# WHEEL-RAIL CONTACT: EXPERIMENTAL STUDY OF THE CREEP FORCES-CREEPAGE RELATIONSHIPS

A. Alonso*[1]), A. Guiral[1]), L. Baeza[2]), S.Iwnicki[3])

*1) CEIT, San Sebastián, Guipúzcoa, Spain;*

*2) CITV - Universidad Politécnica de Valencia, Spain;*

*3) University of Huddersfield, U.K;*

## Abstract

The wheel-rail contact problem plays an important role in the simulation methods used to solve railway dynamics problems. As a consequence, many different mathematical models have been developed in order to calculate wheel-rail contact forces. However, most of them tackle this problem from a purely theoretical point of view and need to be experimentally validated. Such validation could also reveal the influence of certain parameters not taken into account in the mathematical developments. This paper presents the steps followed in building a scaled test-bench to experimentally characterize the wheel-rail contact problem. The results of the longitudinal contact force as a function of the longitudinal creepage are obtained and the divergences with respect to Kalker's Simplified Theory are analysed. The influence of lateral creepage, angular velocity and certain contaminants such as cutting oil or high positive friction modifier is also discussed.

## 1. Introduction

The accuracy obtained when solving most railway dynamics problems depends greatly on the modelling of wheel-rail contact. For this reason, in recent decades several research projects have been carried out in order to develop precise mathematical models that allow the forces transmitted through the contact area to be obtained. Among these projects, the ones of Kalker [1], Polach [2], Chollet [3] and Alonso [4] are particularly noteworthy.

One fact that stands out when analysing the different published papers on the wheel-rail contact problem is that most of them address the problem from a purely theoretical point of view. The causes of this fact can be the high degree of complexity of obtaining experimental measurements. As a result, there are some uncertainties in the development of wheel-rail contact models.

The overall objective of this work is to gain experimental knowledge regarding the wheel-rail contact problem. To this end, a test-bench was designed and constructed. Subsequently, the test-bench was used to experimentally obtain the relationships between creep forces and creepage and their dependence on certain factors such as angular velocity, surface conditions, etc. Finally, the divergences between the experimental results and the theoretical ones were analysed.

## 2. Wheel-Rail Test-Bench Development

In the framework of this project, a test-bench was designed and developed in order to analyse the contact conditions that are produced in a real railway vehicle and replicate them in a scaled manner (see Figure 1). The use of the scaled test-bench is driven by the high cost of reproducing the real conditions in a laboratory due to the large dimensions, high forces and power required. The scale of the different magnitudes (forces, lengths, etc.) should be determined in such a way that the conditions obtained in the test-bench are as similar as possible to the conditions that exist in a real railway vehicle.

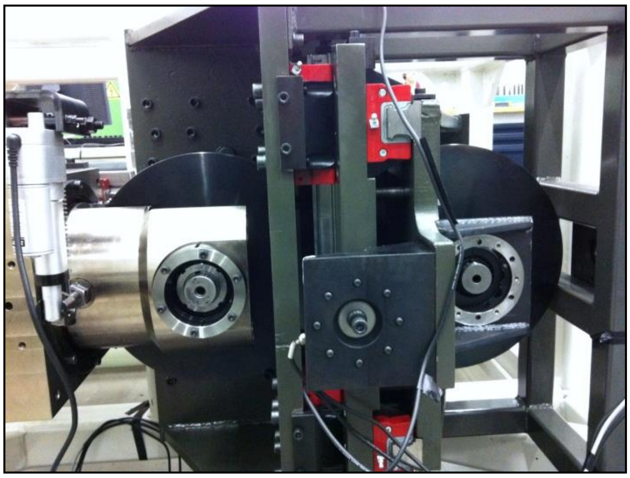

*Figure 1. Test-bench.*

The scale determination criterion has been established in such a way that the friction coefficient in the test conditions has a value that is similar to the one existing in the track. It is known that the friction coefficient between two bodies depends on, among other factors, normal pressure, tangential tractions, slip velocity, surface conditions, etc. Following this criterion, the following relationships between the different parameters were obtained

| ***Parameter*** | ***Relation*** |
|---|---|
| Lengths: | $L_R = \Psi \cdot L_S$ |
| Forces: | $N_R = \Psi^2 \cdot N_S$ |
| Angular velocity: | $\Omega_R = \Omega_S / \Psi$ |

*Table 1. Relation between the real parameter, 'R', and the scaled parameter 'S'.*

The scale parameter determination requires the definition of real reference conditions. In this case the wheel is supposed to have a radius of 425 mm, the contact is assumed to take place on the conic zone, the rail has a curvature of the head profile of 300 mm and the normal load is assumed to be 90 kN. Moreover the following aspects should be considered:

- The power supplied by the engines is critical from a cost perspective. If the power of the motor engine is fixed the maximum achievable theoretical velocity is also fixed.
- The larger the scale parameter, the higher the maximum achievable velocity. Therefore, the larger the scale parameter, the better the friction coefficient’s dependence on the velocity can be evaluated.
- The contact area decreases by the square of the scale parameter. A large scale parameter value would lead to an excessive reduction in the contact area dimensions moving away from real conditions.

Considering all these aspects, the scale has been given a value of 5. If each motor is able to supply 75 kW, the maximum achievable velocity will be 125 km/h. In order to achieve higher velocities, the motor should be replaced in order to supply greater power.

## 3. Results

This section presents the experimental results. The aim is to study the influence of the parameters listed below on the wheel-rail contact forces:

- Longitudinal creepage
- Lateral creepage
- Angular velocity
- Surface conditions

The results have been compared with the theoretical results given by FastSim [5]. The cleanness of the roller surfaces is assured by means of the application of isopropyl alcohol, which eliminates possible unwanted substances. Under these clean conditions the friction coefficient value is 0.6.

The procedure followed in order to carry out the experiments is basically divided into three steps: application of the normal load, specification of the angular velocity and introduction of the longitudinal relative slip between rollers. The latter can be introduced by means of two types of control: brake torque control (TC), which is used for the evaluation of small longitudinal creepage values (up to 0.4%), and velocity control (VC), which introduces an angular velocity difference between rollers and allows the higher longitudinal creepages to be evaluated (up to 2%).

### *1.1. Longitudinal Creepage*

The first set of experiments evaluates the behaviour of the longitudinal force as a function of the longitudinal creepage when the lateral creepage and spin creepage are zero. The rollers are pressed together with a normal load of 2.1 kN and the angular velocity is set to 500 rpm.

Figure 2a shows the longitudinal force as a function of the longitudinal creepage. Additionally, theoretical results obtained with FastSim have been included. The tangential force, which in this case is reduced to the longitudinal force, can be normalized by means of the normal load and the friction coefficient as shown in Figure 2b.

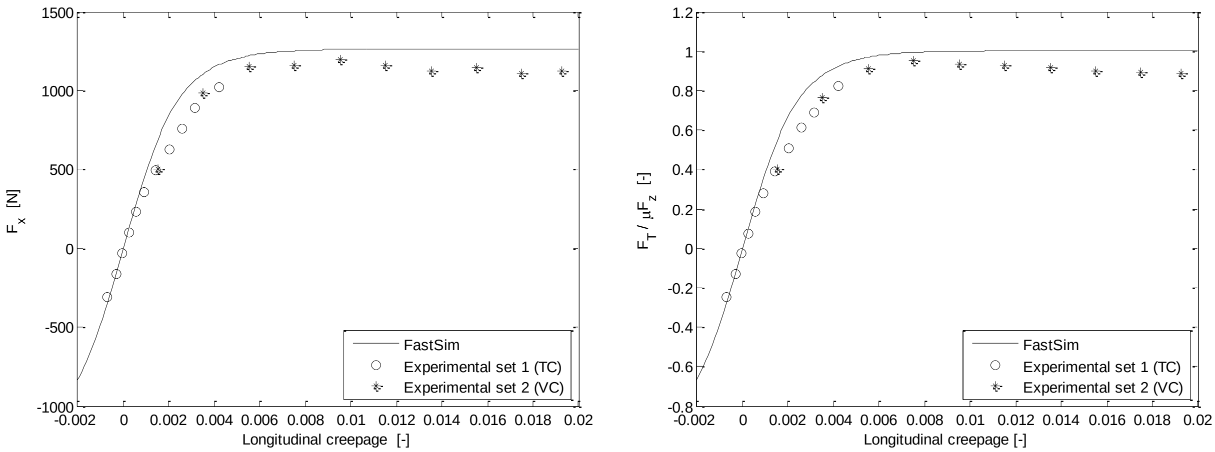


*Figure 2. Theoretical-experimental comparison of a)the longitudinal contact force and b) the non-dimensionalized tangential force vs. longitudinal creepage. $F_z$ = 2.1kN, Ω=500rpm, $\mu$=0.6.*

For small longitudinal creepage values the theoretical results are very close to the experimental ones. For higher values, in the non-linear zone, the theoretical predicted force is higher than that experimentally obtained. In the saturation zone, after a maximum force value is reached, a decay is observed. This reduction may be related to the falling of the friction coefficient for high slip; an effect, that is not taken into account in the simplified theory. However, it is worth pointing out, that the falling of the friction coefficient seems to be almost negligible.

### *1.2. Lateral Creepage*

The lateral creepage between the two rollers is directly related to the angle of attack. In this section, the angle of attack of the left roller will be varied in a discrete manner. The roller on the left (see Figure 1) is attached to a mechanism that has a linear actuator that is able to the angle of attack continuously. The design of this piece is complex since the roller rotates in two directions. Moreover, the desired measurable variation of the lateral creepage is on the order of 0.1 mrad; thus the mechanism must be sufficiently robust to avoid deviations of more than 0.01 mrad.

In the following figures the rollers are pressed together with a normal load of 2.3 kN and the angular velocity is set to 500 rpm. The friction coefficient between surfaces is $\mu$=0.5. As in the previous section, the longitudinal creepage will be varied by means of break torque and angular velocity control, but now three different angle of attack values will be evaluated: 0.37, 1.66 and 2.36 mrad.

*Angle of attack 1: 0.021º = 0.37 mrad*

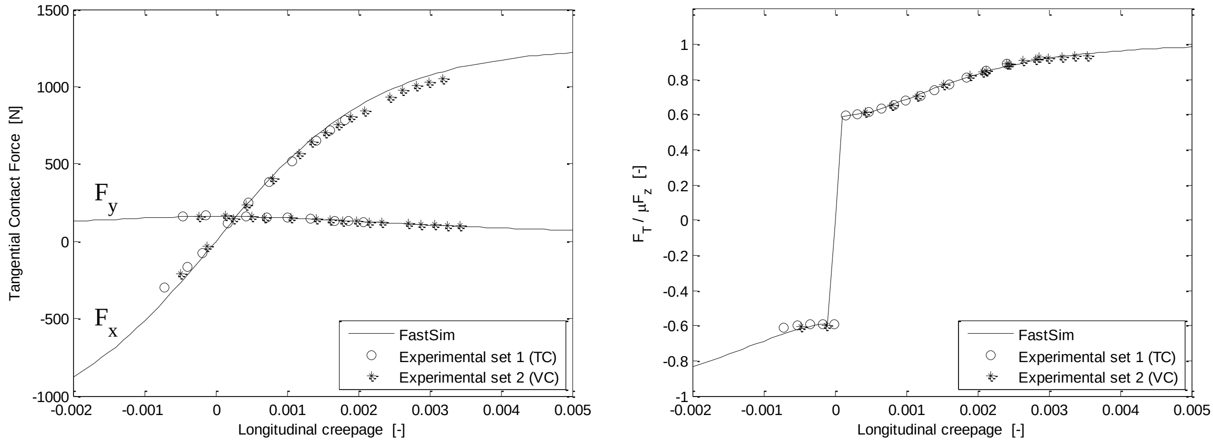


*Figure 3. Theoretical-experimental comparison of a)the tangential contact forces and b) the non-dimensionalized tangential force vs. longitudinal creepage for an angle of attack of 0.37 mrad.*

*Angle of attack 2: 0.095º = 1.66 mrad*

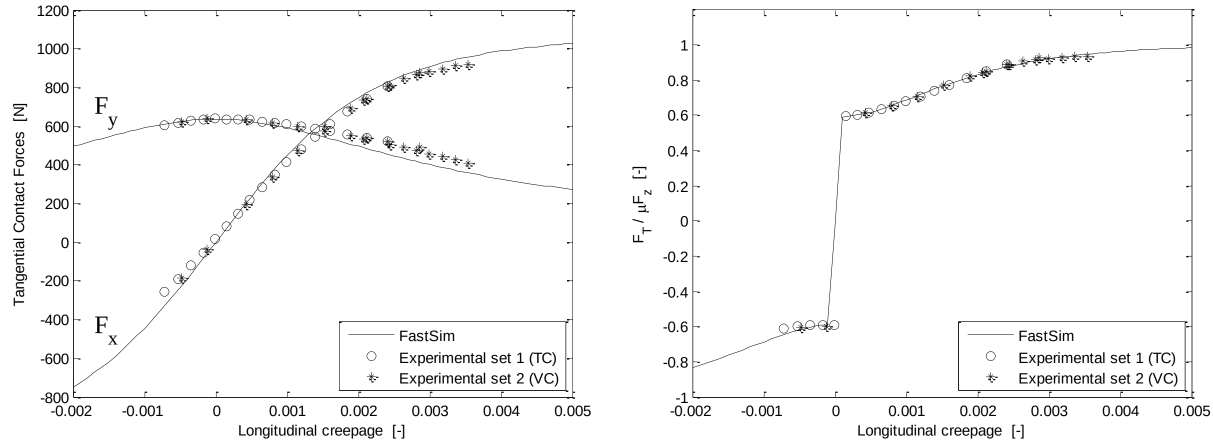


*Figure 4. Theoretical-experimental comparison of a)the tangential contact forces and b) the non-dimensionalized tangential force vs. longitudinal creepage for an angle of attack of 1.66 mrad.*

*Angle of attack 3: 0.135º = 2.36 mrad*

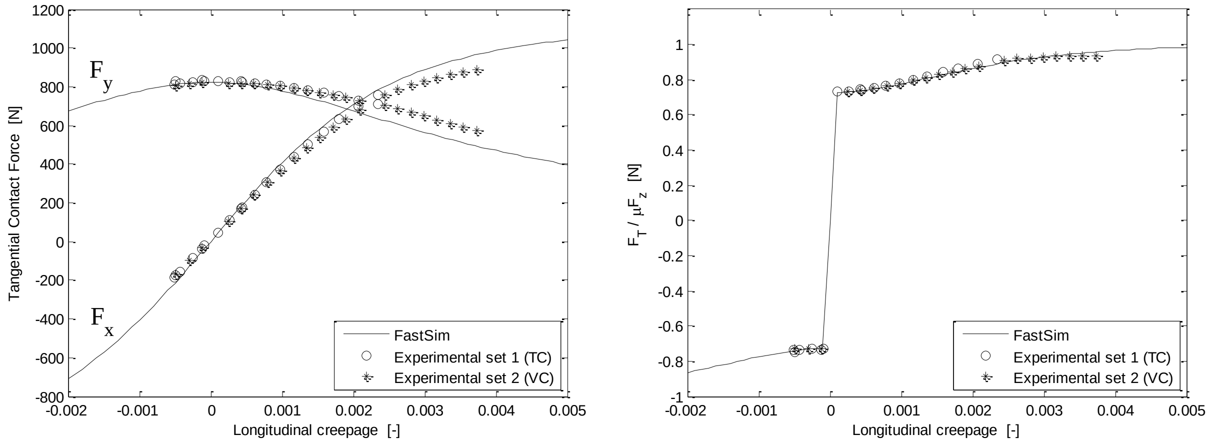


*Figure 5. Theoretical-experimental comparison of a)the tangential contact forces and b) the non-dimensionalized tangential force vs. longitudinal creepage for an angle of attack of 2.36 mrad.*

Figure 3, Figure 4 and Figure 5 present the theoretical-experimental comparison of the tangential contact forces and the non-dimensionalized tangential force as a function of the longitudinal creepage for the discrete values of the angle of attack. For a zero longitudinal creepage the lateral contact force presents its maximum value. The higher the angle of attack is, the higher the lateral force peak value and the discontinuity of the non-dimensionalized contact force are. As the longitudinal creepage increases the longitudinal contact force increases and the lateral force decreases.

In the linear zone, the experimental results coincide with the theoretical prediction. Nevertheless, as was pointed out in the previous section, a small divergence is presented as the longitudinal creepage is increased. The divergence between the theoretical and experimental results is presented for both tangential forces – longitudinal and lateral force – and becomes more noticeable as the angle of attack increases.

### *1.3. Angular Velocity*

The tangential contact force depends on the material properties of the wheel and the rail, the creepages and the friction coefficient between surfaces [6]. The material properties, as well as the creepages, do not depend on the rolling velocity; however, the friction coefficient is thought to vary with the sliding velocity. Thus, as the creep level increases beyond the saturation point the creep force reduces due to the decay in the friction coefficient value. A common assumption made by the wheel-rail contact theories is to neglect the dependency of the friction coefficient on rolling velocity. The entire problem then becomes independent of the rolling velocity. The simplification of the constant friction coefficient could be adopted in the solution of some problems where the wheel-rail creepage is small. However, at high slip the falling characteristic of the friction coefficient is believed to play a significant role in critical railway dynamics problems such as the simulation of the reduced radius curve negotiation, traction torque control on start up, squeal noise, the determination of dynamic instability conditions, etc. [7]. Nevertheless, the experimental results for the longitudinal creep force have shown no evidence of an appreciable decrease in the friction coefficient. Therefore, it can be stated that the assumption of the lack of dependency of the tangential forces on the angular velocity is correct, not only for small creepages but also for larger values of the relative slip between wheel and rail.

Figure 6 shows the results of the longitudinal contact force when the rollers are pressed together with a load of 2.39 kN and a friction coefficient of $\mu$=0.60. Two different angular velocities are tested: 375 and 500 rpm.

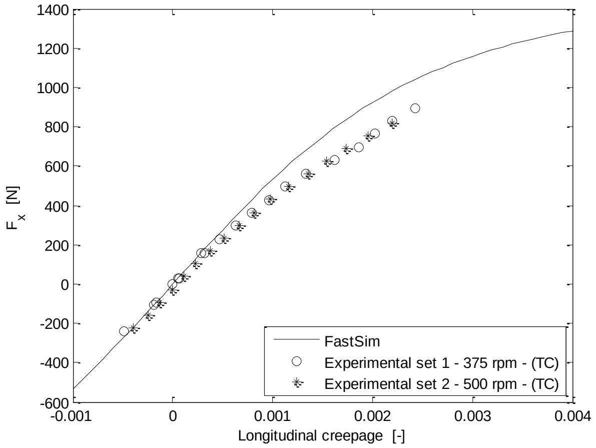


*Figure 6. Theoretical-experimental comparison of the longitudinal contact force vs. longitudinal creepage for two different angular velocities: 375 rpm and 500 rpm. $F_z$ = 2.39kN.*

No difference between the two experimental sets is observed. It can be concluded that, as predicted by the theoretical models, for small slip values the problem is velocity independent. Nevertheless, experimental results for higher values of the longitudinal creepage should be obtained.

### 1.4. Surface conditions

The condition of wheel and rail surfaces plays an important role in the wheel-rail contact problem. Typical contaminants that can interfere at the wheel-rail contact interface are, among other things, water, sand, leaves, grease, oil and friction modifiers. All these contaminants influence the friction coefficient and thus the creep forces values. This section presents some initial results of the tangential contact forces under the influence of interface materials such as friction modifiers, oil and sand.

*Cutting oil*

The surfaces conditions were changed by adding cutting oil to the roller surfaces. Two different friction coefficient values were tested: $\mu$=0.15 and $\mu$=0.45. A thin film of oil was spread onto the profiles at the beginning of the experiments, an amount that was enough to keep the friction value constant during the whole experiment. In both cases the rollers were pressed together with a normal load of 2.3 kN and the angular velocity was set to 500 rpm.

Figure 7 shows the theoretical-experimental comparison of the tangential contact force for the lowest friction coefficient value: μ=0.15.

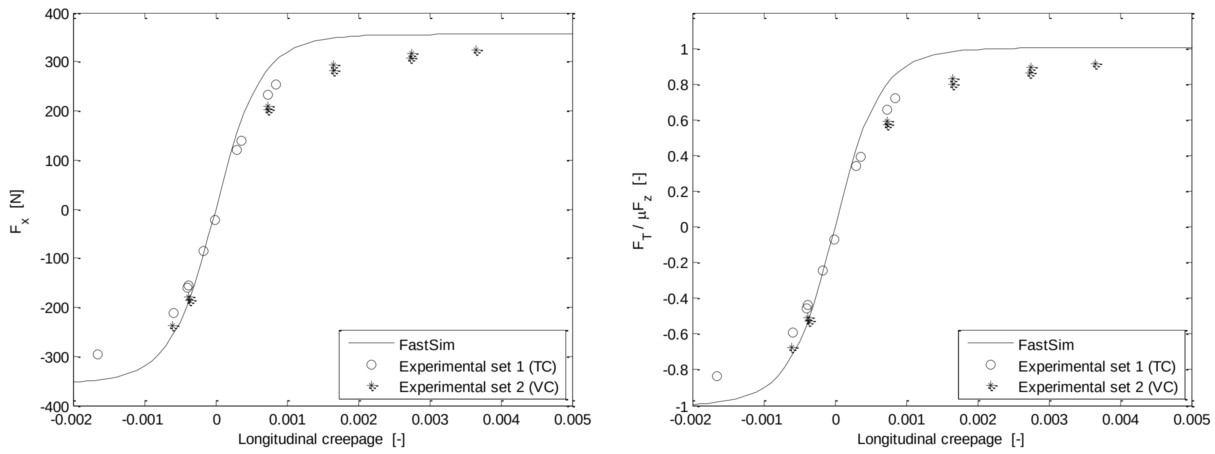


*Figure 7. Theoretical-experimental comparison of a)the longitudinal contact force and b) the non-dimensionalized tangential force vs. longitudinal creepage. $F_z$ = 2.37 kN, Ω=500rpm, μ=0.15.*

As shown, the behaviour of the longitudinal contact force is similar to that in the dry conditions. The theoretical results are in accordance with the experimental results in the linear zone, but show small differences in the non-linear zone. The maximum longitudinal force reached is 323.4 N.

The same experiment was carried out with a higher friction value (μ=0.45). This time, however, higher slip values are tested (see Figure 8).

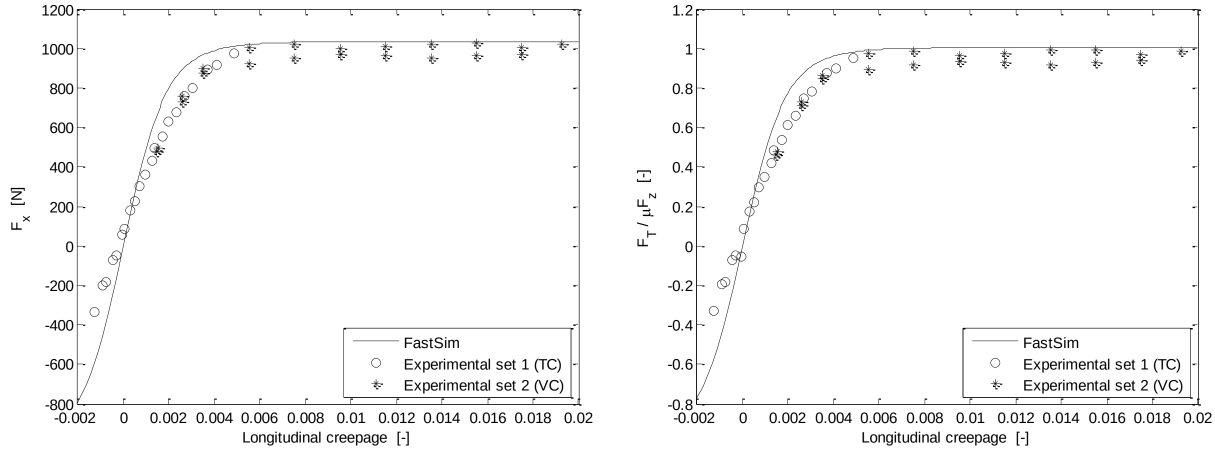


*Figure 8. Theoretical-experimental comparison of a)the longitudinal contact force and b) the non-dimensionalized tangential force vs. longitudinal creepage. $F_z$ = 2.29kN, Ω=500rpm, $\mu$=0.45.*

As expected, the behaviour of the longitudinal contact force does not change. However, it is worth pointing out that no decay of the contact force is observed for high slip when using cutting oil.

*High positive friction modifier (HPF)*

The use of friction modifiers is rapidly increasing due to their advantages when dealing with problems such as squeal noise mitigation, damaging lateral force reduction, wear, corrugation or rolling contact fatigue reduction [8]. These products control the friction level at the wheel-rail interface, providing an intermediate friction coefficient of typically around 0.35. Moreover, they increase the friction coefficient value while increasing creepage values to mitigate the effects of the negative creep slope, which is said to be responsible for the stick-slip mechanism generation leading to squeal.

In order to evaluate the wheel-rail contact forces when HPFs are used a water-based friction modifier was used. It was spread onto one of the rollers and transferred to the other making them roll slowly. The water was evaporated and a dry thin film formed on the roller surfaces.

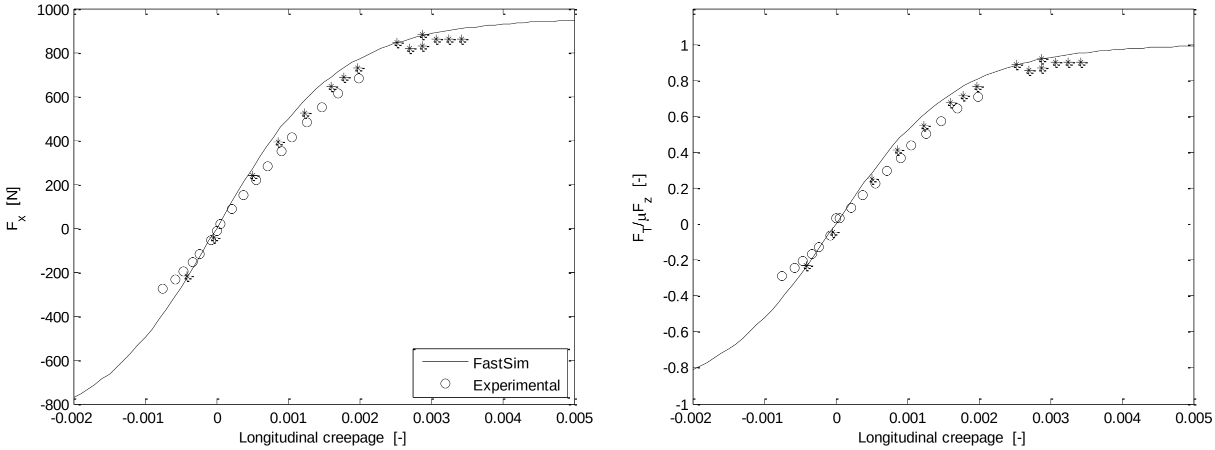


*Figure 9. Theoretical-experimental comparison of a)the longitudinal contact force and b) the non-dimensionalized tangential force vs. longitudinal creepage. $F_z$ = 2.39 kN, Ω=500rpm, $\mu$=0.40.*

Figure 9 shows the longitudinal and non-dimensional tangential contact force as a function of the longitudinal creepage when a normal load of 2.39 kN is applied and the angular velocity is set to 500 rpm. When applying HPF, the friction coefficient reduces from 0.60 (dry conditions) to 0.40, which is in the range specified by the manufacturer [8]. In order to evaluate the positive friction characteristics, experimentsat higher slip level should be carried out.

*Sand*

A common contaminant that can be found on the wheel-rail interface is sand. The reason is that the application of sand to the wheel-rail interface is an effective way of enhancing the adhesion in both braking and traction especially in conditions of inclement weather. Therefore, railways have from train-mounted systems that are activated manually and discharge sand on the wheel-rail contact interface when the adhesion problems are taken place.

There are other cases in which the existence of sand on the wheel-rail interface is uncontrollable and unwanted. This is what happens on sandy environments or dessert zones such as Saudi Arabia, where the high-speed train receive a boost in the last few years.

*Figure 10. Example of track infrastructure covered by sand in China,* [9]

The challenge is now to find a solution to remove the sand that recurrently covers the rail track. The reason is that this is translated into an increase in the track maintenance, the change of track bed damping, the reduction of traffic speeds and safety concerns.

Whether it is controlled or not, the reality is that the use of sand causes roughening and damage of wheel and rail surfaces. Therefore, it is important to gain a scientific knowledge of what happens when sand is applied to the wheel/rail contact. The work presented in [10] is noteworthy, however there are still some uncertainties about the influence of sand in the creep forces vs. slip curve. In addition questions such as what is the type of sand that should be used or how much it needs to be applied are unanswered.

In order to gain scientific understanding of what happens to the force vs. slip curve the following system has been used. Figure 11 shows the device that adds on the wheel-rail interface. The sand flow is controlled by a valve.

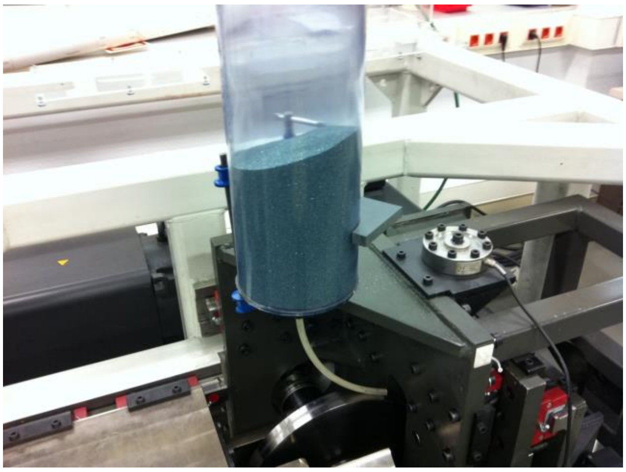

*Figure 11. Device to adds sand on the wheel-rail contact.*

The following three tests are suggested:

1) Constant sand flow and rollers rolling upwards.

2) Worn rollers without sand flow.

3) Constant sand flow with rollers rolling downwards.

In the first measurement test a constant sand flow is set. The roller rolls upwards, which means that the roller on the left rolls anti-clockwise and the one on the left clockwise.

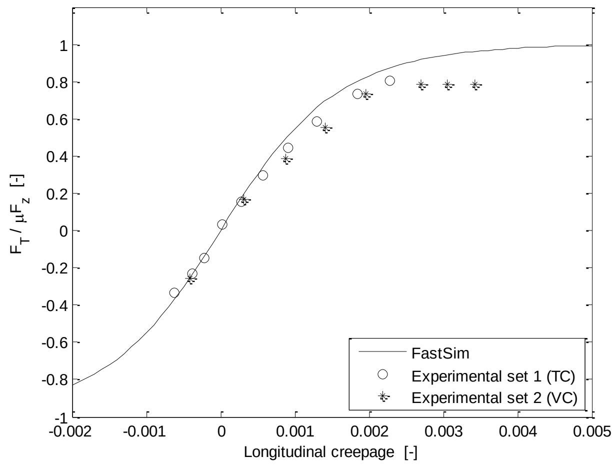


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*Figure 12. Non-dimensionalized tangential force vs. longitudinal creepage (case 1).*

Little sand is able to interfere on the wheel-rail contact since the rolling direction expels the vast majority of grains. As a consequence, the force vs. slip curve does not meaningfully change. However, as shown in Figure 13 the rollers show noticeable surface damage.

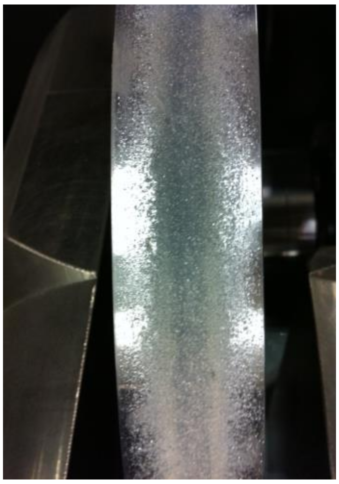

*Figure 13. Worn roller surface after sand test.*

The roller profile seems to conserve its shape. A new test is carried out with the worn rollers.

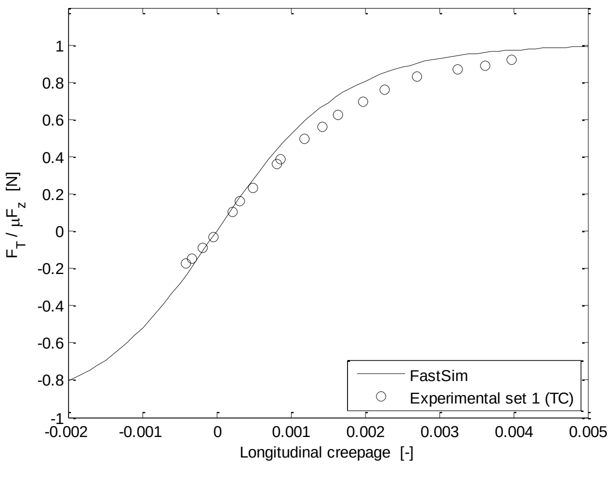


*Figure 14. Non-dimensionalized tangential force vs. longitudinal creepage (case 2).*

Results fit FastSim’s curve, although a change in the initial slope can be appreciated. In the next experiment again sand is introduced into the wheel rail contact. This time, however, the rollers roll downwards boosting the entry of sand into the wheel-rail contact interface.

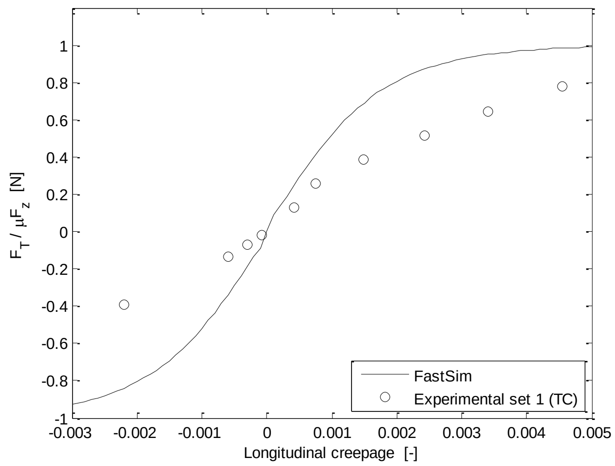


*Figure 15. Non-dimensionalized tangential force vs. longitudinal creepage (case 3).*

Severe roller surface damage and profile change was seen after the addition of sand. The contact between wheel and rail changes considerably under the existence of sand, therefore, FastSim is not able to reproduce the obtained experimental results.

## 4. Conclusions and future research work

The importance of the wheel-rail contact problem in the dynamic behaviour of railway vehicles has led to the publication of different mathematical models that tackle this problem mainly from a purely theoretical point of view. As a result, there are considerable uncertainties concerning the experimental relationships between forces and slips and regarding the influence of some parameters that are not taken into account in the mathematical models. Therefore, the overall objective of the work presented in this paper is to experimentally characterize the wheel-rail contact problem. For that purpose a test-bench was built, and the technical specifications and conceptual design were presented.

The results presented provide a theoretical-experimental comparison of the tangential contact forces as a function of the longitudinal creepage. The comparison with Kalker's Simplified Theory theoretical results has shown good agreement for small longitudinal creepage values. In the non-linear zone, however, the theoretical results give an overestimation of the contact force. In the saturation zone, the experimental results show a decay of the contact force that is not taken into account by the theory.

The influence of the angle of attack between both rollers has been also evaluated. Three different curves have been obtained. The results are equivalent to those obtained for the longitudinal creepage. In the linear zone, the experimental results coincide with the theoretical prediction. However, as the slip is increased both the longitudinal and tangential force present some differences with respect to the theoretical results.

Wheel and rail surface condition plays an important role in the wheel-rail contact problem. The surface conditions have been modified by adding cutting oil, HPF friction modifier and sand. Some initial results have been shown, in which it was observed that the friction coefficient remained constant when using cutting oil. However, more experiments are required, specialle for high creepage values. In addition, wheel-rail contact forces should be tested with more contaminants such as water, which is a very common substance.

## 5. ACKNOWLEDGEMENTS

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