International Journal of Applied Engineering Research ISSN 0973-4562 Volume 12, Number 23 (2017) pp. 13488-13497

Determination of Wheel-Roller Friction Coefficient on Roller Rigs for Railway Applications

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Abstract

Railway technology is strongly influenced by the friction coefficient between wheel and rail. Friction plays a key role in many aspects of railway dynamic such as derailment, wear, noise, stability and is the most important factor to be considered in traction and braking problems.

The railway system is based on the interaction of the profiles of wheel and rail, and their relative motion is subject to the presence of friction, which produces tangential forces acting on the vehicle and determining its dynamic behaviour. In case of traction and braking, the presence of friction is adopted in order to generate the required traction and braking force, and the study of friction in this field is indicated as adhesion and it is fundamental to define the vehicle performance. Since the railway system is subjected to the environmental effects, the friction coefficient can vary due to the presence of contamination, such as water, rust or weaves on the rail surface.

The determination of the friction coefficient is usually obtained through complex and expensive experimental tests performed on the track using a real vehicle. Those tests are mandatory to verify the vehicle safety especially when active system are adopted to control traction and braking (WSP). An alternative to the use of test on the track, is the adoption of test on a roller-rig, a dynamic track simulator where the track is replaced by rollers. The roller rig allows to recreate in a laboratory environment test conditions comparable with the track, but with better repeatability, lower costs and higher safety. Both full scale or scaled similitude prototypes can be used on roller rig, and this strategy was adopted since the beginning of the 19th century to study traction by Carter.

In the present work, a simple and efficient laboratory methodology for the estimation of the value of static friction coefficient on roller-rig is proposed. An accurate determination of the static friction coefficient is mandatory when a roller rig is adopted to study adhesion, wear or vehicle dynamic, in order to be able to perform different tests in known initial condition.

Keywords: Roller-Rig, wheel-rail adhesion, wheel-rail contact, friction coefficient

INTRODUCTION

The estimation of the maximum adhesion between wheels and rails still has a great importance in the railway field, and many studies are carried on to develop methods and algorithms able to predict its behaviour or experimental techniques in order to better understand the phenomenon.

The study of the wheel/rail contact always involves the presence of friction, and historically the models obtained by researchers to simulate this phenomenon have followed different approaches depending on the aim of their studies. In fact when the study is oriented to investigate railway dynamic, the models refer to the friction coefficient, usually considered as a constant value with respect to the vehicle velocity [1,2,3]. When the study is oriented to investigate traction or braking, the friction coefficient is usually indicated as "adhesion coefficient" and considered dependent from the vehicle velocity [4,5,6]. It is evident that in reality the friction or adhesion between wheel and rail represents the same property of wheel/rail contact. The differences found in literature can be explained with the different approaches used to study the two phenomena. Vehicle dynamic and subsequent modelling of the contact mechanics have been defined using a theoretical approach starting with the hypothesis of steady state motion, with a vehicle running at constant velocity.

Traction and braking studies departed from an empirical approach based on experimental tests on the track. Recent contact models are oriented to reduce the gap between the two approaches and obtain an homogeneous solution [7, 8]. The study of wheel/rail adhesion and the measurement of the friction coefficient in order to refine the models, can be achieved using test on the track on real vehicles [9-12] or tests on prototypes on test-rigs [12-17].

The adhesion phenomenon is complex to understand also due to the problems related with repeatability of experimental measurement that are subjected to large variability related with environmental conditions. Furthermore tests performed on real vehicles on the track require expensive and often complex device to measure the friction forces between wheel and rail. For this reason tests on test rig are often more suitable and useful to develop numerical models. The authors have designed and realized a roller-rig in order to perform tests on scaled prototypes [18]. This device has been used in previous works to perform stability [19], and braking [20, 21, 22], traction [23, 24] and wear [25] tests on prototypes in order to validate numerical models of vehicles and wheel-rail/wheel-roller contact algorithms [26, 27].

The roller-rig allows to perform tests on bogies or suspended wheelset and a wide range of configurations is possible considering the mechanical arrangement (gauge, profiles, wheelset spacing, suspension geometry), the axle-load and the suspension stiffness. Previous experiences demonstrate that tests performed on reduced scale prototypes can be obtained with good repeatability, and measurements can be done easily also when it is requested to use different test conditions.

For this reason the roller rig can be used to investigate the adhesion problem. This test can be useful at first to study the problem by a tribological point of view, by measuring the static friction coefficient or the behavior of the tangential forces as a function of the creepages. Furthermore the roller rig can be used to test and improve braking and traction control algorithms on scaled vehicle prototypes. Benefits obtained from this second case are related to the possibility to create the desired adhesion condition for the track, that is simulated by the rollers.

This work mainly deals with the first part of this activity (tribological study) and in this case the test device has been equipped with a single suspended wheelset (shown in fig.1) since in this way the effect of vehicle dynamic can be easily isolated from the adhesion problem.

When operating on a reduced scale prototype on roller-rig two problems must be considered:

- scaling problem;
- finite curvature of the roller.

The first problem has been deeply investigated by the authors [18] and by many other researchers [28]; from these investigation a set of scaling factors can be defined in order to convert the measured values on the scaled prototype in equivalent quantities for the full scale vehicle. Those considerations are also used to properly define mass, inertia, stiffness of the scaled prototype in order to simulate a specific real vehicle.

The prototype has been designed according to the similitude model proposed by A. Jaschinsky [28]. The second problem (finite curvature of the rollers) has different effects on the contact phenomenon [20], and produces a different behavior of a vehicle running on rollers with respect to the same vehicle running on track.

Two main differences can be observed: the first is related to the modification of the shape of the contact patch and the second leads to different creepages formulation. The first effect can be compensated by reducing the transversal curvature of the rollers, respect to the normal rail curvature [18], in order to achieve again on roller the same shape of the contact patch obtained on the rail.

The second effect cannot be compensated but it is possible to estimate its effect by using a numerical model where it is simulated the behavior both on roller and on rails [19]. When measuring the static friction coefficient the influence of the second effect can be assumed to be negligible, since the creepage is vanishing.

All the tests performed on roller-rig are strongly influenced by the friction coefficient between wheel and rail. Therefore, it is very important to identify the friction coefficient during the test being performed, for all the cases considered: braking, traction, adhesion study, wear. The identification of the friction coefficient on the roller-rig is important to make the results of the test comparable with what happens on the real track. The adhesion condition can also vary during the test, especially when contemned surfaces are considered, therefore it is important to have a procedure able to perform the friction measurement efficiently and as fast as possible.

Aim of this work is to develop dedicated device and a procedure to measure the static friction coefficient on the roller rig. The measure can be performed before and during the test (but the system must be obviously stopped during the measure itself). Figure 1 shows the realized device used to measure the friction coefficient of one of the rollers.

International Journal of Applied Engineering Research ISSN 0973-4562 Volume 12, Number 23 (2017) pp. 13488-13497

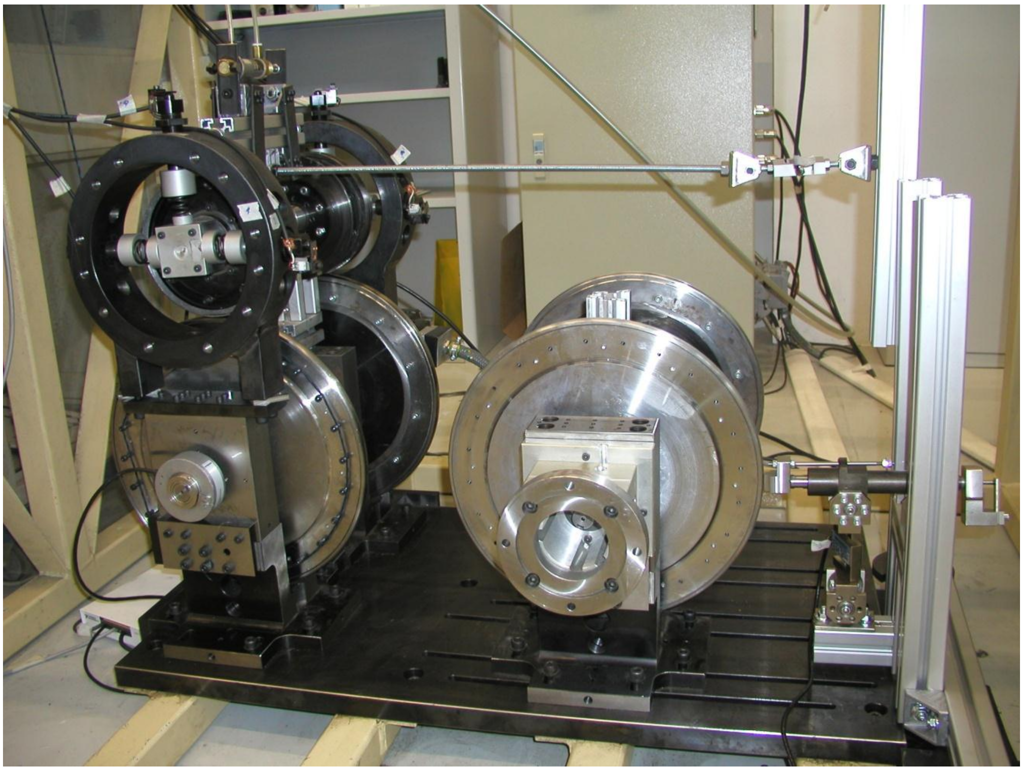
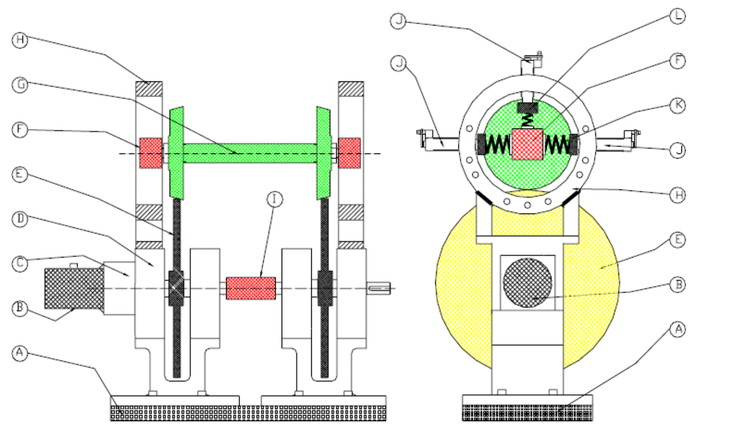


Figure 1: Roller Rig equipped with a suspended wheelset.

EXPERIMENTAL DEVICE

The configuration of the experimental device used for the present work is described in fig. 2. The wheelset (G) is supported by a pair of rollers (E) and it is placed with each wheel tangent to the top of the corresponding roller.

The wheelset is also held in position by four longitudinal springs (K) preloaded (11 Kg in this work) in order to provide an appropriate guidance during the application of a traction or braking torque. The axle load is provided by the proper weight of the wheelset (16,5 Kg) and by an additional preload applied by means of the vertical springs (L). All the loads acting on the suspensions are measured during the test using appositely designed load regulators (J). Fig. 2 shows a detail of the suspension system and preload regulator used during the tests.



A	Main Support.	G	Wheelset
B	Motor.	H	Wheelset suspension support
C	Motor Support.	I	Joint
D	Roller Bearing box.	K	Longitudinal Springs.
E	Roller	J	Load cell.
F	Axle Box.	L	Vertical springs.

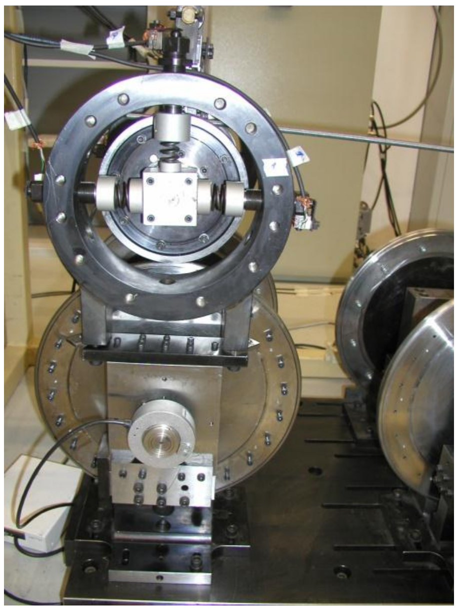


Figure 2: Scheme and picture of the experimental device.

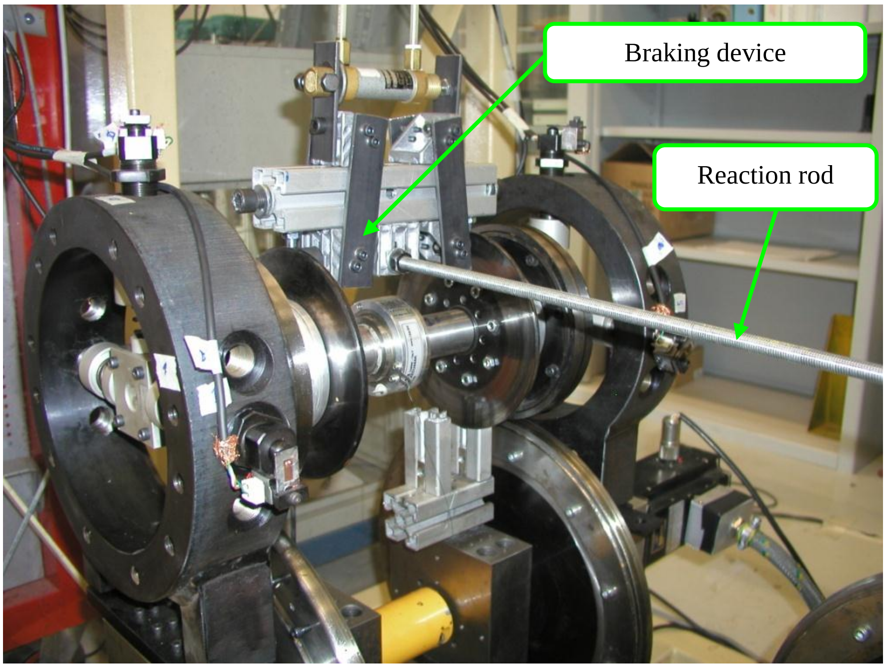


Figure 3: Detail of the braking device.

The wheelset is equipped with a pneumatic braking device that applies a braking force on two discs installed on the wheelset. The braking force is applied with a known distance from the wheelset axis and measured using a load cell (shown in fig. 1) placed on the reaction rod of the braking device, as shown in fig. 3.

The braking force is applied to the disks using two rubber pads pushed against the disk surface by means of a pneumatic cylinder. This system can be used to simulate different level of braking torque by modifying the pressure of the cylinder. During the experimental tests an high value of pressure (8 bar) has been used in order to prevent any movement of the wheelset. Simultaneously a torque has been applied to the rollers using a rope system. The static tests have been performed by braking the wheelset and by increasing in small step the load on the rope (and therefore the torque) until sliding occurs between wheels and rollers. When the static friction limit is reached the rollers start to rotate.

During the test the load on the vertical suspension ($F_{Z,LS}$, $F_{Z,RS}$ for the left and right spring respectively) and the braking force F_b are measured and from this measure the tangential (eq. 1) and the normal load (eq. 2) are calculated.

$$T_{fx} = F_b \cdot \frac{r_b}{r_w} \tag{1}$$

$$N_L = \frac{\left(F_{Z,LS} + F_{Z,RS} + M_w \cdot g\right)}{\cos(\gamma)} \tag{2}$$

Where r_b is the braking radius, r_w the contact radius of the wheel, M_w the non suspended mass of the wheelset, g the gravity acceleration and γ the contact angle.

Eq. 1 assumes that the lateral tangential force can be neglected, and this is suitable if the test is performed as described before. In case the wheelset is rolling and especially with large contact angles the effect of the lateral and spin creepages should be considered [29].

Eq. 2 is valid if the contact angle is the same for the two side, condition achieved with the wheelset centered or with conical profiles.

The maximum value of the ratio between the tangential force and the normal load (which is obtained when sliding occurs) is assumed to be the static friction coefficient, as described by eq. 3.

$$\mu = Max\left(\frac{T_{fx}}{N_L}\right) \tag{3}$$

The determination of the static friction coefficient can be easily performed with the use of constant loads, and this procedure has been preferred in this first phase because it is possible to evaluate accurately the test repeatability. The roller rig and the wheelset are both equipped with dismountable profiles and it is therefore possible to test different materials and surface conditions. For the first test, here described, both the wheel and the roller profiles have been realized in 39NiCrMo5 steel quenched and tempered with a surface hardness of 40-45 HRC. The wheel profile has been grinded with a surface roughness Ra of about 0.2 µm, the roller profile has a roughness of 0.6 µm. It is known that the material properties [29], [30] and the surface roughness [31] can have an influence on the friction coefficient, therefore it is important to be able to verify those parameters during the tests. Once the static friction coefficient has been estimated, it is possible to evaluate the behavior of the tangential forces as a function of the creepages.

Adhesion curve

The behavior of the tangential forces as a function of the creepages is fundamental to study railway vehicles dynamics. Calculation of tangential forces in stationary conditions can be performed according to the theories of Kalker [32], but when traction/braking is involved it is necessary to modify the friction law to better describe the behavior with respect to the experimental results [7].

The scaled roller rig could be used also to perform a direct measure of the creep law in traction/braking conditions. It is possible to operate in two different ways to obtain the measures required to define the creep law. The easiest method consists in imposing a constant velocity to the rollers (the roller rig is motorized) and a constant braking torque to the wheelset. In order to calculate the creepage high resolution encoder have been installed both on the wheelset and on the rollers (the encoder can be observed in fig. 3, it is a 1024 pulses/revolution quadrature encoder) to measure the angular velocities of roller (ω_W) and wheelset (ω_R). The total creepage in case of pure longitudinal sliding can be calculated according to eq. 4.

$$\xi_X = \frac{\omega_W \cdot r_W + \omega_R \cdot r_R}{\omega_R \cdot r_R} \qquad (4)$$

The creep force is instead evaluated according to eq. 1.

In order to build the whole creep law, it is necessary to increase the braking torque increasing the creepage as a consequence, till the wheelset is completely locked (ξ_X=1). This method is simple but during the test it is not possible to keep the creepage constant when the braking force is constant. In fact, also if a constant torque is applied to the braking system, a variation of the friction force along the wheel or roller surface due to stick slip phenomena or to different properties of the surface is possible. The velocity of the wheelset would be therefore modified. The alternative method requires to motorize also the wheelset and to control both the velocities of the wheelset and the rollers. Eq. 4 can be used to define the desired creepage by keeping a steady roller velocity and increasing the wheelset velocity. In this case it is necessary to measure the applied torque with a torquemeter, installed between the roller and the motor or between the wheelset and motor (or both). Also in this case a variation of the tangential force due to different adhesion properties or stick-slip is possible and the consequence would be a variation in the applied torque. Both motors must be driven with a closed loop control and a very high resolution is required, since the creepage value are usually very small (complete sliding occurs with value less than 0.1). This second method is useful to measure the creep law simulating both traction or braking conditions. The method cannot be implemented directly on the existing scaled roller-rig since an high torque is required for the motors. The actual motor has been sized for coasting and stability simulations. In any case in the second part of this project more powerful motor will be installed on the roller-rig in order to provide the torque required to simulate the entire creep law for different railway applications. In order to choose the new motors for the roller rig the values indicated in table 1 have been considered, where *N* is the axle load and *V* the velocity of the vehicle being simulated. In the scaled (1:5) device, according to the similitude model the values of torque *T* and power *P* required can be estimated.

Table 1: Determination of the motor performances.

N	V	μ	T	P
Unscaled			Scaled	
[tonn]	[Km/h]	[/]	[Nm]	[KW]
25	300	0,5	185	37

Since before the test it is important to verify that the condition of rollers and wheel surface is as expected, it has been studied a device in order to measure the static friction coefficient of wheel and roller before the complete test is executed. This preliminary measure must be accurate and fast and it is important to be able to perform it without modifying the setup of roller-rig. For this reason a specific device has been designed to measure the static friction, this device can be applied externally to the wheels and rollers in order to simplify the test.

Device for fast static friction measurement

The device for the fast measurement of the friction coefficient has been realized according to the layout shown in fig. 4. This device is used in alternative to the standard procedure, because it has the possibility to perform a friction measurement when the roller rig is running other tests (traction control, wear, braking), during short suspension of the experimental activity. In this way it is possible to keep under control the variation of the friction coefficient during the test. The operating concept is based on a special head (1) which is pressed into contact with the wheel or roller surface. The head can be easily replaced and therefore it can be realized using different materials and modifying the profile curvature.

The head is pushed in contact with the desired normal load acting on the preload nut (4), which embeds a load cell. In order to keep the normal load constant during the test and not influenced by the run-out error of the rollers, the load is not transferred directly between the nut and the head but through a spring located inside the main body of the device (2).

Once the head is preloaded against the roller (or the wheel) a torque is applied to the roller till it starts to rotate. In this condition a tangential force is generated between the roller and the head and it is transferred to the device supports (5). Under one of these supports a load cell (3) allows to measure the reaction force which is proportional to the tangential force.

Fig. 5a shows different heads realized with the material of wheel/roller and of a real rail. The head used to measure the friction coefficient on the wheel, has the curvature and material corresponding to the roller or of the real rail (scaled), the head used to measure the friction coefficient on the rollers has the curvature and material of the wheelset.

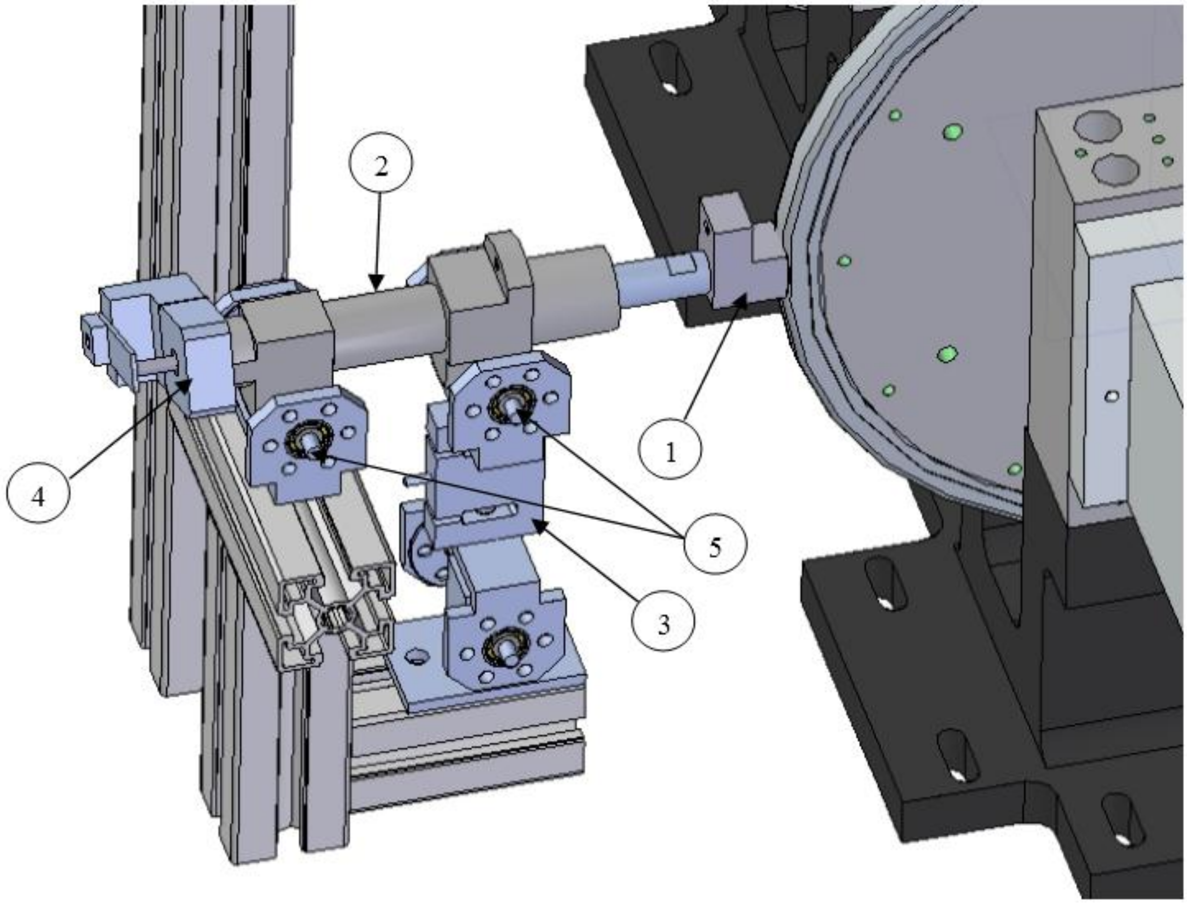


Figure 4: Drawing of friction measurement device.

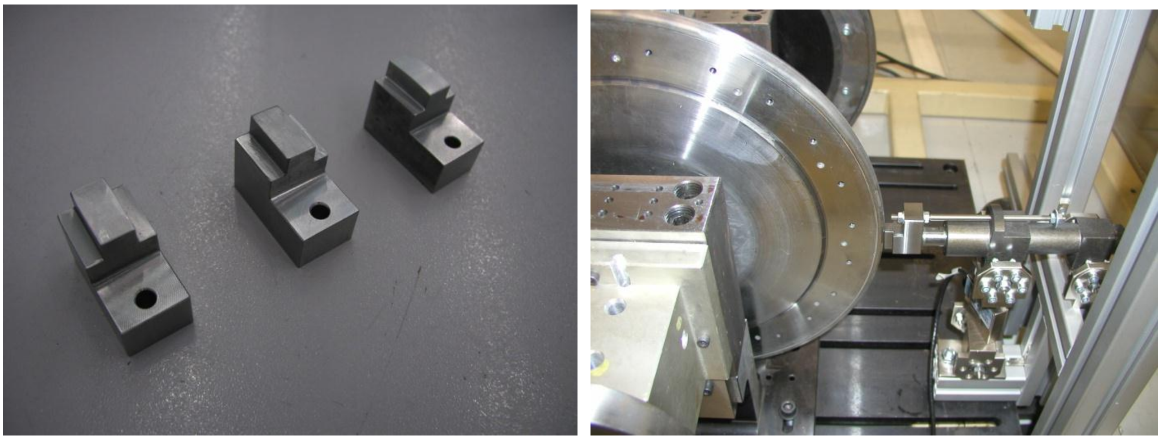


Figure 5: a) Different heads for the friction device. **b)** Detail of friction measurement device.

The friction coefficient can be calculated as a ratio between the normal and tangential force. It is necessary to repeat the measurement operation twice for the roller and for the wheel, using the appropriated head; the two values obtained should be very similar if the condition is the same for the surface of wheel and roller. In any case the mean value can be used as a reference for the static friction coefficient. Fig. 5b shows the realized measuring device installed on the roller.

EXPERIMENTAL RESULTS

The experimental tests are executed using a scaled roller-rig with a single suspended wheelset mounted on it. Aim of the tests is to measure the static friction coefficient in different environmental conditions (dry, wet and reduced adhesion), and to analyze its variability and transient behavior during the initial movement of the wheel/roller. The test has been carried out applying a braking torque to the wheelset in order to prevent its rotation. This torque is accurately measured using the load cell located on the reaction rod of the braking device. The wheelset used for the tests has conical profiles with a conicity corresponding to tread of a new S1002 profile (0.05 rad). The roller profiles have been realized according to the UIC60 rail profile canted 1/20. This profile has been scaled 1:5 and then its transversal curvature has been modified in order to reproduce the same shape of the contact patch considering the finite radius of the rollers (185 mm).

Table 2 shows the main characteristics of the system used for the test. Spring stiffness has been provided only for the axial direction, but the shear/bending stiffness is also important to define the overall property of the vehicle in the different directions. In this work the dynamic of vehicle is not under investigation and therefore only those values are provided.

Table 2: Main characteristics of the test device.

Symbol	Description	Q	unit
Rw	Wheel nominal radius	91.75	[mm]
Rr	Roller nominal radius	185	[mm]
Rb	Braking radius	80	[mm]
Mw	Wheelset mass	16,5	[Kg]
Fx0	Long. spring preload	11	[Kg]
Fz0	Vertical spring preload	5/10/15	[Kg]
Kx	Long. spring stiffness	8622	[N/m]
Kz	Vertical spring stiffness	39805	[N/m]

The wheelset is loaded considering different preloads on the vertical spring in order to investigate the influence of the axle load. Then a torque is applied to the rollers, this torque is generated using constant loads suspended using the rope system previously described. The suspended load is increased in steps until sliding occurs between wheels and rollers. The ratio between vertical and tangential force is calculated for each step during the test (examples are shown in fig. 6, 7 and 8) and its maximum is the local value of the static friction.

Several experiments have been performed in different conditions, and in each case the test has been repeated since an entire rotation of the rollers has been obtained. The tests have been done for three different loads on the vertical springs: 5, 10 and 15 Kg, which correspond to full scale axle loads of 3.3, 4.5 and 5.8 tons.

The considered condition includes:

- dry conditions: roller and wheel surfaces accurately cleaned;
- wet conditions: roller and wheel surfaces wet with water;
- thin soap layer on the roller surface;
- oil layer deposed on the roller surface (ISO VG 64).

The results obtained are summarized in table 3, where mean value and standard deviation is obtained over 4 tests for each condition. The tests have been also confirmed with the fast measuring device obtaining results falling inside the dispersion of the measures performed with the standard technique.

Table 3: Summary of test results

Condition	Vertical spring load	Mean Friction value	Standard deviation
Dry	5 Kg	0,371	0,026
Dry	10 Kg	0,353	0,030
Dry	15 Kg	0,384	0,002
Wet	5 Kg	0,269	0,037
Wet	10 Kg	0,224	0,026
Wet	15 Kg	0,234	0,024
Soap	5 Kg	0,245	0,004
Soap	10 Kg	0,233	0,002
Soap	15 Kg	0,233	0,006
Oil	5 Kg	0,107	0,012
Oil	10 Kg	0,128	0,016
Oil	15 Kg	0,131	0,011

Fig. 6, 7 and 8 show the behavior of the friction coefficient obtained during one of the tests for the dry (fig. 6), wet (fig. 7) and oil contaminated (fig. 8) conditions.

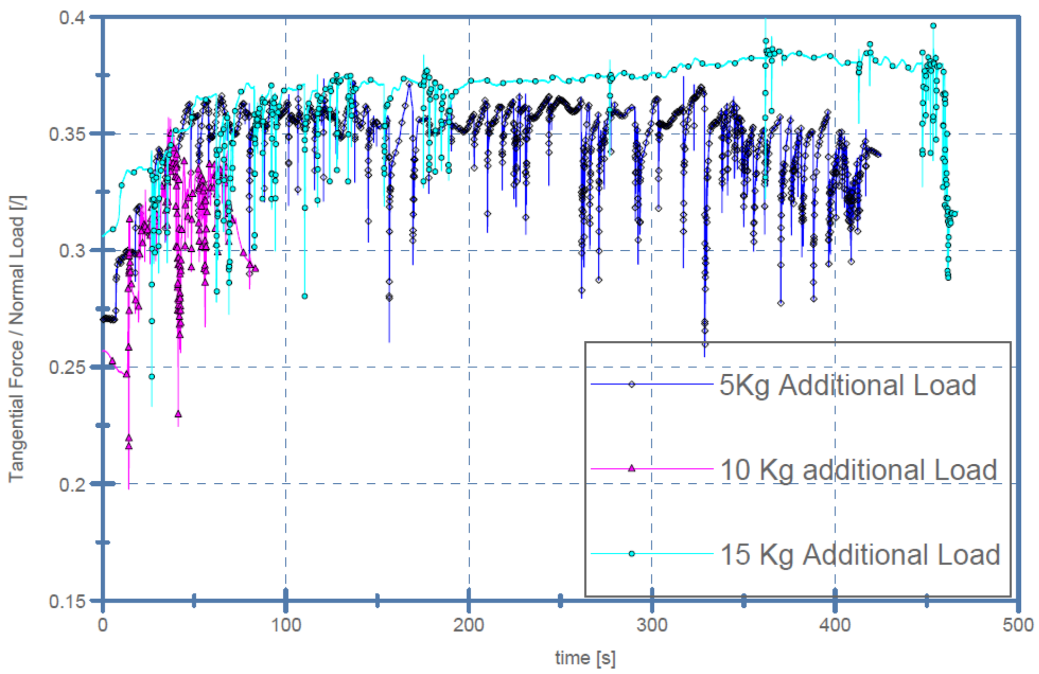


Figure 6: Behavior of the ratio between tangential force and normal load in dry condition.

In each plot at the beginning the load has been increased till the rollers begin to move, since the torque applied to the roller depends from fixed loads, it is not possible to reduce it during the test. Therefore after the static friction limit has been reached, the roller is subject to an angular acceleration since the dynamic coefficient is lower than the static one. In case of dry condition the difference between static and dynamic coefficient is not large (maximum to minimum in fig.6 it is less then 0.1) and it is possible to detect a stick-slip phenomenon.

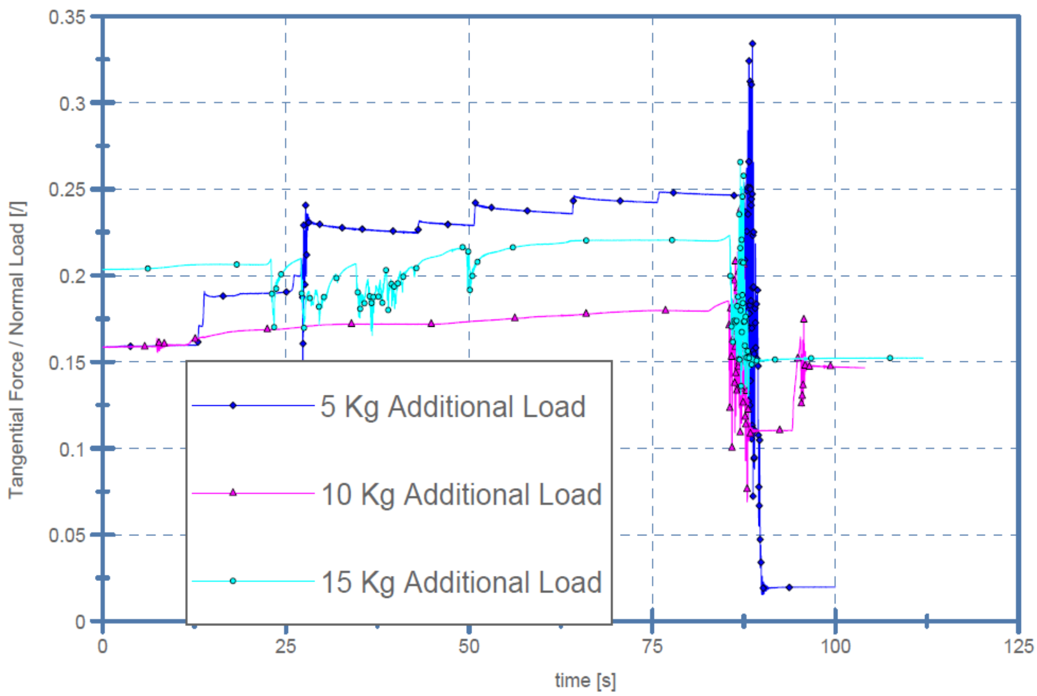


Figure 7: Behavior of the ratio between tangential force and normal load in wet condition.

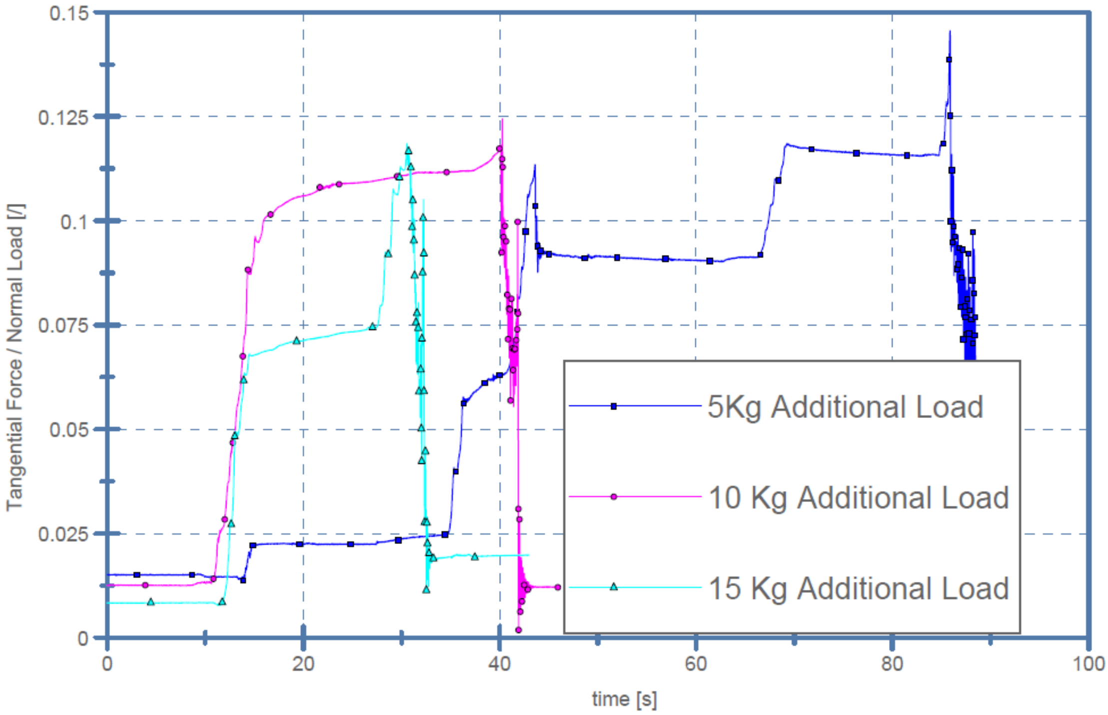


Figure 8: Behavior of the ratio between tangential force and normal load for oil lubricate contact condition.

In case of the wet and oil contamination the difference between the static and dynamic friction coefficient is greater, since before the test begin the load on the wheelset removes part of the oil or of the water in the contact patch. But after the roller starts to slide the contact occurs where more oil or water is present. From data analysis it is shown that the dynamic friction coefficient is reduced more than 75% for oil and 50% for water/soap with respect to the static value. For this reason it is not possible to detect stick slip phenomenon with oil, wet or soap condition.

CONCLUSIONS

The work shows a methodology to estimate the static friction condition on a roller-rig. The determination of the static friction coefficient is important to define the reference friction environmental condition on the roller before to start with more complex tests aimed to investigate traction, braking, wear or adhesion. The test device and the relative methodology have been studied and realized and preliminary tests have been performed in different conditions of the surfaces. The tests performed in dry and wet condition are in good agreement with those obtained on real vehicles where 0.36-0.4 is indicated for dry and 0.28-0.31 for wet condition.

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