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Measurements of friction coefficients between rails lubricated with a friction modifier and the wheels of an IORE locomotive during real working conditions

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ABSTRACT

The real friction coefficients between the rails and the wheels on a 360 t and 10,800 kW IORE locomotive were measured using the locomotive's in-built traction force measurement system. The locomotive consisted of two pair-connected locomotives had a CoCo+CoCo bogie configuration, and hauled a fully loaded set of 68 ore wagons (120 t/wagon). The measurements were performed both on rails in a dry condition and on rails lubricated with a water-based top-of-rail (ToR) friction modifier on the Iron Ore Line between the cities of Kiruna and Narvik in Northern Sweden and Norway, respectively. Since full-scale measurements like these are costly, the friction coefficients were also measured at the same time and place using a conventional hand-operated tribometer, with and without the ToR friction modifier. The most important results are that the real friction coefficient is definitely not constant and is surprisingly low (0.10–0.25) when the ToR friction modifier is used, and that it is also significantly dependent on the amount of ToR friction modifier. A large amount will reduce the friction coefficient. Furthermore, it is concluded that the real friction coefficients are in general lower than the friction coefficients measured with the hand-operated tribometer. A final remark is thus that the use of a water-based ToR friction modifier can give excessively low friction, which can result in unacceptably long braking distances.

1. Introduction

The aim of this study was to measure the real friction coefficients between the wheels of a typical locomotive connected to ore transport wagons for heavy transports, and the rail, when a top-of-rail (ToR) friction modifier is used, and to measure the friction coefficients in the same conditions with a hand-held tribometer to ascertain whether such a simplified and less costly system is trustworthy.

In Sweden, the state rail infrastructure owner, Trafikverket, and the mining company LKAB are considering the eventual introduction of a ToR friction modifier in order to reduce the contact fatigue, cracks and related wear on the rail and wheels. This is extra important since it is planned that the traffic load will increase significantly in the near future. It is naturally suspected that a ToR friction modifier will also influence the friction coefficient and, because the friction between the wheel and rail will have a significant impact on the braking distances, both Trafikverket and LKAB consider it important to increase our knowledge of this topic. It is also well known to all tribology experts that there is almost no correlation between the friction coefficient and crack propagation, wear, etc. [1], or weak correlation [2], which increases the need to investigate the friction coefficient independent of the cracks and wear characteristics when using a ToR friction modifier. Beside the eventual advantage of ToR from the wear point of view, there is also advantages regarding noise reduction, see [3].

Traditionally the friction coefficient is measured in small laboratory rigs such as pin-disc or twin disc machines, see, for instance, [3], with full-scale laboratory rigs [4], or by means of hand-held field measurement tribometers, see, for instance, information from a commercial supplier [5]. Numerical models of twin disc test rigs have been developed for the evaluation of railway wheel wear prediction methods see [6]. However, the disadvantage of small laboratory rigs is that the scaling factor and the problem of achieving realistic surface roughness in the laboratory set-up, as well as the lack of realistic ambient parameters such as humidity, deposits from brakes, etc., will make the friction coefficients unrealistic and thus more or less useless. The scale

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factor is not a problem for full-scale test rigs, but the disadvantages of such rigs are both the huge associated costs and, as in the case of small test rigs, the difficulty of achieving proper surface roughness and correct ambient parameters; this difficulty makes the friction coefficients produced in full-scale rigs unrealistic too, even though they may be more realistic than the values obtained from small conventional laboratory rigs. Hand-held tribometers in the field have the advantage of considering the ambient parameters, as well as the surface roughness, but, on the other hand, the scaling factor can be a big problem. Thus it is of interest to measure the real friction coefficients directly in the locomotive to increase our knowledge of this subject. However, since this type of direct measurement is also very costly and difficult to arrange practically due to the real traffic conditions and the business needs of the ore traffic company, it would be of interest to ascertain whether a hand-held field tribometer can produce trustworthy results by comparing friction coefficients obtained with a tribometer with real friction coefficients.

ToR friction modifiers are today used by some railway owners, while others have rejected this approach due to the contradictory results that have been reported. Tests in the United States [7] and China [8] have found that there is a significant reduction in the wear of the rail and wheels when ToR friction modifiers are used, while, for instance, the German railway infrastructure owner, DB, recently reported contradictory results regarding both wear and excessively low friction causing braking problems. A large amount of research has been performed involving laboratory tests concerning wear and friction when ToR friction modifiers are used (see, for instance, [9–14]), but very little or no research has been conducted on friction coefficient measurements performed directly on a typical locomotive connected to a typically loaded set of wagons in real traffic.

2. Test site and ambient parameters

The measurement session was systematically planned so that it could be performed in exactly 30 min, since that was the time available due to the current traffic situation. The test was then executed exactly according to the plan on 25 March 2014 at the small village of Krokvik, which is situated about 15 km north of the city of Kiruna in Northern Sweden, on the Iron Ore Line between Kiruna and the city of Narvik in Norway. The air temperature was about $-5\,^{\circ}C$ and the temperature of the rail was $-4\,^{\circ}C$. The weather was bright with only small clouds in the sky. There was no snow and no other disturbing factors affecting the rail. A 330 m long section of rail was used as the test rail, located at a small inclination in order to make it more easy for the test train to achieve fully developed friction. The surface roughness of the test rail was not measured due to the high traffic intensity, leading to a shortage of available measurement time.

3. Thickness of the tested friction modifier

Our aim was that the tested ToR friction modifier should be applied on the rail as uniformly as possible, see Fig. 1, where Photo A was taken before lubrication and Photo B after lubrication.

However, in practice the thickness of the friction modifier varied. An estimation of the thickness was performed through optical measurement of a typical section of the lubricated rail before the locomotive approached, see Fig. 2.

Fig. 2 shows a typical thickness variation on a section width of 30 mm in the middle of the rail head. It can be concluded that the average thickness of the friction modifier was about 0.2–0.5 mm.

4. Technical properties of the friction modifier

The ToR friction modifier (Whitmore TOR Armor LT) used in this study is based on a glycol–water solution together with thickener, solid lubricant and additives and adapted to high demands regarding environmental friendliness. It was not possible to obtain any technical data from the supplier, e.g. the water content, thickener type, etc.

Chemical analyses were thus performed to make a detailed characterisation of the studied friction modifier. Representative samples were analysed with gas chromatography using a flame ionisation detector. The analysis revealed a paraffin-like material

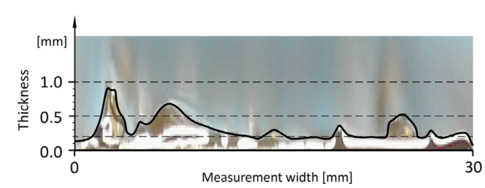

Fig. 2. Estimation of the thickness of the friction modifier.

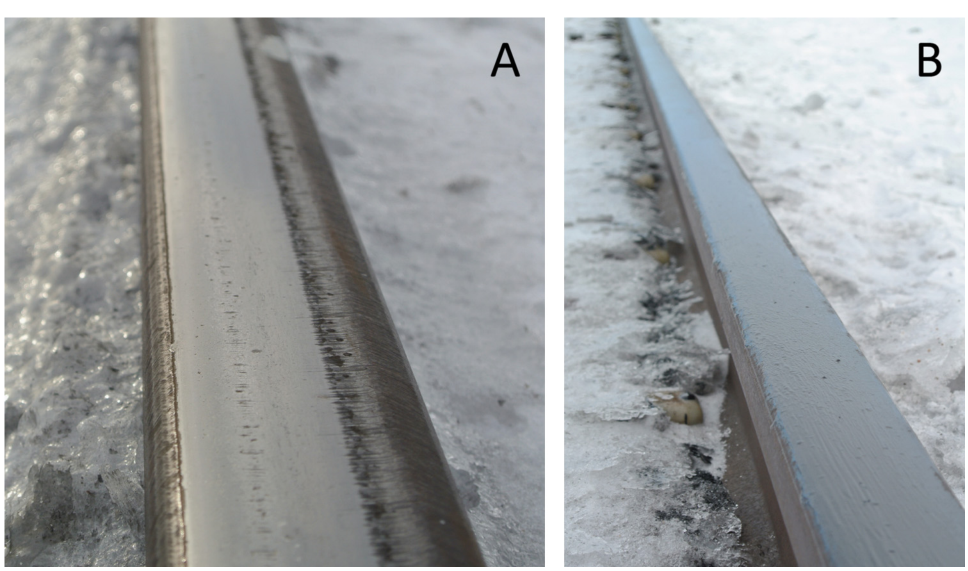

Fig. 1. Dry test rail (A) and lubricated test rail (B).

Table 1
Major (> 1 mg/kg) metal content in studied friction modifier.

Element	Al	As	Cd	Cu	Fe	Mn	Pb	V	Zn
Conc. (mg/kg)	510	1.2	4.8	11.6	343	43.3	8.0	1.4	3.4

consisting of saturated, aliphatic hydrocarbons, mostly in the C15–C40 carbon chain fraction, with respect to n-alkanes. Samples were also analysed for metals (Al, As, Cd, Co, Cr, Cu, Fe, Hg, Mn, Ni, Pb, V, and Zn) using ICP-SFMS (Inductive coupled plasma sector field masspectrometry) in order to ascertain metallic additives in the lubricant. Chemical analyses were performed at the laboratories of ALS Scandinavia AB in Luleå and Stockholm, Sweden. Analytical results for elements present in amounts higher than 1 mg/kg are presented in Table 1.

Fig. 3. IORE locomotive used for real friction coefficient measurements.

5. Friction measurements with the IORE locomotive

The new IORE locomotive, used by LKAB for transporting iron ore, was employed for the measurements of real friction coefficients, since this is the standard locomotive for these heavy transports and has an in-built measurement system capable of continually measuring the friction forces and sliding velocity of all of its four bogies. The locomotive automatically avoids uncontrolled slippage (sliding) and adjusts the delivered traction force according to the friction properties between the rail and the wheels. The measurement system is not capable of performing the measurements individually for each of the wheels, but on average for each of the four bogies. Each of the axes is driven by one electric motor and a measurement system is comparing the linear velocity of the locomotive with the peripheral speed of each of the wheels. If these speeds are not the same as the velocity of the locomotive, then the electric power to the individual electric motors will be adjusted in order to control the sliding. The first axes (in driving direction) of each of the bogies are given slightly more sliding then the rest of the wheels in order to improve the steering performance. This entire together, means that the individual sliding of each of the axis is unknown, but the average sliding speeds of each of the boogie is known. The average friction forces from each of the bogies are then presented in the cabin by means of calculating these forces based on the torque and the radius of the wheels. The fact that only the average friction force from each bogie is measured is a limitation, but the present authors are of the opinion that the method presented in the present paper is a novel and perhaps today the most realistic measurement method for real locomotives, since the mentioned drawback has to be compared with the drawbacks by using laboratory tests or down-scaled tribometers in the field. Due to the technicians at the locomotive work shop at company LKAB also the individual slippage of the individual axis in the bogies have been studied by means of using more inside information from the measurement system. These studies showed that in almost all of the cases, all axes in the bogies are always sliding when the system are reporting an average sliding for the whole bogie. The average sliding velocity (relative velocity between the wheel and the rail) during the tests was 0.15–0.3 m/s when slipping occurred for each of the bogies. The friction coefficient in lubricated conditions is often dependent on the sliding velocity. Increased sliding velocity (slipping velocity) will often increase the friction coefficient since the shear forces increases in the lubricant, see for instance [15], where the friction coefficient is measured as a function of sliding velocity. However, since the sliding velocity is almost constant at the locomotive and as low as possible, the influence of the sliding speed is neglected. The velocity and the acceleration of the locomotive were also recorded at the same time, as well as the difference between the driver-requested drag force and the required drag force due to slipping. In order to ensure that fully developed friction coefficients would be measured; only circumstances when sliding occurred were reported as friction coefficients. These circumstances led, in some cases, to only one bogie sliding and, in other cases, to two, three or all the bogies sliding at the same time.

The IORE locomotive was manufactured by Bombardier between the years 2000 and 2010 and consists of two pair-connected locomotives with a total of four bogies with three axles in each bogie (in a CoCo + CoCo configuration), the total power is 10,800 kW, the total starting force 1400 kN, the total length 45.8 m and the total mass 360 t, see Fig. 3.

During the measurements, the locomotive was connected to 68 wagons, each of which consisted of two bogies (with two axles in each bogie), was fully loaded with ore, and weighed 120 t. Thus the total weight of the train was 8520 t. The locomotive part at the front of the train, directly behind the driver's cabin, is denoted as 116 and its bogie closest to the front cabin is denoted as 116 B1, while its rear bogie is denoted as 116 B2. The rear locomotive part is mirror-connected to locomotive part 116 and is denoted as 108. The front bogie of locomotive part 108, closest to locomotive part 116, is denoted as 108 B2, while the rear bogie of locomotive part 108 is denoted as 108 B1.

Since the connexion point between locomotive parts 116 and 108 is 1.04 m above the rail, there will be a torque striving to increase the normal force on bogie 116 B2 slightly, and decrease the normal force on bogie 116 B1 slightly. By using torque and force equilibrium calculations, normal-force-compensating equations were performed on each of the calculations of the friction coefficients. For the rear part of the locomotive, this compensation could be totally neglected, since the difference between the drag forces on the two parts is small. With a centre-distance of 12.89 m between bogie B1 and B2 and the total drag force F, the compensating equations for each wheel in the front locomotive become as follows:

$$NB2 = F \times 0.0135 + 147.15 \tag{1}$$

where F = the total drag force between the two locomotive parts [kN] and NB2 = the compensated normal force on each of the wheels in bogie 116 B2 [kN], and

$$NB1 = 294.3 - NB2 \tag{2}$$

where NB1 = the compensated normal force on each of the wheels in bogie 116 B1 [kN].

Finally the friction coefficient μ in the length-direction of the rail is calculated as

$$\mu = \mathrm{Fd}/\mathrm{NB} \quad (3)$$

where Fd = the friction force, measured by the locomotive's in-built system, and NB = the compensated normal forces (NB1 or NB2).

The effect of the coupling forces on the second locomotive part on the normal force on the wheels can be totally neglected since the differences between the couplings forces (between the two locomotive parts and between the second locomotive and the rest of the train) on the second locomotive are very low. Thus this net force will give an insignificant torque.

The following three measurement cases were utilised.

5.1. *Case of the train before lubrication*

The tests in this case were performed in a dry condition without the ToR friction modifier on the 330 m long test rail. The speed was 13 km/h at the starting point and 20 km/h at the end point of the test rail because of the acceleration of the train. Full power was requested.

5.2. *Case of the train after lubrication, the first run*

The tests in this case were performed on the test rail after the first application of the ToR friction modifier. The friction modifier was applied on the rail by means of manual brushing, see Fig. 4.

The velocity of the train was initially 10 km/h at the starting point of the test rail and then continuously decreased to 1 km/h due to severe sliding, which led to the testing being stopped because of too low a velocity.

5.3. *Case of the train after lubrication, the second run*

The tests in this case were carried out after the locomotive on the train, having passed the ToR-lubricated test rail once (in the *case of the train after lubrication*, *the first run*), had moved backwards to a position 100 m before the starting point of the manually lubricated test rail. The locomotive then approached the test rail once again and continued along the test rail until the measurements had to be stopped because of a very low speed. This means that these measurements were performed both before the beginning of the test rail (on a rail section lubricated by friction modifier dragged onto the section by the locomotive) and within the test rail section initially lubricated manually with the ToR friction modifier.

Fig. 4. Method for applying the ToR friction modifier.

Fig. 5. The tribometer utilised for the friction coefficient measurements.

6. Friction measurements with a hand-held tribometer

The tribometer utilised was designed by a commercial company to measure ToR friction coefficients. It uses a spring-loaded wheel made of steel with a maximum top diameter of 89.0 mm (+/−0.1 mm), a width of 9.2 mm (+/−0.1 mm), and a radius of contact curvature of 29 mm (+/−1 mm). The steel wheel is connected to a magnetic clutch in such a way that the wheel is free to rotate the clutch. A manually adjusted variable resistor controls the clutch sliding. As the slippage is reduced, the resulting force is transferred to an analogue weight scale. By increasing the resistance of the clutch, the longitudinal rolling resistance of the wheel also increases. The friction at the top of the rail controls the point at which the wheel will slip. A scale then shows the force at which the wheel sliding has occurred, see Fig. 5.

When the operator reaches a steady walking speed, the tribometer starts a 3– s measurement sequence. At the end of each sequence, the friction coefficient of the rail at the desired location is displayed on the tribometer's digital read-out on the rail head. The wheel speed is determined by measuring the pulse duration generated by an optical encoder mounted on the support shaft for the measuring wheel. As the wheel speed increases, the duration or period of the pulse decreases. With all the initial conditions met, the main board's central processing unit (CPU) will begin a six step test cycle by applying a ramping braking force to the measuring wheel. The braking force is provided by an electromagnetic brake. An automatic ramping control circuit immediately senses the point at which wheel sliding occurs and automatically reduces the braking action to the measuring wheel to prevent the wheel from digging into the lubricant on the rail and generating artificially high friction readings. For more information, see data from the supplier [12]. Due to the instructions of the tribometer, it should be pushed with “walking speed”. During the experiments, the walking speed of the tribometer operator was 5 km/h (measured by using a known distance and measuring the time). Increasing or decreasing this speed by changing the walking speed was not resulting in any significant changes of the measured friction values. The following two measurement cases were utilised.

6.1. *Case of the tribometer before lubrication*

The tests in this case were performed in a dry condition without the ToR friction modifier on the test rail, just before the passage of the IORE train to ensure that the surface roughness would be unchanged by traffic.

6.2. *Case of the tribometer after lubrication*

The tests in this case were performed with the ToR friction modifier applied on the same test rail by means of brushing, and

Table 2
Measurement time steps.

Step	Case	Description
1	*Tribometer before lubrication*	The friction was measured using the tribometer.
2	*Train before lubrication*	Measurement of the forces in the bogie when travelling the test distance. After the measurement the train was reversed to a starting point about 500 m before the starting point M1.
3	*Lubrication*	Manual lubrication of the test rail (330 m) with the ToR friction modifier.
4	*Tribometer after lubrication*	Measurement of the friction coefficients with the tribometer on the test rail.
5	*Train after lubrication, first run*	Measurement of the forces in the bogie when travelling the test distance, from the starting point M1 of the test rail.
6 A	*Train after lubrication, second run (A)*	Measurement of the forces in the bogie when travelling the test distance from a point 100 m before the starting point of the test rail to the starting point of the test rail.
6 B	*Train after lubrication, second run (B)*	Measurement of the forces in the bogie when travelling the test distance from the starting point of the test rail. Thus this is a continuation of step 6 A.

Table 3
Case of the train before lubrication.

M	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Dist. [m]	123	124	128	132	136	140	144	149	154	158	162	166	192	219
116 B1	**o**	**–**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**–**	**–**	**–**	**o**	**o**
116 B2	**o**	**–**	**–**	**–**	**–**	**o**	**–**	**–**	**o**	**o**	**–**	**–**	**–**	**–**
108 B1	**–**	**–**	**–**	**–**	**–**	**–**	**–**	**–**	**–**	**–**	**–**	**–**	**–**	**–**
108 B2	**–**	**–**	**–**	**–**	**–**	**–**	**–**	**–**	**–**	**–**	**–**	**–**	**–**	**–**

Table 4
Case of the train after lubrication, the first run.

M	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
Dist. [m]	22	50	78	100	117	131		180	188	196	204	210	218	240	268	296
116 B1	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**
116 B2	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**
108 B1	**–**	**–**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**–**	**o**	**o**	**o**	**o**
108 B2	**–**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**

took place just before the measurements performed with the IORE locomotive's built-in measurement system, see Fig. 4.

6.3. *Summary of the measurement steps*

The measurement steps are summarised in Table 2.

7. Results and discussion

To start with, sliding tables were compiled, since this is fundamental to the definition of friction coefficients (fully developed friction). Accordingly, only fully developed real friction coefficients (sliding conditions) were reported for all the test cases with the in-built locomotive system. For the *case of the train before lubrication*, data for 21 measurement points were collected and no slippage in any bogie was noted for the measurement points M1–6, 8, 17, 18 and 21, see Table 2; M1 represents the starting point of the 330 m long test rail. One reason why it was mostly bogie 116 B1 that slides may be that the normal force was slightly less on this bogie, because the drag force between the torque of the two locomotive parts strove to decrease the normal force at bogie 116 B1 slightly. By using Eqs. (1) and (2), this torque will decrease the normal force by a maximum of 10%, approximately. In Tables 2–4, the following legend applies:

o=slippage, **–** =no slippage, Dist. [m]=distance in metres from the starting point of the test rail.

In Table 4, the results for the *case of the train after lubrication, the first run* are presented in the same way. Data for 17 measurement points were collected. Sliding occurred at all the points, except at the starting point M1, and it can also be concluded that sliding occurred for all the bogies at most of the measurement points, indicating that the friction is reduced by the friction modifier.

Table 5
Case of the train after lubrication, the second run, Part A.

M	1	2	3	4	5	6
Dist. [m]	−100	−93	−76	−57	−35	−8
116 B1	**o**	**o**	**o**	**o**	**o**	**o**
116 B2	**–**	**–**	**–**	**o**	**o**	**–**
108 B1	**–**	**o**	**o**	**o**	**o**	**o**
108 B2	**o**	**o**	**o**	**–**	**o**	**o**

In Table 5, the results for the *case of the train after lubrication, the second run*, *Part A* (i.e. the test sequence when the locomotive started at a point 100 m before the starting point of the test rail and rolled with the ToR friction modifier on its wheels) are presented in the same way. Data for six measurement points were collected. The minus sign before the distance indicates that the tests were performed before the manually lubricated test rail. Note that M1 here indicates the position 100 m before the starting point of the test rail.

Table 5 shows that bogie 116 B1 always slides and bogie 108 B1 and B2 slipped at five measurement points each, while bogie 116 B2 only slipped at measurement point 4 and 5. Thus it can be concluded that the friction modifier reduced the friction in this measurement case too.

In Table 6, the results for the *case of the train after lubrication*, *the second run*, *Part B*, when the locomotive ran on the manually lubricated test rail, are presented in the same way. Data for 14

Table 6
Case of the train after lubrication, the second run, Part B.

M	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Dist. [m]	0	25	60	82	101	118	132	143	151	157	163	166	169	175
116 B1	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**
116 B2	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**
108 B1	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**
108 B2	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**	**o**

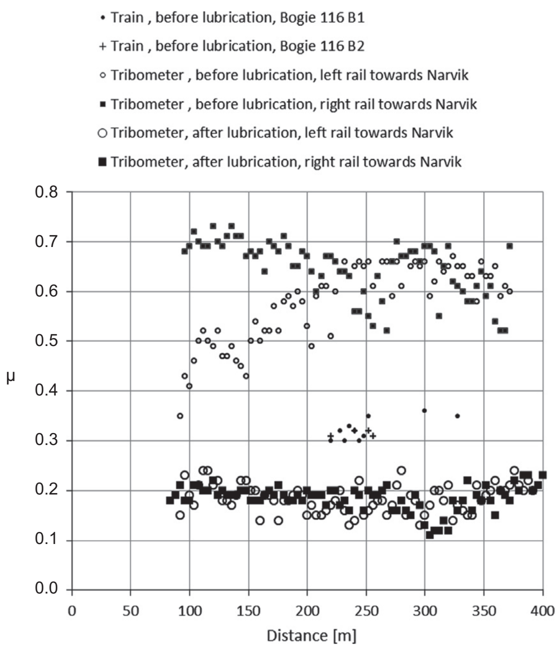

Fig. 6. Friction coefficients as a function of the distance on the rail.

measurement points were collected. It is clearly seen that slippage occurred for all the bogies at all the measurement points.

Table 6 shows clearly the effect of the amount of friction modifier, since all the bogies here slipped all the time.

Fig. 6 shows the friction coefficients as a function of the distance [m]. The starting point of the test rail is defined as being located at a distance of 100 m. The small black squares represent the results for the dry condition without ToR friction modifier, measured with the hand-operated tribometer on the right rail in the direction of the city of Narvik in Norway. The small unfilled circles represent the corresponding results measured for the left rail. The following is the complete legend applicable to Fig. 6.

The measurements presented in Fig. 6 started with measurements performed with the tribometer at a distance point at about 80 m on the left rail and with the measurement wheel cleaned with a cleaning solvent. It can be seen that the friction coefficient was about 0.35 for the left rail at the starting point and then increased up to about 0.6–0.7 at the end for both rails. Since approximately the same value of the friction coefficient was then obtained on the right rail at the end of the test rail, it is reasonable to assume that the discrepancy between the right and left measurements is mainly due to the initial cleaning with the solvent, which significantly reduced the friction coefficient, and the gradual build-up of a third body consisting of deposits from the train or oxides from the rail, which resulted in an increase of the friction coefficient.

The results for the *case of the train before lubrication* are shown in Fig. 6 represented by small filled circles and + signs, showing the real friction coefficients for the dry condition measured on the locomotive in the same test rail section (bogie 116 B1 and 116 B2, respectively). These measurements also started at the starting point of the test rail (a distance point at about 80 m in Fig. 6) and continued until 400 m. Slippage (a relative velocity between the wheel and the rail) occurred between the distance points 219 m and 330 m for locomotive part 116 and at its bogie, 116 B1 (small filled circles), as well as some specific points for bogie 116 B2 (+ sign). The rear locomotive part, 108, did not slide at all, see also Table 2. This indicates that mostly the friction was partially developed, and thus no clear conclusions can be drawn in this specific case regarding the correlation between the real friction coefficients and the measurements performed with the hand-operated tribometer. However, since bogie 116 B1 slipped through almost half of the whole test rail, it is reasonable to assume that the real friction coefficients are generally lower that the friction coefficients measured with the hand-operated tribometer.

The large filled squares and the large unfilled circles represent the *case of the tribometer after lubrication*, which is the case where the ToR friction modifier had been applied on the rails (the large unfilled circles represent the left rail and the large filled squares represent the right rail), and where the measurements were performed with the hand-operated tribometer. It can clearly be seen that the friction coefficient has decreased significantly to approximately 0.18 and is also relatively constant on average. The variation of the friction coefficient is also somewhat smaller than that in the *case of the tribometer before lubrication*. According to the company manufacturing the ToR friction modifier, the friction coefficient should always be about 0.3, unless the amount of modifier is too large or too small. The results indicate that too much modifier was applied in the test, but then the following important questions arise. What is the correct amount of modifier, and how did the manufacturer of the friction modifier conduct the measurements of the friction coefficient resulting in a value of 0.3?

The next step was to measure the friction coefficients on the same ToR-lubricated rail section directly using the locomotive's in-built measurement system; this corresponds to the *case of the train after lubrication, the first run*, see Fig. 7. The following legend applies to Fig. 7.

The filled curves shown in Fig. 7 are composed of straight lines drawn between the tribometer measurements obtained after lubrication (according to Fig. 6), to make comparisons with the real friction coefficients measured by the system built into the locomotive. The small filled circles represent the results for the friction coefficients for locomotive part 116, bogie 116 B1. The value was about 0.03 at the beginning of the test (130 m) and then almost steadily increased to 0.18 at the end. It can be seen that this bogie (116 B1) has the lowest average friction coefficients compared with all the other bogies. The dependency of the friction coefficient on the sliding distance is not surprisingly, it is valid for many machine elements; see for instance [2], where the friction coefficient for brakes is found to depend significantly on the braking distances. A possible explanation is that bogie 116 B1 was the first bogie to approach the friction modifier and thus faced the largest amount of friction modifier, giving an average friction

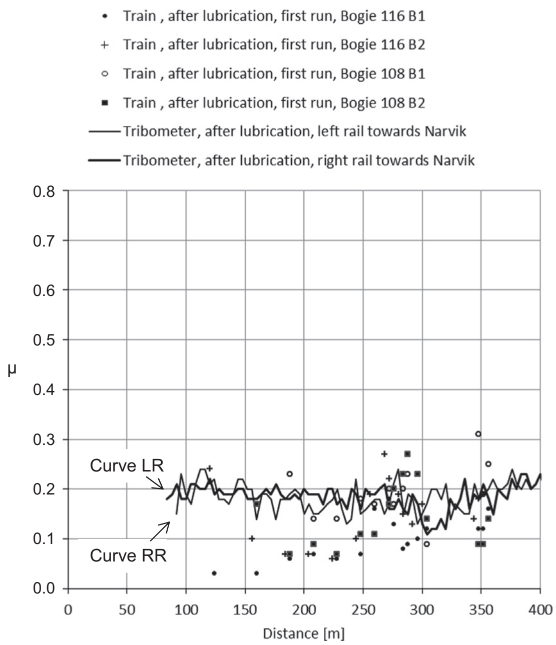

Fig. 7. Friction coefficients in the lubricated condition as a function of the distance.

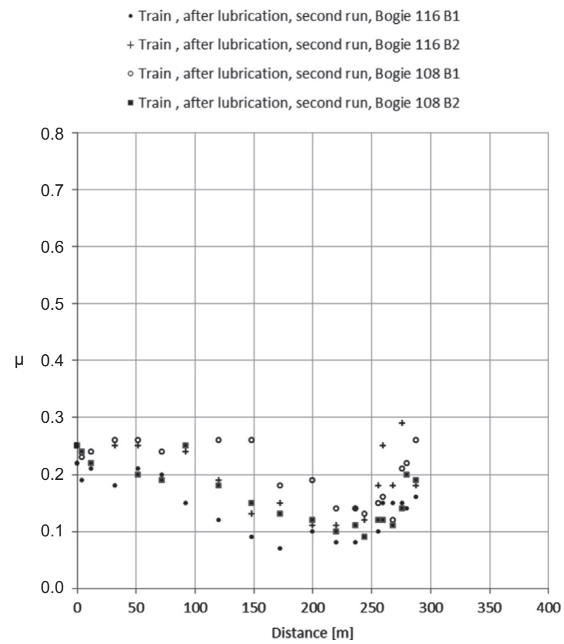

Fig. 8. Real friction coefficients as a function of the distance on the rail.

coefficient of only about 0.10. However, more research is needed to explain the increase in the friction coefficient as a function of the travelled distance and the amount of lubricant. The + sign, and filled squares represent the corresponding results for locomotive part 116, bogie 116 B2, and locomotive part 108, bogie 108 B2, respectively. These measurements indicate, in comparison with those for bogie 116 B1, a much larger variation (+/−0.1) and average values that are somewhat higher (0.14, and 0.15, respectively). Especially for 108 B1, the variation in friction coefficients is too high. The large variation is also significantly larger than the variation of the measurements obtained with the hand-operated tribometer. Moreover, the overall values of the real friction coefficients are significantly smaller than the friction coefficient values (0.18) obtained with the hand-operated device. The discrepancy between the real friction coefficients and the friction coefficients measured with the hand-operated tribometer is not surprising and there are several reasons for this discrepancy. *Firstly* the scale factor is very significant, which makes the influence of the border effects of the wheels much more dominant in the case of the hand-operated device. The border effects from the comparatively small wheel on the hand-operated tribometer will increase the friction force on the small wheel and thus increase the friction coefficient. Scaling is a very important factor; see [16] where this is experimentally concluded for unlubricated sliding contacts. *Secondly* the maximum Hertzian pressure in the contact point (calculated by using the Hertz contact pressure theory in combination with the development of theory by Hamrock [17]) is much smaller for the hand-operated tribometer (1000 MPa) compared with the Hertzian pressure for the real locomotive wheels [18] (1500–2700 GPa) and it is well known that the friction coefficient increases when the Hertzian pressure decreases, see, for instance, [1,2,15,19].

In Fig. 8, the symbols unfilled circle, filled square, small filled circle and + sign represent the results for the *case of the train after lubrication*, *the second run* for locomotive part 108, bogie 108 B1 and bogie 108 B2, and locomotive part 116, bogie 116 B1 and bogie 116 B2, respectively.

The two parts of the measurements in this case can be recognised in Fig. 8. The points at distances from 0–100 m show the *case of the train after lubrication*, *the second run*, *Part A* (see also Table 1). In *Part A* the locomotive, having passed the ToR-lubricated test rail once (see Fig. 6) and having moved backwards to a position 100 m *before* the starting point of the test rail, then travelled to the starting point of the test rail. This sub-case then represents the condition when the ToR friction modifier, which was to some degree attached to the wheels of the locomotive (since it had already passed the lubricated test rail), had been spread by the wheels over the initially dry rail. It can be seen that the friction coefficient is on a relatively constant level (0.17–0.26) until distance point 100 m, compared with the curves between the distances 100 m and 300 m, and also that the first bogie on the leading locomotive part (bogie 116 B1) has the lowest friction coefficient (0.20). It is reasonable to assume that this bogie, because of its position, faced the largest amount of friction modifier, resulting in lower friction.

Between the distances 100 m and 300 m, when the locomotive is within the manually ToR-lubricated area (the *case of the train after lubrication*, *the second run*, *Part B*) (see also Table 2), it is interesting to note the clear shape of all the curves, which more or less follow each other. The general dip of the friction coefficients between 100 m and 300 m is most probably due to the larger amount of friction modifier in this area, but the reason for the shape with the clear minima at about 230 m in the curvature is harder to explain. One possible reason is the surface roughness of the rail, but more research is needed to give a more accurate explanation. By comparing these real friction coefficients with those from the hand-operated tribometer (see Fig. 7), it can be concluded that the hand-operated device utilised is not capable of delivering reliable data. In many cases the error can be as much as 100%. The average values of all of the measured friction coefficients are shown in Table 7, where "Real" signifies the real friction coefficients measured with the locomotive's in-built system and "Tribo" the friction coefficients measured with the hand-operated tribometer (the average for the left and right rails together, except in the case of the dry condition, where only the average for the right rail is considered).

Table 7 and Figs. 6–8 clearly show that the real friction coefficients in general are significantly lower than the friction coefficients measured with the hand-operated tribometer (a factor of 2.2 lower for the dry case and a factor of 1.2 lower in the

Table 7
Summary of the averages of all the measured friction coefficients.

	Non-lubricated dry condition Tribo: **0.65**	**After lubrication, first run** Tribo: **0.18**	**After lubrication, second run (A)** Tribo: not measured	**After lubrication, second run (B)** Tribo: not measured
Front locomotive 116, 116 B1	Real: **0.29** when slipping	Real: **0.10**	Real: **0.20**	Real: **0.12**
Front locomotive 116, 116 B2	Real: **0.32** when slipping	Real: **0.14**	Real: **0.25**	Real: **0.18**
Rear locomotive 108, 108 B2	Real: no slipping	Real: **0.15**	Real: **0.22**	Real: **0.13**
Rear locomotive 108, 108 B1	Real: no slipping	Real: **too high variation**	Real: **0.25**	Real: **0.19**
Average for the locomotives in the slipping condition	**0.30**	**0.13**	**0.23**	**0.16**

lubricated case), and also that the tribometer is almost incapable of measuring accurate tendencies of the values (see Fig. 7). The tested hand-operated tribometer is designed in such a way that it tries to measure the friction coefficient just after sliding occurs, which can partly explain the high values compared with the real values, but there are many other reasons why down-scaled tribometers produce excessively high friction values. It is also clear that the real friction coefficient was unacceptably low when the friction modifier was used, at least when too much modifier was applied on the rail. In the cases where the rail was lubricated with friction modifier, this caused severe problems during the tests in that the locomotive had great difficulty in starting to move from the starting point on the test rail. This can, of course, be catastrophic from the point of view of the braking distance and these problems occur in both *Part A* and *B* of the *case of the train after lubrication*, *the second run*, see Table 6. This is in accordance with findings obtained by the German rail owner, DB, which recently reported problems with the braking distance at some sites when a ToR friction modifier was used. However, since the slippage between the rail and wheels is larger in the case of braking, compared with the sliding in this study, it is not clear to what degree the braking distances are affected. Since the layer of the friction modifier was as thick as 0.2–0.5 mm initially, it can be argued that this is the reason for the unacceptably low friction coefficients in this study. This can partly be confirmed by studying Table 6 and comparing the results for *Part A* and *B* of the *case of the train after lubrication*, *the second run*, since the layer was much thinner in *Part A* compared with *Part B*. On the other hand, it can be concluded that the friction coefficient was still unacceptably low in *Part A*, even though the 100 m long rail section before the test rail had not initially been lubricated and the only lubrication that occurred was achieved by the wheels of the train dragging friction modifier across that section.

8. Conclusions

The following conclusions can be drawn concerning the use of water-based ToR friction modifiers:

- The real friction coefficient seems to be highly dependent on the amount of applied friction modifier; too much modifier results in unacceptably low friction coefficients (on average 0.13–0.16).
- Conditions when the thickness of the layer of friction modifier is low also seem to result in unacceptably low friction coefficients (on average 0.23).
- The tested hand-operated down-sized friction coefficient measuring equipment (tribometer) generally measures excessively high friction coefficients compared with the real values (which were lower for the dry case by a factor of 2.2 and lower in the lubricated case by a factor of 1.4) and is not capable of indicating accurate tendencies.
- More research is needed regarding the optimal amount of friction modifier and the different types of ToR friction modifier. Other issues that require investigation include the length of the stretch of rail that will be affected by friction modifier, and the effect of friction modifier on the braking distances of trains and the wear of wheels and rails.

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