Jose A. Romero Navarrete & Frank Otremba, Int. J. Transp. Dev. Integr., Vol. 4, No. 2 (2020), 142–151

A TESTING FACILITY TO ASSESS RAILWAY CAR INFRASTRUCTURE DAMAGE. A CONCEPTUAL DESIGN

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ABSTRACT

Wheel forces generate stresses in the rail as a function of several vehicle and infrastructure characteristics and operating conditions. The different components of the wheel forces develop strains in the rail which contain an elastic and hysteretic (irreversible) components. The irreversible deformations of the rail would be associated with locomotive energy losses. In this paper, a testing facility is proposed to indirectly characterize the level of stresses in the rail, in terms of the energy that is lost during turning manoeuvres. Different potentially influential factors are considered, including the friction at the centre plate, the wheelbase length, the distance between bogies and the radius of the curved track. The change in the potential energy during a U-turn displacement is measured. In this respect, an experimental model under this operating principles, aimed at validating such a principle of operation, reveals a significant effect of the friction at the centre plate on the energy lost during turning manoeuvres, and consequently, on the level of stresses in the rail.

Keywords: energy losses in transportation, experimental methods, friction energy, rail damage, turning forces, wheel forces.

1 INTRODUCTION

Transportation energy efficiency in terms of energy per tonne-km depends on a multitude of factors, involving the vehicle, the operator, the infrastructure and the environment. Energy demands for transportation is expected to have a steeping growth, where the freight in Europe, for example, is expected to grow by 58% from the level in 2010 [1]. While road transportation has been the main transport mode for the transportation of goods in Europe, with 49.8% of the total, while the railway takes 12.3% of the cargo, it is expected that in 2050, current road freight over 300 km be switched from road to railways [1]. Any increase in the use of the railway transportation instead of the road transportation would derive from the lower rolling resistance of the railway versus the roadway transportation. In this respect, Fig. 1(a) illustrates the force necessary to displace a rail- or roadway vehicle, per unit ton, in absolute terms [2], and Fig. 1(b) of this figure describes the roadway situation but normalized to the railway situation. According to these data, the roadway transportation implies a larger energy consumption to generate the motion of the cargo, as a function of the displacement speed, with a maximum value that is 6.5 times the energy consumption of the railway transportation. Other factors influencing the value of the needed displacement force in the case of these two transport modes, include the aerodynamic resistance, which would be less in the case of the railway transportation, due to the single vehicle front. On the other hand, it should be noted that the railway damage concentrates in small areas of the rail, while the pavement damage spreads on a larger area. However, the greater efficiency of the railway transportation is clear. While this energetic advantage of the railway transportation with respect to the roadway transportation is not arguable, there are some phenomena in the railway transportation that affect such energetic efficiency. One of such situations concerns the turning manoeuvres of the railway cars, where due to both the yaw resistance and the friction at the centre plate or pivot, the tangential wheel–rail forces significantly increase during such operations. This situation has led to upgrading the material specification for those track portions [3].

ISSN: 2058-8305 (paper format), ISSN: 2058-8313 (online), http://www.witpress.com/journals
DOI: 10.2495/TDI-V4-N2-142-151

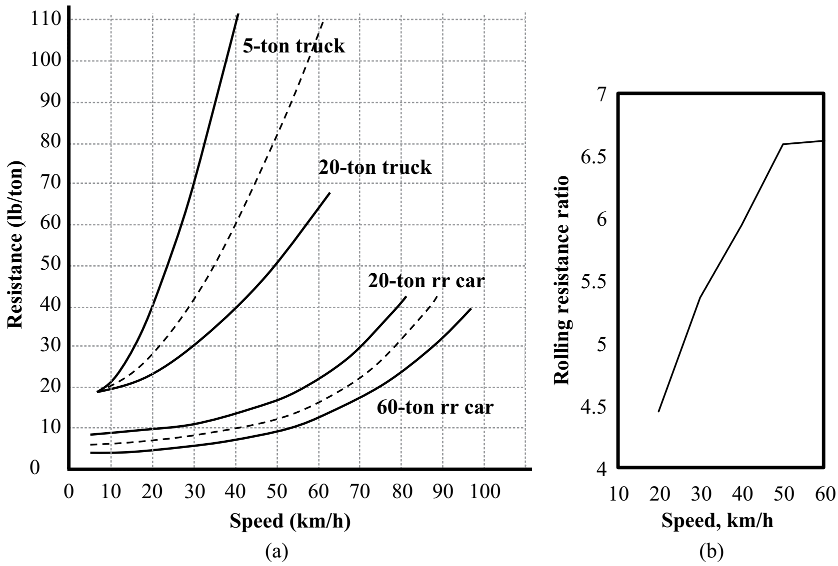

Figure 1: Rolling resistance force per unit ton: (a) absolute values and (b) roadway values, normalized to railway values (Source: part (a) adapted from [2] and (b) own).

Concerning the turning manoeuvres, the reactional vertical forces in the case of the railway transportation also become relatively greater than in the case of the roadway transportation, as the position of the cargo's centre of gravity is higher in the case of the railway transportation, while the track width is shorter. Figure 2(a) illustrates this situation in the case of tankers, where a unit horizontal force exerted along the centre of gravity of the cargo, would create a 48% larger vertical reaction in the rail than in the pavement. Figure 2(b) of this figure illustrates situations concerning the railway transportation, where lateral forces would generate vertical reactions in the rail that would depend on the height of the centre of gravity of the cargo and of the whole vehicle. The two vehicles shown in this figure would represent extreme situations for the heights of the centre of gravity of the cargo in the railway transportation. Considering the different types of cargoes, another influencing factor affecting the level of reaction forces developed in the rail is identified when it is considered the motion of the cargo within its container, that is, for example, when the carried cargo is liquid. The mobility of the cargo in the vehicle would have both a dynamic and a steady state effects. While the steady state would involve the shifting of the cargo when a lateral or a longitudinal acceleration is exerted on it, the dynamic condition would involve the interaction of the cargo with the body-chassis of the vehicle. Figure 3 schematically describes this situation, for both the steady and the dynamic state conditions, where the dynamic case is illustrated through the sloshing cargo–pendulum analogy.

The resultant force at the wheel–track interface has components along the normal and tangential directions. In general, the relationship between the magnitude of the wheel–rail forces and the consequential friction force and potential damage of the rail's material, keeps

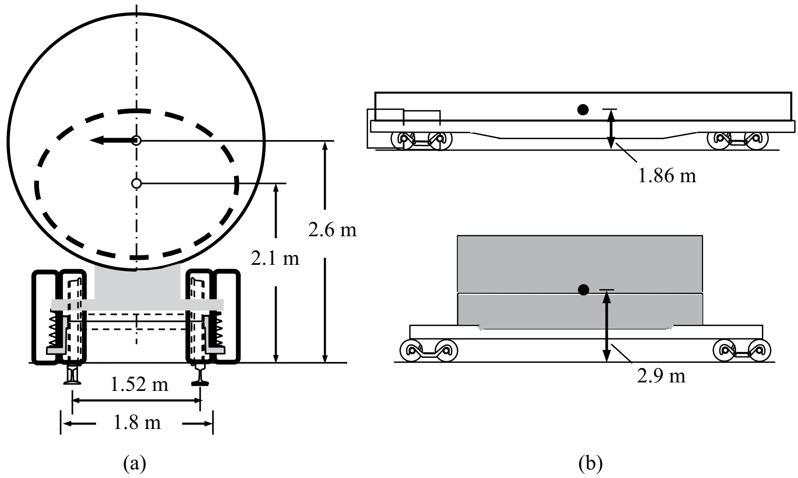

Figure 2: (a) Comparison of the geometry of a road- and rail-tanker to calculate the reactions in the rails and (b) centre of gravity of two cargos representing potential extreme positions for this height (gondola and double-deck cars).

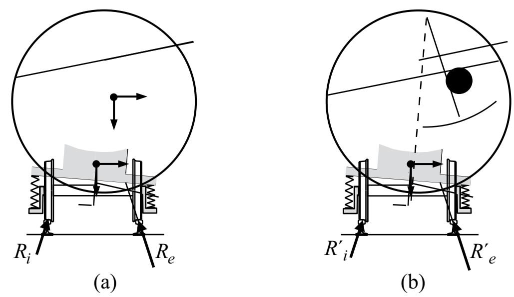

Figure 3: (a) Steady state condition of a liquid cargo during turning and (b) dynamic condition with sloshing-pendulum analogy.

a nonlinear relationship, which is associated with the deformation of the surfaces that are in contact [4], in such a way that the friction coefficients increase with the magnitude of the forces. Consequently, in a vibrational environment such as a car body–axle dynamic interaction, where the magnitude of the forces between these bodies oscillate around their static or steady state values, the forces above such static values will have an effect on the magnitude of the friction, which will not be compensated by the effect of the forces below such static or steady state force magnitude. The wheel–track forces represent a three-dimensional force situation, where the resultant forces have steady and dynamic components associated with the modal vibration of the vehicle (roll, pitch, bounce and yaw), with both normal and tangential components, where these last ones can be decomposed in lateral and longitudinal directions, as it is illustrated in Figure 4. The railway stresses are thus the result of the complex variation of these different forces, whose value will depend on the traveling situation.

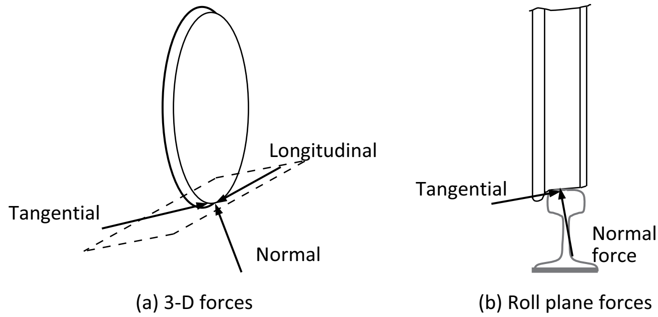

Figure 4: Force decomposition at the wheel–track interface.

In the case of the longitudinal force, its magnitude depends on whether it is a traction wheel or a driven wheel. If it is a driving wheel, the traction force will have the opposition of the rolling resistance force. Another longitudinal force will derive from the yaw vibration of the vehicle. In the case of the normal force at the wheel–track interface, its value will depend on the vibration characteristics of the vehicle, involving lateral and longitudinal load transfers. The magnitude of the tangential forces depends on the lateral displacement of the vehicle, and on the contact condition of the wheel flange with the rail. While in the long term these forces originate some plastic deformation in the rail, in the form of wear, the elastic recovery of the rail material implies that in the short term, the irreversibility of such deformation process implies that some locomotive energy is lost. Consequently, measuring such a loss of energy during the displacement of the vehicle would imply a characterization of the level of stress in the rail. This is the primary effect considered in this paper, that is, a greater locomotive energy loss will imply higher stresses and rail damage.

It is the purpose of this paper to describe the conceptual design of a scaled-down testing facility to indirectly assess the stresses in the rail through measuring the energy loss during the displacement of a railway car during a turning manoeuvre, as a function of different vehicles and track characteristics, including the friction at the centre plate, the height of the centre of gravity of the cargo and the curving radius of the track. Such conceptual design involves the definition of the different modules confirming the facility, as well as the respective principles of operation.

2 CONCEPTUAL DESIGN

The different stages for equipment's traditional conceptual design, begins with an explanation of what is expected from the equipment, trying to describe as precisely as possible the needs and the potential restrictions [5]. This is a crucial stage from which the definition of the modules and their respective operational principles will be set. Once the conceptual design is done, the next stage consists in providing some elements that validate such design, generating feedbacks for correcting/improving the design. The basic idea for this device involves the measuring of the lost energy as an objective indicator of how the level of stress in the rail is. Figure 5 illustrates the basic idea for this testing facility, which is to measure the potential energy that is lost due to the friction of the vehicle with the rail during turning. The vehicle is released at one side of the U-shaped track supported on a tilt table; then, it reaches a final position at different potential energies. The difference in such energy would be linked to the wheel–rail forces/stresses/damage. The testing procedure should consider the parametric effect of each of the factors to be considered.

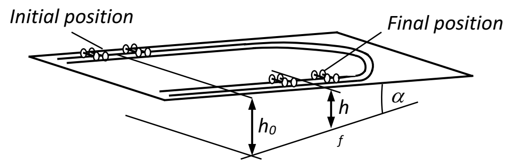

Figure 5: Diagrammatic representation of the testing principles.

2.1 Definition of needs

While there would be a long list of the factors influencing the energy and stresses in the rail, the most relevant for the purpose of this paper are: vehicle speed, the level of friction at the centre plate, the bogie's wheelbase length, the distance between the bogies in the car and the height of the centre of gravity of the cargo. A series of restrictions can also be identified, which will be the result of the circumstances around the design, implying research policies and economy and time limitations. In this case, one of the most important restrictions concerns the available space for this testing facility. In this respect, a surface of 10 m × 10 m × 3 m is the maximum volume for this equipment and includes all the space required for the primary and secondary equipment. It should be noted that the purpose of this testing device is in no way to obtain absolute values for the level of stress at the rails, but to provide a table of the comparative effect of the different parameters on the level of stress, measured in terms of the loss of potential energy. As a continuation of this work, however, a series of different wheel threads could be considered, according to the current standards, as an additional parameter whose effect on the energy losses should be assessed [6].

2.2 Dimensional characteristics of the testing rig

From different perspectives, the most important characteristic of the testing rig consists of the dimensions of the scaled-down model. While a properly scaled model can describe realistic situations under the right calculations of the scaling constants, some practical issues about the testing rig instrumentation and the handling of the equipment, definitively affect the decision about the right size of the device. In this respect, there is a background information that concerns a bogie-type vehicle that was used to study the effect of a liquid cargo on the wheel–track forces in the roll plane [7]. It represented a 1/10 scale down with respect to the full-size equipment. The wheelbase of such an equipment is 210 mm.

The wheelbase length of the vehicle is the most important vehicle dimension, as it further defines the minimum radius for the curved track, and from there the other dimensions of the vehicle. However, according to the design needs, different values for the wheelbase length were needed, starting from this minimal value for *WB*. The relationship between the wheelbase length and the curved track's minimum radius is calculated in terms of the chord that the bogie represents when going along a curved track. Figure 6 illustrates such a relationship, together with the outputs of the resulting equations:

$$b^2 + \left(\frac{WB}{2}\right)^2 = r^2 \,;\ \frac{WB}{2} = r\sin\theta \,;\ r^2\sin^2\theta - \left(\frac{WB}{2}\right)^2 = 0 \tag{1}$$

These data will be used to define the other characteristics of the testing facility.

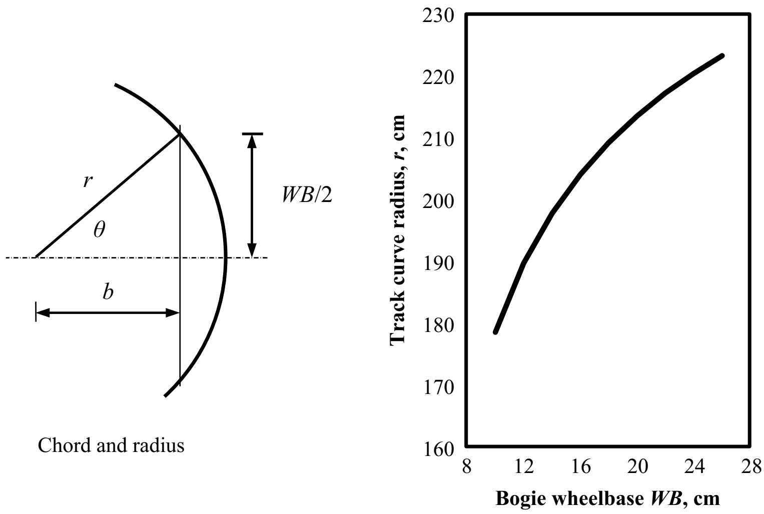

Figure 6: Bogie's wheelbase *(WB)* and track minimal radius relationship.

2.3 Module definitions

The different components of the testing facility will be selected or dimensioned based on a modularization approach. The different modules and their interactions are presented in Fig. 7. These are: vehicle (*VE*), track (*TR*), tilt table (*TT*), handling system (*HS*) and measuring system (*MS*). Within *TT* are the frame sub-module (*FR*) and the positioning system (*PS*). The principles of operation for each of these modules are now defined.

Module vehicle (*VE*): To change the friction at the centre plate, different principles could be considered, including a spring actuated upper dish, or a dead mass acting on this component. In this respect, the spring actuated upper tray is the simplest and easy way to adjust the friction at the centre plate. This equipment would not need a primary suspension, so that only the secondary suspension would be necessary. In this respect, however, there is a need to maintain the alignment of the lateral frames and the bogie bolster, together with the lateral positioning of the rolling bearings between the lateral frames and the wheelset. To maintain the alignment of the bolster and the lateral frames, some lateral plates will be considered. Figure 8 illustrates the resulting conceptual design of the vehicle.

Module track (*TR*): This critical component should provide a smooth rolling surface to the vehicle, while not showing deformations that affect or introduce some noise into the experiment. That is, the surface should not have preferably any joint that could affect the rolling

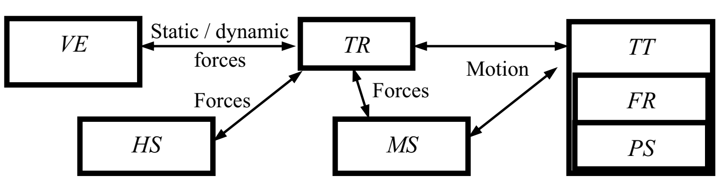

Figure 7: Modules and their interactions.

smoothness of the vehicle and potentially affect the aim of the experiment. In this respect, it can be considered a certain level of simplification for the track profile, by substituting the real shape of the track rail for a simple rounded hollow bar. This is illustrated in Fig. 9, where the actual rail profile and the rounded one are shown together with a fastening device of the rail to a rigid frame. In this respect, the location of the cross hole to support the supporting pin must avoid any potential contact with the wheel.

Tit table frame (*FR*): This component must support the vehicle–track systems, providing a sufficiently stiff infrastructure to avoid any unacceptable noising vibration going into the ride of the vehicle. Concurrently, this frame should be light in order to avoid much power going into the positioning mechanism. Consequently, an aluminium plate is proposed which is supported by a steel open-section truss. That is, no welding should be involved in order to reduce the level of distortion and lack of planarity at the aluminium surface. Figure 10 illustrates this construction concept, which allows the fastening of the rounded rails on the aluminium surface. The positioning mechanism and the arrangement of the aluminium plates and the three tracks with different radiuses are described in Fig. 11. A hydraulic actuator is considered to give the different angles for the tilt table.

Measuring system (*MS*): This system will consist of linear dimensions measuring instruments. In this respect, the surface of the tilt table would be equipped with a scaled illustration in order to relate the vertical position with the longitudinal position, to eliminate any potential lack-of-perpendicularity in the measurements.

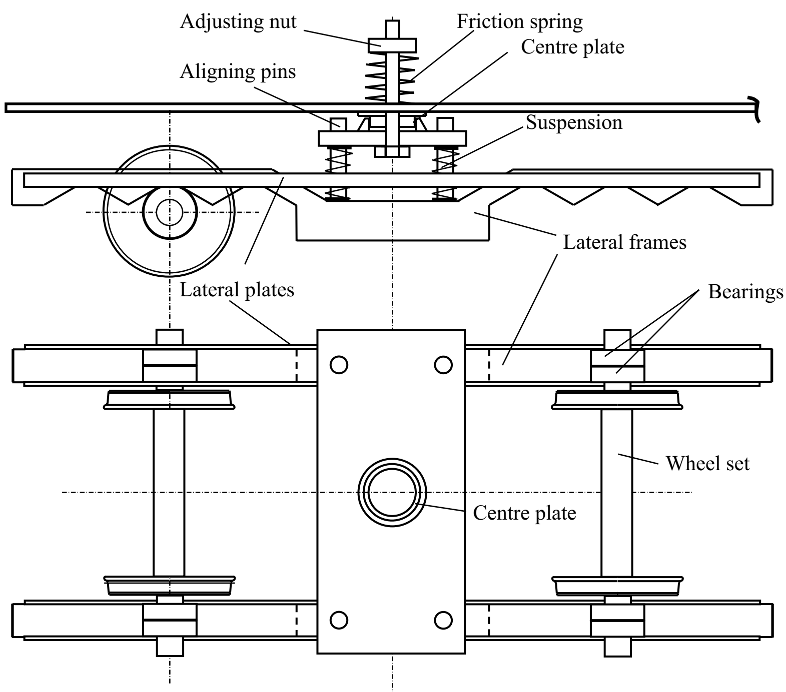

Figure 8: Vehicle configuration.

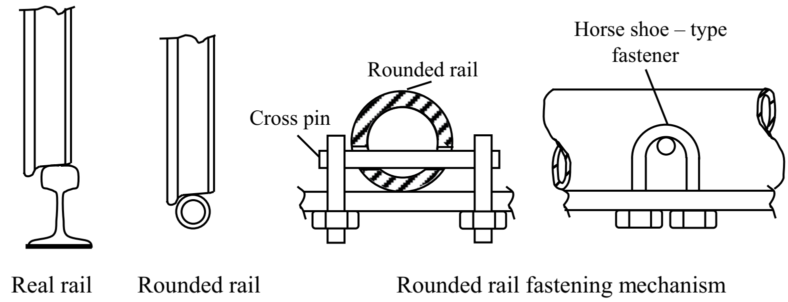

Figure 9: Simplified hollow rounded track and its fastening system.

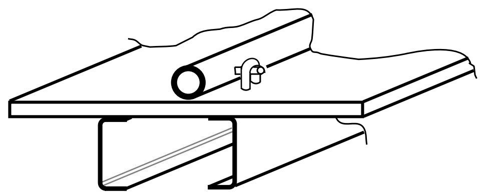

Figure 10: Detail of rounded rail attached to the aluminium plate and the tilt table frame.

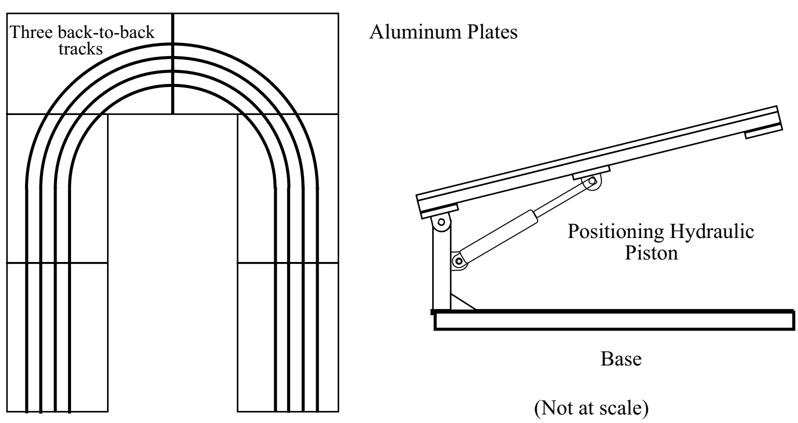

Figure 11: General configuration of the tilt table and positioning system.

2.4 Experimental modelling and validation of the operational principles

To assess the potentials of the proposed operational principles for indirectly measuring the stresses in the rail during turnings as a function of a variety of parameters, a preliminary testing rig was built and tested. It included a very small version of the testing rig proposed herein. Figure 12 illustrates some details of the device. The results in this figure suggest that there is a significant effect of the friction torque on the energy that is lost during the displacement of the vehicle along a U-turn manoeuvre. Concerning the validation of the operational principles proposed for the testing device, it is noted that a mathematical model should be associated with the experiments. In this respect, however, there is available a family of logical supports that could be used to simulate the vehicle–track interaction under the testing conditions proposed in this paper. The experimental outputs would thus be used to calibrate the respective logical tools to estimate the outputs at full-scale environments. As it was explained above, during turning, part of the vehicle's kinetic energy is transformed into heat due to the friction at the centre plate, as a function of the dynamic response of the vehicle to the infrastructure inputs. Consequently, the simulation counterpart for the experimental scheme proposed in this paper would involve a combination of logical supports. One of such potential simulation schemes consists of using a multibody-dynamics systems simulation tool, such as Vampire, and then using Track-ex to calculate the energy that is dissipated in the contact patch due to the tangential force and creepage, as reported by Joy and Tournay [8]. Such dissipated energy would thus correlate with the kinetic energy that is lost during the testing manoeuvres. Another computational scheme would combine the outputs of Simpack as the multibody simulation software [9], with contact mechanics models, including the FASTSIM algorithm, as reported by Trummer [10].

3 CONCLUSIONS

Under the assumption that the lost energy during turning manoeuvres correlates with the level of stresses in the rail due to vehicle wheels, it has been proposed in this paper a testing facility to characterize the effect of several vehicle, infrastructure and operational conditions, on the level of stress developed in the rail. The outputs are to be considered in a relative basis, that is, no absolute values are expected from these testing, but rather an objective ranking of

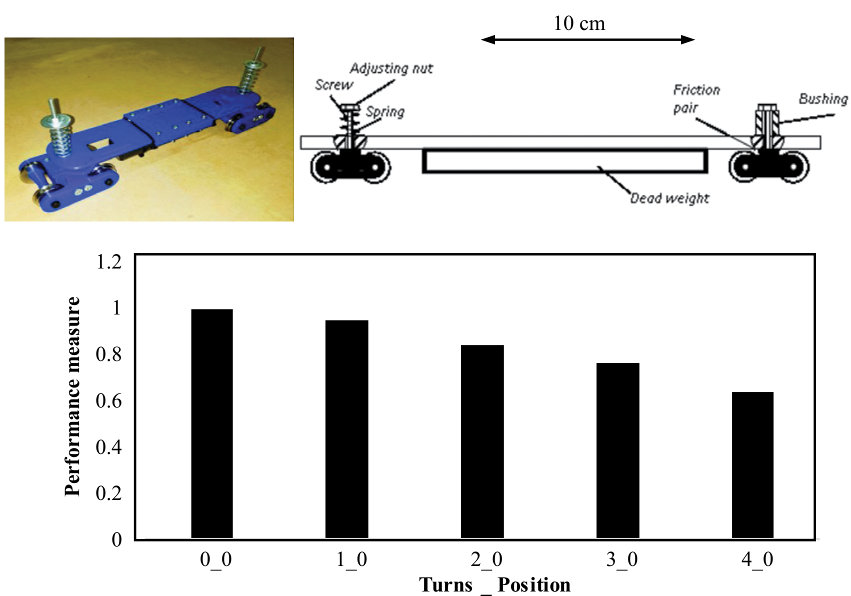

Figure 12: Preliminary results concerning the effect of the friction at the centre plate (turns of the screw actuator) on the relative loss of energy.

the riel unfriendliness of the parameters considered. The preliminary testing of this testing principles suggests that the energy lost due to friction plate, is significant. The theoretical counterpart could be developed on the basis of calibrated existing models that could allow the estimation of the outputs at full-scale situations.

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