test profile and the pre-test profile is the amount of material lost during the test. From the change in area the weight loss per cycle was calculated. Creep and traction were not able to be controlled or measured, so VAMPIRE® simulations and field tribometer measurements were used when calculating $T\gamma/A$ values, with allowances for extremities of conditions, hence the large error bars presented in the Results section.

The following assumptions have been made to calculate the wear rate for the full-scale data [19]:

- The contact patch dimensions were generated using the VAMPIRE® Rail Vehicle Dynamics Software, see Figure 4.
- The test rail length for each pass was 0.5 m

- The creep was estimated to be 0.5 %. This value was obtained from evaluating a creepage distribution vector plot as shown in Figure 5. The creepage plot was simulated using VAMPIRE® [19].
- The coefficient of friction was assumed to be 0.5-0.6 for dry tests and 0.28-0.35 for FM tests. Friction was not measured during the tests, but the range specified is typical of coefficient of friction tribometer measurements in the field.

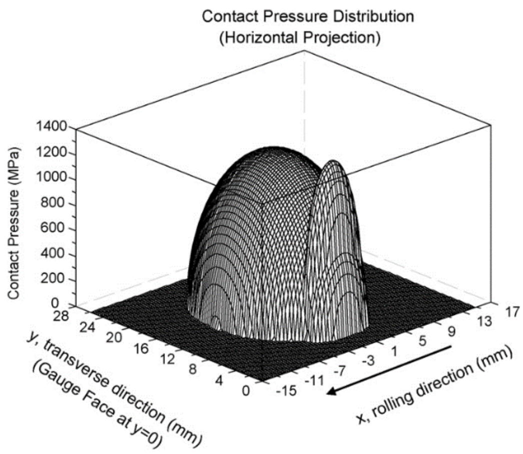

Figure 4- Pressure distribution to approximate conformal contact conditions [19]

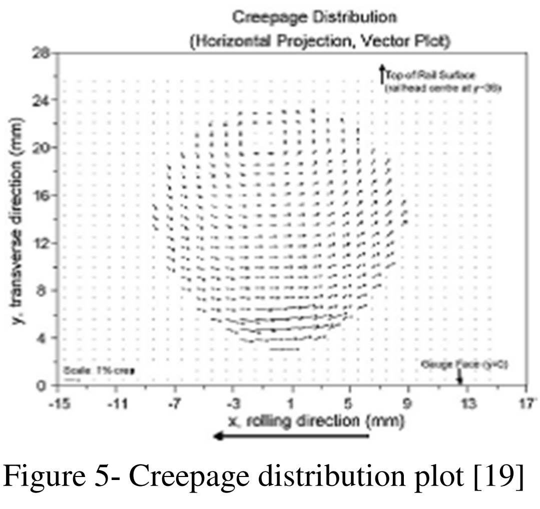

Figure 5- Creepage distribution plot [19]

Twin disc tests were performed for which the maximum Hertzian contact pressure was predicted to be 900 MPa, with creep values ranging from 0-5 % in dry conditions and with FM. These values were chosen to be representative of wheel tread/rail head contact. The nominal rail disc speed was set at 400 rpm which gave a surface speed of 1 m/s. FM was reapplied every 250 cycles. Tests were run for 25,000 cycles where one cycle is one revolution of the disc.

3.2 Study B

The Sheffield full-scale rig, as shown in Figure 3, comprises of a section of rail on a slide bed, which can be brought into contact with a fixed-axle-location wheel (nominal diameter 900 mm), which is free to rotate in bearing housings. Three hydraulic actuators are used to control the normal load, rail velocity and slip of the contact. Figure 6 shows a schematic of how the different actuators work. The normal actuator (1) is set vertically above the wheel, and a 'pancake' load cell is used to measure the applied load. The rail velocity is controlled through a horizontal actuator (3) which moves the slide bed with the mounted rail - velocity is measured using a linear variable differential transformer (LVDT). The final actuator is mounted on the slide bed (2), and is linked, via a chain, to the rim of the wheel. This actuator moves at a set velocity relative to the slide bed actuator to produce a slippage at the wheel-rail contact. The force required to produce this relative movement is equal to the frictional force within the contact and is measured by a load cell.

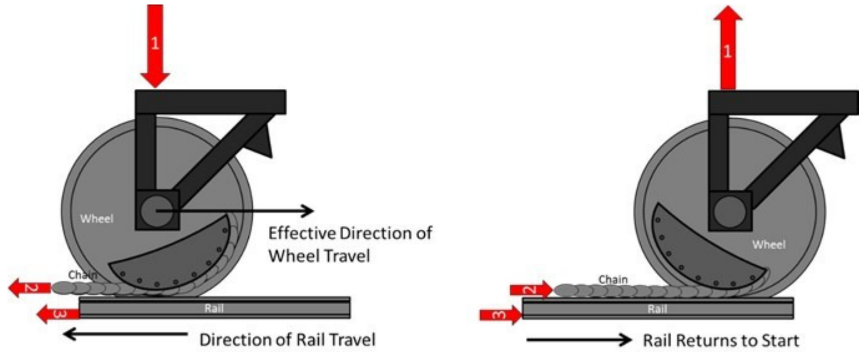

Figure 6- Full-scale rig diagram

FM was applied evenly to a section of the rail head using a brush. A normal load application of 86 kN was applied, which equates to a maximum contact pressure of about 1,500 MPa. Due to limited actuator pressure the rail velocity was restricted to 40 mm/s. The low velocity is one of the main limitations of this test rig when comparing its operation to field operation. Retention tests were run for 800 wheel passes with a fixed creep of 2 %. The wheel always travels in the same direction. The wheel and rail are separated at the end of each pass and the rail returned to its starting position to begin the next wheel pass.

In the twin disc tests a comparable contact stress was used, 1,500 MPa maximum Hertzian contact pressure, and tests were run at 2 % slip. Tests were run at a nominal rail disc speed of 400 rpm, with the driven wheel disc at a higher speed to generate the slippage. Before testing, 0.1 g of the FM product was evenly applied to the rail disc only. The traction coefficient was measured over 5,000 cycles of testing for measurement of a creep curves, and ran with a slippage of 2 % until the traction coefficient reached 0.5 (that of a typical dry test).

4 Results

4.1 Study A

Figure 7 shows the traction curves from two twin disc tests at different slip levels with FM reapplied every 250 passes. It is clear in both graphs that traction levels sharply drop when FM is reapplied. This could be due to the nature of the product which is applied wet, after which the contact dries out/is worn away leading to an increase in traction, although the traction coefficient never reaches the level where it is designed to operate in (0.3-0.4). Another interesting observation is that during the first few applications of FM, the maximum traction coefficient decreases. Both of these observations seen in this twin disc test have been observed previously in other twin disc research [20]. This type of test is useful in analysing what happens when the FM is first applied, but it is difficult to draw other conclusions due to it not representing field conditions closely enough.

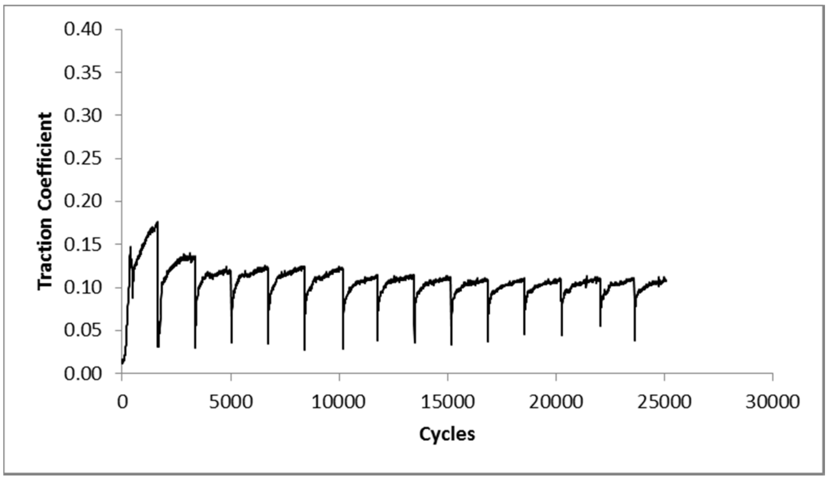

Figure 7- Traction coefficient curve for twin disc test with FM at 1% slip at 900MPa contact pressure

Figure 8 displays wear rate data from previous twin disc tests for dry, wet and grease conditions [21] with the results from the twin disc FM tests overlaid. It shows that the FM has a significantly lower wear rate at all slip values tested when compared to other conditions.

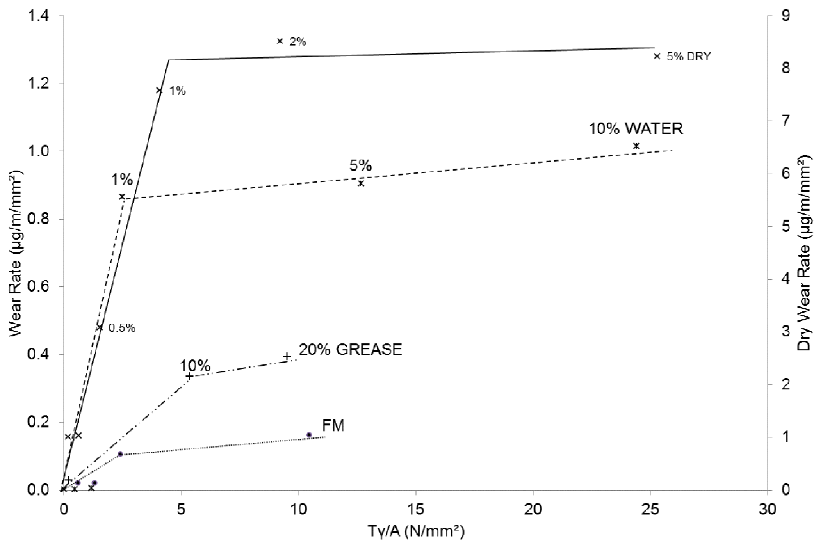

Figure 8- Ty/A wear rate data for twin disc tests with different contaminants [22]

Tγ/A versus wear rate for both twin disc and full-scale in both lubrication conditions is shown in Figure 9. Error bars show the range of values when variation in full-scale contact data is accounted for, as discussed in the Test Methodology section above.

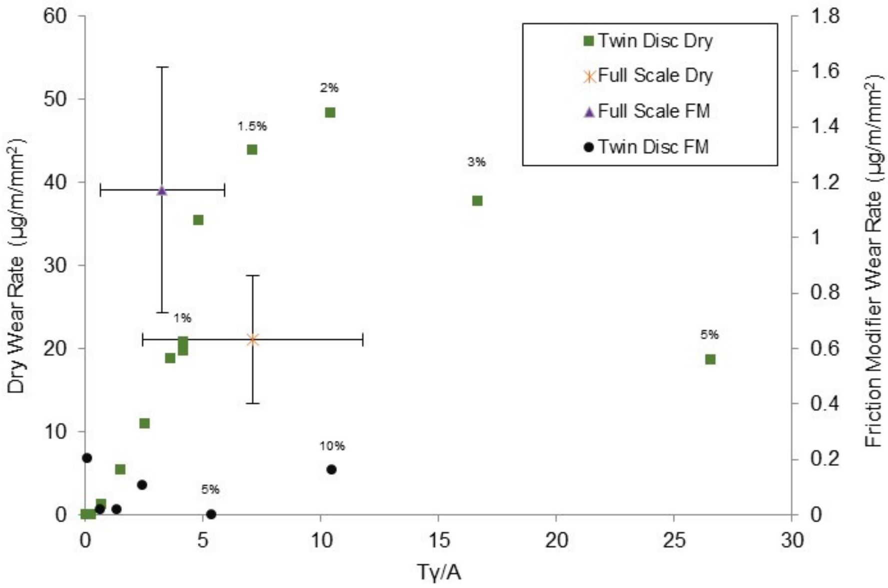

Figure 9- Ty/A wear rate data twin-disc/full-scale comparison for dry and applied friction modifier conditions.

It is clear that applying FM significantly reduces the wear rate. The wear rate when the FM is applied is much higher in the full-scale test than in the twin disc test. This is because even though the amount of product used was scaled down to be appropriate for the size of the discs; all of the product on the disc ends up in the contact whereas on the full-scale rig (FSR) not all the product applied ends up in the contact. Additionally the size and shape of the contact is different in the two different test rigs.

4.1.1 Surface Appearance

The rail discs from the twin disc results presented in Figure 8 were analysed to show the differences in wear features. Figure 10 shows surface images of the rail disc after testing at differing slip levels with FM. A dry comparison for 5 % slip from previous work [21] is also included (Figure 10B). At the lower slip level (Figure 10C) the machining marks are still clearly visible indicating low wear. At 5 % slip (Figure 10A) there are abrasive scratches present, but there are still machining marks visible. There are also abrasive scratches in the dry (Figure 10B) case, but no machining marks present, which indicates that the wear is more severe in the dry case. This is due to the way the friction modifier works. It dries very quickly forming a solid third body layer on the surface of the discs and leads to a lower traction coefficient compared to a dry contact. At 10 % slip (Figure 10D) there is larger material loss than the lower slip levels indicated by the black areas. There are also areas of grey indicating some form of third body layer is present, likely to be a mixture of dried friction modifier product, oxide and wear debris.

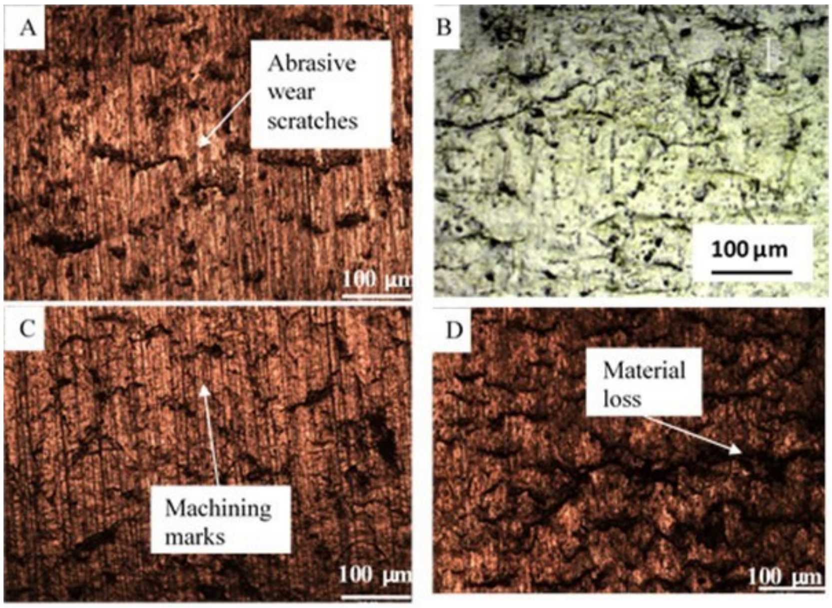

Figure 10- Surface image of rail disc after testing with friction modifier, 1500 MPa, 10% slip

4.1.2 Subsurface Morphology

Figure 11 shows the subsurface deformation of the rail disc after testing at 10 % slip with friction modifier. This depth of deformation is less than 10 µm. This is considerably less than the depth of deformation in the dry condition reported in previous work which is a minimum of 420 µm for the conditions tested [21]. The rail discs were sectioned perpendicular to the rolling direction, polished, and nital solution was applied to show the microstructure.

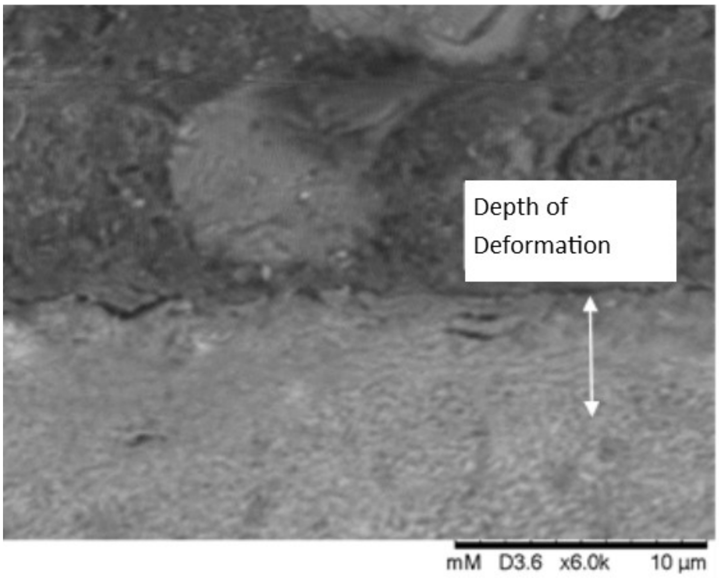

Figure 11- Subsurface deformation of rail disc after testing with friction modifier 1500MPa, 10% slip

4.2 Study B

Retention curves for FM for both types of testing are shown in Figures 12-13 for twin disc and full-scale tests respectively. Figure 12 shows a much lower baseline coefficient of traction than that of the full-scale tests. Figure 13 shows a rapid evolution to a stable traction coefficient (0.3-0.35) that is more in-line with the level required to ensure optimum traction. However, the full-scale tests were not run for long enough to see a return to dry levels of traction, therefore the test should in future be extended until a dry level traction coefficient is reached.

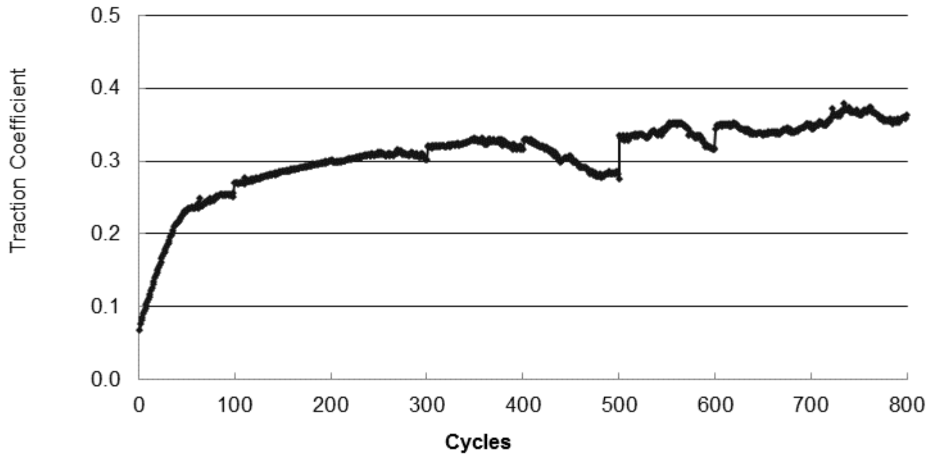

Figure 12- Retention curve for Fm at 2% slip and 1500MPa in a twin-disc test

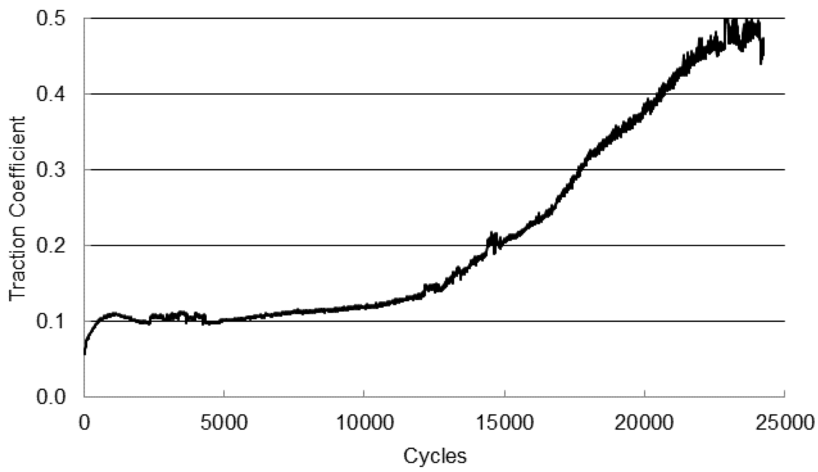

Figure 13- Retention curve for FM at 2% slip and 1500 MPa in FSR test

The initial evolution of traction and longevity of FM retained in the contact is similar in both cases. The lower baseline traction coefficient shown in Figure 12 is believed to be caused by too much product being present in the contact. This is again due to all of the product ending up in the contact in the twin disc test, whereas in the full-scale test less FM ends up in the contact.

Neither test is completely representative of the field. Table 1 summarises the main differences between the FSR, twin-disc and field conditions. These differences have been identified in other published work [22]. For operating speed some twin-disc rigs could be representative of field operations, however, SUROS is slower than typical field operation.

Variable	Full-Scale Rig	Twin-Disc Rig
Contact geometry	Representative of field	Line contact, smaller
Contact Pressure	Representative of field	Representative of field
Climate conditions	Stable	Stable
Operating speed	Up to 100 times slower than field	Slower than field

Table 1- Differences between FSR and twin-disc approach compared to field conditions

There are also a number of differences that affect all lab testing when trying to replicate field conditions. They are:

- The same wheel contacts the same section of rail whereas in the field a wheel travels down a long section of 'fresh' track. This has an effect on the surface condition and geometry as well as the temperature of the contact. High temperatures can build up due to the cyclic reloading of the test specimens and lack of heat transfer away from the contact (in particular in twin-disc testing). Additionally the use of one 'wheel' means that the steering forces acting in the rigs are always the same. Whereas, in the field the steering forces are constantly changing as a wheelset self-steers during curving.

- The contact point and load is always the same, whereas in the field different profiled wheels in a variety of worn conditions with different axle loads run on the same track.
- The longitudinal forces provided by a train's traction system will vary the slip level in the contact as the train changes its levels of braking/acceleration. This will cause changes to the wear and damage mechanisms/rates as the amount of slip changes. Whereas in lab tests, the longitudinal forces are controlled via determining the slip level and is kept constant for the duration of the test.
- The environment within the lab is relatively constant when compared to normal track conditions, which can vary greatly in time and location.
- Contamination of the wheel and rail, for example by leaves, ballast dust etc., has not been simulated in these lab tests.

The differences outlined above will result in a discrepancy between actual performance in the field and performance in the laboratory. However, these differences (in particular controlling the load and slip level in contact, and only using one wheel) are necessary in order to simplify the component being tested (in this case wheel/rail contact). This allows an increase in controllability of the tests in the laboratory and different parameters investigated (in this case the effect of FM on traction coefficient and wear rate). Whilst the differences will result in changes between the absolute values in the laboratory and the field, the trends and relationships are expected to be the same.

The ‘retentivity’ measured in these tests, could give an indication of product “carry down” and how durable it is, i.e., how many wheel passes occur before the effects of the product are no longer seen. Further work is required to prove these links. Unlike lubricants [23] there are no ‘certification’ tests to define the performance of a friction modifier. Therefore, if the ‘retentivity’ is shown to be linked to performance then these tests could form the basis of an approval process.

5 Conclusions

5.1 Study A

- Taking account of the assumptions made with respect to the full-scale data (contact patch size, traction coefficient, creepage) it can be said that reasonable correlation exists between small-scale and full-scale tests.
- For dry contact conditions, it can be seen that the full-scale data sits within the bounds of the twin disc data (see Figure 9).
- When friction modifier is applied, the full-scale wear rate is higher than in the twin-disc tests. This is due to proportionally more FM ending up in the contact in the twin-disc case, protecting the rail disc from damage.

5.2 Study B

- Absolute/baseline friction coefficients differ between twin disc (0.11) and full-scale (0.31) tests.
- Evolution of friction modifier traction coefficient shows similarities between the two test methods used.
- Further testing is needed to fully evaluate the retention in a full-scale contact. This would be done by increasing the number of cycles until the traction coefficient reaches 0.5
- The tests described in this paper could be used as a basis to define approval tests for FM's, there are currently no standards for approval for these products.

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